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Abstract

In the present work an outdoor experimental investigation for solar air heater with arc-shaped apex-upstream flow by the use of circular cross section wires as roughness elements has been carried out. The roughness-element have been expressed in non-dimensionalizing geometric parameters as relative roughness-pitch (P/e), relative roughness-height (e/D) and flow attack-angle ($\alpha/60$), and the range of these parameters varies from 8 to 15, 0.0454, and 0.75 to 1.25, respectively.

For evaluation of performance of the roughened SAH, a novel parameter has been proposed and introduced in the present investigation which is Thermo-Hydraulic Improvement Parameter (THIP). With the use of present roughness geometry, considerably Nusselt number enhancement ratio (NNER) and friction factor enhancement ratio (FFER) have been observed. The maximum NNER and FFER values obtained experimentally is about 2.83 and 1.79 times, respectively. While, the maximum THIP has been obtained 157.49 % higher than the smooth SAH. Using the experimental results correlations for the output parameters (Nusselt number and friction factor) as a function of input parameters (flow and roughness) have been developed.

Keywords: Solar air heaters; Nusselt number Enhancement Ratio; Friction factor Enhancement Ratio; Thermo-hydraulic Improvement Parameter.

65 Nomenclatures

A_c	Surface area of absorber plate (m^2)	p_a	Atmospheric pressure (N/m^2)
A_o	Cross sectional area of orifice (m^2)	Q_u	Useful heat gain (W)
C_{pa}	Specific heat of air (J/kg K)	V	Velocity of air (m/s)
C_d	Coefficient of discharge for orifice meter	T_{sun}	Sun temperature (K)
e	Height of roughness (m)	T_g	Cover glass Temperature ($^{\circ}C$)
G	Mass velocity of air (m/s)	T_{fo}	Outlet air temperature ($^{\circ}C$)
h	Heat transfer coefficient (W/m^2K)	T_s	Sky temperature (K)
h_w	Convective heat transfer coefficient due to wind (W/m^2K)	T_{fi}	Inlet Air Temperature ($^{\circ}C$)
H	SAH Duct depth (m)	T_a	Ambient temperature ($^{\circ}C$)
I	Intensity of global solar radiation (Insolation) (W/m^2)	T_{pm}	Mean absorber plate temperature ($^{\circ}C$)
K_a	Thermal conductivity of air (W/mK)	T_{fm}	Mean air temperature ($^{\circ}C$)
K_g	Thermal conductivity of glass cover (W/mK)	T_{bm}	Mean temperature of bottom plate ($^{\circ}C$)
K_i	Thermal conductivity of glass wool insulation (W/mK)	e/D	Relative roughness height
L	Length of solar air heater duct (m)	f	Friction factor
L_1	Spacing between glass cover and absorber plate (m)	Pr	Prandtl number
L_g	Thickness of glass cover (m)	P/e	Relative roughness pitch
$\dot{m}$	Mass flow rate of air (kg/s)	Re	Reynolds number
ΔP	Pressure drop across the collector duct (N/m^2)	Nu	Nusselt number

66

67 Greek Symbols

68

μ	Absolute viscosity of air ($N\ s/m^2$)	η_{MTHIP}	Modified Thermo-hydraulic Improvement Parameter
ρ	Air density (kg/m^3)	Suffix	
ρ_m	Manometric fluid density (kg/m^3)	T	Temperature

α	Angle of attack ($^{\circ}$)	GI	Galvanized Iron
σ	Stefan-Boltzmann's constant ($\text{W/m}^2 \text{ K}^4$)	SAH	Solar Air Heater
δ_i	Thickness of insulation (m)	R	Roughened
β	Tilt angle of collector surface ($^{\circ}$)	S	Smooth
ν	Air kinematic viscosity (m^2/s)	r	Roughened
$\alpha/90$	Relative angle of attack	s	Smooth
ε_p	Emissivity of absorber plate		
η_{th}	Thermal efficiency		
β_R	Ratio of orifice diameter (D_2) to pipe internal diameter (D_1).		

Introduction

Solar energy is the emergence source of energy that's abundant in nature and cheaper to work with and acquire monumental utilizations, moreover directly or by turning it into useful convenient forms of energy (Duffie and Beckman 1974). It can be utilized by conversion into useful heat or thermal in many manners with using solar energy conversion systems (Duffie and Beckman 1974; Garg and Prakash 2000). Flat plate solar collectors are in use globally for solar energy conversion systems and processes due to the simple construction framework, simple functionality and initial low price for the installation.

Various researchers have noticed that the energy conversion system's performance is low because of inadequate heat transfer/exchange between the absorber plate and working fluid and hence higher heat loss to the ambient (Sukhatme and Nayak 2011).

Investigators have integrated several heat transfer enhancement methods like artificial roughness, packed bed materials, extended surfaces and baffles, V-groove, wavy and corrugated profile plates, winglets and turbulators on the absorber plate to enhance the flat plate solar air heater performance (Kabeel et al. 2017; Sahu et al. 2019; Sahu et al. 2016).

Among the above stated heat transfer enhancement techniques, artificial roughness improves the convective heat transfer between the air and absorber plate in a better way and hence we obtained

improvement in the thermal performance of the SAH. It breaks the laminar sub-layer formulation and effectively reattaches the airstreams in the boundary layer for better heat transfer. These artificial roughness also increase in the friction factor and thus requires more pumping power compare with conventional SAH.

Sahu and Prasad (2016) have reported a review article on the use of various types of artificial roughness elements, its geometries, and working mechanics for rectangular SAH ducts.

Artificial roughness of different roughness elements is being arranged systematically in a repeated form on the absorber plate called roughness geometries. The arrangement of repeated roughness elements, guides the secondary flow in the stream-wise direction and helps to obtain the uniform augmentation of heat transfer all over the absorber plate.

Due to use of roughness, the turbulence gets produced in the core (main) flow and also there is development of secondary swirls /vortices just behind of these roughness elements (Sahu and Prasad 2016) . These phenomena resulted in better augmentation of convective heat transfer from absorber plate to the fluid. Many investigators used the circular cross-section small wires for fabricating the various geometry designs of the artificial roughness. These roughness geometries are named transverse, continues inclined, continues V-shape, arc shape and further it is modified into broken, multiple broken and discrete designs , which have been reported by the investigators.

Aharwal et al. (2008) modified the inclined ribs with providing a sinle gap and by this they obtained enhancement in heat transfer and friction factor values of 2.59 and 2.87, respectively.

The performance of W-up and W-down ribs are compared by Lanjewar et al. (2011). Maximum thermohydraulic performance has been attained for W-down ribs at an angle of attack of 60°.

Momin et al. (2002) and Lanjewar et al. (2011) have developed the correlations for Nusselt number and friction factor for the V and W shaped roughness configurations of the SAHs, respectively.

Singh et al. (2014) experimentally investigated effects of multiple arc ribs on heat transfer and friction factor and reported developed correlations for (Nu) and (fr).

Multiple-arc with symmetrical gaps geometry has been investigated by Kumar et al. (2021). They found 5.57 times enhancement in Nusselt number and 4.46 times enhancement in friction factor.

Kumar et al. (2019) optimized the thermal performance by response surface methodology (RSM) for discrete multiple arc-shaped rib roughness SAH.

Kumar et al. (2020) modified the multiple arc shape ribs into multiple arc with gaps ribs. Pandey et al. (2016) introduced gaps in multiple arc shape and experimentally investigated the effects of broken arc roughness designs on heat transfer and friction factor.

Gill et al. (2017) proposed the novel staggered-rib attached with the broken arc roughness which improves the Nusselt number and friction factor by 2.63 and 2.44 times compared to smooth duct SAH. All the researchers have attached the ribs with a circular cross-section as the artificial roughness over the absorber plate and developed the Nusselt number and frictional factor correlations as a function of design and operating conditions.

Kumar et al. (2021) have compared the exergetic efficiency of triangular SAH with semicircular and square roughness geometries and reported that semicircular ribbed triangular SAH is 26% more than square roughness.

Saini and Saini (2008) carried out the experimental analysis and developed the correlations for arc-shape with apex downstream flow SAH. Saini and Saini (2008) have analyzed the influence of relative roughness height (e/D) and relative angle of arc ($\alpha/90$) while the relative roughness pitch (P/e) was constant for the arc shape with apex downstream flow SAH.

Sahu et al. (Sahu and Prasad 2016; Sahu and Prasad 2016; Sahu and Prasad 2017; Sahu and Prasad 2017) further optimized the arc-shaped down stream roughened SAH based on the effective, exergy and entropy generation based analysis respectively and obtained the design plots as a function of the temperature rise parameter.

Further (Ghritlahre et al. 2020; Sahu et al. 2021) have reported their experimental work to explore the consequence of different orientations, i.e., apex-downstream and apex-upstream flow of arc shape roughness on heat transfer, friction factor, thermal efficiency and pumping/mechanical work of SAH duct and found that apex-upstream flow has better performance over to apex-downstream.

From the above literature survey it is found that no authors have developed and reported the correlations for Nusselt number and friction factor having apex upstream- arc-shape roughened absorber plate on the rectangular SAH.

The aim of present work is to develop the correlations for Nusselt number and friction factor by using the experimental data.

Furthermore, a novel parameter, Thermo-hydraulic Improvement Parameter (THIP) has been presented for comparing the enhancement by using the roughness on absorber with respect to smooth plate SAH for same operating parameters.

Experimental program

The experimental test rig has been designed and made up of according to the guidelines of ASHRAE standards (1977). The experimental setup consists of a wooden duct in a rectangular cross-section, and it comprises entrance, test and exit sections as shown in Fig.1. An induced draft blower with a capacity of 3.5 kW is used to suck the air from the ambient.

Fig. 1 Schematic view of the Experimental Set-up.

The duct is designed with a span of 2150 mm, the width of 330 mm and height of 30 mm. In test section of 1200 mm is covered with transparent glass to permit solar radiation as a test section among this total length. As per the guidelines, the ASHRAE duct should have minimum entry and exit sections; therefore, the duct has 650 mm entry and 300 mm exit length, respectively, with an aspect ratio (W/H) of 11. The base and sides of the duct have been made up of 19 mm thick plywood. Both ducts contain a 4 mm glass sheet on top and a 3 mm G.I. sheet as a back plate. Glass wool with the thickness of 50 mm has been given between the bottom plate and plywood to minimize the bottom heat loss to the ambient. The volumetric flow rate of air for both SAH ducts are controlled to desirable values with the help of the main control valve.

Roughness and flow parameters

A 1 mm thick G.I. sheets with 1200 mm× 330 mm specifications was utilized in smooth and roughened SAH ducts. In addition, a Circular wire having a 2.5 mm diameter was used for the roughness element.

Fig. 2 shows the roughened absorber plate with arc-shaped roughness with Apex-up flow. Table-1 shows the details of roughness geometry and operating parameters that have been considered in the present experimentation.

Test data were recorded in outside conditions from February to May in clear sky and on bright days between 10.00 to 14.00 hours at an interval of half-hour.

For each test conducted, the information such as absorber plate temperatures, outlet and inlet air temperatures, pressure drop across the duct were taken once the system obtained a steady state.

The steady-state was considered that when the temperature remains the same at least 30 min.

Table. -1. Range of roughness and operating parameters

Parameters	Name of Parameter	Symbol	Values/ Range
Roughness	Relative roughness pitch	P/e	8-15 (3 Values)
	Relative roughness height	e/D	0.0454
	Angle of attack	α	45°- 75° (3 Values)
Operating	Reynolds number	Re	2983-13955 (6 Values)

Fig. 2. Schematic and Photograph of arc shape roughness on absorber plate with **Apex-up stream** flow.

Data reduction

The air mass flow rate ' $\dot{m}$ ', was measured by a calibrated orifice meter which is calculated by (Lanjewar et al. 2011; Pandey et al. 2016).

$$\dot{m} = C_d A_o \sqrt{\frac{2\rho_a(\Delta P)}{1 - \beta_R^4}} \quad (1)$$

The pressure drops (ΔP_o) across the orifice meter has been calculated by using the Eq. (2) which is given below (Lanjewar et al. 2011; Pandey et al. 2016).

$$\Delta P_o = \rho g \Delta h \left(\frac{\rho_m}{\rho} - 1 \right) \quad (2)$$

To assess the air temperature at the inlet and outlet of the test segment and the absorber plate, digital temperature indicators that provide output/results in °C with the precision of 0.1 °C have been utilized. Fig. 3 shows the places and locations of temperature indicators.

Fig. 3. Digital temperature indicators positioning on Absorber Plate and at entry and exit sections of the duct.

Average temperatures values of air (T_{fm}) and average temperatures values of absorber plate (T_{pm}) are calculated by Eq. (3) and (4) respectively.

$$T_{pm} = \frac{T_1 + T_2 + T_3 + T_4 +T_{15}}{15} \quad (3)$$

$$T_{fm} = \frac{T_{fi} + T_{fo}}{2} \quad (4)$$

The heat gained by the air passing through the SAH duct was calculated by using the Eq. (5)

$$Q_u = m.C_{pa}(T_{fo} - T_{fi}) \quad (5)$$

205 By applying the Newton's law of cooling the convective heat transfer coefficient between air and
206 absorber plate is evaluated by Eq. (6)

$$h = \frac{Q_u}{A_c(T_{pm} - T_{fm})} \quad (6)$$

207 Using the Eq. (7) the Nusselt number is evaluated which is given by

$$Nu = \frac{hD_h}{K_a} \quad (7)$$

208 Where

209

$$D = \frac{2.W.H}{(W+H)} \quad (8)$$

210

211 Friction factor (f) for both the collectors test sections were calculated by using the Eq. (9) (Lanjewar
212 et al. 2011; Pandey et al. 2016; Kumar 2019).

$$f = \frac{2.\rho.(\Delta P_D)D}{4L_f G^2} \quad (9)$$

213

214 Where

215

$$G = \left(\frac{\dot{m}}{W.H} \right) \text{ is mass velocity of air inside the duct.}$$

216 The Reynolds number (Re) was evaluated by using the Eqn. (10)

$$Re = \frac{GD}{\mu} \quad (10)$$

217 **Comparison of Performance Parameter**

218 Equations 11 to 15 and Table-2 detail the various performance parameters used for the
219 analysis and comparison of SAH.

220 **Table 2:** Various performance parameters used for SAH and its development and formula
221 given by different authors.

S.No	Year	Investigator	Performance Parameters	Parameters Mathematical Relations	
1	1974	Duffie and Beckman	Thermal efficiency (η_{th})	$\frac{\text{Solar energy absorbed by flowing air}}{\text{solar energy received on the absorber area}}$	$(\eta_{th}) = \frac{mC_p(T_{fo} - T_{fi})}{IA_c}$
2	1990	Cortes and Piacentini	Effective efficiency (η_{eff})	$\frac{\text{Useful Heat Gain} - \left[\frac{\text{Pump Work}}{\text{Conversion Factor}} \right]}{\text{solar energy received on the absorber area}}$ <p>Where</p> <p>Useful Heat Gain (Qu) = $mC_p(T_{fo} - T_{fi})$</p> <p>Pump Work (W_p) = $\frac{m(\Delta P_D)}{\rho}$</p> <p>Conversion factor (C_f) = 0.18</p>	$(\eta_{ieff}) = \frac{Qu - \left(\frac{W_p}{C_f} \right)}{IA_c}$
3	1972	Webb and Eckert	Thermo-hydraulic Performance Parameter (η_{THPP})	$\frac{\text{Nusselt Number Enhancement Ratio (NNER)}}{\text{Friction Factor Enhancement Ratio (FFER)}}$ <p>Where</p> $NNER = \frac{(Nu)_R}{(Nu)_S}$ $FFER = \left(\frac{f_R}{f_S} \right)^{1/3}$	$(\eta_{THPP}) = \left[\frac{(Nu)_R / (Nu)_S}{(f_R / f_S)^{1/3}} \right]$
4	1975	Lewis	Thermo-hydraulic Performance Parameter (η_{THPP})	$\frac{\text{Stanton Number Enhancement Ratio (SNER)}}{\text{Friction Factor Enhancement Ratio (FFER)}}$ <p>Where</p> $SNER = \frac{(St)_R}{(St)_S}$ $FFER = \left(\frac{f_R}{f_S} \right)^{1/3}$	$(\eta_{THPP}) = \left[\frac{(St)_R / (St)_S}{(f_R / f_S)^{1/3}} \right]$
5	2017	Sahu and Prasad	Thermal Efficiency Improvement Factor (TEIF)	$\frac{(\text{Thermal Eff. of Rough Surface}) - (\text{Thermal Eff. of Smooth Surface})}{(\text{Thermal Eff. of Smooth Surface})}$	$(TEIF) = \left[\frac{(\eta_{th})_R - (\eta_{th})_S}{(\eta_{th})_S} \right]$

6		Present Study	Thermo-hydraulic Improvement Factor (ε_{THIF})	$\frac{\text{Nusselt Number Improvement Factor (NNIF)}}{\text{Friction Factor Improvement Factor (FFIF)}}$ <p>Where</p> $NNIF = \left[\frac{(Nu)_R - (Nu)_S}{(Nu)_S} \times 100 \right]$ $FFIF = \left[\frac{(f_R) - (f_S)}{(f_S)} \times 100 \right]$	$(\varepsilon_{THIF}) = \left[\frac{(NNIF)}{(FFIF)} \right]$
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Artificial roughness in the SAH duct augments the heat transfer; at the same time, besides this, it is also responsible for increases in the friction or pumping power as a penalty; hence, the roughness geometry ought to be able to enhance the heat transfer with least raise in friction factor (Cortes and Piacentini 1990). Therefore for comparison of augmentation in thermo-hydraulic performance parameter (THPP), i.e., modified surface as compared to reference (smooth) surface, a parameter has been investigated and reported by Webb and Eckert (1972).

$$\eta_{THPP} = \frac{\left(\frac{Nu_R}{Nu_S} \right)}{\left(\frac{f_R}{f_S} \right)^{1/3}} \quad (11.a)$$

The Eq. 11 (a) given by Webb and Eckert (1972) for evaluating thermo-hydraulic performance parameter (THPP) and it can be stated as:

$$THPP = \frac{\text{Nusselt Number Enhancement Ratio (NNER)}}{\text{Friction Factor Enhancement Ratio (FFER)}} \quad (11.b)$$

Where

(NNER) is Nusselt Number Enhancement Ratio which is evaluated by

$$NNER = \frac{(Nu)_R}{(Nu)_S} \quad (12)$$

and

(FFER) is Friction Factor Enhancement Ratio which is evaluated by

$$FFER = \left(\frac{f_R}{f_S} \right)^{1/3} \quad (13)$$

Another parameter has been investigated and proposed by Lewis (1975) which is given as:

$$\eta_{THPP} = \frac{\left(\frac{St_R}{St_S} \right)}{\left(\frac{f_R}{f_S} \right)^{1/3}} \quad (14.a)$$

240
 241 The Eq. 14 (a) given by Lewis (1975) for evaluating THPP can be stated as:

242

243

$$THPP = \frac{\text{Stanton Number Enhancement Ratio (SNER)}}{\text{Friction Factor Enhancement Ratio (FFER)}} \quad (14)$$

244 Where (SNER) is Stanton Number Enhancement Ratio which is evaluated by

$$SNER = \frac{(St)_R}{(St)_S} \quad (15)$$

245 While the (FFER) is evaluated by Eq. (13).

246

247 It is observed by authors that THPP is less sensitive towards the FFER. The deviation between the
 248 terms (f_R/f_S) and $(f_R/f_S)^{0.33}$ has been found 48 % to 64 % for the range of roughness and other
 249 parameters with (Re) 2900 – 14,000, which has been used for the present study.

250 Therefore, in order to evaluate in equal proportions of enhancement in heat transfer (Nu) and friction
 251 factor (f) a new parameter has been proposed and introduced which is more realistic named as
 252 Thermo-hydraulic Improvement Parameter (THIP), and it can be evaluated as

$$(\varepsilon_{THIP}) = \left[\frac{(NNIF)}{(FFIF)} \right] \quad (16)$$

253 Where

254 (NNIF) is Nusselt Number Improvement Factor which is calculated by

$$NNIF = \left[\frac{(Nu)_R - (Nu)_S}{(Nu)_S} \times 100 \right] \quad (17)$$

255 and

256 (FFIF) is Friction Factor Improvement Factor which is calculated by

$$FFIF = \left[\frac{(f_R) - (f_S)}{(f_S)} \times 100 \right] \quad (18)$$

257 Where

258

$(\varepsilon_{THP}) > 1$ means NNIF values is higher and dominated the FFIF, which is desirable (19)

Validity test

The experimental Nu_{exp} and f_{exp} for smooth plate SAH has been calculated and compared with values obtained from the Dittus-Boelter equation for (Nu) and modified Blasius equation for (f) as (Rohsenow et al. 1998; Bhatti and Saha 1987)

$$Nu_s = 0.024 Re^{0.8} Pr^{0.4} \quad (20)$$

$$f_s = 0.085 Re^{-0.25} \quad (21)$$

The comparison plots are shown in Figs 4 and 5. The comparison results shows that the Nusselt number values have a mean deviation of $\pm 2.3\%$. Likewise, experimental values of friction factor (f_{exp}) with data obtained by modified Blasius equation shows the average of deviation of $\pm 2.5\%$.

Fig. 4. Comparison of experimental and theoretical values of (Nu_s) for smooth SAH.

Fig. 5 Comparison of experimental values and theoretical values of (f) for smooth SAH.

An uncertainty or error analysis has been carried out as per the procedure described by Kline and Mc Clintok (1953). The uncertainties associated with various parameters have been calculated, and the values are ranges from 1.8 to 2.10% for Reynolds number, 1.42% to 4.78% for Nusselt number and 3.6% to 6.89% for friction factor. The evaluation of uncertainties for different parameters have been given in Appendix-B.

Results and Discussion

The Nusselt number and friction factor for smooth and arc-shaped roughened (Apex up streamflow) SAH have been evaluated, and their performances were compared based on THPP and THIP. The detailed results have also been presented in the following subsections.

Influence of Relative Roughness Pitch (P/e)

The influence of relative roughness parameter on the Nusselt number and friction factor have been presented in figures 6 to 9, for the Reynolds number which is ranging from 2900 to 14,000

with fixed relative roughness height ratio of $(e/D) = 0.0454$, and relative angle of arc $(\alpha) = 60^\circ$. The figures show that the Nusselt number has been increasing, and the friction factor has decreased with an increase in Reynolds number for all the values of P/e and smooth plate SAHs. The presence of artificial roughness makes a significant improvement in Nusselt number and friction factor compared with the conventional system.

From Figs. 6 and 7, it is observed that the maximum value of Nusselt number and Nusselt number enhancement ratio was attained at the relative roughness pitch (P/e) value of 8. The maximum value of NNER were found 2.83 for $(Re) = 13955$, $(P/e) = 8$, $\alpha = 60^\circ$ and $(e/D) = 0.0454$. Further increasing in P/e value decreases the heat transfer rate. This happens due to that increase in P/e values not reattach the primary flow with secondary flow, which reduces the number of reattachment points and formulate the boundary layer that acts as a barrier to heat transfer. This result agrees with the results reported by (Lanjewar et al. (2011); Singh et al. (2014); Hans et al. (2010); Kumar et al. (2017); and Hans et al. (2010).

Fig. 6 Variation of Nusselt number with Reynolds number for different relative pitch (P/e).

Fig. 7 Nusselt Number Enhancement Ratio (NNER) with Reynolds number for different relative pitch (P/e).

From the Figs. 8 and 9, it is observed that the maximum value of friction factor and friction enhancement ratio was attained at the relative roughness pitch (P/e) value of 8. The maximum value of FFER was found 2.25 for (Re) =13955, (P/e) =8, $\alpha = 60^\circ$ and $(e/D) = 0.0454$. The presence of rib roughness creates flow turbulence and offers more flow resistance hence friction factor for roughened surface is more as compared to smooth SAH.

Fig. 8 Variation of Friction factor with Reynolds number for different relative pitch (P/e).

Fig. 9 Friction factor enhancement ratio (FFER) with Reynolds number for different relative pitch (P/e).

Influence of angle of arc (α)

The influence of angle of attack (α) on the Nusselt number and friction factor have been presented in Figs. 10 to 13, for the Reynolds number ranging from 2900 to 14,000 with fixed relative roughness height ratio of $(e/D) = 0.0454$, and relative roughness pitch $(P/e) = 8$.

The figures show that the Nusselt number has been increasing, and the friction factor has been decreasing with an increase in Reynolds number and for all the values of (α) and smooth plate SAHs.

From the Figs. 10 and 11, it is observed that the maximum value of Nusselt number and Nusselt number enhancement ratio was attained at the angle of arc (α) value of 60° . The Nusselt number increases with a rise in the angle of arc from 45° to 60° . Further increment in an angle of arc reduces heat transfer.

This happened due to fact that the angling of arc at this value in the form of artificial roughness improves the Nusselt number and friction factor due to the formulation of more moving vortices as compared to other values of (α).

When all the design parameters remain constant, the maximum value of NNER was found as 2.83.

Fig.10 Nusselt number with Reynolds number for different angle of attack (α).

Fig. 11 Nusselt Number Enhancement Ratio (NNER) with (Re) for different values of (α).

By angling the roughness elements flow vortices effectively move from the leading edge of the rib to the trailing edge and formulate the high heat transfer zones. Further raise in an angle of arc (α) from 60° restrict the flow vortices and creates the dead zones.

This result agrees with the results reported by (Lanjewar et al. 2011; Singh et al. 2014; Hans et al. 2010; Kumar et al. 2017 and Hans et al. 2010).

From Figs.12 and 13, are plotted to shows the effect of different (α) on friction factor and friction enhancement ratio. It is observed that the maximum values of friction factor and friction enhancement ratio was attained at the angle of arc (α)= 60°.

Fig. 12 Friction factor with Reynolds number for different angle of attack (α).

Fig. 13 Friction factor Enhancement Ratio (FFER) with Reynolds number for different angle of attack (α).

The friction factor increases by increasing (α) values from 45° to 60° then it starts reduces while further raising in arc angle values. It happens because when the angle of arc is equal to 60° ,

the flow vortices offer more resistance over the leading edge of the rib and improve the friction factor associated with pressure drop.

Thus, it can be seen from Fig.13 that the friction factor enhancement ratio ranges from 1.6 -2.1 in the range of operating conditions.

Thermo-Hydraulic Performance Parameter (THPP) based Comparison

The thermo-hydraulic performance parameter (THPP) based comparison between present work and other roughness configurations of SAHs (Momin et al. 2002; Lanjewar et al. 2011; Kumar et al. 2009; Gupta et al. 1997; Saini and Saini 2008) is shown in Fig. 14.

It can be seen from this Fig.14 that the arc-shape apex upstream roughened SAH curve follows the similar pattern to the SAHs, having absorber plate with different roughness geometries, which validates the precision of the results obtained from the present experimental work.

It can also be seen from Fig.14 that arc-shaped apex upstream roughened SAH had a maximum value of THPP as compared to the THPP attained by other roughened SAHs for $Re > 7459$ (Approx.).

Fig.14 Thermo-hydraulic Improvement Parameter (THPP) comparison of present roughness geometry with others roughness geometries.

The THIP values with Reynolds numbers (Re) for present roughened SAH and other SAHs with different roughened absorber plate geometrical configuration under identical conditions have been evaluated and presented in Appendix-A, Tables- A1 to A6 for comparison.

It has been observed from Fig.15 that Thermo-hydraulic Improvement Parameter of present roughened SAH increases with (Re). It also observed that the obtained THIP value for the present (arc shape) roughened SAH is having greater than one for $Re > 5000$ (Approx.)

Fig.15 Comparison between Nusselt Improvement factor (NNIF), Friction factor Improvement factor (FFIF) and Thermo-hydraulic Improvement Parameter (THIP) for present roughness geometry SAH.

Fig. 15 also exemplifies the relationship between NNIF, FFIF, and THIP with Reynolds number for optimum values of roughness parameters. All these three parameters are increases with increases in (Re). The NNIF and FFIF increase from 0.555 to 1.834 % and 0.816 to 1.108 %, respectively. The present Thermo-Hydraulic Improvement Parameter represents the actual performance of the SAH duct by considering the pressure loss enhancement ratio to the Nusselt

number enhancement ratio. For present apex upstream SAH, THIP is found in the range of 65.82 % to 157.49 %.

Correlations for Nusselt number and friction factor

It is found in previous sections that for present roughened SAH, (Nu) and (fr) are closely determined by roughness and flow parameters. Thus, (Nu) and (fr) concerning roughness and flow parameters could also be written as:

$$Nu = Nu(Re, P/e, \alpha / 90) \quad (22)$$

$$f = f(Re, P/e, \alpha / 90) \quad (23)$$

Based on statistical regression analysis, correlations have been developed for (Nu) and (fr) in relation to roughness geometry and flow parameters by using the recorded experimental data.

Nusselt number Correlations

A functional relationship between Nusselt number and Reynolds number is accessed by plotting a graph for (Nu) and (Re). All the data points of the Nusselt number acquired from experimental results are plotted on a log-log scale against (Re), as depicted in Fig.16.

Fig.16 $\ln (Nu)$ Vs $\ln (Re)$

The curve fitting with a least-square fit for all data points, and it is identified that (Nu) has a linear association with (Re) (Kumar et al. 2020; Kumar V 2019). Further, it is revealed that the average slope is 1.1589. Consequently, it can be written as

$$Nu = A_0 Re^{1.1589} \quad (24)$$

The coefficient A_0 in the above equation influences the remaining roughness parameter that is (P/e) . Therefore, $Nu/Re^{1.1589}=A_0$ was computed and plotted in log-log scale as a part of (P/e) , as shown in Fig.17.

Fig.17 $\ln (Nu / Re^{1.1589}) = \ln (A_0)$ VS (P/e)

The regression analysis shows that linear relationship between the data and the variation of the function is obtained as

$$\frac{Nu}{Re^{1.1589}} = B_0 \left(\frac{P}{e} \right)^{-0.1467} \quad (25)$$

Similarly, the variation of coefficient B_0 in the Eqn. (25) is derived as a function part of other roughness parameters such as relative angle of arc ($\alpha/90$). Consequently, $Nu/Re^{1.1589} (P/e)^{-0.1467} = B_0$ was computed and plotted in log-log scale as a part of relative arc angle ($\alpha/90$), as shown in Fig.18.

Fig.18 Plot of $\ln (Nu / Re^{1.1589} (P/e)^{-0.1467}) = \ln (B_0)$ VS $(\alpha/90)$

By carried out the second-order regression analysis on the data points yields the relationship as follows.

$$\frac{Nu}{Re^{1.1589} \left(\frac{P}{e} \right)^{-0.1467}} = C_0 \left(\frac{\alpha}{90} \right)^{-2.66} \exp \left(-2.7723 \left(\ln \left(\frac{\alpha}{90} \right) \right)^2 \right) \quad (26)$$

The regression analysis obtains the value of the coefficient C_0 . Further, by rearranging the Eqn.26, the relationship for Nusselt number is developed, and it is stated as

$$Nu = 1.340 \times 10^{-3} Re^{1.1589} \left(\frac{P}{e} \right)^{-0.1467} \left(\frac{\alpha}{90} \right)^{-2.66} \exp \left(-2.7723 \left(\ln \left(\frac{\alpha}{90} \right) \right)^2 \right) \quad (27)$$

Friction factor Correlation

The identical techniques of statistical analysis have been utilized once again to derive the friction factor correlations. Further, the relationship between the parameters is shown in Figs. 19-21.

Fig.19 $\ln(fr)$ VS $\ln(Re)$

Fig. 20 $\ln(Nu / Re^{-0.1603}) = \ln(A_1)$ VS $\ln(P/e)$

Fig. 21 $\ln(Nu / Re^{-0.1603} (P/e)^{-0.1505}) = \ln(B_1)$ VS $\ln(\alpha/90)$

The regression analysis results the final correlation as follows.

$$fr = 8.113 \times 10^{-2} \text{Re}^{-0.1603} \left(\frac{P}{e} \right)^{-0.1505} \left(\frac{\alpha}{90} \right)^{-1.0365} \exp \left(-1.0716 \left(\ln \left(\frac{\alpha}{90} \right) \right)^2 \right) \quad (28)$$

Comparison of experimental and predicted value of (Nu) and (fr)

Fig. 21 and 22 show the deviation between experimental data (Nu) and (fr) with predicted values by developed co-relations via Eqn. 27 and 28, respectively.

Fig.22. Comparison between Experimental and predicted Nusselt number

Fig.23. Comparison between Experimental and predicted friction factor

The predicted Nusselt number and friction factor deviation vary from the original values by a maximum of $\pm 8\%$ and $\pm 5\%$, respectively. Further, the regression coefficient for the correlations are 0.99 and 0.98, respectively. From these statistical results, it is concluded that the correlations can predict the values of Nu and fr with the required accuracy for the lower and upper bound values of design and operating conditions. The suggested novel parameter can be utilized to compare the various design of solar air heaters and optimize its design parameters. Due to higher values of MTHIP, the arc-shaped roughness designs can be integrated with jet impingement and packed bed solar air heaters to enhance its performance.

Conclusions

The following main conclusions were drawn based on the present experimental work and are given as:

- For both the artificially roughened and flat plate SAHs, the heat transfer augmented with the increase in the values of Reynolds number at the same time friction factor for both ducts reduces with increase in Re.

- The maximum value of the thermal performance of the SAH was obtained at relative roughness pitch $(P/e) = 8$, angle of attack $(\alpha) = 60^\circ$ and relative roughness height $(e/D) = 0.0454$.
- Compared to the smooth absorber plate present, apex upstream roughened SAH has enhanced the NNER by 2.83 times and FFER by 1.83 times for the range of design and operating conditions as investigated.
- Nusselt Number Improvement factor (NNIF) and Friction Factor Improvement factor (FFIF) are 1.83 and 1.10, respectively, for the investigated range of parameters.
- The maximum thermo-hydraulic performance parameter (THPP) for present roughened SAH has been found 2.21 times, while the Maximum Thermo-Hydraulic Improvement Parameter (THIP) has been found 157.49 %.
- Statistical Correlations have been developed for the Nusselt number, and friction factor as a function of roughness geometry and flow parameter of SAH roughened with arc-shape apex-upstream flow.

510 Ethical Approval

511	Not Applicable
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533 **Consent to Participate**

534	Not Applicable
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556 **Consent to Publish**

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579 **Authors Contributions**

580 All authors contributed to the study conception and design of the manuscript. Material preparation,
581 data collection and analysis were performed by all authors: Mukesh Kumar Sahu^{a,b}, Manjeet
582 Kharub^c, M. M. Matheswaran^d

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648 **Availability of data and materials**

649	Not Applicable
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APPENDIX-A

VARIOUS PERFORMANCE PARAMETERS AND IMPROVEMENT FACTORS

Table A 1: Various performance parameters and Improvement Factors for Arc Shape rib SAH, Saini and Saini (2008)

	Reynolds Number	Nusselt No. Enhancement Ratio $[(Nu)_R/(Nu)_S]$	Friction Factor Enhancement Ratio $[(f_R)/(f_S)]$	Thermohydraulic performance Parameter	Nusselt Number Improvement Factor	Friction Factor Improvement Factor	Thermo-hydraulic Improvement Parameter
S.NO	Re	(NNER)	(FFER)	THPP	NNIF	FFIF	THIP
1	2983	1.1332	1.6218	0.9660	0.1332	0.6218	0.2140
2	4717	1.4370	1.6816	1.2105	0.4370	0.6816	0.6412
3	7459	1.8225	1.7436	1.5171	0.8225	0.7436	1.1062
4	9435	2.0587	1.7762	1.7032	1.0587	0.7762	1.3640
5	11556	2.2870	1.8049	1.8221	1.2870	0.8049	1.5990
6	13955	2.5221	1.842	2.0554	1.5221	0.832	1.8295
7	18500	2.9052	1.8832	2.3730	1.9192	0.8732	2.1978
8	22000	3.1936	1.8990	2.5844	2.1936	0.8990	2.4400

Table A 2: Various performance parameters and Improvement Factors for V-Shape rib SAH, Momin et al. (2002)

	Reynolds Number	Nusselt No.Enhancement Ratio $[(Nu)_R/(Nu)_S]$	Friction Factor Enhancement Ratio $[(f_R)/(f_S)]$	Thermohydraulic performance Parameter	Nusselt Number Improvement Factor	Friction Factor Improvement Factor	Thermo-hydraulic Improvement Parameter
S.NO	Re	(NNER)	(FFER)	THPP	NNIF	FFIF	THIP
1	2983	1.8309	3.1677	1.2515	0.8309	2.16	0.38834
2	4717	1.9063	2.9236	1.3379	0.9063	1.9236	0.4712

3	7459	1.9847	2.6983	1.4303	0.9847	1.6983	0.5798
4	9435	2.0262	2.5896	1.4802	1.0262	1.5896	0.6456
5	11556	2.0642	2.4993	1.5246	1.0627	1.4994	0.7088
6	13955	2.097	2.4181	1.5671	1.0972	1.4181	0.7737
7	18500	2.1499	2.3017	1.6328	1.1498	1.317	0.8834
8	22000	2.1829	2.233	1.6746	1.1829	1.233	0.9594

Table A 3: Various performance parameters and Improvement Factors for W-Shape rib SAH, Lanjewar et al.(2011).

	Reynolds Number	Nusselt No. Enhancement Ratio $[(Nu)_R/(Nu)_S]$	Friction Factor Enhancement Ratio $[(f_R)/(f_S)]$	Thermohydraulic performance Parameter	Nusselt Number Improvement Factor	Friction Factor Improvement Factor	Thermo-hydraulic Improvement Parameter
S.NO	Re	(NNER)	(FFER)	THPP	NNIF	FFIF	THIP
1	2983	1.8198	2.1207	1.42	0.8198	1.1207	0.7315
2	4717	1.9121	2.1447	1.4864	0.9121	1.1447	0.7967
3	7459	2.009	2.169	1.556	1.009	1.169	0.8631
4	9435	2.0606	2.1816	1.5929	1.0606	1.1816	0.8976
5	11556	2.1062	2.1925	1.6255	1.1062	1.1925	0.9276
6	13955	2.1495	2.2027	1.656	1.1495	1.2027	0.92557
7	18500	2.2159	2.2181	1.7036	1.2159	1.2181	0.9982
8	22000	2.2577	2.2275	1.7334	1.2577	1.2275	1.0246

Table A 4: Various performance parameters and Improvement Factors for Inclined Continuous rib SAH, Gupta et al. (1997).

	Reynolds Number	Nusselt No. Enhancement Ratio $[(Nu)_R/(Nu)_S]$	Friction Factor Enhancement Ratio $[(f_R)/(f_S)]$	Thermohydraulic performance Parameter	Nusselt Number Improvement Factor	Friction Factor Improvement Factor	Thermo-hydraulic Improvement Parameter
S.NO	Re	(NNER)	(FFER)	THPP	NNIF	FFIF	THIP
1	2983	1.0137	1.9542	0.8126	0.0137	0.9542	0.0143
2	4717	1.1546	2.0318	0.9138	0.1546	1.0318	0.1498
3	7459	1.3151	2.1125	1.0274	0.3152	1.1125	0.2832
4	9435	1.4058	2.1552	1.0912	0.4058	1.1552	0.3513
5	11556	1.4892	2.1926	1.1494	0.4892	1.1926	0.4101
6	13955	1.5711	2.2281	1.2061	0.5712	1.2281	0.465
7	18500	1.7021	2.2821	1.2064	0.7021	1.2821	0.5476
8	22000	1.7879	2.316	1.3552	0.7879	1.316	0.5987

Table A 5: Various performance parameters and Improvement Factors for Discrete W-Shape rib SAH, Kumar et al.(2009).

	Reynolds Number	Nusselt No. Enhancement Ratio $[(Nu)_R/(Nu)_S]$	Friction Factor Enhancement Ratio $[(f_R)/(f_S)]$	Thermohydraulic performance Parameter	Nusselt Number Improvement Factor	Friction Factor Improvement Factor	Thermo-hydraulic Improvement Parameter
S.NO	Re	(NNER)	(FFER)	THPP	NNIF	FFIF	THIP
1	2983	2.3266	3.3492	1.5614	1.3266	2.3492	0.5647
2	4717	2.4057	3.1267	1.6515	1.4057	2.1267	0.6610
3	7459	2.4876	2.9190	1.7468	1.4876	1.9190	0.7752
4	9435	2.5306	2.8179	1.7978	1.5306	1.8179	0.8419
5	11556	2.5683	2.7335	1.843	1.5683	1.7335	0.9047
6	13955	2.6040	2.6581	1.8862	1.6040	1.6572	0.9678
7	18500	2.6581	2.5472	1.9524	1.6581	1.5472	1.0717
8	22000	2.6919	2.4819	1.9944	1.6919	1.4819	1.1418

Table A 6: Various performance parameters and Improvement Factors for **Present Study**

	Reynolds Number	Nusselt No. Enhancement Ratio $[(Nu)_R/(Nu)_S]$	Friction Factor Enhancement Ratio $[(f_R)/(f_S)]$	Deviation Between the term $[(f_R)/(f_S)]$ & $[(f_R)/(f_S)]^{0.33}$	Thermo-hydraulic performance Parameter	Nusselt Number Improvement Factor	Friction Factor Improvement Factor	Thermo-hydraulic Improvement Parameter
S.NO	Re	(NNER)	(FFER)	(%)	THPP	NNIF	FFIF	THIP
1	2983	1.5555	1.8166	48.8849	1.2748	0.5555	0.8166	0.6582
2	4717	1.8997	1.8841	52.5483	1.5381	0.8997	0.8841	0.9860
3	7459	2.3015	1.9856	57.9828	1.8312	1.3015	0.9856	1.2816
4	9435	2.4576	2.0205	59.8263	1.9440	1.4576	1.0205	1.3734
5	11556	2.7310	2.0587	61.8340	2.1469	1.7310	1.0587	1.5582
6	13955	2.83495	2.1008	64.0333	2.2135	1.8349	1.1008	1.5749

APPENDIX-B

UNCERTAINTY ANALYSIS

A methodology for evaluation of the uncertainty in experimental results which has been suggested by Kline and McClintock (1953), is used in the present work. The procedure is as below:

If a parameter is calculated using certain measured quantities as,

$$Y = y(x_1, x_2, x_3, \dots, x_n)$$

Then uncertainty in the measurement of “y” is given as follows:

$$\frac{\delta y}{y} = \left[\left(\frac{\partial y}{\partial x_1} \delta x_1 \right)^2 + \left(\frac{\partial y}{\partial x_2} \delta x_2 \right)^2 + \left(\frac{\partial y}{\partial x_3} \delta x_3 \right)^2 + \dots + \left(\frac{\partial y}{\partial x_n} \delta x_n \right)^2 \right]^{0.5}$$

(1). Area of absorber plate (A_c):

$$A_c = W \times L$$

$$\frac{\delta A_c}{A_c} = \left[\left(\frac{\delta L}{L} \right)^2 + \left(\frac{\delta W}{W} \right)^2 \right]^{0.5}$$

(2). Cross sectional area of air flow duct (A)

$$A = W \times H$$

$$\frac{\delta A}{A} = \left[\left(\frac{\delta W}{W} \right)^2 + \left(\frac{\delta H}{H} \right)^2 \right]^{0.5}$$

(3). Perimeter of duct (P)

$$P = 2(W + H)$$

$$\frac{\delta P}{P} = \left[\left(2 \cdot \frac{\delta W}{W} \right)^2 + \left(2 \cdot \frac{\delta H}{H} \right)^2 \right]^{0.5}$$

(4). Hydraulic diameter of duct (D)

$$D = \frac{2 \cdot W \cdot H}{(W + H)}$$

$$\frac{\delta D}{D} = \left[\left(\frac{\delta A}{A} \right)^2 + \left(\frac{\delta P}{P} \right)^2 \right]^{0.5}$$

(5). Area of orifice meter (A_o)

$$A_o = \frac{\pi}{4} \times D_2^2$$

$$\frac{\delta A_o}{A_o} = \left[\frac{2\delta D_2}{D_2} \right]$$

(6). Density (ρ)

$$\rho = \frac{P_{atm}}{R.T_{fo}}$$

$$\frac{\delta \rho}{\rho} = \left[\left(\frac{\delta P_{atm}}{P_{atm}} \right)^2 + \left(\frac{\delta T_o}{T_o} \right)^2 \right]^{0.5}$$

(7). Mass flow rate (m)

$$m = C_d \cdot A_o \sqrt{\frac{2\rho(\Delta P_o)}{1 - \beta_R^4}}$$

$$\frac{\delta m}{m} = \left[\left(\frac{\delta C_d}{C_d} \right)^2 + \left(\frac{\delta A_o}{A_o} \right)^2 + \left(2 \frac{\delta \rho}{\rho} \right)^2 + \left(\frac{\delta(\Delta P_o)}{(\Delta P_o)} \right)^2 \left(\frac{2 \cdot \beta_R^3 \cdot \delta \beta_R}{1 - \beta_R^2} \right) \right]^{0.5}$$

(8). Reynolds number (Re)

$$Re = \frac{\rho V D}{\mu}$$

$$\frac{\delta Re}{Re} = \left[\left(\frac{\delta V}{V} \right)^2 + \left(\frac{\delta \rho}{\rho} \right)^2 + \left(\frac{\delta D}{D} \right)^2 + \left(\frac{\delta \mu}{\mu} \right)^2 \right]^{0.5}$$

(9) Useful heat gain (Qu)

$$Qu = \dot{m} C_{pa} \cdot \Delta T$$

$$\frac{\delta Qu}{Qu} = \left[\left(\frac{\delta \dot{m}}{\dot{m}} \right)^2 + \left(\frac{\delta C_{pa}}{C_{pa}} \right)^2 + \left(\frac{\delta(\Delta T)}{\Delta T} \right)^2 \right]^{0.5}$$

(10). Heat transfer coefficient (h)

$$\frac{\delta h}{h} = \left[\left(\frac{\delta Qu}{Qu} \right)^2 + \left(\frac{\delta A_c}{A_c} \right)^2 + \left(\frac{\delta(T_{pm})}{(T_{pm})} \right)^2 \right]^{0.5}$$

(11). Nusselt number (Nu)

$$\frac{\delta Nu}{Nu} = \left[\left(\frac{\delta h}{h} \right)^2 + \left(\frac{\delta D}{D} \right)^2 + \left(\frac{\delta(K_a)}{(K_a)} \right)^2 \right]^{0.5}$$

(12) Friction factor (f)

$$\frac{\delta f}{f} = \left[\left(\frac{\delta(P_D)}{(P_D)} \right)^2 + \left(\frac{\delta D}{D} \right)^2 + \left(\frac{\delta(L_f)}{(L_f)} \right)^2 + \left(\frac{\delta(\rho)}{(\rho)} \right)^2 + \left(\frac{\delta(G)}{(G)} \right)^2 \right]^{0.5}$$