Asian Journal of Applied Science and Technology (AJAST)
(Open Access Quarterly International Journal) Volume 2, Issue 3, Pages 197-210, July-September 2018

Numerical Investigation of Heat Transfer Enhancement in Rectangular Duct with Square Ribs and Optimization of Flow and Geometrical Parameters-Taguchi Approach

Sudeep Singh Rawat[1], Vikas Chauhan[2] and Vikalp Semwal[3]

[1]*Faculty of Technology, UTU, Dehradun, Uttarakhand, India.*
[2]*Faculty of Technology, UTU, Dehradun, Uttarakhand, India.*
[3]*Faculty of Technology, UTU, Dehradun, Uttarakhand, India.*

Article Received: 26 April 2018 *Article Accepted: 30 July 2018* *Article Published: 07 September 2018*

ABSTRACT

The thermo-hydraulic performance was analyzed computationally for the stationary channels with rib turbulators situated at right angles i.e. 90 degrees and then optimized by employing Taguchi approach. Ribs were arranged in a linear manner one after another and numerical values of the friction factor and convective heat transfer coefficient were computed. A L_{16} (4^3) orthogonal array has been taken for optimization purpose for maximization of thermal performance. The concept of thermal performance includes the twofold effects at a same time i.e. it maximizes the heat transfer coefficient and at the same time minimizes the friction factor. Maximization of thermal enhancement factor is considered as the ultimate aim of present work. The rib relative pitch (p/e) is varied as 3, 6, 9 and 12. The inclination angles are varied from 45˚ to 90˚ in a step of 15 degree thereby leading different rib shapes i.e. right angled triangle, trapezoidal and square ribs. Different values of heat transfer coefficient and friction factors were computed by varying Reynolds numbers 4000, 8000, 12,000, 16,000. By combined effect of rib pitch-to-height ratio, inclination angles and flow parameters, the optimal cooling configuration was obtained. A right triangular shaped rib with optimum parameters (α) =45˚, Re=4000, and p/e=9 has found best for thermo-hydraulic performance against square rib.

Keywords: Heat Transfer, Friction Factor, Thermal Enhancement Factor, Taguchi Method.

1. INTRODUCTION

Expanding interest of energy required either more generation of energy or finds an elective way which decreases the energy utilization. The change, usage and recovery of energy in each field for example either business or local involve a heat transfer process. Some significant zones in which forced convection heat transfer happen are recorded as power sector, refrigeration and air conditioning, thermal processing of chemicals, electrical machines and electronic gadgets, squander heat recuperation in manufacturing, inside cooling of engine; turbo-machinery frameworks and some more. Rib turbulators are playing a significant role in this regard. There are a few strategies, for example, jet impingement, film cooling, rib turbulators, shaped internal cooling sections, dimple cooling, appeared in Figure below, to cool gas turbine sharp edged blade. The jet impingement is utilized to cool the leading edge, pin fin cooling at the trailing edge and rib turbulators are utilized cool the internal entries. The present investigation centres on the thermal enhancement of heat utilizing rib turbulators or turbulence promoters.

2. LITERATURE REVIEW

A lot of research movement has been committed to the comprehension of heat transfer and fluid flow in rib turbulated conduits. There are significant quantities of test and investigative examinations on flows through conduits with a basic cross area. The vast majority of these investigations depend on the flows from pipes with either smooth walls or with rib roughened walls, or with in part roughened walls or with both smooth and harsh walls. Recently, the researchers have paid special attention towards various geometric parameters of a roughened surface along with varying flows**. (**J.C. Han, 1978) studied first the examination of rib-roughened surface to decide the impacts of rib shape, angle of attack and pitch to height proportion or ratio on friction factor and connective heat transfer coefficient comes about. . Parallel plate geometry was utilized. Ribs at a 45˚ approach were found to have prevalent heat enhancement at a given grating force when contrasted with ribs at a 90˚ approach angle or when contrasted with sand-grain roughness. (J. S. Park & J.C. Han, 1988) experimentally investigated the joint impact of attack angle (α) of ribs and aspect ratio on the dissemination of convective heat transfer coefficient for creating streams in short rectangular channels using opposite faced ribbed surfaces at Reynolds number (Re) between 10,000 to 60,000. The results indicated that the local Nusselt number attained a constant value at X/D > 3 after a entrance decrement for the α=90˚ while for the other angle values i.e. 60˚, 45˚and 30˚ it showed a continuous increment after X/D > 3 due to introduction of secondary induced flow produced by rib angle. (Lockett and Collins, 1989) used some different approach by the use of holographic interferometery for the diagnosis of thermal flow field using square and rounded shaped geometry and found that the heat transfer distribution was predominantly high for rounded ribs than the later and was dependent on the local Reynolds number (Re). (G. Rau, 1998) experimented detailed streamlined and heat exchange estimations in a square channel with ribs showing a noteworthy blockage proportion ($e/D_h = 0.1$). Reynolds number (Re) was held at 30,000.The after-effects of the nearby estimations were talked about for three diverse p/e proportions (6, 9, 12) in a one-side-ribbed channel.. (Hibbs et al., 2000) examined tentatively the impact of vortex-generators on the heat transfer from the inside ribbed sections of a turbine blade coolant channel utilizing a mass-transfer system. Reynolds number was taken between 5000 to 30,000.It was demonstrated that small generator-rib relative pitch (s/e=0.55) lead to a retardation of the shear layer development past the reattachment point, usually linked with lower heat transfer values. (Z.X.Yuan and W.Q.Tao, 2003) experimented Heat exchange and friction characteristics for another kind of improved rectangular channel with winglets have been explored. The outcomes show that, in the scope of Reynolds number from 5×10^3 to 4.7×10^4, heat exchange characteristic of the upgraded conduit with winglets is unrivalled to the enhanced duct with transverse disturbances. (Chandra and Han, 2003) conducted a trial investigation of thermal exchange and friction factor of a completely developed turbulent air flow in a square channel with transverse ribs on 1,2,3, and 4 walls was accounted for. Tests were done for Reynolds numbers going from 10,000 to 80,000. The pitch-to-rib height proportion, p/e, was fixed at 8 and rib height-to-channel water hydraulic diameter, e/D_h was freezed at 0.0625. The channel length-to-hydraulic diameter, L/D_h, was 20. The test results indicated that the average Nusselt number ratio was varying inversely with the Reynolds number (Re) and reaches a constant figure in a fully developed region for $X/D_h > 6.0$.The heat transfer was enhanced with the increment of ribbed channel walls 2.16(1 ribbed wall) to 2.57(4 ribbed wall) case at Re=30,000. (Tariq et al., 2004) experimented study of flow and convective heat transfer coefficient on a rib mounted at the surface along with slit inside the wind tunnel. They took the open area ratios in the step of 10% (i.e. 10,20,30,40 and 50%) with respect to total area of projection of the rib with the Reynolds number (Re) being 32,100. Results demonstrated that the slit inside the rib improved thermal exchange and lessens friction factor, with an ideal execution seen at an open region proportion of 20%. Heat transfer expansion of a slit rib comes out to be higher than the strong rib with 20% open region proportion being an ideal. (Sahin and Yakut, 2005) performed experiment by the qualities of winglet edge, height, width, heat exchange and pressure loss are researched for the plate compose converging-diverging ducts by utilizing Taguchi Design of Experiment technique According to the ideal outcomes the fluid flow is acquired as 4 m/s, winglet width as 15 mm, winglet height as 100 mm and winglet angle as 15°. It was resolved that the winglet height and width, and fluid velocity are the best parameters which provided ideal configurations as per optimization technique. (Wang and Sunden, 2005) explored tentatively with series of experiments on the turbulent heat transfer and friction in a square duct roughened by consistent and truncated ribs on one wall. For the two cases, the rib height to-hydraulic diameter across proportion was 0.15, the rib pitch-to-height proportion was fixed at 12, and the Reynolds number shifted from 8,000 to a maximum of 20,000. The experimental results confirmed that the localization of heat transfer coefficient was strongly influenced by the shape of the rib. Their final conclusion remarked the better thermal performance of continuous ribs than that to truncated ribs. (Kamali et al., 2008) carried out the CFD analysis of various shaped ribs i.e. square, triangular, trapezoidal rib with diminishing height and trapezoidal rib with escalating height with respect to flow direction. The pitch-to-height (p/e) varied from 8 to 12. For the Reynolds number(Re)=20,000 ,trapezoidal ribs with diminishing height yielded maximum heat transfer and the friction penalty values were minimum for the trapezoidal ribs with escalating height with respect to flow also for the different (p/e) ratios, the maximum heat enhancement was found at p/e=12. (L.M. Wright, 2008) performed a trial investigation on a 3:1 rectangular channel with calculated ribs. The rib dispersing of 10 and 20 are taken into account. Rib widths were adjusted and their impacts on the performance were analyzed. The examination infers that the angled ribbed channel has more heat exchange impact than the smooth channel. (Kim and Lee, 2009) an optimum design for improvement in the thermal (heat) transfer and thermal performance for a stationary channel with angular rib turbulators was investigated to find the foremost appropriate rib configuration. Among numerous design parameters, only two viz. rib angle of attack (α)[$30^\circ \le \alpha \le 80^\circ$] and pitch-to-rib height (p/e)[$3.0 \le p/e \le 15$], were chosen. The optimization was done through RSM (Response surface method).As a result, the thermal performance attained maximum value when the two parameters were (α) [$50^\circ \le \alpha \le 60^\circ$] and [$6.0 \le p/e \le 7$].The maximum heat transfer occurred at α=53.31˚ and p/e=6.50. (Aharwal et al., 2010) carried out experimental investigation to find out the heat transfer enhancement and fluid flow characteristic by varying the gap between a pair of inclined ribs. They took a rectangular conduit with aspect ratio of 5.83. The gap width varied from 0.5 to 2.0 and the Reynolds number (Re) was in the range of 3000 to 18,000. For relative gap width (g/e) =1.0, the value of Nusselt number was maximum compared with other values of g/e. (Tanda et al., 2011) Experimented the investigation of forced convection heat transfer on an inclined ribbed geometry placed at 45˚ inside the rectangular conduit (aspect ratio=5).The ultimate aim was to investigate the influence of rib spacing on thermal enhancement factor. Reynolds number (Re) was taken in the range from 9000 to 35,500.He took rib height-to-hydraulic diameter (e/D_h =0.09) and relative pitch (p/e) were taken as 6.66, 10.0, 13.33 and 20.0. Maximum heat transfer corresponded at p/e =13.33(one-ribbed wall channel) and p/e=6.66-10(two-ribbed wall channel).(Ali &Tariq, 2012) experimented on the rectangular duct channel considering the trapezoidal ribs and the flow was varied (Re=9400-61,480) at different chamfering angles at the step of 5˚ (α=5˚, 10˚, 15˚ and 20˚).Their observations on the basis of experimental results concluded that there were formations of large recirculation bubbles for all the configurations in downstream of rib. The most striking component was the reduced impact of this optional distribution rise at higher Reynolds number, which blurred advance concerning the expansion in trapezoidal angle, and has been viewed as the potential reason for forestalling the hotspots in the leeward region of the sold rib. (Yang et al., 2014) conducted numerical investigation accompanied with optimization with RSM (Response Surface Method) and GA (genetic algorithm) for the two dimensional ribbed channel for maximum heat transfer enhancement. Their prime focus was on the 3 geometrical parameters wiz. Height of rib (4 mm $< e <$ 10 mm), thickness of rib (5 mm $< t <$ 25 mm) and pitch of rib (25 mm $< p <$ 40 mm). Reynolds number taken for the experiment was in the range of 5000, 10,000 and 15,000. On the basis of optimization through GA (Genetic Algorithm) and RSM (Response Surface Method) they came to the conclusion that for the in-line ribbed channel, the thermal enhancement factor expanded to 1.1 to 1.5 after the ideal plan. In the unsteady ribbed channel, the performance factor comes to around 2.7. (Rahimi et al., 2015) performed numerical investigation on the ribbed channel solar air heater system along with optimization with the help of Taguchi approach. Their prime objective was to develop such rib with best thermo-hydraulic performance i.e. maximum heat transfer characteristic and minimum friction penalty. They considered four major factors for this investigation; rib pitch-to-height (p/H), rib relative height (e/H), rib front projection(s) and rib relative tip width (a/H) and CFD simulation were performed at constant Reynolds number (Re) as 10,000. Their outcome indicated (e/H) and (p/H) as the major dominating factor for thermal enhancement. (Mayo & Arts, 2015) has done experimental investigation on a rotating ribbed channel setup with varying Reynolds number(Re) 15,000, 20,000, 30,000 and 40,000 .They took the aspect ratio to be 0.9 and 8 equidistant ribs normal to the flow direction. The Rib-to-pitch height (p/e) was 10. Their finding indicated maximum rotation number values were falling in the range of 0.12 to 0.30 for Reynolds number (Re) 40,000 and 15,000 respectively. (Sunil Chamoli, 2015) performed an experimental investigation along with optimization of flow and geometrical parameters with Taguchi approach on V down perforated baffles. He took Reynolds number (Re), rib pitch-to height (p/e), relative roughness height (e/H) and open area ratio (β) as the constraints and Nusselt number friction factor as the performance factor. His results portrayed Reynolds number as a major performing asset with 73.97% contribution ratio and the optimum parameters were p/e= 2, e/H=0.4, β=12% and Re=18,600. (Liu et al., 2017) numerically investigated the 3-D turbulent flow and convective heat transfer coefficient around a square channel with distorted cylindrical ribs with aim of gaining best thermo-hydraulic performance. The incoming Reynolds number was taken from 10,000 to 25,000. His concluding remarks enlightened better thermal performance of rounded transitioned cylindrical ribs over regular ribs since the former was resulting in effective reduction in re-circulating flow and better re-attachment lines. (Sharma et al., 2017) conducted an experimental investigation on pentagonal shaped rib with maximization of thermo-hydraulic performance as the prime motive by varying pitch-to-height ratio (p/e) from 6 to 12 and chamfering angle (α) from 5˚ to 20˚. They have chosen Reynolds number (Re) from 9400 to 58,850. Liquid Crystal Thermography (LCT) technique yielded surface temperatures and eventually the Nusselt number and predicted better performance of pentagonal ribs over square ones.(Arjumand et al., 2018) performed numerical investigation on two-pass square ribbed channel for maximizing the thermo-hydraulic performance. The performance parameter considered was the varying the shape of ribs at relative rib pitch of 10 mm and Reynolds number ranging from 5000 to 52,000. Their outcome stated the 1.19 times increment in Nusselt number of investigated rib with respect to square rib and decrement in friction penalty by 1.3. Finally best thermal enhancement was achieved by boot-shaped ribs.

These works were helpful in concentrating the impact of rib spacing, rib width and the flow parameters on the heat transfer enhancement. The works gave a thought on how the heat transfer changes when the channel is roughened.

2.1 Objective of Present Research:

1. To ponder the impact of variety in rib geometry, rib pitch-to-height proportion and Reynolds number on general Nusselt number and friction factor qualities.
2. To research the capability of Taguchi Method in foreseeing the ideal arrangement of plan parameters in the event of ribbed duct.
3. To anticipate the outline parameters i.e. geometrical and flow parameters that have the best effect on heat transfer and friction factor.

3. METHOD

3.1 Mathematical Modelling

The stream in the computational space has been thought to be 2-D, turbulent, in-compressible and consistent in nature. In Cartesian tensor framework, the basic governing equations including the Continuity Equations, Reynolds-Averaged Navier–Stokes (RANS) Momentum Equations, and Energy Equations stated below as (1), (2) and (3) respectively;

$$\frac{\partial}{\partial x_i}(\rho u_i) = 0 \tag{1}$$

$$\frac{\partial}{\partial x_j}(\rho u_i u_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3}\delta_{ij}\frac{\partial u_k}{\partial x_k}\right)\right] + \frac{\partial}{\partial x_j}\left(-\rho\overline{u_i' u_j'}\right) \tag{2}$$

$$\frac{\partial}{\partial x_i}[u_i(\rho E + p)] = \frac{\partial}{\partial x_j}\left[\left(k + \frac{C_p \mu_t}{Pr_t}\right)\frac{\partial T}{\partial x_j}\right] \tag{3}$$

Where turbulent viscosity$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon}, C_\mu = 0.009,$, E is the total energy, Pr_t is the turbulent Prandtl number for energy.

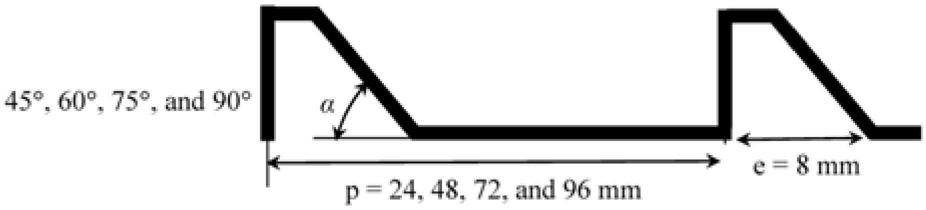
3.2 Numerical Method

3.2.1 Geometry and Computational grid

A square geometry with side (e=8mm) is taken and the inclination angle (α) is varied thereby resulting in different configurations wiz. Square, trapezoidal and right triangle as these are most studied geometries. The meshing of the geometry has been performed utilizing GAMBIT 2.3.16. Surface cross sections were created by triangular and quads work components. For grid independence test, a solution adaptive refinement strategy is utilized. At the point when adaption is utilized properly the subsequent work is most good for the stream arrangement, after 20, 8815 quantities of cells, the variety in the estimation of heat transfer coefficient is irrelevant. Subsequently, the further investigation utilizing networks with 20, 8815 number of cells.

3.2.2 Solution Procedure

The design of rectangular channel rib turbulators with square ribs is created in the FLUENT WORKBENCH 15.0. A 2-Dimensional CFD study of fluid flow and heat transfer characteristics in a rectangular duct with square ribs of changing shapes from square, trapezoidal to right angled triangle along flow direction mounted on bottom surface. The bottom wall is provided with to a constant heat flux, while the remaining walls kept insulated.



The Realizable k-ε display is generally late improvement and not the same as the other model. Since it is probably going to give better execution than stream including pivot, boundary layer under strong pressure gradient, separation and recirculation. Henceforth, this model has been selected for the present examination.

The average Nusselt number is being calculated as:

$$\boldsymbol{Nu = \frac{1}{L}\int \frac{h(x).D_h}{k}\,dx} \tag{4}$$

Where h(x) is local heat transfer coefficient, D_h being hydraulic diameter (64mm), L is being test duct length. Friction factor (f) is also calculated by the below equation:

$$\boldsymbol{f = \frac{\Delta P.D_h}{\frac{1}{2}\rho u^2 L}} \tag{5}$$

Where ΔP stands for pressure drop across the duct, u being average flow velocity over entire test section length L. The Thermal enhancement factor (η) shows the two fold effect of Nusselt number and friction factor being calculated by:

$$\eta = \frac{\frac{Nu}{Nu_s}}{\sqrt[3]{\left(\frac{f}{f_s}\right)}} \tag{6}$$

Where the subscript s shows respective value for smooth channel. Also friction factor for smooth duct is calculated using Modified Blassius equation; $\boldsymbol{f_s = 0.085Re^{-0.25}}$ and Nusselt number for smooth flow is calculated by using Dittus Boelter equation i.e. $\boldsymbol{Nu_s = 0.024Re^{0.8}Pr^{0.4}}$ for prandtl value of 0.7.

3.3 Taguchi Method

The Taguchi method is widely used in various industrial and engineering applications for optimizing the design parameters, i.e. flow and geometrical parameters. This methodology is based on two fundamentals concepts of statistical engineering; First, the quality losses must be defined as deviations from the targets, not conformance to arbitrary specifications, and the second, achieving high system quality levels economically requires quality to be designed into the product. In the Taguchi method the orthogonal array facilitates the experimental design process and caters a method for fractional factorial experiments. The orthogonal array contributes to study the effect of main and interacting parameters via minimizing the number of experimental trials. Taguchi analysis is performed with **Minitab 17.0** software. In the current work, a L_{16} (4^3) orthogonal array is used, which implied carrying out 16 tests with 3 factors of 4 levels that listed in Table 3.1 and Table 3.2 respectively. As indicated in the table, the observed values of the Reynolds number (A), rib pitch to height (B), and Inclination angle (C).

Table 3.1 Chosen L_{16} (4^3) Experimental Plan

Number of test	Control factors and their levels		
	A(Re)	**B(p/e)**	**C(α)**
1	1	1	1
2	1	2	2
3	1	3	3
4	1	4	4
5	2	1	2
6	2	2	1
7	2	3	4
8	2	4	3
9	3	1	3
10	3	2	4
11	3	3	1
12	3	4	2
13	4	1	4
14	4	2	3
15	4	3	2
16	4	4	1

Table 3.2 Parameters and Their Values Corresponding to Their Levels to be studied in Experiments

Code	Parameters	Levels			
		1	**2**	**3**	**4**
A	Reynolds Number, (Re)	4000	8000	12000	16000
B	Rib pitch to height ratio, (p/e)	3	6	9	12
C	Inclination angle, (α)	45	60	75	90

Taguchi strongly recommends the use Signal-to-Noise (S/N) ratios. By maximizing the S/N ratio, the loss associated with the process can be minimized. The S/N ratio determines the most robust set of operating conditions from variation within the results. The experimental observations are transformed into a signal-to-noise (S/N) ratio. There are several S/N ratios available depending on the type of characteristics. The S/N ratio characteristics can be divided into three categories given by Equation below when the characteristic is continuous:

Smaller is the better characteristic: $\frac{S}{N} = -10\log\frac{1}{n}\left(\sum y^2\right)$ (7)

Nominal the better characteristic: $\frac{S}{N} = -10\log\frac{1}{n}\left(\sum\frac{\overline{Y}}{S_Y^2}\right)$ (8)

Larger the better characteristic: $\frac{S}{N} = -10\log\frac{1}{n}\left(\sum\frac{1}{y^2}\right)$ (9)

Where $\overline{Y}$ the average of observed data, S_Y^2 is the variation of y, n is the number of observations, and y is the observed data. “Higher is better” characteristic and “Lower is better” characteristic, respectively, with the above S/N ratio transformation is suitable for maximization of Nusselt number and minimization of friction factor.

4. RESULTS AND DISCUSSION

For the validation of present model, the Nusselt number outcomes acquired with single square rib at Re = 12000 from the present examination has been contrasted with outcomes reported by Tariq et al. (2004) at Re = 13400 as shown in table below.

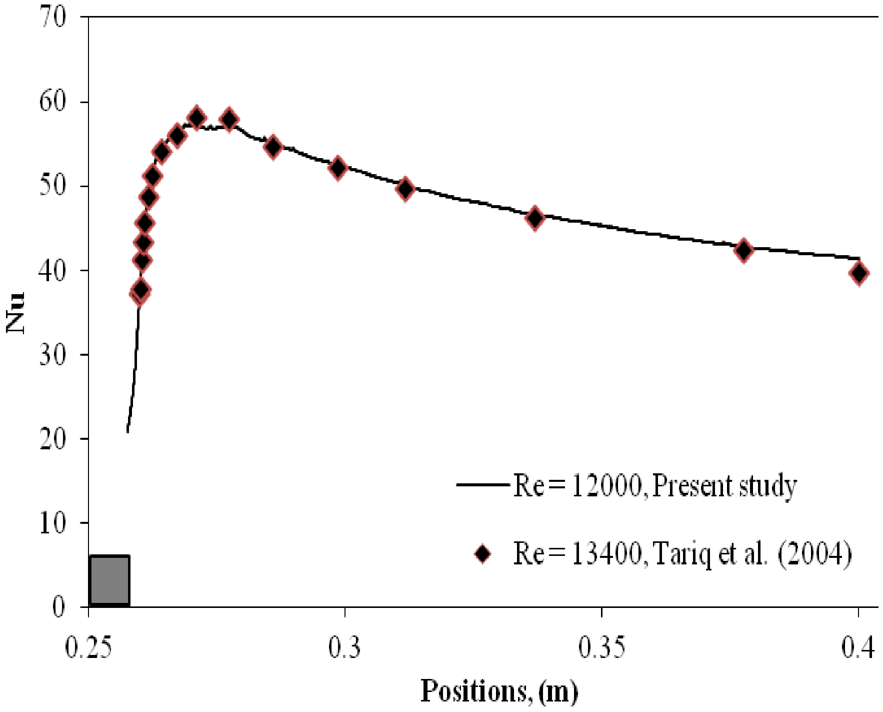


Figure 4.1 Validation of numerical analysis with Experimental results Tariq et al. (2004).

Asian Journal of Applied Science and Technology (AJAST)
(Open Access Quarterly International Journal) Volume 2, Issue 3, Pages 197-210, July-September 2018

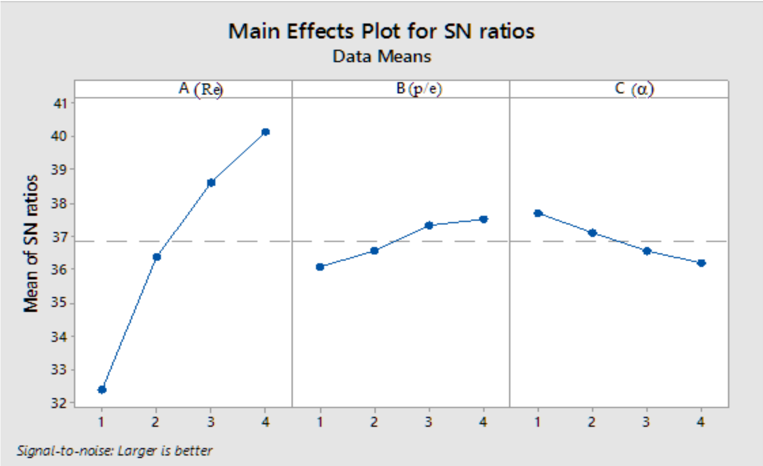
The response table of SNR for individual parameters at different levels, after implementing the Taguchi Method, in the case of heat transfer is presented in tabular form. These tables and figures successfully illustrate the importance of the design factors on the heat transfer, friction factor and thermo-hydraulic performance, and their respective rank is reported in the penultimate row of the tables. The factors with the higher difference between maximum and minimum values of SNR have the higher influence.

4.1 Effect on Nusselt number

For parameter A (Re), the Nusselt number increases with the increase of mean fluid velocity, as expected (Table 4.1). The Nusselt number increases with increasing B (p/e), because with increasing rib pitch to height ratio the flow reattaches on the heat transferring surface, which leads to high heat transfer rate. For parameter C (α), the Nusselt number decreases as the rib geometry changes from triangular to square.

Table 4.1 Response Table of S/N ratios for Nusselt number and Main effects Plot

Level	A	B	C
1	41.73	65.7	82.08
2	65.84	70.4	76.8
3	85.49	78.77	70.16
4	102.65	80.84	66.67
Delta	60.92	15.14	15.41
Rank	1	3	2
Contribution ratio, (%)	66.60	16.55	16.85



The analysis of the results gives the combination factors resulting in maximum heat transfer rate among the investigated test plate configurations are as follows: Re = 16000 (A4), p/e = 12 (B4), and α = 45° (C1). Therefore, $A_4B_4C_1$ is the optimum set of design parameters associated with heat transfer as per the "higher is the better" condition.

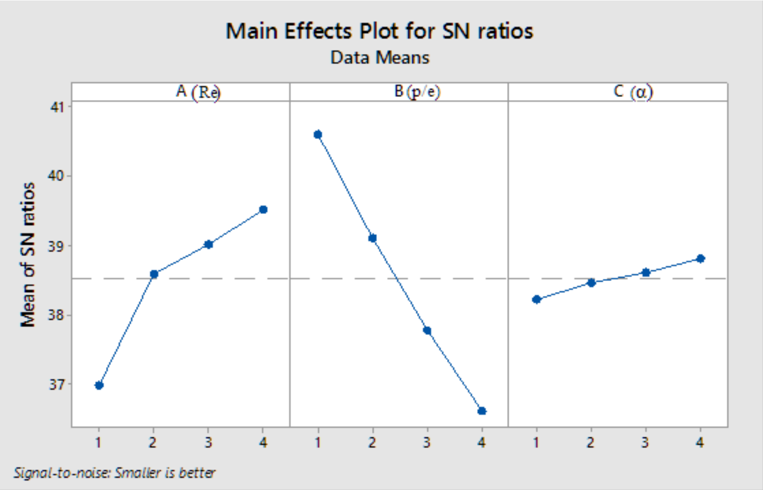
4.2 Effect on friction factor

Table 4.2 illustrates the effect of control parameters on friction factor. Friction factor increases with increase in control factor A (Re). The friction factor tends to decrease with the increase of parameter B (p/e). The values of friction factor slightly decreases with increase the control factor C (α). The optimum values of the control factors for minimum friction factor condition as follows: Re = 16000 (A4), *p/e* = 3 (B1), and α = 90° (C4). Therefore, $A_4B_1C_4$ is the optimum set of design parameters associated with friction factor as per the "lower is the better" condition.

 Online ISSN: 2456-883X

Table 4.2 Response Table of S/N ratios for friction factor and Main effects Plot

Level	A	B	C
1	36.99	40.61	38.22
2	38.58	39.1	38.45
3	39.02	37.78	38.61
4	39.51	36.61	38.82
Delta	2.52	3.99	0.6
Rank	2	1	3
Contribution ratio, (%)	35.44	56.12	8.44

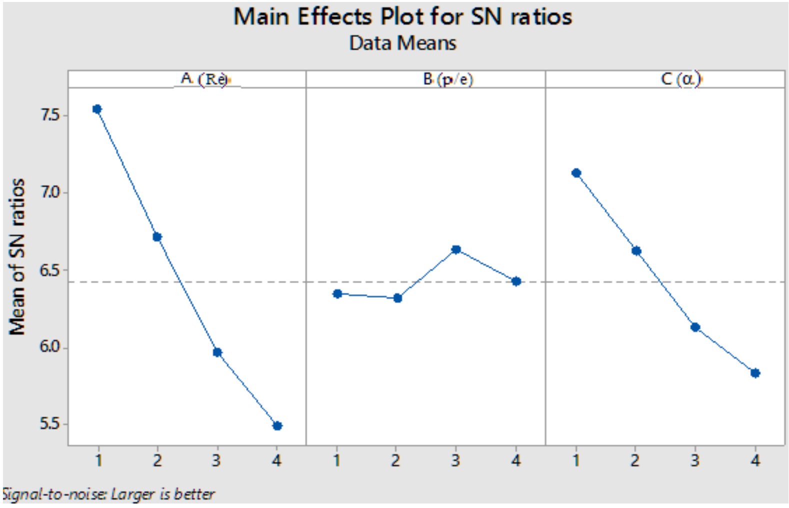


4.3 Effect on thermo-hydraulic performance

Table 4.3 shows the influence of control parameters on thermo-hydraulic performance factor. Thermo-hydraulic performance decreases with increase in control factor A (Re). While no particular trend of thermo-hydraulic performance factor has been observed for parameter B (p/e). The value of thermo-hydraulic performance significantly decreases with increase in the control factor C (α). The optimum values of the control factors for maximum thermo-hydraulic performance are as follows: Re = 4000 (A1), *p/e* = 9 (B3), and α = 45° (C1). Therefore, $A_1B_3C_1$ is the optimum set of design parameters associated with friction factor as per the "higher is the better" condition.

Table 4.3 Response Table of S/N ratios for Thermo-Hydraulic Performance and Main effects Plot

Level	A	B	C
1	7.55	6.345	7.134
2	6.714	6.316	6.627
3	5.968	6.634	6.129
4	5.489	6.426	5.831
Delta	2.06	0.318	1.303
Rank	1	3	2
Contribution ratio (%)	55.96	08.64	35.40



The *delta values*, defined as the difference of maximum and minimum SNR for every control factor, and *contribution ratios*, defined as the ratio of the delta values of each factor to the total delta value of all factors, are presented respectively in the penultimate and last row of Tables 4.1, 4.2, and 4.3 respectively. From Table 4.1, it is observed that the Reynolds number contributes to 66.65% of the total effect; this means that the parameter A (Re) is the most effective one on heat transfer and the factors B and C have almost same effectiveness on heat transfer. Table 4.2 confirms that parameter B is the most dominant factor that controls the friction factor with a contribution ratio of 56.12% of the total effect followed by factor A and C respectively. Furthermore, Table 4.3 predicts that parameter A is the most significant having a contribution ratio of 55.96% of the total effect, followed by parameter C and then by B.

The optimum level of control factors for Nusselt number, friction factor and thermo-hydraulic performance factor are$A_4^a B_4^c C_1^b$, $A_4^b B_1^a C_4^c$ and $A_1^a B_3^c C_1^b$ respectively, shown in Table 4.4. Here the coefficients a, b, and c symbolize the importance level of each parameter and indicate the first, second, and third effective parameter, respectively.

Table 4.4 Optimum Conditions And Performance Values, Where Superscript A, B, And C Are The Sequence Of Effective Parameters.

Parameters		**Factors**			**Value**
		A (Re)	**B (*p*/*e*)**	**C (*α*)**	
Nusselt number(Nu)	Optimum level	4^a	4^c	1^b	119.94
	Optimum value	16000	12	45°	
Friction factor(f)	Optimum level	4^b	1^a	4^c	0.00824
	Optimum value	16000	3	90°	
Thermal enhancement factor(η)	Optimum level	1^a	3^c	1^b	2.88
	Optimum value	4000	9	45°	

5. CONCLUSION

Heat transfer coefficient, friction factor and thermal enhancement factor has been numerically investigated and then Optimized by Taguchi Method and (L_{16}) orthogonal arrays. Following Outcomes is worth noting as stated below:

(1) For Nusselt number, the results indicated that Reynolds number (Re) was major contributing factor (66.60%).Also, Nu was increasing with rib pitch-to-height(p/e) because of the reattachment of flow attaining optimum value at Re=16000, p/e=12 and α=45˚.

(2) For friction factor, major contributing factor comes out was the rib pitch-to-height ratio (56.12%) and optimum parameters were Re=16000, p/e=3 and α=90˚.

(3) For thermo-hydraulic performance, it shows a downgrade as the Re values were going high; was almost uninfected by changing p/e values and reflected a descending trend as the inclination angle (α) was increasing. Optimum parameters for maximum performance comes out as Re=4000, p/e=9 and α=45˚.

The above results indicated that with the change in value of p/e, α and Re, there does not exist a unique rib configuration, which provided the maximum heat transfer enhancement with minimum friction penalty and the best thermo-hydraulic performance as compared to that of square rib**.** This information can provide reliable information to those researchers who are looking forward to carry out their work in this field.

REFERENCES

[1] Han, J. C., Park, J. S., and Lei, C. K., Heat Transfer Enhancement in Channels with Turbulence Promoters, ASME Journal of Engineering for Gas Turbines and Power, vol. 107, no. 3, pp. 628–635, 1985.

[2] Yuan, Z. X., and Tao, W. Q., Experimental Study on Heat Transfer in Ducts with Winglet Disturbances, Heat Transfer Engineering, vol. 24, no. 2, pp. 76–84, 2003

[3] Akcayoglu, A., and Nazli, C., A Comprehensive Numerical Study on Thermo-hydraulic Performance of Fluid Flow in Triangular Ducts with Delta-winglet Vortex Generators, Heat Transfer Engineering, 2017. DOI: 10.1080/01457632.2017.1288046

[4] Chamoli, S., and Thakur N. S., Thermal Behavior in Rectangular Channel Duct Fitted with V-Shaped Perforated Baffles, Heat Transfer Engineering, vol. 36, no. 5, pp. 471-479, 2015.

[5] Han, J. C., Dutta, S., and Ekkad, S., Gas Turbine Heat Transfer and Cooling Technology, 2nd ed., CRC Press, Taylor and Francis group, New York, 2012.

[6] Han, J. C., and Park, J. S., Developing Heat Transfer in Rectangular Channels with Rib Turbulators. International Journal of Heat and Mass Transfer, vol. 31, no. 1, pp. 183–195, 1988.

[7] Hwang, J. J., Heat Transfer-Friction Characteristic Comparison in Rectangular Ducts with Slit and Solid Ribs Mounted on One Wall, ASME Journal of Heat Transfer, vol. 120, no. 3, pp. 709–716, 1998.

[8] Chandra, P. R., Alexander, C. R., and Han, J. C., Heat Transfer and Friction Behaviors in Rectangular Channels with Varying Number of Ribbed Walls, International Journal of Heat and Mass Transfer, vol. 46, no. 3, pp. 481–495, 2003.

[9] Chang, S. W., Yang, T. L., and Wang, W. J., Heat Transfer in a Rotating Twin-Pass Trapezoidal-Sectioned Passage Roughened by Skewed Ribs on Two opposite Walls. Heat Transfer Engineering, vol. 27, no. 10, pp. 63–79, 2006.

[10] Lockett, J. F., and Collins, M. W., Holographic Interferometry applied to Rib-Roughness Heat Transfer in Turbulent Flow. International Journal of Heat and Mass Transfer, vol. 33, no. 11, pp. 2439–2449, 1990.

[11] Liou, T. M., and Hwang, J. J., Effect of Ridge Shapes on Turbulent Heat Transfer and Friction in a Rectangular Channel. International Journal of Heat and Mass Transfer, vol. 36, no. 4, pp. 931–940, 1993.

[12] Wang, L., and Sunden, B., Experimental Investigation of Local Heat Transfer in a Square Duct with Various-Shaped Ribs. Heat and Mass Transfer, vol. 43, no. 8, pp. 759–766, 2007.

[13] Ali, M. S., Tariq, A., and Gandhi, B. K., Flow and Heat Transfer Investigation behind Trapezoidal Rib using PIV and LCT Measurements. Experiments in Fluids, vol. 54, no. 5, pp. 1520:1–15, 2013.

[14] Sharma, N., Tariq, A., Mishra, M., Detailed Heat Transfer Investigation inside a Rectangular Duct with an Array of Ventilated Rib Turbulators, in Fluid Mechanics and Fluid Power – Contemporary Research, Eds. Saha, A. K., Das, D., Srivastava, R., Panigrahi, P. K., Muralidhar, K., Book Chapter-73, Springer, India, 2017.

[15] Sharma, N., Tariq, A., Mishra, M., Detailed Heat Transfer and Friction Factor Characteristics in a Rectangular Duct with Rib Turbulators, Proc. 42nd National Fluid Mechanics and Fluid Power Conf., NITK Surathkal, Karnataka, India, FMFP 2015-69, December 14-16, 2015

[16] Tanda, G., Effect of rib spacing on heat transfer and friction in a rectangular channel with 45˚ angled rib turbulators on one/two walls. International Journal of Heat and Mass Transfer, vol. 54, no. 5–6, pp. 1081–1090, 2011.

[17] Sharma, N., Tariq, A., Mishra, M., Effect of Pentagonal Ribs on Local Heat Transfer and Friction Factor in a Rectangular Channel, Proc. 23rd Natl. Heat Mass Transfer Conf. & 1st Intl. ISHMT-ASTFE Heat Mass Transfer Conf., Thiruvananthapuram, India, Paper 671, December 17-20, 2015.

[18] Yang, Y. T., Chen, P. J., Numerical Optimization of Turbulent Flow and Heat Transfer Characteristics in a Ribbed Channel, Heat Transfer Engineering, vol. 36, no. 3, pp. 290–302, 2015.

[19] Park JS, Han JC, Huang Y, Ou S. Heat transfer performance comparisons of five different rectangular channels with parallel angled ribs. Int. J. Heat Mass Transfer 1992; 35(11):2891–903.

[20] Chandra PR, Alexander CR, Han JC. Heat transfer and friction behaviors in rectangular channels with varying number of ribbed wall. Int J Heat Mass Transfer 2003; 46(3):481–95. [34] Lau SC, Rd Mcmillin, Han JC. Turbulent heat transfer and friction in a square channel with discrete rib turbulators. J Turbomach 1991; 113(3):360–7.

[21] Lau SC, Rd Mcmillin, Han JC. Heat transfer characteristics of turbulent flow in a square channel with angled discrete ribs. J Turbomach 1991; 113(3):367–74.

[22] Zhang YM, Gu WZ, Han JC. Heat transfer and friction in rectangular channels with ribbed or ribbed-grooved walls. J Heat Transfer 1994; 116(1):58–65.

[23] Prasad K, Mullick SC. Heat transfer characteristics of a solar air heater used for drying purposes. Appl Energy 1983; 13(2):83–93. [38] Chaube A, Sahoo PK, Solanki SK. Analysis of heat transfer augmentation and flow characteristics due to rib roughness over absorber plate of a solar air heater. Renew Energy 2006; 31(3):317–31.

[24] Chaube A, Sahoo PK, Solanki SK. Effect of roughness shape on heat transfer and flow friction characteristics of solar air heater with roughened absorber plate. WIT Trans Eng Sci 2006; 53:43–51.

[25] Karmare SV, Tikekar AN. Analysis of fluid flow and heat transfer in a rib grit roughened surface solar air heater using CFD. Sol Energy 2010; 84(3): 409–17.

[26] Gandhi BK, Singh KM. Experimental and numerical investigations on flow through wedge shape rib roughened duct. Inst Eng (India) J MC 2010; 90 (January):13–8.

[27] Sharma AK, Thakur NS. CFD based fluid flow and heat transfer analysis of a V-shaped roughened surface solar air heater. Int J Eng Science Technology 2012; 4(5):2115–21.

[27] Kumar S, Saini RP. CFD based performance analysis of a solar air heater duct provided with artificial roughness. Renew Energy 2009;34(5):1285–91.

[28] Yadav AS, Bhagoria JL. A CFD based performance analysis of an artificially roughened solar air heater having equilateral triangular sectioned rib roughness on the absorber plate. Int J Heat Mass Transfer 2014; 70:1016–39.
[29] Yadav AS, Bhagoria JL. Heat transfer and fluid flow analysis of solar air heater: a review of CFD approach.Renew Sustain Energy Rev2013; 23:60–79.
[29] Yadav AS, Bhagoria JL. A CFD based heat transfer and fluid flow analysis of a solar air heater provided with circular transverse wire rib roughness on the absorber plate. Energy 2013; 55:1127–42.
[30] Yadav AS, Bhagoria JL. Modeling and simulation of turbulent flows through a solar air heater having square sectioned transverse rib roughness on the absorber plate. Sci World J 2013; 2013:12.