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Enhancement of Heat Transfer Coefficient in DPDGSAH having V Ribs with Symmetrical Numbers of Gaps in a Rectangular Duct: A Robust Design Approach

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Abstract: ***Enhancement of heat transfer coefficient has been carried out through a solar air heater (DPDGSAH) having artificially roughened V-type geometry with symmetrical gaps and then optimized by use of Robust design method. A L25 orthogonal array is used to optimize the geometry factors for the maximum thermal performance of the ribbed channel. Thermal performance concept includes maximization of heat transfer coefficient and its thermal efficiency. Maximization of heat transfer coefficient is taken as the criteria of optimization. The investigation encompassed Reynolds number (Re) ranging from 4000 to 12000, relative roughness pitch (P/e) values of 4–12, angle of attack (α) range of 30^0–90^0, relative gap width (g/e) values of 1–5, Number of gaps (Ng) values of 1–5, and relative roughness height (e/D) values of 0.043. Results show that Reynolds Number (Re), relative roughness pitch (P/e), angle of attack (α) have the greatest influences parameters on maximization of heat transfer coefficient.***

Keywords: V-Rib with symmetrical gaps, Taguchi method, ANOVA, Heat transfer coefficient

1. Introduction

It is observed that the solar energy is the ultimate source of energy because renewable energy sources are generated from the solar energy [1], and hence can be applicable as a source of energy in many engineering applications such as heating systems, solar water heaters, drying systems for agricultural products etc. Non-renewable sources are depleting at a very fast rate therefore, employing new sources of energy like renewable sources which design efficiently in the field of engineering. Solar air heaters which collect solar radiation and convert it into heat energy but with low thermal efficiency because of low convective heat transfer coefficient between duct channel due to maximum thermal losses. In order to achieve the maximum efficiency and enhance the heat transfer coefficient, it is recommended to break laminar sub layer by introducing artificial roughness on one side of absorber plate.

Creating artificial roughness in the form of wires and ribs in various arrangements were studied by various investigators [2–5]. Artificial roughness can be introduce for the enhancement of heat transfer coefficient and thermal efficiency between the absorber plate and air hence improving the thermal performance of solar air heater. However, as the laminar sub layer is broken and turbulence is created resulting in an increase of pressure drop, more friction will occur and more pumping power required to blowing the air through the duct, therefore great considerations should be taken in modeling and designing such structures.

Numerous experimental and numerical investigation have been carried out in design and optimization of solar air heater with artificial roughness. Prasad and Saini [4,6] has investigated the roughness geometry effects such as relative roughness pitch and relative roughness height on friction factor and convective heat transfer of fully developed turbulent flow in a solar air heater. They conclude that maximum heat transfer reached near the reattachment points. They also found that reattachment of free shear layer was not formed if the ratio of relative roughness pitch to height is smaller than 8-10. They achieved an enhancement of 2.38 and 4.25 times on a smooth duct in heat transfer and friction factor, respectively. Karwa [7] have experimentally studied the thermo-hydraulic performance of air flow through a rectangular duct with repeated chamfered ribs mounted on one side and five different aspect ratios. They proposed empirical correlations for friction factor and heat transfer coefficient. Sahu and Bhagoria [25] has investigated experimentally the effect of pitch varying from 10 to 30 by taking height of the rib to be 1.5 mm and aspect ratio 8 on friction factor and heat transfer coefficient for 90˚ broken transverse ribs. They have been found that the highest Nusselt number attained for relative roughness pitch (P/e) of 13.33 and decreased with an increase in relative roughness pitch. The maximum thermal efficiency of 83.5 % has been found for 13.33 relative roughness pitch.

Kumar [8] have investigated heat transfer and friction factor with discrete W-shaped roughness. The maximum enhancement of Nusselt number was found to be 2.16 times that for smooth duct for angles of attack of 60° and relative roughness height of 0.0338. Muluwork [9] have investigated the thermal performance of a SAH with staggered discrete V-apex up and down ribs. Result shown that the V-down discrete ribs have higher Stanton number as compared to transverse discrete ribs and V-up ribs. In case of smooth duct the Stanton number was found to be 1.32 to 2.47 folds. It was seen from the results that V-down discrete ribs produces the maximum friction factor among all the investigated configurations.

Kenan Yakut [10] have optimized the design parameters on double side delta-winglet under different geometrical and flow factor, by use of Taguchi method. Demir, [11] have studied the heat transfer enhancement and corresponding pressure drop over a flat surface equipped with circular cross section perforated pin fins in a rectangular channel, by use of Taguchi method. Isak Kotcioglu [12] have experimentally investigated to optimize the design geometry parameters of a rectangular duct with plate-fins heat exchanger by applying Taguchi method. Gunes [13] have applied Taguchi method to determine the optimum values of the design parameters. An optimum design parameter of the concentric heat exchanger with injector turbulators using Taguchi experimental design method was carried out by Turgut [14]. Çakmak [15] have identified optimum parameters affecting pressure losses and heat transfer for a concentric heat exchanger using Taguchi's method. The authors used the L16 orthogonal arrays for five different factor and found Re number to be the most effective parameter. Aghaie [16] have numerically examined the thermal performance of a solar air heater and optimized it by using the Taguchi method. Result shows that height and rib relative pitch has the most impact parameter on enhancing thermo-hydraulic behavior.

Isak Kotcioglu [12] have experimentally investigated to optimize the design geometry parameters of a rectangular duct with plate-fins heat exchanger by applying Taguchi method. Pawar [17] have experimentally investigated heat transfer enhancement in a SAH with diamond shape ribs in its absorber plate. Effecting factor, includes Reynolds number, angle of attack rib, relative height and pitch, were taken in to consideration. Kumar [8] have also investigated a similar research on a rectangular duct having repeated V-shaped ribs with gap. Their results showed that Nusselt number enhanced 6.74 times as compared to smooth channel, while friction factor also increased by 6.37 times to the smooth channel. Boulemtafes and Benzaoui [18] has carried out a CFD numerical investigation on heat transfer enhancement in a solar air heater with transverse rectangular ribs. A comprehensive review on thermo-hydraulic performance of roughened surface SAH can be found in Refs. [1,19-21].

In the current study, Enhancement of heat transfer coefficient has been carried out through a double-pass double glaze solar air heater having artificially roughened V-type geometry with symmetrical gaps and then optimized by use of Robust design method. Heat transfer coefficient (h), Nusselt number (Nu) and thermal efficiency are numerically calculated. Experiment setup is introduced by Taguchi method and roughness geometry is then optimized by Taguchi method and applying ANOVA in order to determine the most effective parameter on heat transfer coefficient.

2. Experiment setup and procedure

In this experimental study, the tests are constructed in a double-pass double glaze solar air heater based on the particular design parameters. An experimental setup as shown in Fig. 2.1(a) and (b) has been designed and fabricated to study the enhancement of heat transfer coefficient having V-rib with symmetrical gap as a artificial roughness, made of circular wire on the absorber plate in rectangular duct channel. The wooden rectangular duct having internal size of 1500mm X 200mm X 30mm. The entrance section, test section and exit section are 400mm, 1500mm, and 200mm respectively as recommended in ASHRAE standards 93-77 [26].The exit section of 200mm length is used after test section in order to reduce the end effects in the test section. An electric heater of size 1500mm X 200mm was fabricated by Ni-chrome wire of 25 SWG on 5 mm asbestos sheet to supply constant heat flux of 1000 W/m^2. A 25 mm cotton wool of thermal conductivity 0.029W/ m K was applied as insulation. The top side of entry and exit section of the duct is covered with smooth face of 12 mm thick plywood. The heated plate is 1.5 mm thick G.I. plate on which 60° inclined V-rib with symmetrical gap made of circular wire has been pasted with the help of flex glue. The mass flow rate of the air is measured by means of an orifice plate connected with a U tube manometer with distilled water as manometric fluid and flow is controlled by the control valve provided in the pipe line. The air is sucked through the duct by means of a blower driven by a 3-phase 440V, 2.3KW and 1420 rpm, A.C. motor.

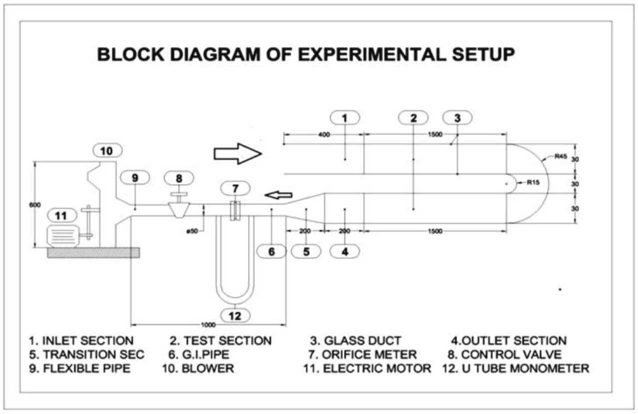

Figure 2.1(a): Schematic diagram of experimental set-up

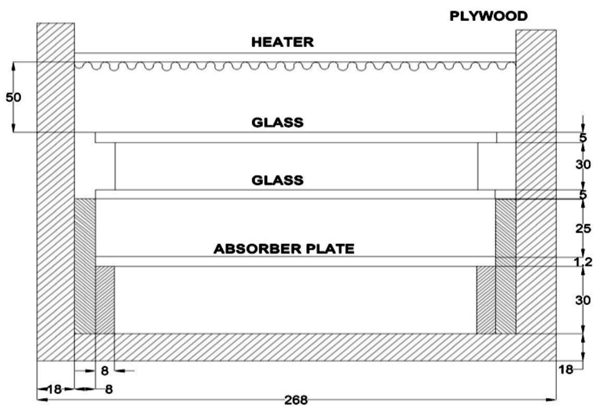

Figure 2.1(b): Cross sectional view of duct

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2.1 Instrumentation

The calibrated (K-type) Chromel Alumel 28 SWG thermocouples with digital temperature indicator instrument were used to measure the air and heated plate temperature at different locations. Twelve thermocouples were inserted in the test section of the duct at equal interval.

The mass flow rate through the duct was measured by a flange type calibrated orifice meter which was fabricated and fitted in a 50 mm diameter pipe carrying air from the duct to the blower.

The pressure drop across the test –section of the duct was measured by means of an air flow meter and micro manometer having a least count of 0.01 mm. The movable reservoir is mounted on a sliding arrangement using a lead screw having a pitch of 1.0mm and a graduated dial having 100 divisions; each division showing a movement of 0.01 mm of the reservoir.

2.2 Procedure

In the present experimental study, twenty five roughened surfaces, in addition to smooth surface, having different values of roughness parameters, were investigated under similar flow conditions for the purpose of comparison with the roughened absorber surfaces. Five values of mass flow rates were considered for each roughened surface at a fixed heat flux of 1000 W/m2.

The experimental data for air and roughened plate temperatures at different locations in the duct were recorded under quasi-steady state conditions, where mass flow rate of fluid and heat flux are given. The quasi-steady state condition is assumed to be reached, when the temperature change does not occurs for about 10-12 minutes. When a change in operating condition is made, it takes about 20-30 minutes to reach in quasi-steady state condition except when new set-up is started at ambient temperature, it takes about 3-4 hours to reach quasi-steady state condition. Five values of flow rates were used for each set at a fixed heat flux. After each change of mass flow rate, the system has to maintain a steady state before the data were recorded. Twenty five of roughened absorber plates, as shown in Table 2 have been tested.

The test set up was validated by conducting experiments for a smooth duct before conducting test for roughened duct. The heat transfer coefficient for smooth duct were determined from experimental data. These values were then compared with the values obtained from the correlations of Dittus Boelter equation [22] for heat transfer coefficient for smooth duct. The experiment data is also validate and optimized through Taguchi method followed by ANOVA technique. These optimized values were also compared with the values obtained from the predicted equations [23].

Dittus Boelter equation : $N_u = 0.023Re^{0.8}Pr^{0.4}$

The comparison of experimental and predicted values from equation of Nusselt number has been shown in Fig. 2.

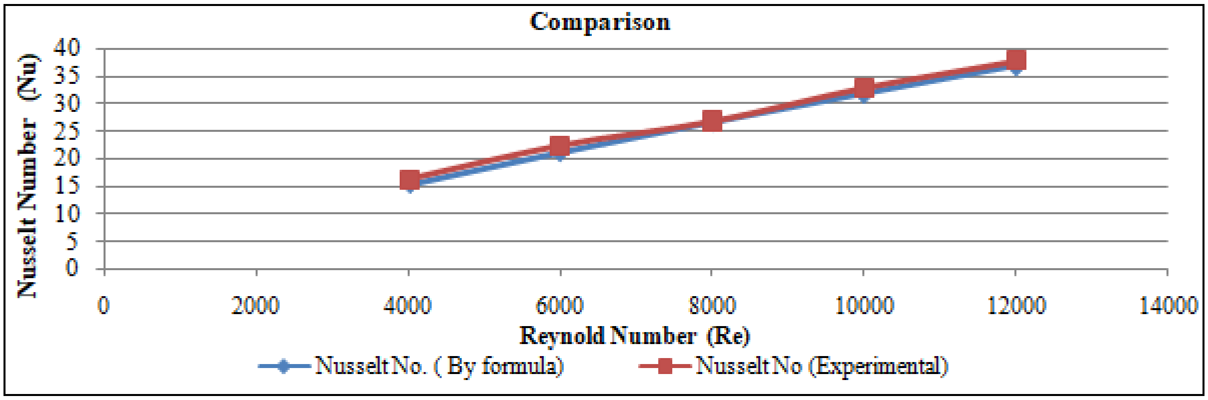

Figure 2: Comparison of experimental and predicted values of smooth rectangular duct

2.3 Data reduction

The experimental data have been reduced to obtain the mass flow rate, average plate temperature, average air temperature, and Reynolds number. These data were then used to determine the heat transfer coefficient, Nusselt number and thermal efficiency.

The mass flow rate was then calculated from the following expression:

$$\dot{m} = cd\ A_0\sqrt{\frac{2\rho\,(\Delta P_o)}{1-\beta^4}} \qquad (1)$$

The average air temperature was then calculated from the following expression:

$$T_f = (T_i + T_o)/2 \qquad (2)$$

The average heat transfer coefficient was then calculated from the following expression:

$$h = \frac{\dot{m}c_P[t_o - t_i]}{A_P(T_P - T_f)} \qquad (3)$$

The Nusselt number was then calculated from the following expression:

$$N_u = \frac{hD_h}{k} \qquad (4)$$

The Reynold number was then calculated from the following expression:

$$Re = \frac{\rho V D_h}{\mu} \qquad (5)$$

The friction factor was determined from the measured values of pressure drop (DP) across the test section as given below;

$$f = \frac{2(\Delta P)_d D_h}{4\rho L_f V^2} \qquad (6)$$

The Thermal efficiency was then calculated from the following expression :

$$\eta = \frac{G C_P (T_0 - T_i)}{I} \qquad (7)$$

Where, $G = \frac{\dot{m}}{A_p}$

Table 1: Levels of the variables used in the experiment

Control Factor		Levels				
		I	II	III	IV	V
A	Reynolds No.	4000	6000	8000	10000	12000
B	Relative roughness pitch (p/e).	4	6	8	10	12
C	Angle of attack (α),	30^0	45^0	60^0	75^0	90^0
D	Relative gap width (g/e)	1	2	3	4	5
E	Number of gaps (Ng).	1	2	3	4	5

Table 2: Orthogonal array for L25 (3125) Taguchi Design [24].

RUN	A	B	C	D	E
1	1	1	1	1	1
2	1	2	2	2	2
3	1	3	3	3	3
4	1	4	4	4	4
5	1	5	5	5	5
6	2	1	2	3	4
7	2	2	3	4	5
8	2	3	4	5	1
9	2	4	5	1	2
10	2	5	1	2	3
11	3	1	3	5	2
12	3	2	4	1	3
13	3	3	5	2	4
14	3	4	1	3	5
15	3	5	2	4	1
16	4	1	4	2	5
17	4	2	5	3	1
18	4	3	1	4	2
19	4	4	2	5	3
20	4	5	3	1	4
21	5	1	5	4	3
22	5	2	1	5	4
23	5	3	2	1	5
24	5	4	3	2	1
25	5	5	4	3	2

Table 3: Experimental plan of L25 for heat transfer coefficient with their SNR values

Re	P/e	A	g/e	Ng	h	S/N ratio
4000	4	30	1	1	16.12	24.14
4000	6	45	2	2	21.04	26.46
4000	8	60	3	3	33.76	30.56
4000	10	75	4	4	32.92	30.349
4000	12	90	5	5	12.28	21.784
6000	4	45	3	4	13.54	22.632
6000	6	60	4	5	41.06	32.268
6000	8	75	5	1	39.2	31.866
6000	10	90	1	2	46.83	33.41
6000	12	30	2	3	19.13	25.634
8000	4	60	5	2	34.98	30.876
8000	6	75	1	3	36.07	31.143
8000	8	90	2	4	48.17	33.656
8000	10	30	3	5	61	35.707
8000	12	45	4	1	26.07	28.323
10000	4	75	2	5	37.02	31.369
10000	6	90	3	1	47.25	33.488
10000	8	30	4	2	50.95	34.143
10000	10	45	5	3	72.43	37.198
10000	12	60	1	4	38.43	31.693
12000	4	90	4	3	43.75	32.82
12000	6	30	5	4	59.25	35.454
12000	8	45	1	5	68.36	36.696
12000	10	60	2	1	111.3	40.932
12000	12	75	3	2	45.3	33.122

3. Experimental Design and Taguchi Method

Taguchi methods are statistical methods, or sometimes called robust design methods. Taguchi has developed a methodology for the application of designed experiments based on "Orthogonal Array" which gives much reduced "variance" for the experiment with "optimum settings" of control parameters. Taguchi method is known as one the widely-used tools in engineering applications, with the minimum number of experiments. The choice of a suitable orthogonal array is critical for the success of an experiment and depends on the total degrees of freedom required, where the degrees of freedom for the array design should be greater than or at least equal to those for the design parameters [12]. For this reason L25 are suitable for the considered design to study the main and interaction effects, goal of experiment, resources and budget available and time constraints. Before selecting the orthogonal array, the minimum number of conducted experiments (N_{exp}) can be fixed by using the following relation, [12].

$$N_{\text{exp}} = (N_{\text{level}} - 1)Np + 1 \qquad (8)$$

where Np is the number of parameter and N_{level} is the number of levels. These orthogonal arrays prescribe specific arrangement of fractional experiments with regard to the influencing factors and their levels. In the current work, a L25 (5^5) orthogonal array was used, which implied carrying out 25 tests with 5 factors of 5 levels that listed in Table 1.

Under the present study an attempt has also been made to optimize the heat transfer coefficient. Therefore, a large number of factors were included so that non-significant variables can be identified at the earliest opportunity. These studies revealed that parameters viz. Reynolds number (Re), relative roughness pitch (P/e), angle of attack (α), Number of gaps (Ng), relative gap width (g/e), etc. had influenced solar air heater performance. The impact of such parameters was studied using L25 (5^5) orthogonal design. The operating conditions under which the performance output carried out are as given in table 1.

Table 2 each column represents a test parameter whereas each row stands for a treatment or test condition which is a combination of parameter levels. In conventional full factorial experiment design, 3125 (5^5) runs were required to study five parameters each at five levels whereas, Taguchi's factorial experiment approach reduces it to 25 runs only offering a great advantage in terms of experimental time and cost. The experimental observations were further transformed into signal-to-noise (S/N) ratio), which are log functions of desired output, serve as objective functions for optimization, help in data analysis and prediction of optimum results [23]. The Signal-to-Noise ratios are also used to measure the effect of the noise on the system. In this system while the signal is defined as the mean, the noise is defined as the standard deviation. There are several S/N ratios available depending on the type of performance characteristics. The S/N ratio for maximum heat transfer coefficient can be expressed as ''Higher is better'' characteristic, which was calculated as logarithmic transformation of loss function as shown below.

Higher is a better characteristic; [12]

$$\frac{S}{N} = -10\log\frac{1}{n}\left(\sum\frac{1}{y^2}\right) \tag{9}$$

where 'n' is the number of observations, and y is the observed data. ''Higher is better (HB)'' characteristic, with the above S/N ratio transformation, is suitable for maximization of heat transfer coefficient. The standard linear graph, as shown in Fig. 4, was used to assign the factors and interactions to various columns of the orthogonal array. In table 2, each column represents a test parameter whereas each row stands for a experiments or test condition which is a combination of parameter levels. The first column was assigned to Numbers of Run or Experiments, the second column to Reynolds number (A), the third column to Relative roughness pitch (B), the forth column to Angle of attack (C), the fifth column to Relative gap width (D), the sixth column to Number of gaps (E).

3.1 Analysis of variance (ANOVA) and the effect of factor

The Analysis of Variance is used to analyze the results of the orthogonal array experiment and to determine how much variation each quality-influencing factor contributes [24]. Table 3 shows the experimental design for L25 orthogonal array. In order to find out statistical significance of various factors like Reynolds number (A), relative roughness pitch (B), angle of attack(C), relative gap width(D) and number of gaps(E) on heat transfer coefficient, analysis of variance (ANOVA) was conducted on experimental data.

Table 4 shows the results of the ANOVA with the heat transfer coefficient. This analysis was undertaken for a level of confidence of significance of 5%. The total sum of squared deviations SS_T from the total mean S/N ratio η_m can be calculated as [23]:

$$SS_T = \sum_{i=1}^{m}\eta_i^2 - \frac{1}{m}\left[\sum_{i=1}^{m}\eta_i\right]^2 \tag{10}$$

Where, η is calculated S/N ratio and m is number of experiments.
The sum of square due to each parameter is calculated as:

$$\text{SSp}=\sum_{i=1}^{t} n_i(mf_i - m)^2 \tag{11}$$

Where,
t = number of levels =5;
n_i is the number of experiments with i^{th} level;
mf_i is the mean S/N ratio for the i^{th} level;
m is the mean of all means of S/N ratios.

From the Table 4 it is found that all the five parameters Reynold Number, Relative roughness pitch, Angle of attack, Relative gap width and Number of gaps are the significant parameters for affecting rise in heat transfer coefficient. Therefore, based on the S/N ratio and ANOVA analyses, the optimal parameters for maximum heat transfer coefficient are Reynold number at level 5 (12000), Relative roughness pitch at level 4 (10), Angle of attack at level 3 (60^0), Relative gap width at level 5 (2) and Number of gap at level 5 (1). Similarly, Table 4 shows the major contribution order of the parameters for maximum heat transfer coefficient is Reynold number(51.3%) then Relative roughness pitch(40.57%) and then Angle of attack(5.03%). Relative gap width and Number of gap having least contribution for maximum heat transfer coefficient. So, the optimal parameters are $\mathbf{A_5B_4C_3E_1D_2}$.

Table 4: AVOVA table for heat transfer coefficient

No	Source of Variation	DOF	Seq SS	Adj. MS	F Ratio	Percentage contribution
1	Reynold Number	4	259.47	64.86	22.05	51.30%
2	Relative roughness pitch	4	205.21	51.3	17.44	40.57%
3	Angle of attack	4	25.44	6.36	2.16	5.03%
4	Relative gap width	4	0.81	0.2	0.068	0.16%
5	Number of gaps	4	3.03	0.757	0.257	0.60%
6	Error	4	11.76	2.94		2.33%
7	Total	24	505.73			100%

Table 5: Results of the confirmation experiments for output performance

Performance output	Optimal control Factor settings	S/N Ratio predictive values (db)	S/N Ratio experimental values (db)	Error (%)
1. Heat transfer Coefficient	$A_5B_4C_3E_1D_2$	42.23	40.93	3.08%

Confirmation Experiment

The optimal combination of control parameter has been determined in the analysis discussed above. However, The confirmation experiment is the final step in the design of experiment process. The confirmation experiment is conducted to validate the interference drawn during the analysis phase.

The confirmation experiment is performed by considering the new set of factor settings $\mathbf{A_5B_4C_3E_1D_2}$ to predict the heat transfer coefficient. The estimated SNR for heat transfer coefficient can be calculated with the help following predictive equation [23]:

$$\eta 1 = \bar{T} + (A_5 + \bar{T}) + (B_4 + \bar{T}) + (C_3 + \bar{T}) + (E_1 + \bar{T}) + (D_2 + \bar{T})$$

Where η is the predicted average, $\bar{T}$ is the average result of 25 runs. The SNR value of heat transfer coefficient by the predictive equation was found to be 42.23 db and from the experimental results value of SNR is found to be 40.93 db. The resulting model seems to be capable of predicting heat transfer coefficient to a reasonable accuracy. An error of 3.078% for the SNR of heat transfer coefficient is observed as shown in Table 5.

4. Results and Discussion

The heat transfer coefficient in double-pass double glazed solar air heater with different roughness and operating factor has been evaluated. For the experimental analysis regarding the solar air heater constructed the considered parameters are the are Reynolds number (Re), relative roughness pitch (P/e), angle of attack (α), relative gap width (g/e) and number of gaps. Each parameter has five levels as given in Table 1. In order to establish these conditions, L25 orthogonal array is organized and the collected experimental data are analyzed for determination of the effects of each design parameter on heat transfer coefficient.

Comparison of experimental and predicted values for smooth duct from equation of Nusselt number is given by

Dittus Boelter equation : $\boldsymbol{N_u = 0.023Re^{0.8}Pr^{0.4}}$

and shown in fig. 2. which shows good agreement between experiment and predicted value for smooth duct in solar air heater. The overall mean for the S/N ratio of the heat transfer coefficient is found to be 40.93 db. as given in Table 3. Analysis of experimental data is carried out manually as well as using the popular software specifically used for design of experiment applications known as MINITAB 18. The effect of the five control factors on the performance measures heat transfer coefficient are shown graphically in figs. 3 respectively.

In the present study, the optimal parameters have been designed to maximize the heat transfer coefficient by Taguchi method followed by ANOVA. The most important parameter with the aspect of heat transfer coefficient is Reynolds number, as the Reynolds number shows the contribution ratio of the order of 51.03% as shown in Table 4. Thus heat transfer can be improved by controlled change of flow Reynolds number. Optimum condition of design parameters is A5 B4 C3 E1 and D2 and the optimum values of the parameters for maximum heat transfer condition are given as follows: Re= 120027, P/e= 10, α= 60^0, g/e= 2 and Ng= 1. The major contribution order of the parameters for maximum heat transfer coefficient is Reynold number having 51.03% then Relative roughness pitch having 40.57% and then Angle of attack having 5.03% as shown in Table 4. The analysis of variance ANOVA was performed to determine the variance of each control factor on the overall results and it was found that the most important parameter for variance is Reynolds number, Relative roughness pitch, Angle of attack, relative gap width and number of gaps corresponding to heat transfer coefficient, respectively. The confirmation of experimental test was performed and an error of the order of 3.078% between experimental and predicted values of optimum performance statistics (SNR) for heat transfer coefficient was obtained, as shown in Table 5 respectively.

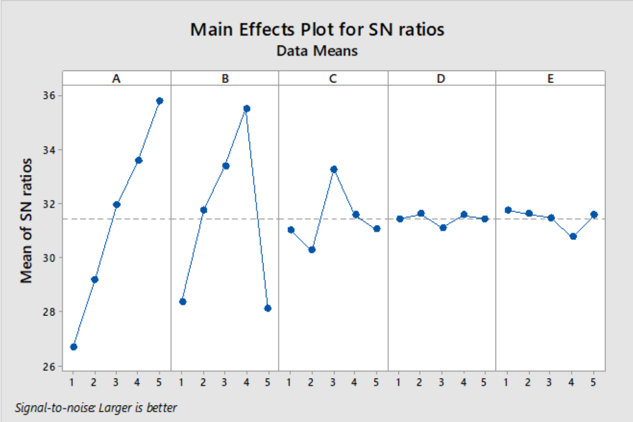

Figure 3: Effect of control factor on heat transfer coefficient.

5. Conclusion

Analysis of heat transfer coefficient in the solar air heater duct enhanced with V-ribs having symmetrical gaps in the absorber plate is carried out, with the aim to improve the overall performance of the system.

The present study has been done by using Taguchi method because it gives all the possible levels of factors with the minimum number of experiments. The author concludes that for maximization of heat transfer coefficient is concerned, the factors such as Reynolds number (Re), relative roughness pitch (P/e) and angle of attack (α) have major significant effect on the process of heat transfer on solar air heater with rib having symmetrical gaps as shown in fig.3. Factors such as number of gaps (Ng) and relative gap width (g/e) have negligible effect on the whole process. The maximum S/N ratio of the heat transfer coefficient is found to be 40.93 db, which shows the loss associated with this process is minimum. The experimental result is than validated with the predicted values of optimum performance statistics (SNR) with an error of 3.078%, as shown in Table 5 respectively.

Nomenclatures

Ac	surface area of absorber plate, m^2
C_p	specific heat of air, J/kg K
D, D_h	equivalent or hydraulic diameter of duct, m
e	rib height,
h	heat transfer coefficient, W/m^2 K

H	depth of air duct, m
I	intensity of solar radiation, W/m^2
K	thermal conductivity of air, W/m K
L	length of test section of duct or long way length of mesh, m
m	mass flow rate, kg/s
P	pitch, m
Q_u	useful heat gain, W
Q_l	heat loss from collector, W
Q_t	heat loss from top of collector, W
To	fluid outlet temperature, K
Ti	fluid inlet temperature, K
Ta	ambient temperature, K
Tp	mean plate temperature, K
W	width of duct, m
e/Dh	relative roughness height
e/H	rib to channel height ratio
Nu	Nusselt number
p/e	relative roughness pitch
Pr	Prandtl number
Re	Reynolds number
W/H	duct aspect ratio
SNR	signal to noise ratio

Greek symbols

ηth	thermal efficiency
μ	dynamic viscosity, Ns/m^2
ρ	density of air, kg/m^3
α	angle of attack, degree

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