Optimization of Design Parameters for Wire Mesh Fin Arrays as a Heat Sink Using Taguchi Method

Kavita H. Dhanawade, Hanamant S. Dhanawade

***Abstract*—**Heat transfer enhancement objects like extended surfaces, fins etc. are chosen for their thermal performance as well as for other design parameters depending on various applications. The present paper is on experimental study to investigate the heat transfer enhancement through wire mesh fin arrays equipped with horizontal base plate. The data used in performance analysis were obtained experimentally for the material (mild steel) for different heat inputs such as 40, 60, 80, 100 and 120 watt, by varying wire mesh diameter, fin height and spacing between two fin arrays. Using the Taguchi experimental design method, optimum design parameters and their levels were investigated. Average heat transfer coefficient was considered as a performance characteristic parameter. An L_9 (3^3) orthogonal array was selected as an experimental plan. Optimum results were found by experimenting. It is observed that the wire mesh diameter and fin height have a higher impact on heat transfer coefficient as compared to spacing between two fin arrays.

***Keywords*—**Heat transfer enhancement, finned surface, wire mesh diameter, natural convection.

NOMENCLATURE

A_T	Total surface area, m^2
A_f	Area of all horizontal wire, m^2
A_f	Area of all vertical wire, m^2
A_f	Total area of fin, m^2
A_b	Base surface area, m^2
D	Diameter of wire mesh, m
H	Height of fin, m
L	Length of fin, m
W	Width of fin, m
K_b	Thermal conductivity of insulation brick, W/mK
K_a	Thermal conductivity of air, W/mK
Ts	Average surface temperature, °C
T_a	Ambient temperature, °C
T_{mf}	Mean film temperature, °C
T_f	Fins temperature, °C
dtb	Difference of temperature at bottom of fin base to temperature in brick, °C
dxb	Distance of thermocouple from base plate to insulation brick, m
Q_{cond}	Conduction heat loss, W
Q_{cov}	Heat transfer by convection, W
Q_{Net}	Net heat transfer, W
h_a	Average heat transfer coefficient, W/m^{2}K
N_{uf}	Nusselt number with fin
G_r	Grashof number
P_r	Prandalt number
R_a	Rayleigh number
N_{uwf}	Nusselt number without fin
$\beta 1/T_{mf}$	Coefficient of volumetric thermal expansion (1/K)
μ	Dynamic viscosity, Ns/m^2
ν	Kinematic viscosity, m^2/s
C_p	Specific heat capacity, J/kg K
g	Acceleration due to gravity, m/s^2

Kavita H. Dhanawade is with the Lokmanya Tilak College of Engineering, Department of Mechanical Engineering, University of Mumbai, Maharashtra, India (phone: 9224410022; e-mail: kavitadhanawade2@gmail.com).

Hanamant S. Dhanawade is with the Smt. Indira Gandhi College of Engineering Department of Mechanical Engineering, University of Mumbai, Navi Mumbai, Maharashtra, India (e-mail: dhanashri_hd@rediffmail.com).

I. INTRODUCTION

IN the field of mechanical engineering the enhancement of heat transfer is an important subject. The removal of excessive heat from system components is essential to avoid the damaging effects of burning or overheating. The heat transfer from a surface is enhanced by increasing the heat transfer coefficient between the surface and its surrounding or by increasing the heat transfer area of the surface. To enhance heat transfer in several industries the extended surface or fins are used. For both natural and forced convection heat transfer, fins with different geometries such as square, circular pin-fins, plate fins, perforated fins etc. are used. Mahmoud et al. [1] worked experimentally on the micro fin geometry made up of copper material for varying fin height and fin spacing by natural convection heat transfer. They concluded that the heat transfer coefficient is directly proportional to fin spacing and inversely proportional to fin height. Sabale et al. [2] worked experimental on multiple V-fin arrays with vertical heated plate by natural convection heat transfer. For experimentation V-Fins are arranged in two different arrangements varying bottom spacing and fin height for different heat input. They observed that V-type partition plate fin enhances heat transfer rate as compared to horizontal partition plate fins. Goshayeshi and Ampofo [3] worked on vertical and horizontal surface using vertical fins by natural convection for heat transfer enhancement. They use three configurations: 1) Vertical plate with vertical fins; 2) horizontal face with fin spacing up; 3) vertical plate with horizontal fin. They observed that vertical plate with vertical fins gives the best performance for natural cooling than other configurations. AIEssa et al. [4]-[7] studied the heat dissipation from a horizontal rectangular fin by using finite element techniques under natural convection for different perforations like equilateral triangular of bases parallel and towards its fin tip, square and rectangular perforations with an aspect ratio of two. They showed that heat dissipation rates were enhanced for perforation in the fins as compared to solid fins.

Suryawanshi and Sane [8] worked experimentally on normal and inverted notched fin arrays (INFAs) and obtained up to 40% increase in average heat transfer coefficient for

INFAs as compared to normal array. AI-Widyan and AI-Shaarawi [9] worked numerically on perforated fins under natural convection. They found that the heat transfer from the perforated fins increased with Grashoff number as well as by decreasing spacing between the holes. Wadhah [10] investigated experimentally that the heat transfer coefficient and heat dissipation rate increases due to increase in the number of perforation. Dhanawade et al. [11] investigated the solid fin arrays and external dimensionally equivalent perforated fin arrays experimentally by natural convection and validated the results by CFD simulation. They noted that the enhancement in the heat transfer of perforated fins than that of the solid fins for all heat inputs and it is increasing with the increase in size of perforations. Also, they worked on the CFD simulation to investigate heat transfer flow visualization of perforated fin arrays.

Shaeri et al. [12]-[14] worked numerically on a three-dimensional array of rectangular fins with lateral and longitudinal square perforation. The RNG k-Ɛ model and Navier Stroke's equations were used for investigation. They showed that heat dissipation rate is more due to perforation as compared to solid fins. Yaghoubi et al. [15] worked numerically on three dimensional laminar steady flow conjugate heat transfer around solid and rectangular fin arrays. They used FORTRAN code based on the SIMPLE algorithm with staggered grid for the analysis. The range of Reynolds number was 100 to 250. Dhanawade et al. [16]-[18] experimentally the turbulent fluid flow and convective heat transfer around arrays of solid and perforated fins. The main objective of their study was to determine the thermal performance of a new design of perforated fins with circular perforations and without perforations by forced convection. It was noted that the enhancement in the heat transfer of perforated fins is better than that of the solid fins for heat inputs, for tested range of Reynolds number and for sizes of perforations. They also experimentally investigated the optimum design parameters such as Reynolds number, porosity and thickness of fin and their level for lateral circular perforated fin arrays under forced convection. They found that the maximum heat transfer rate was observed at the 8.7×10^4 Reynolds number and porosity (Ø) = 0.22. Fuskele and Survivya [19] worked on double pipe heat exchanger with twisted dense wire mesh. They determines friction factor and heat transfer coefficient having different twist ratios (Y=5, Y=7) for twisted wire mesh. The comparison has been carried out in between twisted wire mesh and plain tube. They observed that friction factor and heat transfer coefficient increases with decrease in twist ratio. An investigation by Pismenny [20] was based on results of experimental data dependencies for calculations of heat transfer convective coefficient of flat surfaces with meshed wire finning by natural convection for different orientations in the space. He studied three positions of surface in the space, which differed by condition of the fins. Positions corresponding to the surface washing at vertical base and vertical fin position two at horizontal base and vertical fins; Position at vertical base and horizontal fins. Results of experiments were approximated by the degree like dependence, widely used in theoretical and applied problem free convective heat transfer.

From the literature survey, it has been observed that many investigators worked on the plate fin as well as perforated fins. They studied parameters affecting the heat transfer enhancement. But none emphasized the experimental investigation on wire mesh as heat sink because it requires a vast number of experiments which enormously increase the experimental cost and time. For many practical applications, it is necessary to determine the economic benefits for the heat transfer enhancement. Thus, the aim of this study is to minimize the experimental trials using Taguchi experimental design for determining the heat transfer characteristics of the wire mesh fin arrays as a heat sink, and to determine new design parameters and their levels.

II. Experimental Set up and Experimentation

The experimental set up is shown in Fig. 1. Fig. 1 consists of wooden box with insulation brick, heating coil, bottom base plate, upper base plate, wire mesh fin geometry, control panel, enclosure. A 450 watt electric heating coil was used as the heat source and was placed between the upper base plate and the bottom base plate of material mild steel. The upper base plate is 160mm x 120mm x 8mm in size. The slots are made on the surface of upper base plate to fix the wire mesh fin arrays. Slots are made according to the diameter of wire such as 1 mm, 1.2 mm, 1.6 mm at constant width and length of fin (160 mm x 120 mm). Heights of fins are 50 mm, 45 mm, and 40 mm.

To minimize heat loss from the base, the bottom and upper base plate was placed inside insulating firebrick of material alumina in size of 220 mm x 220 mm x 70 mm in such a way that only the upper surface of the base plate was exposed to the enclose environment. A total of 27 k type Copper-constantan thermocouples were located on different points on the base and wire mesh fin arrays to measure the temperatures. To calculate heat loss two thermocouples were fixed between the insulating bricks and bottom base plate. One thermocouple was used to measure the air temperature inside the enclosure. Four thermocouples were attached to the base plate to measure the temperature of upper base plate. Reaming twenty thermocouples were attached to the fin arrays at different places to measure the surfaces temperature of fins. All thermocouple wires were numbered to recognize the position of the connection. To control natural convection heat transfer, the enclosure is made outside of the whole assembly of test section to avoid disturbance from the outside environmental conditions. Removable acrylic sheet was placed on the front of the enclosure. On the bottom of the enclosure small holes were made to pass the air inside the enclosure. Upper side of the enclosure is opened to the atmosphere. Cold air is coming from bottom and passes out from the top of enclosure.

For achieving constant heat flux along the test section, a variac transformer was used to control the electric power input of the heating coil. The experiments were conducted at 40, 60, 80, 100 and 120 watts heat input. The digital temperature indicator was calibrated to $\pm$ 0.1 degree, having a range from 0 °C to 600 °C to measure the temperatures.

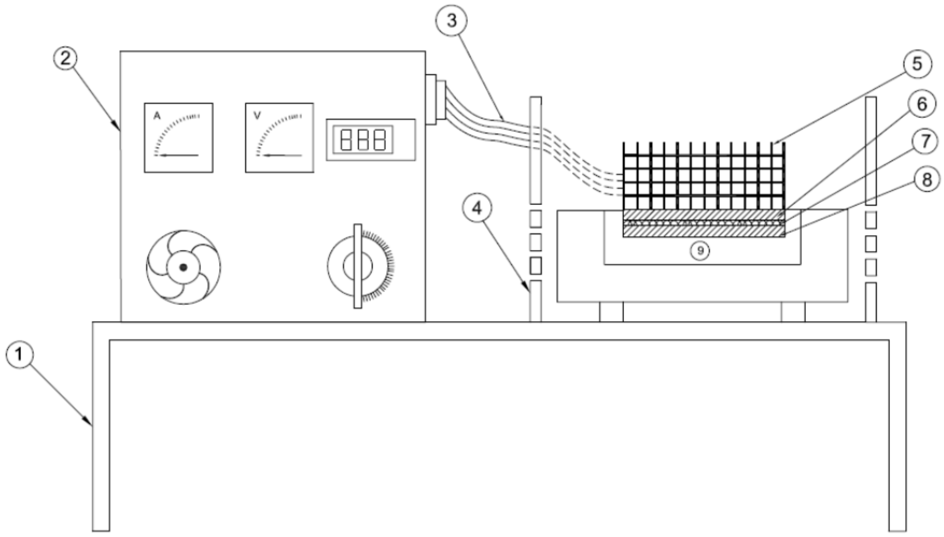

Fig. 1 Experimental set up: 1-Table, 2- Panel board with ammeter, voltmeter and temperature indicator, 3- Thermocouple wire, 4- Enclosure, 5- Wire mesh fin arrays, 6- Upper base plate,7- heating coil, 8 – Bottom plate, 9- Wooden box with insulated bricks

III. Data Processing

A. Heat transfer

The net heat transfer rate Q_N is the heat given to the flow by the forced convection at the steady state conditions and can be calculated by following energy balance equation:

$$Q_N = Q_{(electrical)} - Q_{(loss)} \tag{1}$$

$$Q_{(loss)} = Q_{(conduction\ side)} + Q_{(conduction\ bottom)} + Q_{(radiation)} \tag{2}$$

The electrical heat input is calculated from the electrical potential and current supplied to the surface. The conduction heat released from the bottom and side of the test section to the surrounding was calculated by using (3):

$$Q_{(conduction)} = -K_b A_b \frac{dt_b}{dx_b} \tag{3}$$

Also,

$$Q_N = Q_{(convection.)} \tag{4}$$

The steady state rate of convection heat transfer from the test section with fin arrays and without fin arrays can be calculated by using (5)

$$h_a = \frac{Q_N}{A_T\left(T_S - T_a\right)} \tag{5}$$

$$A_b = W_b + L_b \tag{6}$$

A_{fh} = Area of one horizontal wire (πdl) x Total number of horizontal wire (N_h); A_{fv} = Area of one vertical wire (πdl) x Total number of vertical wire (N_v); $A_f = (A_{fh} + A_{fv})$ x Number of fin arrays (N_f).

$$A_T = A_b + A_f \tag{7}$$

where W_b and L_b are the width and length of the base plate and N_f is the number of fins.

The dimensionless groups are calculated as follows:

$$Nu = \frac{h_{av} H}{K_a} \tag{8}$$

$$G_r = \frac{g\beta(\Delta T)H^3}{\upsilon^2} \tag{9}$$

$$P_r = \frac{\mu C_P}{K_a} \tag{10}$$

$$R_a = G_r x P_r \tag{11}$$

$$\Delta T = (Ts-Ta) \tag{12}$$

The values of thermo physical properties of air are taken at the bulk mean temperature,

$$T_m = \frac{T_S + T_a}{2} \tag{13}$$

B. Experimental Plan (Control Parameters and Orthogonal array)

Dr. Genichi Taguchi developed the Taguchi method for designing experiments which described how different parameters affect the mean and variance of process performance characteristics [22]. The Taguchi method gives the S/N ratio as the performance index to evaluate the characteristics of the product or process. It can be defined as the ratio of the mean (signal) to the standard deviation (noise). The S/N ratios may be dependent on the particular type of performance characteristics, including smaller-is-better, nominal-is-better, and larger-is-better. In the present study, the average heat transfer coefficient is a performance characteristic and it is larger-is-better *(Z_L)*.

$$z_L = -10\log\left(\frac{1}{n}\sum_{i=1}^{n}\frac{1}{Y_i^{\,2}}\right) \tag{14}$$

where η is the number of tests in trials, Y_i is the performance value of the i^{th} experiment.

In this study, wire mesh diameter, height of fin and spacing between two fin arrays are selected as control parameters with their levels, as shown in Table I. Geometry of fin arrays tested for different heat inputs was selected by Taguchi method Orthogonal array L_9 (3^3) as shown in Table II. This method is the most suitable for the optimum working conditions being investigated. According to the Taguchi method, an orthogonal array can provide an effective experimental performance with a minimum number of experimental trials. Each experiment is repeated three times under the same condition as per L_9 (3^3) Orthogonal array [22].

TABLE I
CONTROL PARAMETERS AND THEIR LEVELS

Symbol	Control parameter	Level I	Level II	Level III
A (D)	Wire mesh diameter	1 mm	1.2 mm	1.6 mm
B (H)	Height of fins	50 mm	45mm	40 mm
C (S)	Spacing between two fins	16.5mm	21.5mm	30mm

TABLE II
GEOMETRY OF FIN ARRAYS AS PER TAGUCHI METHOD ORTHOGONAL ARRAY L_9 (3^3)

Expt. no	A	Wire diameter D	B	Height of fin H	C	Fin Spacing S
1	1	1	1	50	1	16.5
2	1	1	2	45	2	21.5
3	1	1	3	40	3	30
4	2	1.2	1	45	2	21.5
5	2	1.2	2	40	3	30
6	2	1.2	3	50	1	16.5
7	3	1.6	1	40	3	30
8	3	1.6	2	50	1	16.5
9	3	1.6	3	45	2	21.5

B. Experimental Uncertainties

By using the estimation method of Moffat [21], the maximum uncertainties of the investigated non dimensional parameters are as follows: Average heat transfer coefficient = 2.2%, Nusselt number = 2.4%, Raleigh number = 3%. The individual contributions to the uncertainties of the non-dimensional parameters for each of the measured physical properties are summarized in Table III.

TABLE III
UNCERTAINTIES FOR THE RELEVANT VARIABLES

Variables	Uncertainty (%)
Electric heat input	0.98
Mean temperature	0.42
Thermal conductivity	0.118
Dynamic viscosity of air	0.210
Density of air	0.252

IV. RESULTS AND DISCUSSION

A. Analysis of Signal to Noise Ratio (S/N)

The average heat transfer coefficient was calculated via the experimental design for each combination of the control parameters by using (5). The rule: Larger-is-better is used for calculating the S/N ratio using (13). The average Heat transfer coefficient and S/N of each combination are given in Table IV.

TABLE IV
RESULT OF ORTHOGONAL ARRAY L_9 (3^3) AND S/N RATIO FOR AVERAGE HEAT TRANSFER COEFFICIENT

Expt. trial	D	H	S	ha	S/N
1	1	1	1	20.55	26.26
2	1	2	2	19.05	25.60
3	1	3	3	18.03	25.12
4	2	1	2	19.94	25.99
5	2	2	3	18.53	25.06
6	2	3	1	17.50	24.86
7	3	1	3	18.27	25.23
8	3	2	1	17.50	24.86
9	3	3	2	16.01	24.14

The average S/N value for heat transfer coefficient is calculated for each parameter at different levels. The main effect of each parameter is nothing but difference between high (S/N) ratio value and low (S/N) ratio values. $\Delta R = Max\frac{S}{N} - Min\frac{S}{N}$ of each parameter. The larger the Rank value for a parameter, the larger the effect the variable has on the process as shown in Table V.

TABLE V
RESULT OF S/N RESPONSE TABLE FOR MAXIMUM AVERAGE HEAT TRANSFER COEFFICIENT

Level	Wire mesh dia.	Height of the fin	Spacing between two fins
I	25.66[a]	25.83[a]	25.33[a]
II	25.30	25.17	25.24
III	24.74	24.71	25.14
ΔR	0.92	1.12	0.19
Rank	II	I	III

[a] Optimum level

B. Optimization of Results

Fig. 2 shows the degree of influence of the parameters on the S/N ratio. The numerical value of the maximum point in each graph shows the best value of that particular parameter. The most effective parameter to enhance the heat transfer rate is the wire mesh diameter, height of fin and least effective is spacing between two fins. The design parameter combination for an average heat transfer coefficient is A1B1C1 and the corresponding values of each parameter are: A1 i.e. wire mesh diameter = 1mm, B1 i.e., height of fin = 50 mm and C1 i.e. 16.5 mm. spacing between two fins.

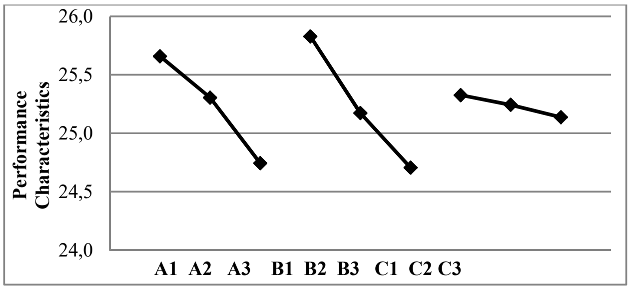

Fig. 2 The effect of each parameter on to enhances the heat transfer

C. Analysis of Variance (ANOVA)

In this study ANOVA is used to analyze the effects of wire mesh diameter, spacing between two fins, and height of fin. ANOVA is the statistical method employed to determine individual interactions of all control parameters. In the analysis, the percentage contribution of each control parameter is used to measure the corresponding effects on the performance characteristics. Table VI summarizes the ANOVA results for a maximum heat transfer coefficient of fin arrays and shows percentage contribution and variance of parameters A, B, and C characteristics. Fig. 3 shows the percentage contribution of each parameter, i.e. A, B and C. The results indicate that the optimal design parameter combination is B1A1C1 and corresponding percentage of each parameters for maximum average heat transfer coefficient is height of fin = 59.21%, wire mesh diameter = 38.62% and height of fin = 2.16%

TABLE VI
ANOVA RESULT OF AVERAGE HEAT TRANSFER COEFFICIENT WIRE MESH FIN ARRAYS

Parameters	SS	DOF	MS	F-actual	%
A	5.6069	2	2.80347	37.608	38.62
B	8.5961	2	4.29807	57.6579	59.26
C	0.3140	2	0.15701	2.10627	2.16
Error	0.1490	2	0.2391	0.03727	-
Total	14.666	8	-	-	100

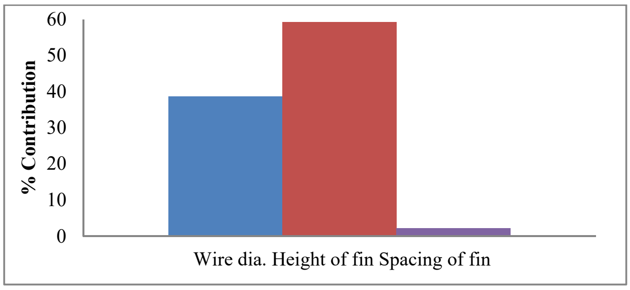

Fig. 3 Percentage contribution of each parameter to enhance the heat transfer

D. Confirmation Test

The final step of the Taguchi method is performing the confirmation experiments for examining the performance characteristics. The purpose of the confirmation experiment is to validate the conclusion drawn during the analysis phase. This analysis is carried out for a level of significance of 5%, i.e. for a 95% level of confidence. The confidence interval (CI) for the predicated optimal values of average Nusselt number is calculated by using (15):

$$CI=\sqrt{F_{aV_1V_2}V_{ep}\left(\frac{1}{\eta_{eff}}+\frac{1}{r}\right)} \tag{15}$$

where, $F_{aV_1V_2}$ is the F-ratio required for α = 0.05 with a confidence of 95%, V_1V_2 are the number of degree of freedom of the mean and number of degree of freedom of error respectively, V_{ep} is the error of variance, r is the number of repetitions in confirmation experiments. And η_{eff} is the number of effective measured results i.e.

$$\eta_{eff} = \frac{N}{1 + DOF_{opt}} \tag{16}$$

where N is the total number of trials and DOF_{opt} is total degree of freedom that are associated with items used to estimate $\eta_{\text{efficiency}}$. Using optimum levels arrived by the Taguchi method of optimization, a full range experiment was conducted. For the A1B1C1 combinations, the values of average heat transfer coefficient are Optimum predicated = 20.55 W/m^{2}k and Optimum CI = 19.02 to 22.16 W/m^{2}k.

E. Heat Transfer Analysis

The optimal design parameters are the combination of B1, A1 and C1. The effective parameters enhancing the heat transfer are given as height of fin (B1), wire mesh diameter (A1), and the spacing between two fins (C1). Hence, the experiments conducted are for different heat input 40, 60, 80, 100, 120 watt for 1, 1.2 mm and 1.6 mm wire mesh diameter, keeping 50 mm height of fin and 16.5 mm spacing between two fins. Fig. 4 shows the variation of the heat transfer coefficient with ΔT for different wire mesh diameter. Increase in the average heat transfer coefficient of different wire mesh fin arrays depends on the increase in difference of the surfaces temperature of fin arrays and enclosure temperature (surrounding). Also, it is observed that the highest average heat transfer coefficient is observed at the 1 mm wire mesh diameter.

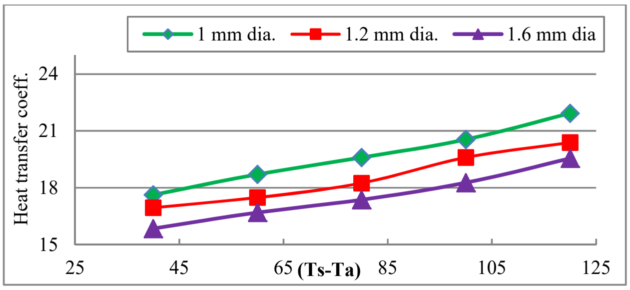

Fig. 4 Variation of h_a with ΔT for different diameters

Fig. 5 shows the variation of N_{uf}/N_{uwf} with different heat inputs for the different wire mesh diameter keeping 50 mm height of fin and 16.5 mm distance between two fins. It can be seen from this figure that the ratio of Nusselt number with fins and Nusselt number without fins N_{uf}/N_{uwf} decreases with increase in difference between fins and surrounding temperature; after some temperature difference it is constant. Also N_{uf}/N_{uwf} ratio is more in 1 mm wire diameter fin arrays.

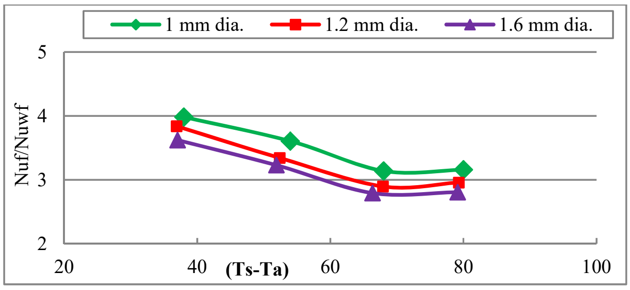

Fig. 5 Variation of ratio N_{uf}/N_{uwf} with ΔT for different diameters

In evaluating the effectiveness of the fin, it is required to determine fin performance for different wire mesh diameters Effectiveness of the fin can be determined by the ratio of the actual heat transfer rate from the fin (Qf) having the convection heat transfer coefficient and average surface temperature; to the heat transfer rate that would exist without the fin (Qsd) at the base temperature.

$$Effectiveness\ of\ fin(\varepsilon_f) = \frac{Q_f}{Q_{sd}} \tag{17}$$

The variation of fin effectiveness with difference between surface temper of fins and enclosure temperature (surrounding) is plotted in Fig. 6. The effectiveness of the fin decreases with an increase in diameter of wire mesh, the effectiveness of all the wire mesh diameter of fins decreases with increase in ΔT. Also, it is observed that the higher fin effectiveness is observed at the highest for 1 mm wire mesh dia and H= 50 mm and S= 16.5 mm. For net energy gain, the value of enhancement efficiency must be greater than unity [13]. In the present study, the effectiveness of fins is more than one; this means that the use of newly designed fins is advantageous as far as heat transfer enhancement is concerned.

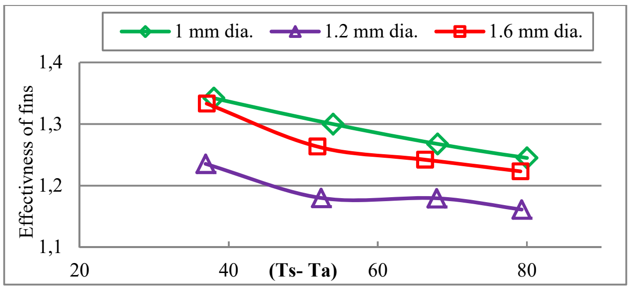

Fig. 6 Variation of effectiveness of fin with ΔT for different wire mesh diameter

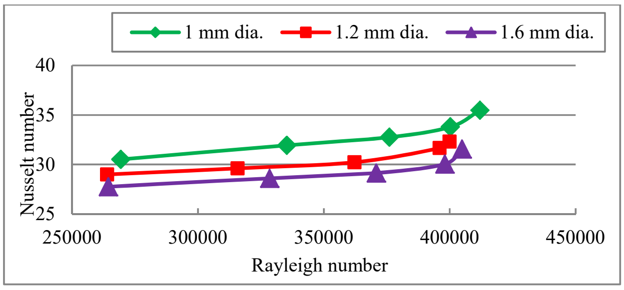

Fig. 7 Variation of Nusselt number with Rayleigh number for different wire mesh diameter

Fig. 7 is the graph between Nusselt number and Rayleigh number for different wire mesh diameters. It is observed that Nusselt number increases with increase in Rayleigh number.

VI. Conclusion

In this study, the heat transfer enhancement and the effect of the various design parameters on the heat transfer for different wire mesh diameter fin arrays by natural convection were investigated experimentally.

- From the experimental trials mentioned under Taguchi L_9 (3^3) orthogonal arrays, it was found that the height of fin is the most prominent parameter affecting the heat transfer coefficient; secondly wire mesh diameter and then spacing between two fins. The maximum heat transfer rate is observed for 1 mm wire mesh diameter and 50 mm height of fin and 16.5 mm spacing between fins. Hence, it can be concluded that the heat transfer can be successfully improved by controlling these parameters.
- Heat transfer coefficient increases with increase in difference between average surface temperature of the fin arrays and surrounding temperature (Ts-Ta). The average Nusselt numbers of all the wire mesh diameters increases with increase in Rayleigh number. The Nusselt numbers of 1 mm wire diameter fin arrays are higher than others fin arrays. The fin effectiveness ε_f of wire mesh fin arrays is greater than 1 for all wire mesh diameters. The maximum effectiveness is observed at the 1 mm wire mesh diameter, 50 mm height of fin and 16.5 mm spacing between two fins.
- Utilization of extended surfaces (wire mesh fin arrays) increases the heat dissipation rate. Hence it may be utilized for many industrial applications.

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Dr. Kavita Hanamant Dhanawade has completed Ph. D in Mechanical Engineering from University of Mumbai in Jan 2016. M E Mechanical Engineering from Walchand College of Engineering Sangli in sept. 2004 and B E Mechanical Engineering from Rajarambapu college of Engineering Islampur in 1995.She has total 20 years as UGC approved teaching experience. Around 10 research papers have been published in International Journals and approximately 30 citations have got for research paper. She has got three times Minor Research Grant from University of Mumbai. Total 12 Short Term Training Programs, Seminars and Workshops have been attended. Research area is the heat transfer, Refrigeration and air conditioning. She is the life member of the ISTE and ISHRAE

Dr. Hanamant S. Dhanawade has completed PhD from JJT University, Rajasthan. He is dean of academic of Smt. Indira Gandhi College of Engineering, Navi Mumbai. He has 28 years of teaching experience and more than fifteen research papers in the national, international conferences and Journals in his credit. Life member of ISTE and Associate Member of IIIE.