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Optimization of Convective Heat Transfer Model of Cold Storage with Cylindrical Pin Finned Evaporator Using Taguchi Analysis

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ABSTRACT
This work contains of design of experiments to optimize the various control factors of a cold storage evaporator space inside the cold room, in other words the heat absorption by evaporator will be maximize to minimize the use of electrical energy to run the system. Here we have use cylindrical pin fin to maximize the heat absorption by evaporator. Taguchi orthogonal array have been used as a design of experiments. Three control factors with three levels of each have been chosen for analysis. In the evaporator space the heat absorbs by the evaporator and fins totally a convective heat transfer process. The control factors are Area of the evaporator with cylindrical pin fin(A), temperature difference of the evaporator space (dT), and relative humidity inside the cold room(RH). Different amount of heat gains in the cold room for different set of test runs have been taken as the output parameter. The objective of this work is to find out the optimum setting of the control factors or design parameters so as the heat absorb in the cold room by the evaporator will be maximum. The Taguchi regression analysis have been used as an optimization technique.

Keywords: Taguchi orthogonal array, convective heat transfer co-efficient, cylindrical pin fin area and arrangement, regression analysis, graphical representation of control factors with heat transfer.

I. INTRODUCTION

Cold storages form the most important element for proper storage and distribution of vide variety of perishables like fruits, vegetables and fish or meat processing. India is the largest producer of fruits and second largest producer of vegetables in the world. In spite of that per capita availability of fruits and vegetables is quite low because of post harvest losses that account for about 25 to 30% of production. Besides, quality of a sizable quantity of products also deteriorates by the time it reaches the consumer. As India is the second largest producer (45,343,600 tonnes at 2015) of potato after China and largest producer of the ginger (702000 metric tonnes i.e. 34.6% of the world total) without these there are many kind of food commodities are produce in our country so demand for cold storages have been increasing rapidly over the past couple of decades so that food commodities can be uniformly supplied all through the year and food items are prevented from perishing. Besides the role of stabilizing market prices and evenly distributing both on demand basis and time basis, the cold storage industry provide other advantages and benefits to both the farmers and the consumers. The farmers get the opportunity to get a good return of their hard work. On the consumer sides they get the perishable commodities with lower fluctuation of price. Very little theoretical and experimental studies are being reported in the journal on the performance enhancement of cold storage. Energy crisis is one of the most important problems the world is facing nowadays. With the increase of cost of electrical energy operating cost of cold storage storing is increasing which forces the increased cost price of the commodities that are kept. So it is very important to make cold storage energy efficient or in the other words reduce its energy consumption. Thus the storage cost will eventually come down. In case of conduction we have to minimize the leakage of heat through wall but in convection maximum heat should be absorbed by refrigerant to create cooling uniformity thought out the evaporator space. If the desirable heat is not absorbed by tube or pipe refrigerant then temp of the refrigerated space will be increased, which not only hamper the quality of the product which has been stored there but reduces the overall performance of the plant. That’s why a mathematical modelling is absolutely necessary to predict the performance. In this paper we have proposed a theoretical heat transfer model of convective heat transfer model development of a cold storage using Taguchi L9 orthogonal array. Area of the evaporator with fin (A), Temperature difference (dT), Relative Humidity (RH)are the basic variables and three ranges are taken each of them in the model development. A graphical interpretation from the model justifies the reality.

II. TAGUCHI METHODOLOGY

Our main objective in this project work is to optimize the different parameters so that the heat gain in the cold room will be minimum. It involves examining various combinations of input parameters and their effects on the response. In order to achieve this we need a proper strategy before actually going into test runs. Design of Experiments (D.O.E) serves this purpose. The basic definition of D.O.E is that it is a systematic, rigorous approach to engineering problem-solving that applies principles and techniques at the data collection stage so as to ensure the generation of valid, defensible, and supportable engineering conclusions.

Taguchi method is based on performing evaluation or experiments to test the sensitivity of a set of response variables to a set of control parameters (or independent variables) by considering experiments in "orthogonal array" with an aim to attain the optimum setting of the control parameters.

Classical experimental design methods are too complex and are not easy to use. To solve this problem, the Taguchi method uses a special design of orthogonal arrays to study the entire parameter space with only a small number of experiments. Three parameters which control the heat absorption are considered as controlling factors. They are area, temperature difference and relative humidity. Each parameter has three levels – namely low, medium and high, denoted by 1, 2 and 3 respectively. According to the Taguchi method, if three parameters and 3 levels for each parameters L9 orthogonal array should be employed for the experimentation.

Table no. 1 shows the control factors and their levels considered for the experimentation

Experiment Number	Column 1	2	3	4
1	1	1	1	1
2	1	2	2	2
3	1	3	3	3
4	2	1	2	3
5	2	2	3	1
6	2	3	1	2
7	3	1	3	2
8	3	2	1	3
9	3	3	2	1

Fig.1 L9 Orthogonal Array

Experiment Number	Column 1	2	3	4	5	6	7	8
1	1	1	1	1	1	1	1	1
2	1	1	2	2	2	2	2	2
3	1	1	3	3	3	3	3	3
4	1	2	1	1	2	2	3	3
5	1	2	2	2	3	3	1	1
6	1	2	3	3	1	1	2	2
7	1	3	1	2	1	3	2	3
8	1	3	2	3	2	1	3	1
9	1	3	3	1	3	2	1	2
10	2	1	1	3	3	2	2	1
11	2	1	2	1	1	3	3	2
12	2	1	3	2	2	1	1	3
13	2	2	1	2	3	1	3	2
14	2	2	2	3	1	2	1	3
15	2	2	3	1	2	3	2	1
16	2	3	1	3	2	3	1	2
17	2	3	2	1	3	1	2	3
18	2	3	3	2	1	2	3	1

Fig.2 L18 Orthogonal Array

III. MODEL DEVELOPMENT

1. Range And Parameter Selection

The length, breath and heightof each chamber of cold storage are 87.5m,34.15m and 16.77m respectively.

The three values of area of the evaporator with fin (A) of evaporator space are 8.253m^2, 10.314m^2 and 14.628m^2 respectively. The three values of temperature difference (dT) of evaporator space are 2, 5 & 8 centigrade respectively. The three values of relative humidity (RH) of evaporative space are 0.85, 0.90 & 0.95 respectively.

Table no. 1 Control factors with their range

Notation	Factors	Unit	Levels		
			1	2	3
A	Area of the evaporator with fin	m^2	8.253	10.314	14.628
dT	Temperature Difference	$^{\circ}$C	2	5	8
RH	Relative Humidity	%	0.85	0.90	0.95

In this study, Mohitnagar cold storage (Jalpaiguri) & Teesta cold storage has been taken as a model of observation.

2. Regression Analysis

Regression analysis is the relationship between various variables. Using regression analysis one can construct a relationship between response variable and predictor variable. It demonstrates what will be the changes in response variable because of change in predictor variable. Simple regression equation is,

y=a+ b x

In this problem more than one predictor variable is involved and hence simple regression analysis cannot be used. We have to take the help of multiple regression analysis. There are two types of multiple regression analysis- i) Simple multiple regression analysis (regression equation of first order) ii) Polynomial multiple regression analysis (regression equation of second order or more). Simple multiple regression analysis is represented by the equation of first order regression

$Y=\beta_0+\beta_1X_1+\beta_2X_2+\beta_3X_3+$…………………..+ε ………………………………….(i)

Where β is constant terms & X is the variables & ε is the experimental error.

Polynomial multiple regression analysis equation is

$Y=\beta_0+\beta_1X_1+\beta_2X_2+\beta_3X_3+\beta_{11}X^2_{11}+\beta_{22}X^2_{22}+\beta_{33}X^2_{33}+\beta_{12}X_1X_2+\beta_{13}X_1X_3+\beta_{23}X_2X_3$ ……(ii)

The above equation is second order polynomial equation for 3 variables. Where β are constant, X_1, X_2,X_3 are the linear terms, X_{12} X_{13} X_{23} are the interaction terms between the factors, and lastly X_{11} X_{22} X_{33} are the square terms.

$Q_{\text{(heat due to convection)}}$ = response variable & A, dT, RH= predictor variable.

Polynomial regression equation becomes after replacing real problem variables

$Q_{\text{(heat due to convection)}} = \beta_0+\beta_1(A)+\beta_2(dT)+\beta_3(RH)+\beta_{11}(A)*(A)+\beta_{22}(dT)*(dT)+\beta_{33}(RH)*(RH)+\beta_{12}(A)*(dT)+\beta_{13}(A)*(RH)+\beta_{23}(dT)*(RH)$ …………..(1)

To solve this equation following matrix method is used

Q= [β][X]………………..(2)

$[\beta]=[Q]\ [X^{-1}]$ where [β] is the coefficient matrix, Y is the response variable matrix; $[X^{-1}]$ is the inverse of predictor variable matrix……………….(3)

In this problem there are 3 independent variables and each variable has 3 levels and hence from the Taguchi Orthogonal Array (OA) table L9 OA is best selected.

Table no- 2 Taguchi's L9 Orthogonal Array

	Factorial combination		
Sl. No.	**A**	**dT**	**RH**
1	1	1	1
2	1	2	2
3	1	3	3
4	2	1	2
5	2	2	1
6	2	3	3
7	3	1	3
8	3	2	1
9	3	3	2

Experiments have been carried out using Taguchi's L9 Orthogonal array experimental design which consists of 9 combinations of area, relative humidity and temperature difference. It considers three process parameters to be varied three discrete levels.

In the equation (1) number of unknown constant (i.e. β) is ten and in using Taguchi's L9 Orthogonal array experimental design which consists of 9 combinations of area, relative humidity and temperature difference so to form X as a 10*10 matrix, β as a 10*1 matrix and Q as a 10*1 matrix we have taken a combination of area, relative humidity and temperature difference which does not match with any of combination of the L9 orthogonal array mentioned above.

As we are working on three level value of the factors and the first column of L18 orthogonal is of 2 level so have omit this and took a combination of next three column and we have chosen a combination which which does not match with any of combination of the L9 orthogonal array mentioned above. So the desired combination is,

Table no- 3

	Factorial combination		
Sl. No.	**A**	**dT**	**RH**
1	1	1	1
2	1	2	2
3	1	3	3
4	2	1	2
5	2	2	1
6	2	3	3
7	3	1	3
8	3	2	1
9	3	3	2
10	3	1	2

In the equation (2) X is a 10*10 matrix, β is a 10*1 matrix and Q is a 10*1 matrix. By using the L9 Orthogonal array and its iteration terms we have to find out the betas(β) values.

Now there are nine test runs, so there will be nine values of Q and also nine equations. These equations are-

$Q_1=\beta_0+8.253\beta_1+2\beta_2+.85\beta_3+68.1120\beta_4+4\beta_5+0.7225\beta_6+16.506\beta_7+7.0151\beta_8+1.7\beta_9$

$Q_2=\beta_0+8.253\beta_1+5\beta_2+.90\beta_3+68.1120\beta_4+25\beta_5+.81\beta_6+41.265\beta_7+7.4277\beta_8+4.75\beta_9$

$Q_3=\beta_0+8.253\beta_1+8\beta_2+.95\beta_3+68.1120\beta_4+64\beta_5+.9025\beta_6+66.024\beta_7+7.8404\beta_8+7.6\beta_9$

$Q_4=\beta_0+10.314\beta_1+2\beta_2+.90\beta_3+106.3786\beta_4+4\beta_5+.81\beta_6+20.628\beta_7+9.2826\beta_8+1.8\beta_9$

$Q_5=\beta_0+10.314\beta_1+5\beta_2+.95\beta_3+106.3786\beta_4+25\beta_5+.9025\beta_6+51.57\beta_7+9.7983\beta_8+4.75\beta_9$

$Q_6=\beta_0+10.314\beta_1+8\beta_2+.85\beta_3+106.3786\beta_4+64\beta_5+.7225\beta_6+82.512\beta_7+8.7669\beta_8+6.8\beta_9$

$Q_7=\beta_0+14.628\beta_1+2\beta_2+.95\beta_3+213.9784\beta_4+4\beta_5+.9025\beta_6+29.256\beta_7+13.8966\beta_8+1.9\beta_9$

$Q_8=\beta_0+14.628\beta 1+5\beta_2+.85\beta_3+213.9784\beta_4+25\beta_5+.7225\beta_6+73.14\beta_7+12.4338\beta_8+4.25\beta_9$

$Q_9=\beta_0+14.628\beta_1+8\beta_2+.90\beta_3+213.9784\beta_4+64\beta_5+.81\beta_6+117.024\beta_7+13.1652\beta_8+7.2\beta_9$

$Q_{10}=\beta_0+14.628\beta_1+2\beta_2+.90\beta_3+213.9784\beta_4+4\beta_5+.81\beta_6+29.256\beta_7+13.1652\beta_8+1.8\beta_9$

The values of A, dT, RH, A^2, dT^2, RH^2, A*dT, A*RH, dT*RH can be found out by the following table-

Table no: 4 Observation table with square and interaction terms-

Test runs		A	dT	RH	A*A	dT*dT	RH*RH	A* dT	A*RH	dT*RH
1	1	8.253	2	0.85	68.1120	4	0.7225	16.506	7.0151	1.7
2	1	8.253	5	0.90	68.1120	25	0.81	41.265	7.4277	4.5
3	1	8.253	8	0.95	68.1120	64	0.9025	66.024	7.8404	7.6
4	1	10.314	2	0.90	106.3786	4	0.81	20.628	9.2826	1.8
5	1	10.314	5	0.95	106.3786	25	0.9025	51.57	9.7983	4.75
6	1	10.314	8	0.85	106.3786	64	0.7225	82.512	8.7669	6.8
7	1	14.628	2	0.95	213.9784	4	0.9025	29.256	13.8966	1.9
8	1	14.628	5	0.85	213.9784	25	0.7225	73.14	12.4338	4.25
9	1	14.628	8	0.90	213.9784	64	0.81	117.024	13.1652	7.2
10	1	14.628	2	0.90	213.9784	4	0.81	29.256	13.1652	1.8
	C1	C2	C3	C4	C5	C6	C7	C8	C9	C10

3. Heat Calculation

In this study heat transfer from evaporating space to refrigerant (which are in tube or pipe) only being considered. The transfer heat evaporating space to refrigerant are calculated in terms of Area of the evaporator with fin(**A**), temperature difference (**dT**) & relative humidity RH) . Only convection heat transfer effect is being considered in this study.

Basic equation for heat transfer

$Q_T = Q_{conv} + Q_{condensation}$.

$Q_{conv}=Ah_c dT$ & $Q_{condensation}=Ah_{m(}RH)h_{fg.}$ [3]

Here Q_{conv}=heat transfer due to convection &$Q_{condensation}$=heat transfer due to condensation & Q_T=Total heat transfer or absorb heat into refrigerant.

$Q_T= Ah_c dT + Ah_{m(}RH)h_{fg}$

[As we know,

$h_c/h_m = C_p(Le)^{2/3}$

or, $h_c/h_m = 1.005*(1)^{2/3}$

or, $h_c = h_m$]

or, $Q_T= [Ah_c dT] + [(h_c/1.005).A.RH.h_{fg}]$

or, $Q_T=Ah_c[dT+(RH.h_{fg})/1.005]$

or, $Q_T= A.h_c(dT + RH.h_{fg})$

The heat transfer equation due to area of the evaporator with fin (**A**), temperature difference (**dT**) & relative humidity RH) is **$Q_T=Ah_c$ (dT + 2490 RH)……………….**(4)[8]

Here, **A**=surface area of tubes in evaporator with fin

.**h_c**=convective heat transfer co-efficient.

h_m=convective mass transfer co-efficient,

h_{fg}=latent heat of condensation of moisture 2490 KJ/Kg-K.

C_p=specific heat of air 1.005 KJ/Kg-K.

Le=Lewis number for air it is one.

Now we calcutate the value of convective heat transfer co-efficient (h_c),

We know,

$$\mathbf{Nu} = \frac{\textit{Convective heat transfer}}{\textit{Conductive heat transfer}} = \mathbf{(h_c * L) / k}$$

Where:

Nu = Nusselt number

hc = convective heat transfer coefficient

k = thermal conductivity, W/mK

L = characteristic length, m

The convection heat transfer coefficient is then defined as following:

$$\mathbf{h_c} = \frac{\mathbf{Nu*K}}{\mathbf{L}} \quad \text{......................................(5) [2]}$$

The Nusselt number depends on the geometrical shape of the heat sink and on the air flow. For natural convection on flat isothermal plate the formula is given in table

Table no- 5

Nusselt number formula.

	Vertical fins		Horizontal fins
Laminar flow	Nu = 0.59 * $Ra^{0.25}$	Upward laminar flow	Nu = 0.54 * $Ra^{0.25}$
Turbulent flow	Nu = 0.14 * $Ra^{0.33}$	Downward laminar flow	Nu = 0.27 * $Ra^{0.25}$
		Turbulent flow	Nu = 0.14 * $Ra^{0.33}$

Where:

Ra = Gr * Pr

The **Rayleigh number(Ra)** defined in terms of **Prandtl number (Pr)** and **Grashof number (Gr).**
If , Ra < $10^9$the heat flow is laminar,
while Ra > 106the flow is turbulent.

Grashof number (Gr):

Gr = $\frac{g*L^3*\alpha*(Ta-Tp)}{\eta^2}$ [for natural convective heat transfer from a cold body]

Where:

- g = acceleration of gravity = 9.81, m/s2
- L = longer side of the fin =30 foot =9.144 m
- α= air thermal expansion coefficient. For gases, is the reciprocal of the temperature in Kelvin:

$\alpha = \frac{1}{Ta}$, 1/K = (1/275.15) K

- T_p = Plate temperature, = 272.15 K
- T_a = Air temperature= 275.15 K
- η = air kinematic viscosity = 13.39 * 10^{-6} m^2/s [at air temp.=275.15 K & air pressure= 1 bar]

$$Gr = \frac{9.81*(9.144)^3*\left(\frac{1}{275.15}\right)*(275.15-272.15)}{(13.39*10^{-6})^2}$$

or, Gr = 4.56*10^{11}

Prandtl number (Pr):

Pr = $\frac{\mu*cp}{K}$

Where:

- μ = air dynamic viscosity, is 1.725*10^{-5} kg/m.s at 275.15 K
- cp = air specific heat = 1005 J/(Kg*K) for dry air
- k = air thermal conductivity = 0.0244 W/(m*K) at 275.15 K

Pr = $\frac{1.725*10^{-5}*1005}{0.0244}$

Or**,** Pr = 0.711

So,

Ra = Gr * Pr

Ra = 4.56*10^{11} * 0.711

Ra = 3.24*10^{11}

As, Ra > 10^9 = Turbulent flow

So, Nustle Number for turbulent flow,

Nu = 0.14 * $Ra^{0.33}$

Nu= 0.14 *$(3.24*10^{11})^{0.33}$

Nu= 880.25

So**,** Convective Heat Transfer co-efficient(h_c) :-

$h_c = \frac{880.25*0.0244}{9.144}$

or, **h_c= 2.35**

So, The final Heat Transfer equation when we replace the h_c in equation (1) we get,

Q_T = 2.35*A(dT + 2490 RH) (6)

4. Cylindrical Pin-Fin

The configuration of the pin is shown in Figure 3. The cross section is a 5 mm circle. This diameter was considered as a reference length scale.

If we consider "H" as the height of the cylinder, the surface area can easily calculated from the following formulas:

Surface Area = Areas of top and bottom +Area of the side

Surface Area = 2(Area of top) + (perimeter of top)* height

Surface Area of cylindrical pin fin = [2π(D^2/4) + (π*D) H]

The calculated surface area was kept constant for all different fin morphologies. This ensured that the contact surface areas between fluid and fins were equal in all cases and the effect of fin morphology could be studied more easily. Also the height of the pin (H) were kept constant for the rectangular pin-fin too. This was impossible to do for the drop-shaped pin-fin due to practical matters.

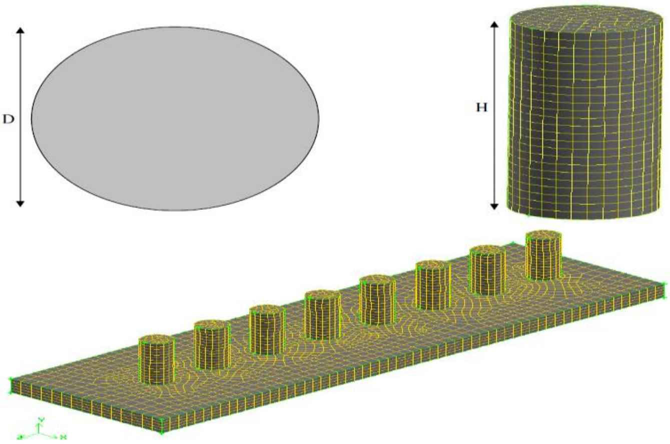

Fig.3 Configuration of Cylindrical pin fin

We know,
A_b = Area of bare tube
A_f = Area of cylindrical pin fin

$$A = A_b + A_f$$
$$A_b = \pi * r^2 * L$$
$$A_f = [2\pi(D^2/4) + (\pi * D)\, H] * n * N$$

Where,

r = Radius of bare tube = 0.038 m
L = Length of bare tube = 1 m
D = Diameter of cylindrical Pin fin = 0.005 m
H = Height of cylindrical Pin fin = 0.02 m
n = Number of bare tube = 1
N = Number of cylindrical Pin fin

Chain ordering Pin fin arrangement:

Fig.4 Arrangement of Pin fin

Where,
D = Diameter of fin
S = Longitudal Fin Spacing (S=3D)
S_b = Breadth wise fin spacing (S=2.5D)
m = Margin (m=1.5D)

Now, From equation number (6) we have $Q_{1......10}$ values by putting the values of Area of the evaporator with fin (A) from the equation of Area(A), Relative humidity (RH) and temperature difference (dT).

Table no: 6 Response variable matrix –

Q
41087.35
43560.16
46032.96
54365.61
57455.94
51493.42
81384.71
72928.26
77311.17
77104.92

Now using the equation number (3) we get the coefficient matrix [β]-

We use MATLAB software to find out the coefficient matrix [β],

Table no: 7 Regression coefficient table

B_0	-36.0811
B_1	0.1920
B_2	0.7257
B_3	73.8010
B_4	-0.0072
B_5	-0.0180
B_6	-39.0099
B_7	2.35
B_8	5851.4863
B_9	-0.6871

After putting the values of regression coefficient in equation no- (1)The final regression equation becomes-

$$\mathbf{Q_{(heat\ due\ to\ convection\ heat\ transfer)} = -36.0811 + 0.1920(A) + 0.7257(dT) + 73.8010(RH) - 0.0072(A)(A) - 0.0108(dT)(dT) - 39.0099(RH)(RH) + 2.35(A)(dT) + 5851.4863(A)(RH) - 0.6871(dT)(RH)}$$

This is the proposed regression equation. It establishes the relationship between the response variable Q and the predictor variables i.e. control variables A, dT, and RH. By the help of above equation the heat transfer evaporator space to evaporator coil can easily be computed. With the help of the multiple regression equation a computer program has been generated to check the effect of variations of control parameters on output variable. With the help of data sets generated by the program various graphs are drawn.

IV. RESULTS AND DISCUSSIONS

The regression equation developed is simulated in computer to find the effects of the control parameters on heat transfer rate within the considered ranges. This is done by varying one parameter at a time within its range and keeping other parameters constant at their mid-level values.

1. Effect Of Area Of The Evaporator On Heat Transfer Rate –

The values of dT and RH remain constant. Enter the value for A within the range 8.253 to 14.253, Start with 8.253 keeping dT=5 and RH=0.90 and incrementing the value of A by 0.5, the corresponding values of Q are Figure 5: shows that heat absorption increase with area of the evaporator increases. By using the above computer program data set the variation of A with Q can be graphically represented as-

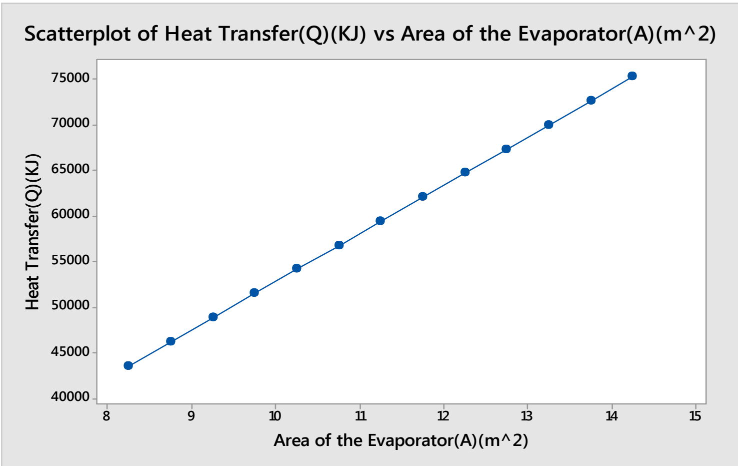

Figure 5: variation of heat transfer with area of the evaporator.

2. Effect Of Temperature Difference On Heat Transfer Rate-

The values of A and RH remain constant. Enter the value for dT within the range 2 to 8, Start with 2.0, Keeping A=10.314 and RH=0.90 and incrementing the value of dT by 0.5, the corresponding values of Q are found by using a computer program data set the variation of dT with Q can be graphically represented as-

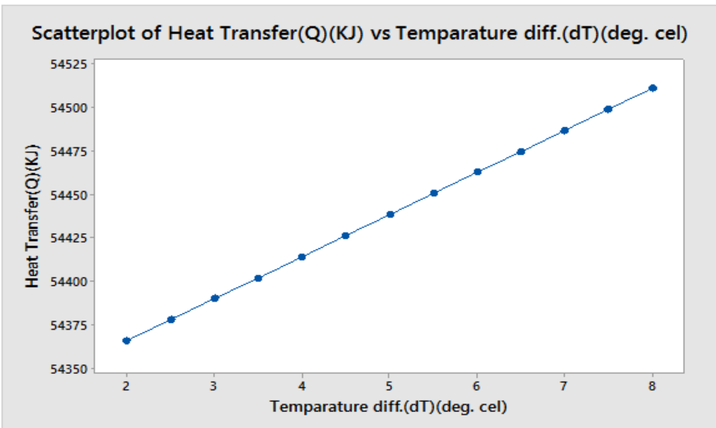

Figure 6: variation of heat transfer with temperature difference.

Figure 6: shows that heat absorption increase with temperature difference increase and at lower temperature difference it is more affected than the higher temperature difference.

3. Effect Of Relative Humidity On Heat Transfer Rate-

First, the values of A and dT remain constant. Enter the value for RH within the range 0.85 to 0.95, Start with 0.85 Keeping A=8.253 and dT=5 and incrementing the value of RH by 0.005, the corresponding values of Q are, By using a computer program data set the variation of RH with Q can be graphically represented as-

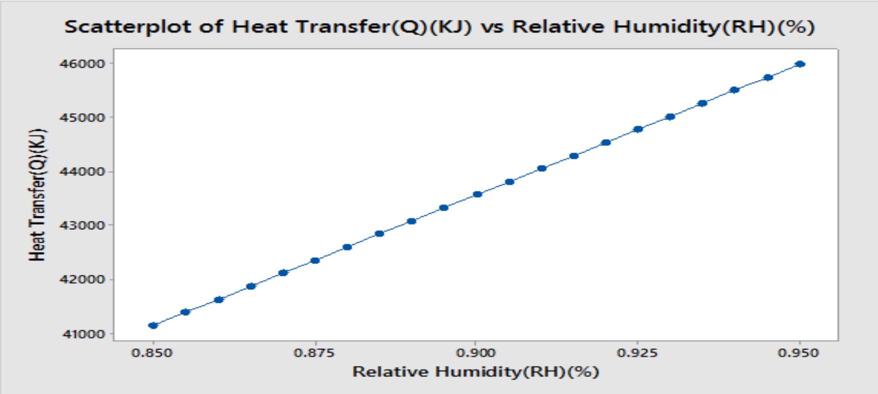

Figure 7: variation of heat transfer with relative humidity.

V. CONCLUSION

The present study discusses about the application of Taguchi methodology to investigate the effect of control parameters on heat gain (Q) in the cold room, and also to propose a mathematical model with the help of regression analysis. From the analysis of the results obtained following conclusion can be drawn-

1. From the graphical analysis the optimum heat absorption by the evaporator when area is 14.628 m^2, temparature difference(dT) is 5 ^0C and relative humidity (RH) is 0.95. This optimality has been proposed out of the range of [A (8.253, 10.314, 14.628), dT (2, 5, 8), RH (0.85, 0 .90, 0.95)].
2. The proposed regression equation establishes the relationship between the response variable heat transfer Q and the predictor variables i.e. control variables A, dT, and RH.
3. Heat absorption capability of pin fin is better compared to other fins. So it is necessary use in cold storage.

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