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 uniformity 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

<table>
<thead>
<tr>
<th>Experiment Number</th>
<th>Column 1</th>
<th>Column 2</th>
<th>Column 3</th>
<th>Column 4</th>
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<td>9</td>
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</table>

Fig.1 L9 Orthogonal Array

Fig.2 L18 Orthogonal Array

III. MODEL DEVELOPMENT

1. Range And Parameter Selection

The length, breadth and height of 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 8.253$m^2$, 10.314$m^2$ and 14.628$m^2$ 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

<table>
<thead>
<tr>
<th>Notation</th>
<th>Factors</th>
<th>Unit</th>
<th>Levels</th>
</tr>
</thead>
<tbody>
<tr>
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<td>Area of the evaporator with fin</td>
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<td>3</td>
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<tr>
<td>dT</td>
<td>Temperature Difference</td>
<td>°C</td>
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<td>8</td>
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<tr>
<td>RH</td>
<td>Relative Humidity</td>
<td>%</td>
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<td>0.95</td>
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</table>
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,

\[ y = a + bx \]

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

\[ Y = \beta_0 + \beta_1 X_1 + \epsilon \]


Where \( \beta \) is constant terms & X is the variables & \( \epsilon \) is the experimental error.

Polynomial multiple regression analysis equation is

\[ Y = \beta_0 + \beta_1 X_1 + \beta_2 X_2 + \beta_3 X_3 + \epsilon \]


The above equation is second order polynomial equation for 3 variables. Where \( \beta \) 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 finally \( X_{11}, X_{22}, X_{33} \) are the square terms.

\( Q \) (heat due to convection) = response variable & A, dT, RH= predictor variable.

Polynomial regression equation becomes after replacing real problem variables

\[ Q = \beta_0 + \beta_1 (A) + \beta_2 (dT) + \beta_3 (RH) + \beta_{11} (A)^1 (A) + \beta_{12} (dT)^1 (A) + \beta_{13} (RH)^1 (A) + \beta_{22} (dT)^2 (A) + \beta_{33} (RH)^2 (A) + \epsilon \]

To solve this equation following matrix method is used

\[ [\beta] = [\beta]^{-1} [Y] \]

The above equation is second order polynomial equation for 3 variables. Where \( \beta \) 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 finally \( X_{11}, X_{22}, X_{33} \) are the square terms.

Table no- 2 Taguchi’s L9 Orthogonal Array

<table>
<thead>
<tr>
<th>Sl. No.</th>
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<th>RH</th>
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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. \( \beta \)) 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, \( \beta \) 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

<table>
<thead>
<tr>
<th>Sl. No.</th>
<th>A</th>
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<th>RH</th>
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In the equation (2) \( X \) is a 10*10 matrix, \( \beta \) 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 beta(\( \beta \)) values.

Now there are nine test runs, so there will be nine values of Q and also nine equations. These equations are-
The heat transfer equation due to area of the evaporator with fin (A), temperature difference (dT) & relative humidity RH is $Q_t = A_h_c (dT + 2490 \text{ RH})$. 

Here, $A_c$=surface area of tubes in evaporator with fin $h_c$=convective heat transfer co-efficient. $h_m$=convective mass transfer co-efficient. $h_g$=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 calculate the value of convective heat transfer co-efficient ($h_c$), We know, 

$$Nu = \frac{\text{Conductive heat transfer}}{\text{Convection heat transfer}} = \frac{(h_c * L)}{K}$$

Where:
$Nu = \text{Nusselt number}$

$h_c$ = convective heat transfer coefficient 
$k = \text{thermal conductivity, W/mK}$ 
$L = \text{characteristic length, m}$ 

The convection heat transfer coefficient is then defined as following:

$$h_c = \frac{Nu*K}{L} \quad \ldots \ldots \ldots \ldots \ldots \ldots (5)$$

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

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

$$Q_t = Q_{conv} + Q_{condensation}$$

$$Q_{conv} = A_h_dT & Q_{condensation} = A_m(RH)h_g$$

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.

$$Q_t = A_h_dT + A_m(RH)h_g$$

[As we know, 

$h_m = C_p(Le)^{2/3}$ or $h_m = 1.005*(1)^{2/3}$ or $h_m = h_g$.

or, $Q_t = [A_h(T) + [(h_c / 0.055)*A_{RH}h_g]]$

or, $Q_t = A_h(T + RH)h_g / 0.005$

or, $Q_t = A_h(T + RH)h_g$]

The heat transfer equation due to area of the evaporator with fin (A), temperature difference (dT) & relative humidity RH is $Q_t = A_h_c (dT + 2490 \text{ RH})$. 

Here, $A_c$=surface area of tubes in evaporator with fin $h_c$=convective heat transfer co-efficient. $h_m$=convective mass transfer co-efficient. $h_g$=latent heat of condensation of moisture 2490 KJ/Kg*°K. 

$C_p$=specific heat of air 1.005 KJ/Kg*°K. 

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Now we calculate the value of convective heat transfer co-efficient ($h_c$), We know, 

$$Nu = \frac{\text{Conductive heat transfer}}{\text{Convection heat transfer}} = \frac{(h_c * L)}{K}$$

Where:
$Nu = \text{Nusselt number}$

$h_c$ = convective heat transfer coefficient 
$k = \text{thermal conductivity, W/mK}$ 
$L = \text{characteristic length, m}$ 

The convection heat transfer coefficient is then defined as following:

$$h_c = \frac{Nu*K}{L} \quad \ldots \ldots \ldots \ldots \ldots \ldots (5)$$

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
Nusselt number formula.

<table>
<thead>
<tr>
<th></th>
<th>Vertical fins</th>
<th>Horizontal fins</th>
</tr>
</thead>
</table>
| Laminar flow        | $\text{Nu} = 0.59 \times \text{Ra}^{0.25}$ | $\text{Upward laminar flow}$  
|                     | $\text{Nu} = 0.14 \times \text{Ra}^{0.33}$ | $\text{Downward laminar flow}$ |
| Turbulent flow      | $\text{Nu} = 0.54 \times \text{Ra}^{0.25}$ | $\text{Nu} = 0.27 \times \text{Ra}^{0.25}$ |
|                     | $\text{Nu} = 0.14 \times \text{Ra}^{0.33}$ | $\text{Turbulent flow}$       |

Where:

$Ra = Gr \times Pr$

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 $Ra > 10^6$ the flow is turbulent.

Grashof number ($Gr$):

$$Gr = \frac{gL^3\alpha(T_p - T_a)}{\eta^2}$$  [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
- $\alpha =$ air thermal expansion coefficient. For gases, is the reciprocal of the temperature in Kelvin:
  $$\alpha = \frac{1}{T_a} = \frac{1}{(1/275.15) \text{ K}}$$
- $T_p =$ Plate temperature, = 272.15 K
- $T_a =$ Air temperature = 275.15 K
- $\eta =$ air kinematic viscosity = $13.39 \times 10^{-6}$ m²/s [at air temp. = 275.15 K & air pressure = 1 bar]

$$Gr = \frac{9.81 \times (0.144)^{1/2} \times \frac{1}{(1/275.15)^2} \times (275.15 - 272.15)}{(13.39 \times 10^{-6})^2}$$

or, $Gr = 4.56 \times 10^{11}$

Prandtl number ($Pr$):

$$Pr = \frac{\mu cp}{\kappa}$$

Where:

- $\mu =$ air dynamic viscosity, is $1.725 \times 10^{-5}$ 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

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

Or, $Pr = 0.711$

So,

$Ra = Gr \times Pr$

$Ra = 4.56 \times 10^{11} \times 0.711$

$Ra = 3.24 \times 10^{11}$

As, $Ra > 10^9$ = Turbulent flow

So, Nuske Number for turbulent flow,

$Nu = 0.14 \times Ra^{0.33}$

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

$Nu = 880.25$

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

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

or, $h_c = 2.35$

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

$$Q_T = 2.35 \times A(dT + 2490 \text{ RH})$$  (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 = $2 \times \text{(Area of top)} + \text{(perimeter of top)} \times \text{height}$

Surface Area of cylindrical pin fin = $[2\pi(D/4) + (\pi*D) \times H]$

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.
We know,

\( A_b \) = Area of bare tube
\( A_f \) = Area of cylindrical pin fin
\( A = A_b + A_f \)
\( A_b = \pi r^2 L \)
\( A_f = [2\pi (D^2/4) + \pi D] H \times n \times N \)

\( r \) = 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:

| \( B_0 \) | -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:

\[ Q_{\text{heat due to convection heat transfer}} = -36.0811 + 0.1920(A) + 0.7257(dT) + 73.8010(RH) - 0.0072(A^2) - 0.0108(dT^2) - 39.0099(RH^2) + 2.35(A)(dT) + 5851.4863(A)(RH) - 0.6871(dT)(RH) \]

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 found by using a computer program data set the variation of \( A \) with \( Q \) can be graphically represented as-

![Scatterplot of Heat Transfer vs Area of the Evaporator](image)

**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-

![Scatterplot of Heat Transfer vs Temperature Difference](image)

**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 anddT remain constant. Enter the value for RH within the range 0.85 to 0.95, Start with 0.85 Keeping A=8.253 anddT=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 m², temperature difference(dT) is 5 °C 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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