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Optimization Of Convective Heat Transfer Model Of Cold Storage With Evaporator Fins Using Taguchi Analysis.

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ABSTRACT:- Energy crisis is one of the most important problems the world is facing now-a-days. 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. In this situation if the maximum heat energy(Q) is absorbed by the evaporator inside the cold room through convective heat transfer process in terms of –heat transfer due to convection and heat transfer due to condensation, more energy has to be wasted to maintain the evaporator space at the desired temperature range of 2- 8 degree centigrade. In this paper we have proposed a theoretical heat transfer model of convective heat transfer in cold storage using Taguchi L9 orthogonal array. Area of the evaporator and plate fins(A), Temperature difference(dT), Relative Humidity(RH) are the basic variable and three ranges are taken each of them in the model development. Graphical interpretations from the model justifies the reality

Keywords:- cold torage1, convective heat transfer model2, heat absorbed by evaporator with fin3, and Taguchi Analysis4 etc…

I. INTRODUCTION

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. India is having a unique geographical position and a wide range of soil thus producing variety of fruits and vegetables like apples, grapes, oranges, potatoes, chillies, ginger, etc. Marine products are also being produced in large quantities due to large coastal areas. The cold storage facilities are the prime infrastructural component for such perishable commodities. 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 comes 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 modeling 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. Velocity of air (**V**), Temperature difference (**dT**), Relative Humidity (**RH**)are the basic variables and three ranges are taken each of them in the model development. Graphical interpretations from the model justifies the reality.

II. MODEL DEVELOPMENT

Relationship between heat gain & energy consumption is given by $E= (Q\ t)/COP$ [M.S.Soylemez, M.Unsal](1997) [1] E=energy consumption of refrigeration system (kw/h), t=equivalent full load hours of operation of refrigeration system(hrs), COP= co efficient of performance of refrigeration plant., Q= heat energy extracted from cold room (Joule) Response variable is heat transfer due to convection and condensation and predictor variables are Velocity of air (**V**), Temperature difference (**dT**), Relative Humidity (**RH**) .With the help of Taguchi methodology we construct our design matrix.

Orthogonal arrays provide a best set of well balanced (minimum) experiments .It was developed by C.R.Rao (1947) Popularized by Gene chi Taguchi (1987).The number of rows of an orthogonal array represents the requisite number of experiments

The font size for **heading is 11 points bold face** and **subsections with 10 points and not bold.** Do not underline any of the headings, or add dashes, colons, etc.

III. INDENTATIONS AND EQUATIONS

Regression analysis

Regression analysis is the relationship between various variables. By 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 the changes 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 can not be used. We have to take the help of multiple regression analysis. There are two types of multiple regression analysis- 1) Simple multiple regression analysis (regression equation of first order) 2) Polynomial multiple regression analysis (regression equation of second order or more) Simple multiple regression analysis is represented by the equation of first order regression $Y= \beta0+\beta1X1+\beta2X2+\beta3X3+\ldots\ldots\ldots\ldots..+\varepsilon$ Where β is constant terms & X is the variables & ε is the experimental error. Polynomial multiple regression analysis equation is $Y = \beta0+ \beta1X1+\beta2X2+\beta3X3+ \beta\ 11X211+ \beta\ 22X222+ \beta33X233+ \beta12X1X2+ \beta13X1X3+ \beta23X2X3$ The above equation is second order polynomial equation for 3 variables. Where β are constant, X1 , X2 , X3 are the linear terms, X12 X13 X23 are the interaction terms between the factors, and lastly X11 X22 X33 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 (heat due to conv+condensation) = $\beta0+\beta1(A)+\beta2\ (dT)+\beta3\ (RH)+ \beta\ 11\ (A)*(A)+ \beta\ 22(dT)*(dT)+\beta33\ (RH)*(RH)+ \beta12(A)*(dT)+ \beta13(dT)*(RH)+ \beta23(RH)*(A)$ To solve this equation following matrix method is used $Y= [\beta][\ X\ ]\ [\beta]=Y\ [X\text{-}1]$ where $[\beta]$ is the coefficient matrix, Y is the response variable matrix; [X-1] is the inverse of predictor variable matrix. 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.
(10)

Cold storage Description

The overall dimensions of cold storage plant are 87.5mx34.15mx16.77m.The cold storage building isof five floors with each floor having 2 cold chambers of 43.25m x 17m sizes operating at different temperature as per the requirements of commodities. For our analysis purpose we only considered zone 1 which is referred as cold room.

Parameter &Range Selection

The one chamber of cold storage Length ,Breath and Height 87.5m,34.15m and 16.77m respectively. The three values of Area (**A**) of evaporator space(bare tube and plate fins) are $10.75m^2/m^2$ face area, $12.44m^2/m^2$ face area and $12.71m^2/m^2$ face area 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.

Heat (heat due to convection**)**

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 bare tube and plate fins(**A**), temp. difference (**dT**) & relative humidity RH) . Only convection heat transfer effect is being considered in this study. Basic equation for heat transfer QT = Qconv+ Qcondensation. $Qconv=Ah_c(dT+RH.h_{fg})$.

Here Qconv=heat transfer due to convection & Qcondensation=heat transfer due to condensation & QT=Total heat transfer or absorb heat into refrigerant .**A=(D-t/DB)□d_0^2+2/D[C-(□d_0^2/4B)]**
A=total area of the fins & bare tube,in m^2,B=vertical spacing between the tubes in a raw,m,C=spacing between the tubes in different raw, in m,t=thinks of the fins in m, D=center to center spacing between fins in m,d_0=outer diameter of bare tube in m.

The final heat transfer equation due to Area of fins & bare tubes(**A**), temp. difference (**dT**) & relative humidity RH) is **QT=7.905A(dT + 2490 RH).** Hera **A**=surface area of tubes & plate fins in evaporator space 1872 m2.**hc**=convective heat transfer co-efficient. **hm**=convective mass transfer co-efficient.**hfg**=latent heat of condensation of moisture 2490 KJ/Kg-K. Cp=specific heat of air 1.005 KJ/Kg-K. Le=Lewis number for air it is one. After getting the full observation table which include all the predictor variables and response variable, values can be computed easily in the following equation- Q (heat due to conv+condensation) = β0+β1(A)+β2 (dT)+β3 (RH)+ β 11 (A)*(A)+ β 22(dT)*(dT)+β33 (RH)*(RH)+ β12(A)*(dT)+ β13(A)*(RH)+ β23(dT)*(RH) We get **nine**equations but number of unknowns are **ten.** Now we can solve these equations-eliminate some of variables in terms of others. Then we get[X] matrix as [9*9]. We also get response variable matrix [Q] which is a [9*1] matrix. With the help of following equation we can get the co efficient values-[β]=[Q]*[X-1] where [x]-denotes the variable matrix. The proposed theoretical mathematical model for heat transfer in cold storage is –
$Q_{\text{(heat due to conv+condenstion)}}$=232927-29739(A)+2666(dT) -36(RH)-15299(A)(A)+1084(dT)(dT)-186(RH)(RH)+167(A)(dT)-148416(A)(RH)+305302(dT)(RH)

IV. FIGURES AND TABLES

L_9 OA Combination Table with notation for Matrix design **$L_9(3^3)$**

Exp. No.	**Control Factors**		
	A	B	C
1	1	1	1
2	1	2	2
3	1	3	3
4	2	1	2
5	2	2	3
6	2	3	1
7	3	1	3
8	3	2	1
9	3	3	2

Table 1

Level	**Area of fins and bare tube(m^2)m^2 per m^2 face area per row(A)**	**Temp. differ. In evaporator space(dT)^{0}c**	**Relative Humidity(RH)**
1	10.75	2	0.85
2	12.44	5	0.90
3	12.71	8	0.95

Table 2

A	**dT**	**RH**
10.75	2	0.85
10.75	5	0.90
10.75	8	0.95
12.44	2	0.90
12.44	5	0.95
12.44	8	0.85
12.71	2	0.95
12.71	5	0.85
12.71	8	0.90

Table 3

Obs. No.	**A**	**dT**	**RH**	**Q**
1	**10.75**	**2**	**0.85**	**179913.61**
2	**10.75**	**5**	**0.90**	**190741.55**
3	**10.75**	**8**	**0.95**	**201569.49**
4	**12.44**	**2**	**0.90**	**220427.75**
5	**12.44**	**5**	**0.95**	**232957.64**
6	**12.44**	**8**	**0.85**	**208782.33**
7	**12.71**	**2**	**0.95**	**237735.14**
8	**12.71**	**5**	**0.85**	**213032.78**
9	**12.71**	**8**	**0.90**	**225032.50**

Variation of Convective Heat Transfer with Area of the bare tube and plate fins of evaporator

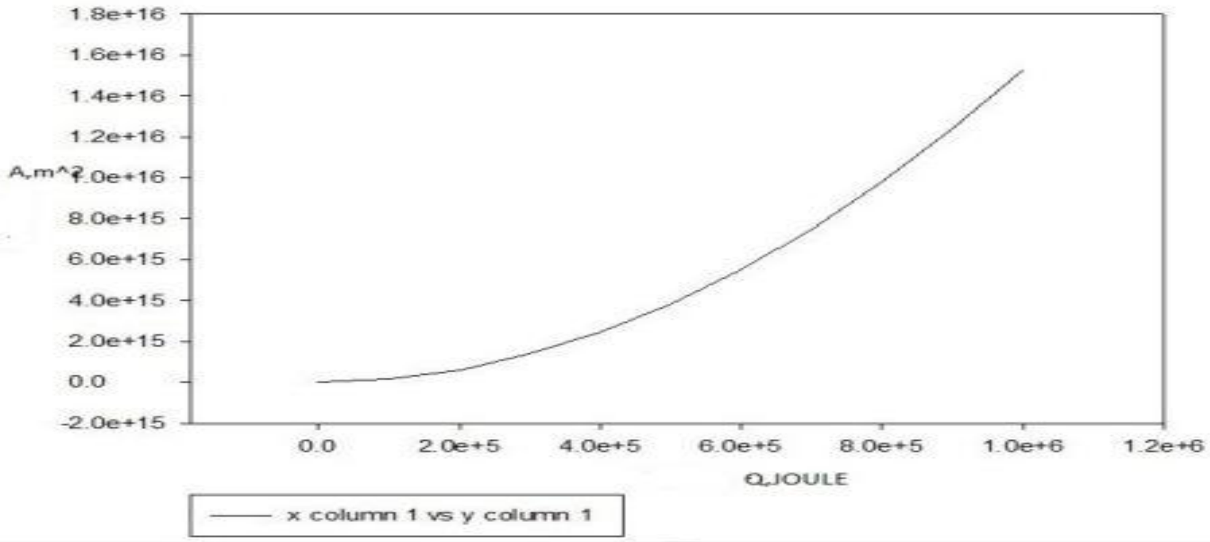

Figure: 1 Variation of heat transfer with Area of the evaporator coil and bare tube.

Figure:1 Shows that heat absorption increase with increase the Area of the Evaporator & evaporator plate fins.

Variation of Convective Heat Transfer with Temperature Difference of Evaporator space

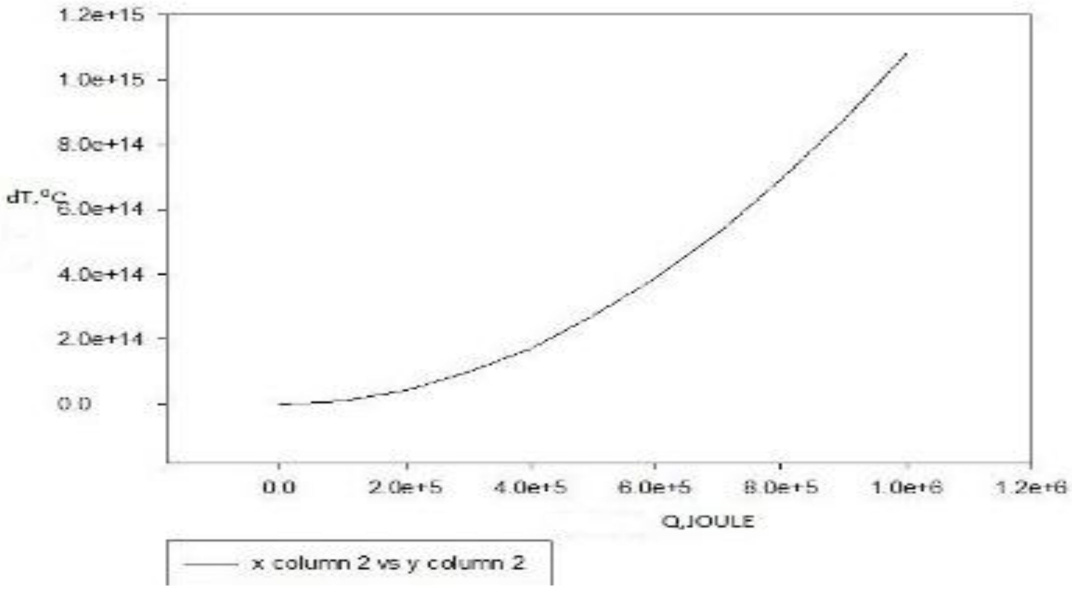

Figure 2: variation of Heat transfer with temperature difference.

Figure 2: Shows that heat absorption increase with temperature difference increase and lower temperature difference is more effect than higher temperature difference

Variation of convective Heat Transfer with Relative Humidity

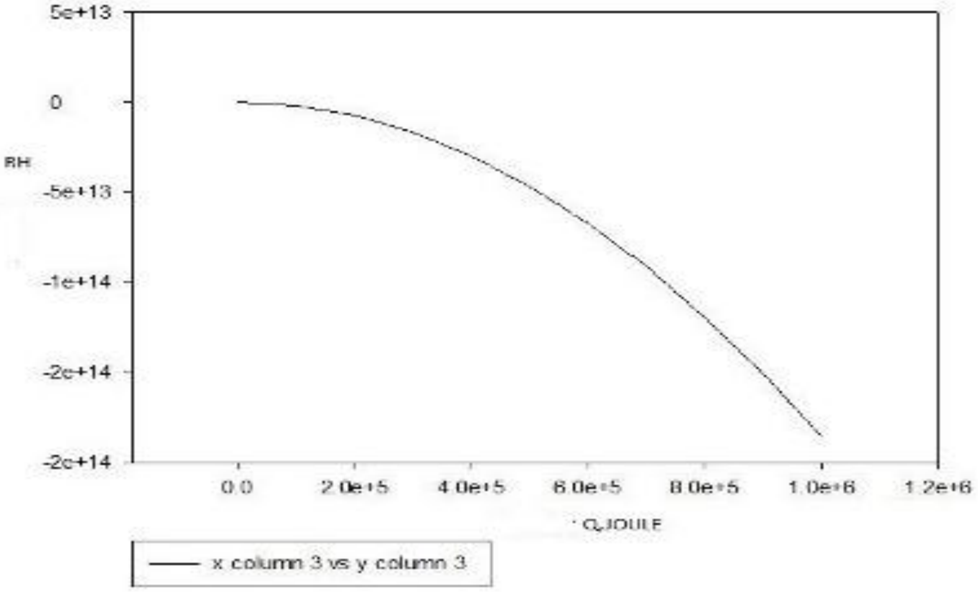

Figure 3: variation of heat transfer with Relative humidity.

Figure 3: Shows that heat absorption increase with increase in relative humidity

V. CONCLUSION

1. With increasing the Area by using plate fins ,temperature difference and relative humidity heat absorb by the refrigerant is increased.

2. To create maximum cooling effects maximum heat transfer is required to increase the Area of the evaporator by using fins , temperature difference and relative humidity are 237732.14 , 2 & 0.95 respectively

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