EXPERIMENTAL STUDY OF MIXED CONVECTION HEAT TRANSFER USING CIRCULAR, SQUARE, RHOMBIC FINS

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ABSTRACT

Pulling Fins are extended surfaces employed to enhance the convective heat transfer from a surface for increasing heat dissipation. Fins with various geometries have been designed and used in various cooling application the selection of particular fins configuration in any heat transfer application is an important state in designed process and takes into account the space, weight, manufacturing technique and cost consideration as well as the thermal characteristics it exhibits. Fins cross section profiles have profound influence on thermal characteristics of Annular Fins and the surface area changes with change of cross section of fins. This study deals with studying the performance of various available fins profiles. Widely used fins profile viz. Rectangular, Triangular, Trapezoidal, Circular, Rhombic, and Elliptical Fins. In Addition to the normal configuration of fins, to new configurations were designed and created.

KEYWORDS: fins, triangular, thermal etc.

INTRODUCTION

1Background and motivation of the Study

Engineering curriculum, practicing engineers and techniques engaged in the design, construction testing and operation of many diverse forms of Heat exchange equipments required in our scientific and industrial technology. Electrical engineers apply their knowledge of heat transfer for the design of the cooling system for motors, generators. Chemical engineers are concerned with the evaporation, condensation, heating & cooling of the fluids. [7]

The Mechanical engineers deals with problems of heat transfer, in the field of internal combustion engines, steam generation, refrigeration and air conditioning and the ventilation.[2] To estimate the cost, the feasibility and size of the equipment necessary to transfer a specified amount of heat in the given time, a detailed heat transfer analysis must be made. The dimensions of boilers, heaters, refrigerators and heat exchanger depend not only on the amount of heat to be transmitted but rather on the rate at which heat is to be transferred under the given condition. The successful operation of equipment such as turbine blades and walls of combustion chambers of gas turbine depends on the possibility of cooling certain metal parts by removing heat continuously at a rapid rate from the surface. These varied examples shows that in almost every branch of engineering, heat transfer problems are encountered, which cannot be solved by thermodynamic reasoning alone but required an analysis based on science of heat transfer .

2 Statement of Problem

In the present work we have to study mixed convection heat transfer from circular, square, rhombic fin arrays on a horizontal surface. In the proposed work it is proposed to carry experimental study on mixed convection heat transfer in circular, square, rhombic fin arrays. The objective of study is that to find different parameters. And observations and comparison of all these parameters.

3 Objective of Study

In the present study we have performed experimental work on circular, square, rhombic fin arrays. The purpose of this study to show maximum heat transfer takes place in mixed convection mode.

4 Modes of heat transfer

The literature of heat transfer generally recognizes three distinct modes of Heat Transmission. Heat transfer is the energy in transits due to temperature difference. Whenever there is exit temperature difference in a body, heat flows from regions of high temperature to the region of low temperature. This heat transfer takes place by three different processes called as modes of heat transfer. There are

1.4.1 Conduction
1.4.2 Convection
1.4.3 Radiation

EXPERIMENTAL SETUP

The objective of this project work on "combined convection heat transfer though circular, square, rhombic fin array" was to determine the $\mathbf{G_r/R_e^2}$ = 1 for assisting mode & opposing mode at the different velocities & power output. It also studies the effect of different velocities on combined convective heat transfer coefficients. It was therefore decided to built fin arrays with hot surface with on vertical base.

The fin array was constituted by three geometrical parameters fin length "L" fin height 'H' & fin spacing 'S'. It was decided to use cartridge type heater. This was inserted at the base of fin array thus the fins & spacer pieces made of 'Mild Steel having small thickness were used, which gives high thermal conductivity. The component of the fin array assembly was put together by using tie rods & nuts. The horizontal & vertical ducts are made up of plywood. The heat transfer by radiation is neglected because black coating is provided inside the duct & heating surface.

2.1 Duct

2.2 Fin array

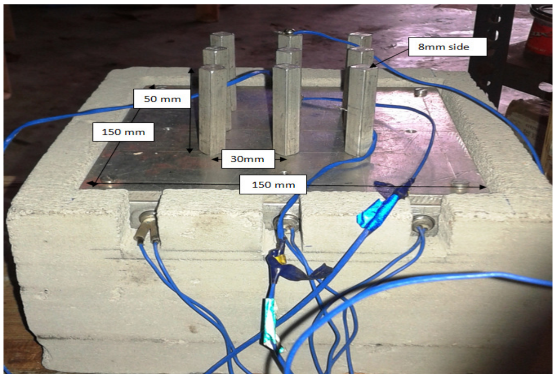
i) Circular fin 50 mm height.
ii) Circular fin 100 mm height
iii) Square fin 50mm height
iv) Square fin 100 mm height
v) Rhombic fin 50 mm height
vi) Rhombic fin 100 mm height

2.3 Input power measurement

2.4 Temperature measurement

2.5 Blower's

2.6 Anemometer



Specimen calculation

1. The sample specimen calculation for one reading is shown here.
From observation table, for "Assisting mode", at v=0.15m/s & power = 63.294 watts.
For heat flow due to natural convection (q_n)

$$= 68.4^0C$$

$$T = 68.4^0C$$

Bulk mean temperature

$$= \frac{T+Tamb}{2}$$

$$= \frac{68.4+27.4}{2}$$

$= 47.9^0c$

$T_B = 47.9^0c$

Properties at 47.9^0c

Pr = 0.699 L = 0.05m

K = 56.5x 10^{-3} w/m^2 -k A = 0.00157m^2

V = 16.96 x 10^{-6} m^2/sec

$$= \frac{1}{47.9+273}$$

$= 3.11 \times 10^{-3}$

$\beta = 3.11 \times 10^{-3}\ K^{-1}$

$$Gr = \frac{9.81\times\beta\times(T-Tamb)\times L^3}{V^2}$$

$$Gr = \frac{9.81\times3.11\times10-3(68.4-27.4)\times0.05^3}{(16.96\times10^{-6})^2}$$

$Gr = 5.443 \times 10^5$

$Gr\times Pr = 3.8 \times 10^5$

Since the fin array has a major portion of vertical plates then use the correlation for the vertical plates, $10^4 < Gr*Pr < 10^9$

$Nun = 0.59(Gr\times Pr)^{0.25}$

$= 0.59(3.8\times10^5)^{0.25}$

$= 14.65$

$h_n = h_{un}\times K/L$

$= 14.65 \times 46.5 \times 10^{-3} /0.05$

$= 13.62$

$Q_{nat} = h_n\times A\times dT$

$= 13.62 \times 1.57\times10^{-3}\times(68.4-27)$

$= 0.876$ Watt

Heat flow due to convection

$R_e = VD/\upsilon$

$= 0.15\times 0.05/16.96\times10^{-6}$

$= 442.21$

But $R_e < 5\times10^5$

Hence flow is laminar

Correlation for cylinder

$N_u = C\ R_e^{\ m} \times P_r^{.33}$

Where m= coefficient,

C = constant

$R_e = 442.21$

ISSN: 2394-3696
VOLUME 2, ISSUE 12, DEC.-2015

$C = 0.683$

$m = 0.466$

$N_u = 0.683\times 442.21^{0.466} \times 0.609^{.38}$

$= 10.36$

$h_f = N_{uf} \times K/L$

$= 10.36\times0.237/0.05$

$= 4.912\ W/m^2K$

$Q_f = h_f\times A\times dT$

$= 4.912\times 1.57\times10^{-3}\times(68.4-27.4)$

$= 1.88\times10^{-4}$ Watt

Heat flow due to radiation

$Qrad = \sigma A_S (T^4 - T^4_{atm})$

$A_S = 1.57\times 10^{-4}$

$\sigma = 5.6 \times 10^{-6}\ W/m^2K$

$c = 1$

$T_{atm} = 27.9+273 = 300.93$ K

$Qrad = 5.6 \times 10^{-6} \times 1.57\times 10^{-4} (310.4^4-300.9^4)$

$= 0.14$ Watt

$Q_{total} = Q_{nat} + Q_{conv} + Q_{rad}$

$=0.876 + 1.88\times10^{-4}+0.14$

$= 1.016$ Watt

$\mathbf{G_r/(R_e)^2} = 5.443 \times10^3/ (442.2)^2$

$\mathbf{G_r/(R_e)^2 = 2.78}$

2. The sample specimen for Circular fin (50mm) calculation for one reading is shown here.

From observation table, for "Opposing mode", at v=0.15m/s & power = 22.54 watts.

For heat flow due to natural convection (qn)

$= 45.86\ ^0C$

$T = 45.86\ ^0C$

Bulk mean temperature

$$= \frac{T+Tamb}{2}$$

$$= \frac{45.86+27.3}{2}$$

$= 36.63\ ^0c$

$T_B = 36.63\ ^0c$

Properties at 36.63^0c

$Pr = 0.699$ $L = 0.05m$

$K = 23.7x\ 10^{-3}\ w/m^2 -k$ $A = 1.57\ x\ 10^{-3}m^2$

$V = 16.96\ x\ 10^{-6}\ m^2/sec$

$$= \frac{1}{36.63+273}$$

$= 3.22 \times 10^{-3}$

$\beta = 3.22 \times 10^{-3}\ K^{-1}$

$$Gr = \frac{9.81\times\beta\times(T-Tamb)\times L^3}{V^2}$$

$$Gr = \frac{9.81\times3.22\times10-3(46-27.4)\times0.1^3}{(16.96\times10^{-6})^2}$$

$Gr = 2.56 \times 10^5$

$Gr\times Pr = 1.79 \times 10^5$

Since the fin array has a major portion of vertical plates then use the correlation for the vertical plates, $10^4 < Gr*Pr < 10^9$

$Nun = 0.59(Gr\times Pr)^{0.25}$

$= 0.59(1.79\times10^5)^{0.25}$

$= 12.13$

$h_n = h_{un}\times K/L$

$= 12.13\times23.7 \times 10^{-3} /0.05$

$= 5.75$ wm/K

$Q_{nat} = h_n\times A\times dT$

$= 5.75\times 0.032\times(46-27.4)$

$= 3.42$ Watt

Heat flow due to convection

$R_e = VD/\upsilon$

$= 0.15\times0.05 /16.96\times10^{-6}$

$= 442.2$

But $R_e < 5\times10^5$

Hence flow is laminar

Correlation for cylinder

$N_u = C\ R_e^m \times P_r^{.33}$

Where m= coefficient,

C = constant

$R_e = 442.2$

C = 0.683

m = 0.466

$N_u = 0.683 \times 442.2^{0.466} \times 0.699^{.33}$

$= 10.37$

$h_f = N_{uf} \times K/L$

$= 10.37\times23.7\times10^{-3}/0.05$

$= 4.915\ W/m^2K$

$Q_f = h_f\times A\times dT$

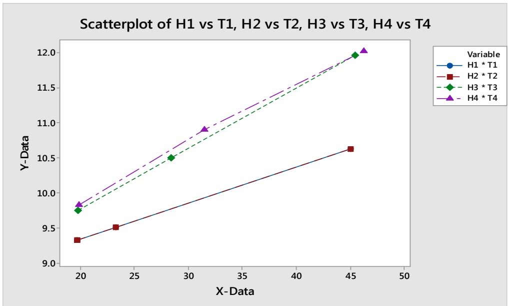
$$=4.915\times 0.032 \times (46-27.4)$$
$$= 2.925 \text{ Watt}$$
Heat flow due to radiation
$$Qrad = \sigma A_S (T^4 - T^4_{atm})$$
$$A_S = 0.032$$
$$\sigma = 5.67 \times 10^{-6} \text{ W/m}^2\text{K}$$
$$c = 1$$
$$T_{atm} = 27.4+273 = 300.3 \text{ K}$$
$$Qrad = 5.67 \times 10^{-8} \times 0.032\ ((46+273)^4 - 300.9^4)$$
$$=4.092 \text{ Watt}$$
$$Q_{total} = Q_{nat} + Q_{conv} + Q_{rad}$$
$$= 3.42+2.925+4.092$$
$$= 10.43 \text{Watt}$$
$$\mathbf{G_r/(R_e)^2} = 2.56 \times 10^5 / (442.2)^2$$
$$\mathbf{G_r/(R_e)^2 = 1.309}$$

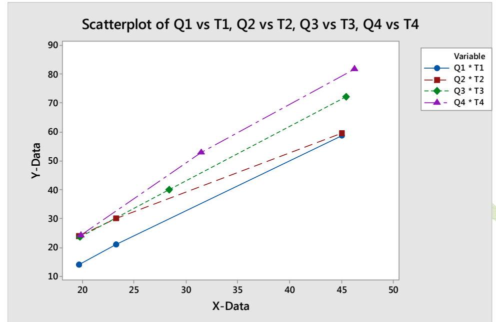
RESULT TABLE AND GRAPHS

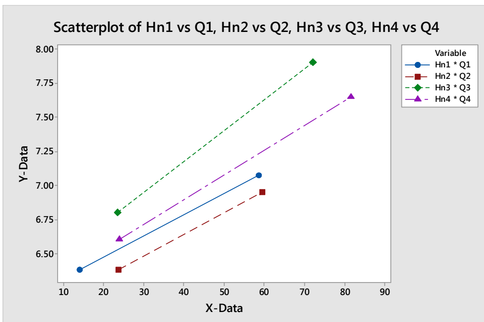
Sr No.	Velocity(V) m/s	Kinematic Viscosity (V) m²/s	Thermal Conductivity (k) w/m-k	Prandtl No. (Pr)	Grashoff No (Gr)	Gr.Pr	Reynolds No. (Re)	Nusselt No (Nun)	Nusselt No (Nuf)	$h_{natural}$ w/m²k	h_{forced} w/m²k	h_{total} w/m²k	Q (natural) W	Q (forced) W	Q (rad) W
1.	0.00	16.96×10^{-6}	23.7×10^{-3}	0.699	2.966×10^{4}	1.5×10^{4}	-	20.65	8.5	4.89	-	6.92	2.760	-	3.81
	0.05	16.96×10^{-6}	23.7×10^{-3}	0.699	2.146×10^{4}	1.5×10^{4}	294.811	20.65	8.5	4.89	2.03	6.92	2.760	1.14	3.81
	0.10	16.96×10^{-6}	23.7×10^{-3}	0.699	2.072×10^{4}	1.448×10^{4}	589.62	20.468	11.86	4.851	2.811	7.662	2.638	15.29	3.65
	0.15	16.96×10^{-6}	23.7×10^{-3}	0.699	2.06×10^{4}	2.069×10^{4}	884.43	22.31	14.32	5.200	3.396	8.726	0.4175	2.675	3.45
2.	0.00	17.45×10^{-6}	24×10^{-3}	0.699	3.169×10^{4}	2.215×10^{4}	-	22.56	8.474	5.43	-	7.463	5.3	-	6.45
	0.05	17.45×10^{-6}	24×10^{-3}	0.699	3.169×10^{4}	2.215×10^{4}	286.53	22.56	8.474	5.43	2.003	7.463	5.3	19.8	6.45
	0.10	17.45×10^{-6}	24×10^{-3}	0.699	2.811×10^{4}	1.965×10^{4}	573.06	21.99	11.70	5.301	2.809	8.11	4.56	24.1	6.84
	0.15	16.96×10^{-6}	23.7×10^{-3}	0.699	2.92×10^{4}	2.046×10^{4}	884.43	22.50	14.32	5.25	3.396	8.676	0.44	0.287	6.47
3.	0.00	17.45×10^{-6}	24×10^{-3}	0.698	3.735×10^{4}	2.607×10^{4}	-	23.708	8.470	5.689	-	7.722	16.07	-	10.02
	0.05	17.45×10^{-6}	24×10^{-3}	0.698	3.735×10^{4}	2.607×10^{4}	286.53	23.708	8.470	5.689	2.033	7.722	16.07	21.7	10.02
	0.10	17.95×10^{-6}	24.3×10^{-3}	0.698	3.679×10^{4}	2.5684×10^{4}	557.103	23.61	11.72	5.826	2.806	8.702	26.409	39.650	6.30
	0.15	17.95×10^{-6}	24.3×10^{-3}	0.698	3.72×10^{4}	2.6×10^{4}	835.65	23.7	14.35	5.756	3.389	9.145	6.51	3.83	5.038

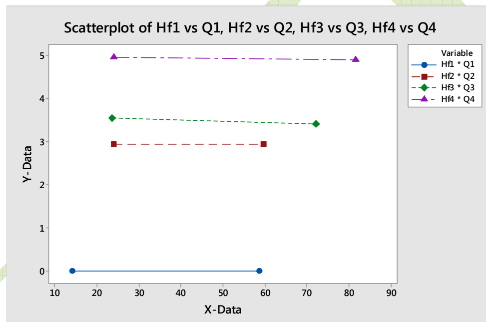
Table 4.1 Result Table (Assisting mode) Circular fin array(100mm)

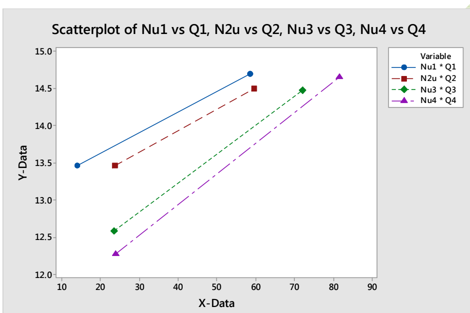
100 mm Circular fin array assisting mode graph result

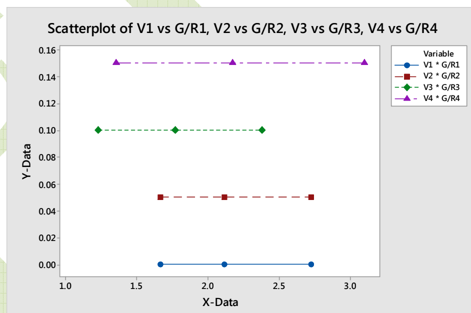












CONCLUSION

The observation, result & discussions made in previous chapter V &VI enable one to predict, the heat transfer rate from fin array, losing by combined convection. In this experiment we use the velocity of air in between 0 m/s to 0.15 m/s for the combined convection.

Overall conclusions

From the experimental analysis of set up, the following conclusion can be summarized.
1. The heat transfer coefficient for natural & forced condition are comparable with each other .indicating the combined convection region was present in the experiments.
2. The temperature of finned system decreases with increase in air velocity, as expected.
3. The specimen temperature are increasing in opposing mode when compare with assisting mode.
4. The observed value of G_r/Re^2 within the prescribed zone i.e. 1 to 10 which is the combined convection effect.

5. From graph, it is clear that the value of heat transfer coefficient increase with increase in air velocity for given heat input.
6. it's seen that value of Reynolds Number increase with increasing in air velocity.
7. The nusselt number for circular, square, rhombic fin array increases with increasing the grashofs Number.
8. The nusselt number for circular, square, rhombic fin array increases with increasing the Heat input.
9. The nusselt number for circular, square, rhombic fin array increases with increasing the Fin length
10. The nusselt number for natural convection is maximum in square fin than circular in for same length and same heat input.
11. The nusselt number for forced convection is maximum in rhombic fin than square fin.
12. Heat transfer coefficient is maximum in rhombic fin than square fin, more in square fin than circular fin.

13. Total heat transfer is maximum in rhombic fin than square and circular fin for similar condition.

FUTURE SCOPE

1) Heat Transfer by radiation is also a factor of consideration. This can be studied by surface of the fin arrays made of polished & dull by providing a black coating etc.
2) The work was concerned with the combined convection heat transfer from circular,square,rhombic fin array. It is worthwhile to carry out the work on vertical fin arrays under forced and natural convection condition also.
3) it may also be possible to change the specimen material from aluminum to copper A alloy cast iron etc. because the heat transfer rate & the thermal conductivity for different material is different .
4) In future the similar experiment may be studied for the various cross sectional specimen such as .triangular trapezoidal etc .by using different material of the specimen & using different working fluids.

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