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Research Article

Effect of Low Aspect Ratio on Convective Heat Transfer from Rectangular Fin Array in Mixed Convection

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A B S T R A C T

Experimental and CFD analysis is conducted in order to establish effect of geometrical fin parameters for mixed convection heat transfer from vertical rectangular fin arrays. Mixed convective heat transfer from rectangular vertical plates has been reviewed. Study revealed that most of the work was carried out considering various configurations. Experimental work carried on steady state mixed convection heat transfer from vertical rectangular fins made of aluminum. Experimental work carried investigates the effect of fin spacing, fin height, fin length on the performance of heat dissipation from the fin arrays.

Keywords: Mixed Convection, Aspect Ratio, Rectangular Fins

Introduction

In many engineering applications heat dissipation rate is important. From many years efforts have been taken to develop heat transfer enhancement techniques for improving perfor- mance of equipments. Enhancement of heat transfer takes place by active and passive method. Among passive method fins are widely use to enhance heat transfer rate. Fins are use in a wide variety of engineering applications of passive cooling of electronic equipment such as compact power supplies, portable computers and telecommunications enclosures.

The optimal combination of geometry and orientation of the finned surface is required in order to achieve the desired rate of heat dissipation, with the least amount of material. Among the various types of fins, rectangular fins are the commonly use fin geometry because of their simple construction, cheap cost and effective cooling ability. Two common orientations of rectangular fin configurations, horizontally based vertical fins and vertically based vertical fins, have been widely used in the applications. However, vertically orientation fins is prefer over the horizontal orientation because of its relatively better ability to dissipate heat.[3]

Combined Natural and Forced Convection

The presence of a temperature gradient in a fluid in a gravity field always gives rise to natural convection currents and thus heat transfer by natural convection. Therefore, forced convection is always accompanied by natural convection.

It is desirable to have a criterion to assess the relative magnitude of natural convection in the presence of forced convection. For a given fluid, it is observed that the parameter Gr/Re2 represents the importance of natural convection relative to forced convection. The convection heat transfer coefficient is a strong function of the Reynolds number Re in forced convection and the Grashof number Gr in natural convection.

Natural convection is negligible when Gr/Re2 < 0.1, forced convection is negligible when Gr/Re2 > 10 and neither is negligible when 0.1 < Gr/Re2 > 10. Therefore, both natural and forced convection must be considered in heat transfer.

When determining heat transfer under combined natural and forced convection conditions, it is tempting to add the contributions of natural and forced convection in assisting flows and to subtract them in opposing flows. However, the evidence indicates differently. A review of experimental data suggests a correlation of the form *Nucombined* = (*Nun forced* ± *Nun natural*) 1/*n* where Nu_{forced} and $Nu_{natural}$ are determined from the correlations for pure forced and pure natural convection, respectively. The plus sign is for assisting and transverse flows and the minus sign is for opposing flows. The value of the exponent n varies between 3 and 4, depending on the geometry involved. It is observed that n = 3 correlate experimental data for vertical surfaces well.[9]

Literature Review

Mixed convection from finned surfaces has been investigated both theoretically and experimentally by many researchers. In all of the previous studies, different geometries and configurations were studied to find the optimum fin structure for maximum heat transfer rate. Rectangular fins are the most popular fin type because of their low production costs and high effectiveness. The rate of heat transfer from fin arrays depends on the geometric parameters and the base-to-ambient temperature differences. Optimum spacing for 100 mm length fin is 7.5 mm and 200 mm length fin is 10 mm. It revealed from literature survey that less work is carried out on low aspect ratio fin array.

The main objective of this study is to find effect of low aspect ratio on performance of vertical rectangular fin array in mixed convection.

Table 1.Literature Review

Year	Authors	Keywords	Observation
2001	A. Ozsunar, S. Baskaya and M. Sivrioglu	Mixed convection	It was found that the buoyancy-driven secondary flow could enhance the heat transfer by as much as 250 percent as the Nusselt number rises to its maximum.
2005	C.J. Kobus, T. Oshio	Mixed, Combined, Convection, Heat sink, Heat transfer	• For a given fin spacing, fin performance, was only a weak function of fin diameter, d. • The optimal fin spacing appears to be 1.8 ≤s≤ 2.3 cm, although it was not as pronounced for the smaller sized base.
2009	Hussam Jouhara, Brian P. Axcell	Fins, Heat sink, Laminar Heat transfer, Parallel flow	Heat transfer coefficients and fin efficiencies vary substantially along the flow path
2010	M. Dogan, M. Sivrioglu	Mixed convection, Fins, Fin spacing, Fin height, Channel, Heat transfer	• The mixed convection heat transfer depends on the fin height and spacing. • The optimum fin spacing was between 8 and 9 mm. • The average heat transfer coefficient increases with the increase in fin height for each fin spacing. • For each of the modified Rayleigh numbers, the average heat transfer coefficient increases with fin spacing up to the optimum value of fin spacing and after that it decreases.
2010	Min-Hsiung Yang, Rong-Hua Yeh;, Jen-Jyh Hwang	Fin Mixed, Convection Channel, Inclination, Optimum aspect ratio	• For a fixed fin profile area, there exists an optimum aspect ratio, γo, of a fin which dissipates maximum heat transfer in a parallel-plate channel. • The optimum aspect ratio of a fin increases with Re at a fixed Gr/Re^2. • For Gr/Re^2=10, γo increases with the inclination angle, whereas the effects of orientation on γo was not pronounced for Gr/Re^2< 1. • The optimum aspect ratio of a fin decreases with increasing K for a mixed convection flow in a parallel-plate channel. • The optimum aspect ratio of a fin for a smaller A* at Gr/Re^2 = 0.1 was close to the analytical result obtained.

2011	F. Bazdidi-Tehrani, H. Nazaripoor	Mixed convection, Radiative heat transfer, Finite volume method	• The occurrence of reversed flow enhances both heat transfer and the fanning friction coefficient • As wall emissivity increases from 0 to 1, effects of radiation on flow and thermal fields rise • As ε increases, the bulk temperature increases more quickly and reaches the wall temperature.
2011	Lyes Boutina, Rachid Bessaïh	Mixed convection, Cooling, Electronic components, Numerical simulation	• For θ=45º, the maximum heat transfer rate was obtained inside the channel • The increase of the Reynolds number and separation distance can enhance the cooling of electronic components inside the channel. • The heat transfer decreases by increasing the dimensions of the heat sources.

Experiemental Setup

Experimental setup is made in order to analyze effect of aspect ratio and CFD analysis is done in ANSYS Fluent 14.0 software. The experimental set-up primarily consists of an aerated concrete case and supporting frame on which the concrete is mounted and various instruments for measuring the ambient temperature, base-plate temperature and the power input for the heater. The dimension of the aerated concrete block is 250 200 100 mm which acts as insulator in one direction. Aerated concrete block is fixed on the frame which ensures one dimensional heat transfer. The front surface of the frame has detachable acrylic sheet so as to replace fin arrays. Heater plate is placed on the concrete block. Base thickness of the array is 5 mm so as to distribute power supply uniformly. Heater covered by cases fully consists of nichrome wire wound around thin mica plate and mica sheet. Heater plate rated for 300 W and 220 V, AC.

Figure 1.Experimental setup

The aerated concrete block has 5 mm depth to fit heater plate into it. Extruded surface is kept over fin array for fitting to aerated concrete block. Concrete block has in-built 4 bolts to tight fin array over heater plate. Thus, air gap between heater plate and fin array is consider to be negligible Aerated concrete block has high insulation quality and high temperature resistance (thermal conductivity, k ~ 0.15 W/m.K). In addition, it can be shaped very easily so that required processes can be performed.

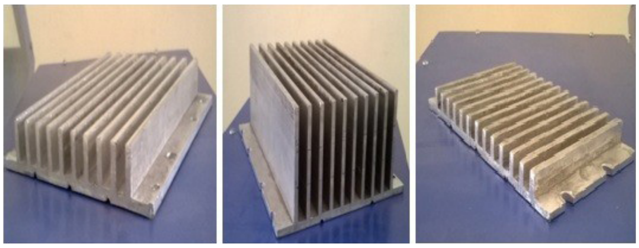

Figure 2.Aluminum Fins

Methodology

While performing experiments power input will be varying through dimmer stat. Power input is varied from 20 W to 100 W. After attaining steady state base plate temperatures is measured. The base plate temperature Tw, the ambient temperature Ta and the power input P to heater is recorded at steady state. Testing procedure mentioned above is repeated for various heater inputs 20 W to 100 W.

For mixed convection, velocity is given from 0.5 m/s to 3 m/s by blower mounted at bottom of the enclosure to

Table 2.Fin array configurations

Block No.	Fin Length	Fin Width	Fin Spacing (mm)	Fin Thickness (mm)	Fin Height (mm)	No. of Fins
1.	200	180	10	5	20	10
2.	200	180	10	5	40	10
3.	200	180	10	5	60	10
4.	100	180	7.5	5	10	13
5.	100	180	7.5	5	20	13
6.	100	180	7.5	5	30	13

assist natural convective current velocity which is measured by anemometer. Speed of blower is adjusted by dimmer stat.

Experimental Nusselt number is validated through Nusselt number obtained from existing correlations.

Nu combined = (*Nun forced* + *Nunnatura* in) 1/*n*

Where n = 3 for vertical surface.

Nu forced = 0.664 X *Re*0.5 X *Pr*1/3

Nu natural = 0.59*Ra*0.25

Comparison of convective Heat Transfer Rate Qc for fin array 10020 with 0.5 m/s velocity.

Flow Visualization

Computational fluid dynamics CFD is the useful to visualize the flow. In this section the variation of temperature and velocity of the flow with different parameters of heat sink is displayed.

Variation of Flow velocity with Fin height

To show the variation of flow temperature with fin length, the following fin configuration is taken as Fin length, L=200 mm, Power input=60W with 2 m/s velocity.

Temperature contours of the flow for two different fin height H=40 mm and 60 mm are shown in Figure 3 and 4.

Table 3.Variation of nu vs power input for fin array 10020 with 0.5 m/s velocity

P	Gr	Re	Gr/Re^2	Nu exp.	Nu mix.
20	2473920	2939.85	0.28624	35.9721	34.9744
40	4052811	2767.17	0.52928	36.2918	35.372
60	5054129	2625.39	0.73326	36.9854	35.4289
80	5771158	2509.59	0.91634	37.3083	35.3965
100	6335927	2391.09	1.1082	36.0451	35.2526

Table 4.Comparison of convectiveheat transfer rate qc forfin array 10020 with 0.5 m/s velocity

Power Input	Qc Experiment	Qc Ansys Fluent
20	17.5337	14.94172
40	34.7795	29.45944
60	51.8463	47.16626
80	68.6205	62.72466
100	84.5069	78.17555

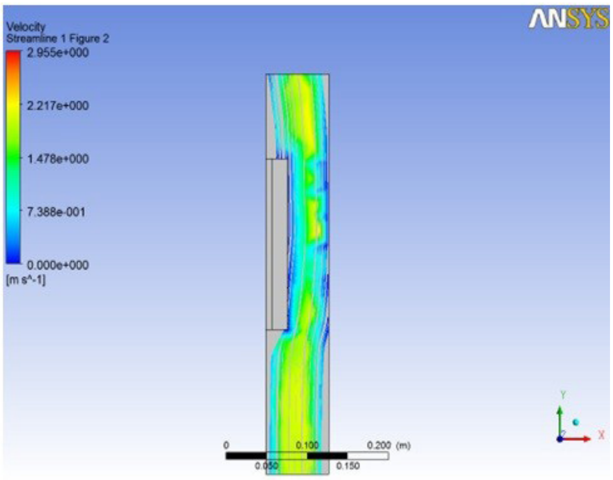

Figure 3.Streamline for fin array with fin length L=200mm and fin height H=20 mm, v =2 m/s

It can be seen from figure that as fin height increases flow is disturbed which lead to increase in heat transfer rate.

Variation in Flow Pattern with velocity

To show the variation of flow type with velocity, the following fin configuration is taken as Fin Length, L=100 mm, Fin height, H=30 mm for Power input 60W Effect velocity over streamline flow and thereby on mixed convection heat transfer rate can be seen through Figure 5 to 7.

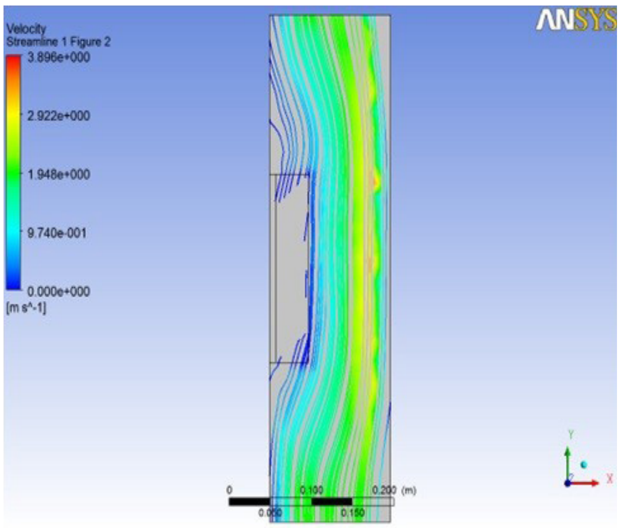

Figure 4.Streamline for fin array with fin length L=200 mmand fin height H=40 mm, v =2 m/s

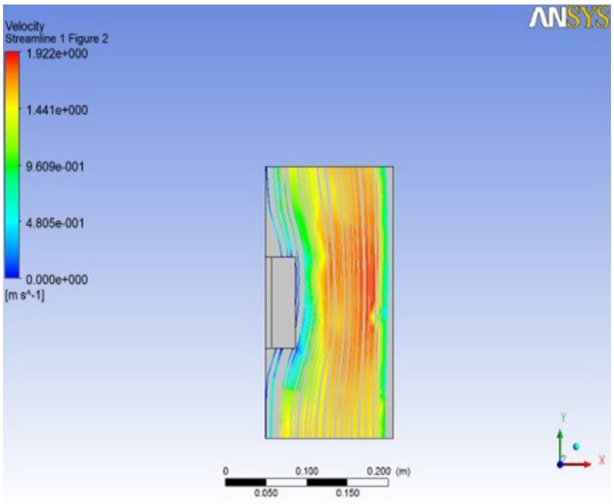

Figure 5.Streamlines for fins array of L=100 mm, H=30mm, v=0.5 m/s

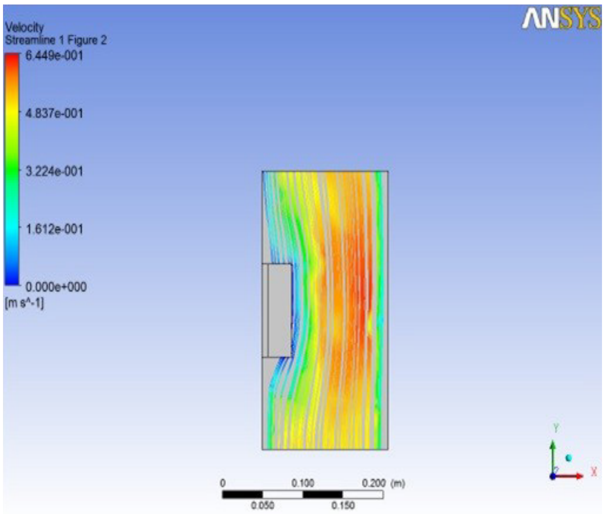

Figure 6.Streamlines for fins array of L=100 mm, H=30mm,v=1.0 m/s

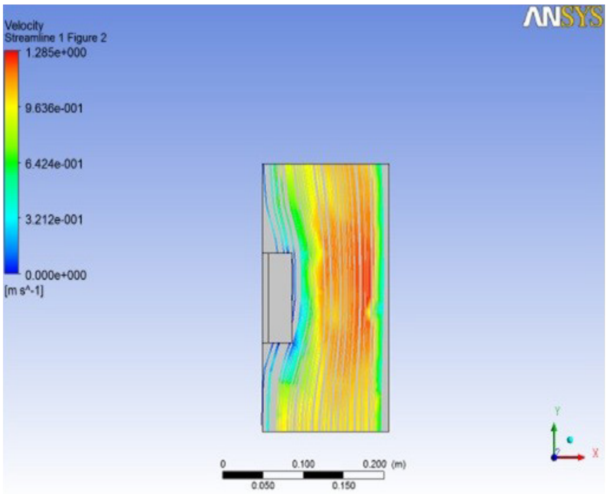

Figure 7.Streamlines for fins array of L=100 mm, H=30mm,v=1.5 m/s

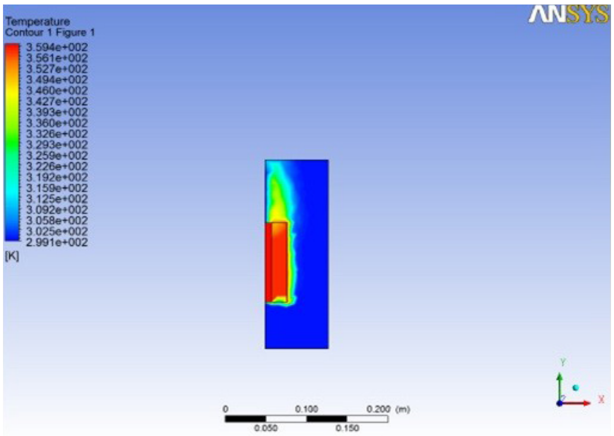

Figure 8.Temperature Contour for fins array 10020, v = 0.5m/s

Variation in Temperature Contour with Velocity

To show the variation of temperature with velocity, the following fin configuration is taken as Fin height, H=20 mm, Fin Length L=100 mm Power input=60W Effect of velocity over temperature distribution and thereby on mixed convection heat transfer rate can be seen through Figure 8 to 10. Temperature distribution in domain and thus temperature contour is studied.

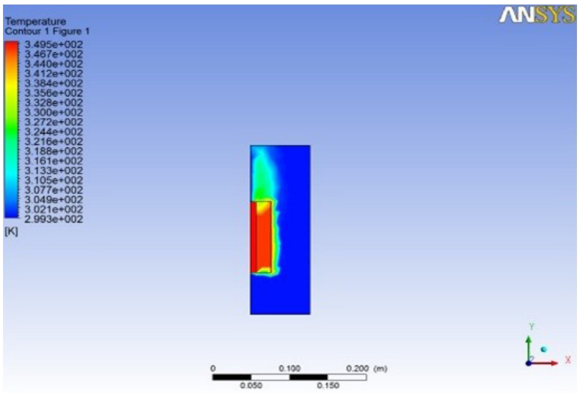

Figure 9.Temperature Contour for fins array 10020, v =1.0 m/s

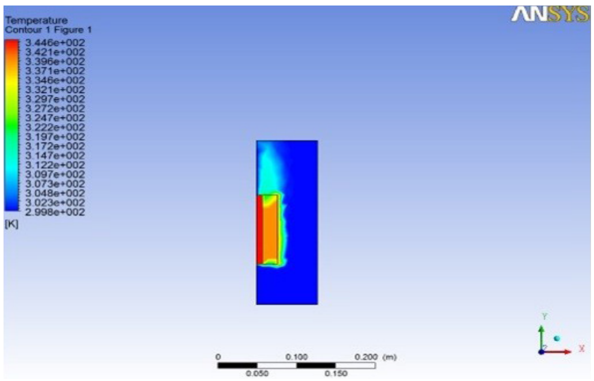

Figure 10.Temperature Contour for fins array 10020, v = 1.5m/s

Result and Discussion

Variation of Nusselt Number with different power input for Array

Figure 11 to 13, shows variation of Nusselt Number with different power input for different array. It can view from the figure that as power input increases Nuseelt number increases for all array configurations. For 0.5 m/s velocity the increment is more for larger fin array to that of smaller one with same aspect ratio. Later on for all velocities Nusselt number slightly decreases as power input decreases. Also Nuseelt number from experiment is near to that of calculated from existing correlation.

Variation of convective heat transfer with different power input for fin array with aspect ratio

Figure 14 to 16, shows variation of convective heat transfer with different power input for fin array with same aspect ratio for different velocities. For all velocities h is more for smaller fin array than larger one with same aspect ratio. This shows smaller fins give better heat transfer rate than larger one for same aspect ratio. Again as the velocity increases h also increases for same power input.

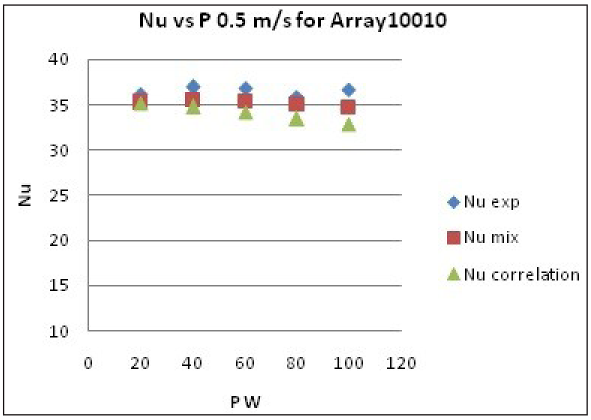

Figure 11.Variation of Nu with different power input for array 10010, v= 0.5 m/s

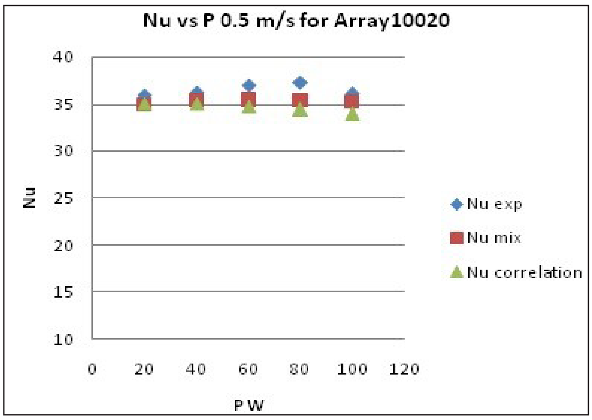

Figure 12.Variation of Nu with different power input for array 10020, v= 0.5 m/s

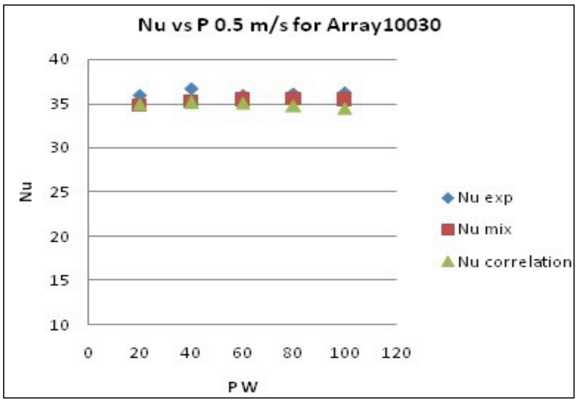

Figure 13.Variation of Nu with different power input for array 10020, v= 0.5 m/s

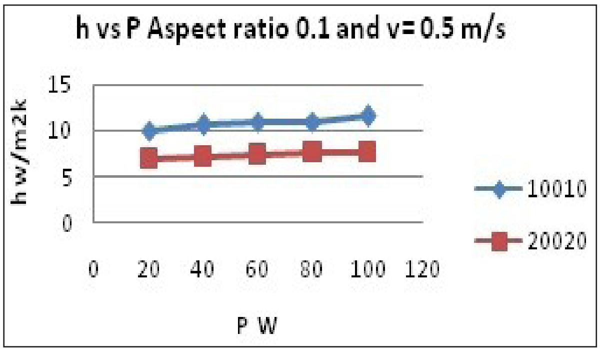

Figure 14.Variation of Nu with different power input for array 10020, v= 0.5 m/s

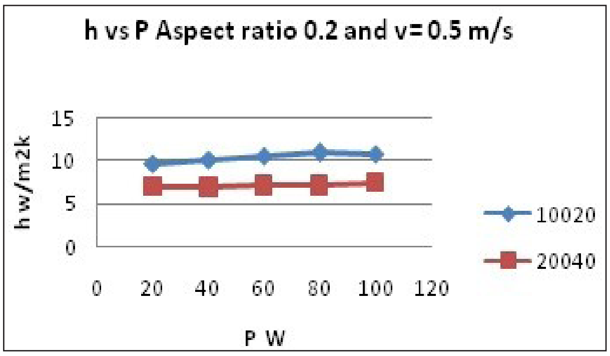

Figure 15.Variation of h with different power input for fin array with aspect ratio 0.2, v=0.5 m/s

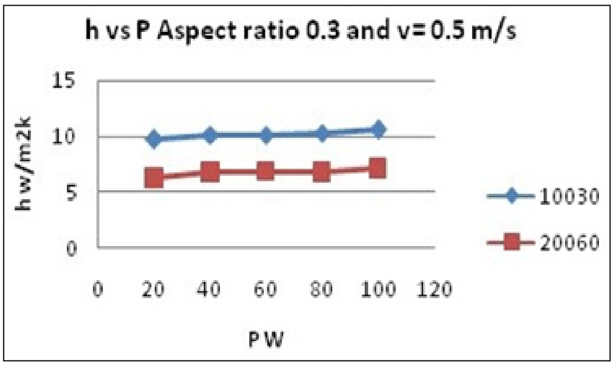

Figure 16.Variation of h with different power input for fin array with aspect ratio 0.3, v=0.5 m/s

Variation of convection heat transfer rate with fin height for different fin length

The convection heat transfer rate from the fin arrays are plot- ted as a function of fin height for different power inputs. From the figure 17 to 18, it can be observed that for every power inputand fin length combination, the convection heat transfer rate from the fin array increases with the increase in the fin height. With an increase in fin height, the total heat dissipation area also increases. Since the convection heat transfer rate directly related to the surface area in contact with air, increasing fin height increases the total heat dissipation.

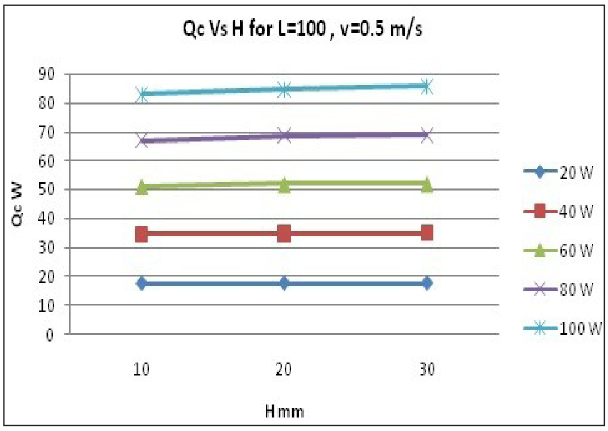

Figure 17.Variation of convection heat transfer rate with fin height for fin length L=100 mm, v=0.5 m/s

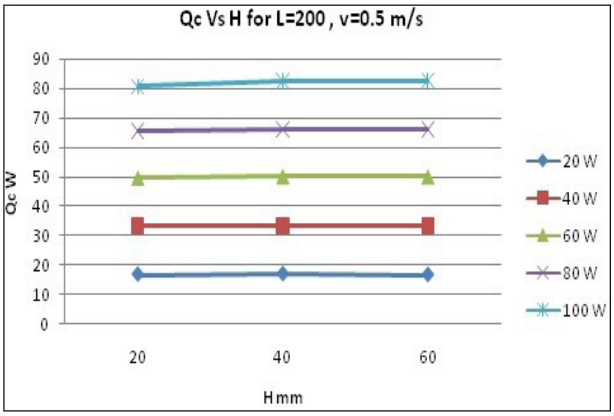

Figure 18.Variation of convection heat transfer rate with fin height for fin length L=200 mm, v=0.5 m/s

Variation convective heat transfer rate with different ve-locity for different aspect ratio array

In figure 19 to 20 convection heat transfer rates are plotted as function velocity for s different aspect ratio fin array.It can be seen from figures that as the velocity increases Qc increases because more heat is dissipated to air. Also Qc is more for smaller length fin array than larger fin length with same aspect ratio. This shows smaller length fin array gives better heat transfer rate than larger one with same aspect ratio.

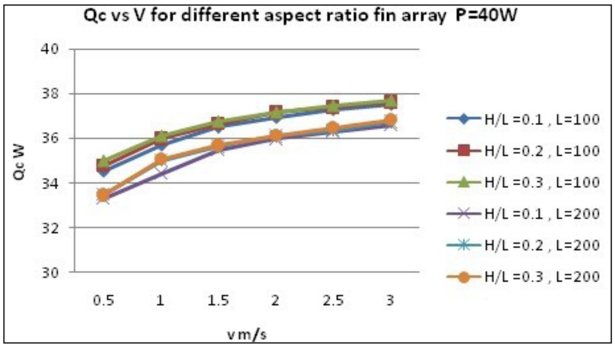

Figure 19.Variation of convection heat transfer rate with different velocity for different aspect ratio array, P=40 W

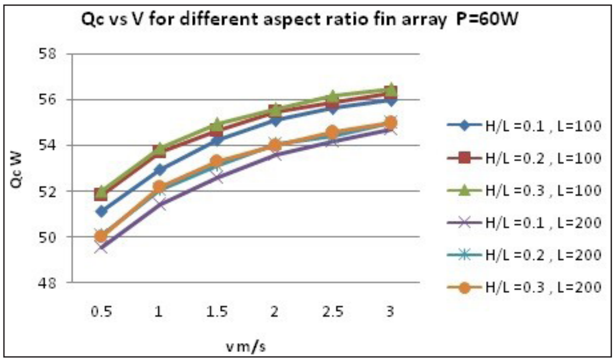

Figure 20.Variation of convection heat transfer rate with different velocity for different aspect ratio array, P- 60 W

Correlation of Nusselt Number in Mixed Convection

One of the objectives of the project is to find correlation for Nusselt number in mixed convection. In mixed or low velocity convection Grashofs number, Reynolds number both affects the heat transfer rate. Therefore Nusselt number correlation founds with these numbers from experimental Nusselt number values.

$Nu = Re^{(0.3625)} * Gr^{(0.045)}$ for $Re < 10000$

$Nu = Re^{(0.4025)} * Gr^{(0.025)}$ for $10000 < Re < 20000$

$Nu = Re^{(0.415)} * Gr^{(0.02)}$ for $20000 < Re < 30000$

These correlations value are accurate with an error of $\pm$ 5% in given range of Re.

Conclusion

Experimental and CFD analysis is conducted in order to establish effect of geometrical fin parameters for natural and mixed convection heat transfer from vertical rectangular fin arrays .Steady state natural and mixed convection heat transfer from vertical fin arrays is experimentally investigated. 6 dif- ferent fin configurations are analyzed. From earlier literature study it is found that for 100 mm fin length optimum fin spacing is 7.5 and for 200 mm is 10 mm. To validate our experiment, results are compared with existing correlations. It is found that experimental Nusselt number is in good agreement with that of calculated from exiting correlations for mixed convection:

- It is found that convection heat transfer rate depends on fin height and fin length as predicted.
- For a given fin spacing, the convection heat transfer rate from fins increases with fin height.
- For a given fin spacing, the convection heat transfer rate from fins increases with fin length.
- For a same aspect ratio it is found that fin array with smaller dimension fin array has more heat transfer rate than larger one.
- For all velocities convective heat transfer coefficient is more for smaller fin array than larger one with same aspect ratio.

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