# Effect of Percentage of Perforation on the Natural Convection Heat Transfer from a Fin Array

**Gaurav A. Chaudhari, Indrajit N. Wankhede, Mitesh H. Patil**

***Abstract*— In this study, the steady-state natural convection heat transfer from vertical rectangular fins extending perpendicularly from vertical rectangular base was investigated experimentally. The effects of perforations and base-to-ambient temperature difference on the heat transfer performance of fin arrays were observed and optimum value of perforation is suggested. The experimental set-up was employed during experiments in order to take measurements from 4 different fin configurations. The base and the ambient temperatures were measured in order to evaluate the heat transfer rate from fin arrays.**

**The results of experiments have shown that the convection heat transfer rate from fin arrays depends on all percentage of perforation and base-to-ambient temperature difference. The effect of these parameters was also examined, and it was realized that for a given base-to-ambient temperature difference, the convective heat transfer rate from the fin array is maximum for 30% perforated fin array.**

***Index Terms*— Fins, Natural convection, Perforation, Steady state**

## I. Introduction

While we are operating many of the engineering systems there is a major problem of generation of heat. This generated heat can play a role in the failure of the system due to overheating. This generated heat in a system should be dissipated in the surrounding so as to maintain the system at its recommended working temperatures. Due to this the system can be operated with good efficiency. In modern electronic systems the packaging density of circuits is much higher; therefore the problem of heating is serious in this case. In order to overcome this problem, thermal systems with effective emitters as fins are desirable [1].

If we want to achieve the required heat dissipation rate, with the least amount of material, the combination of geometry and orientation of the fins should be optimum.

The basic equation of convection heat losses is given by:

$$Q_c = hA\Delta T \tag{1}$$

As seen from Eq. (1), the rate of heat dissipation from the surface can be enhanced either by increasing the heat transfer coefficient, h or by increasing the surface area, A. The value of h can be enhanced by using proper conditions of forced flow over the required surface. Although the forced convection is efficient, it requires an extra space for fan or blower which interns causes the enhancement in initial and maintenance cost. Therefore, forced convection is not always selectable. For increasing the heat transfer, it is more preferable, convenient and easy to use the extended surfaces. To increase the heat transfer area, it is very efficient to use the fins over the surface. As a result of this the rate of heat transfer will be enhanced. However, if the number of fins and the spacing between two fins are not properly designed then the heat transfer rate can be decreased also. Although adding more number of fins increases the surface area, they may resist the air flow and cause boundary layer interferences which affect the heat transfer adversely [3].

The experimental findings related to the thermal performances of rectangular fins were reported in literature [1-7, 8-14, 16]. In this study, the steady-state natural convection heat transfer from a perforated vertical rectangular fin configurations protruding from a vertical base will be investigated experimentally.

The previous studies conducted about the heat transfer performance of rectangular fin arrays. In their experiments, four sets of fin arrays were checked for investigation of heat transfer performance with natural convection. The fin arrays were placed with three types of base, vertical, inclined at 45 degrees and horizontal. From experimental data, it was found that heat transfer rates obtained from the tests with vertical arrays are 10 to 30 per cent below than that of similarly spaced parallel plates. For the 45-degree inclined base position, heat transfer rates were 5 to 20 per cent below from the values with respect to vertical position. [1]

An experimental finding of the rate of heat transfer from an array of vertical rectangular fins on vertical rectangular base has been reported by Leung, Probert and Shilston [3].

To investigate the average heat transfer coefficient, Harahap and McManus [5] observed the flow field of horizontally based rectangular fin arrays for natural convection heat transfer.

D. Abdullah H. AlEssa-The study considered the gain in fin area and of heat transfer coefficients due to perforations. The results showed that, for certain values of rectangular perforation dimension, the perforations lead to an augmentation in heat dissipation of the perforated fin over that of the equivalent solid one. For the fin considered in this study, both perforated and non-perforated, the fin tip is a vertical surface for which the Nusselt number is given by

$$Nu_t = 0.5\left\{\left[\frac{2.8}{ln\left(1+\frac{2.8}{0.515Ra^{0.25}}\right)}\right]^6 + [0.103Ra^{0.333}]^6\right\}^{\frac{1}{6}} \tag{2}$$

$$h_t = Nu_t . K_{air}/L_c \tag{3}$$

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Where,

$$L_c = L \cdot \frac{t}{(2L+2t)} \tag{4}$$

## II. RESOURCES REQUIRED

**Material resources**

Aluminum Blocks, Aerated Concrete, Base metal plate, Heater, Heatlon Insulation.

**Electrical resources**

Dimmer stat, Voltmeter, Ammeter, Electrical wire

**Temperature measuring resources**

K type thermocouples, Temperature indicator

## III. TEST SET UP AND EXPERIMENTAL METHODOLOGY

### 1. Test set up

Experimental set-up will be constructed to test vertically based rectangular fin arrays. The experimental set-up will be developed similar to those used in Refs [13, 14].

The experimental set-up will 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 frames of set-up will be filled with Heatlon in order to maintain the insulation of the aerated concrete cases. The front surfaces of the frames will be covered with metal plates, which have rectangular holes at the centre, so that fin arrays are placed into the cases through these holes. The heaters will be placed into these cases. 3 mm thick aluminium plates will be located between the heaters and base-plates in order to distribute the power input evenly.

Fig 1.1Experimental Setup

Thus, a more uniform temperature distribution at the base of the fin array can be achieved. The geometry of the fin arrays is illustrated in Figure 1.2

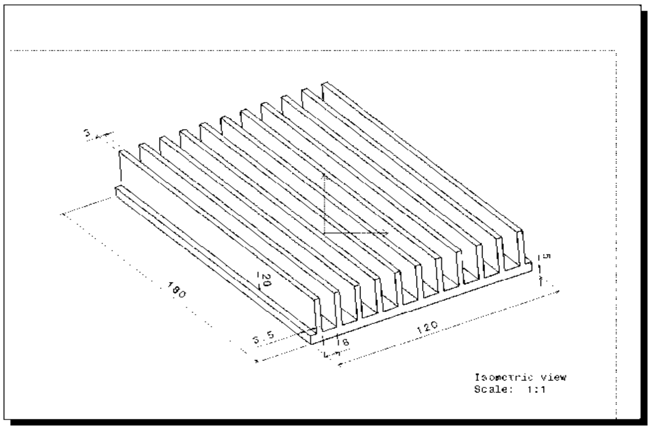


Fig 1.2 Fin Array Geometry

### 2 Test Procedure

In order to be able to determine the convective heat transfer performances of the fin arrays under steady-state conditions, total heat losses from the set-ups should be calculated first. Hence, the experimental set-ups will be calibrated and the calibration method will be verified before starting experiments.

### 2.1 Calibration of Setup:

For the calibration of the set-up, the rate of heat transfer from the heated base-plate should be determined. Since the experimental set-ups had similar properties except dimensions, the solution procedures of heat conduction equations were same. Using a procedure similar to that proposed in Ref. [13, 14], the heat conduction equation was solved with the method of integral transform technique. For the heat transfer rate from heated base-plate, the following equation is obtained:

$$\frac{\dot{Q}_{out}}{\dot{Q}} = \zeta - \tau \, \frac{T_w - T_a}{\dot{Q}} \tag{5}$$

where $\dot{Q}$ is the power input to the heater, Ta is the ambient temperature, Tw is the average surface temperature of the heated plate, $\dot{Q}_{out}$ is the total heat transfer rate from the heated plate and, ζ and τ (W/K) are constants that depend on the geometry and the average thermal conductivity of the system.

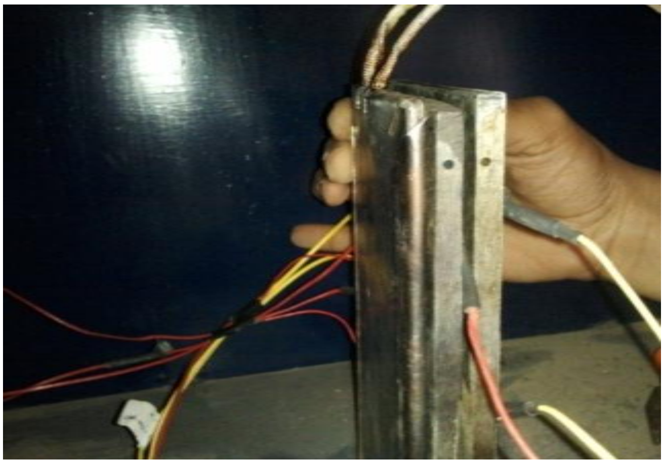

Fig 2.1 Calibration of Plate

At steady state, the heat transfer between the plates will have the following form:

$$\dot{Q}_{out}=\frac{k}{d}A(T_1-T_2)+\sigma\left(\frac{\varepsilon}{2-\varepsilon}\right)A\left(T_1^4-T_2^4\right) \qquad (6)$$

### 2.2 Verification of Calibration Method

In order to determine the validity of the used calibration equations and method, a set of experiments will be conducted on a vertical plate for each set-up. Using the experimental results, experimentally and theoretically estimated Nusselt numbers were will be compared for verification.

For each of the set-ups, predetermined power inputs will be supplied to heat the vertical plates. Under steady-state conditions, the vertical plate temperatures, Tw, the ambient temperatures, Ta and the power inputs, will be measured.

For each power input, the total heat transfer rate from vertical plate will be calculated by substituting the measured data into Eqs. (3.2). Then, the radiation heat transfer rate from vertical plate was estimated by assuming the environment as a blackbody at ambient temperature Ta as:

$$(\dot{Q}_o)_r=\sigma\varepsilon A\left(T_w^4-T_a^4\right) \qquad (7)$$

The convection heat transfer will be calculated for vertical plate as

$$(\dot{Q}_o)_c=\dot{Q}_o-(\dot{Q}_o)_r \qquad (8)$$

Therefore heat transfer coefficient based on the surface area of the vertical plate will be calculated as

$$h=\frac{(\dot{Q}_o)_c}{A\ .\ (T_w-T_a)} \qquad (9)$$

Rayleigh number and Nussult number will be then evaluated as

$$R_a=\frac{g\beta\Delta TL^3\ (T_w-T_a)}{\nu\ \alpha} \qquad (10)$$

$$N_{u\ exp.}=\frac{h_{exp.}\ L}{k} \qquad (11)$$

After determining experimental Nusselt numbers for the set-ups, they will be compared with the Nusselt numbers evaluated by using available vertical plate correlations from literature

The correlations which will be utilized for the comparison are:

1. Churchill and Chu's first relation (for laminar and turbulent flows) [19]:

$$N_{u\ th}=\left\{0.825+\frac{0.387\ .\ (R_a)^{\frac{1}{6}}}{\left[1+\left(\frac{0.492}{Pr}\right)^{\frac{9}{16}}\right]^{\frac{8}{27}}}\right\}^2 \quad \text{for} \quad 10^{-1}<R_a<10^{12} \qquad (12)$$

2. Churchill and Chu's second relation (for laminar flows only) [19]:

$$N_{u\ th}=0.68+\frac{0.67\ .\ (R_a)^{\frac{1}{4}}}{\left[1+\left(\frac{0.492}{Pr}\right)^{\frac{9}{16}}\right]^{\frac{4}{9}}} \quad \text{for } 10^{-1}<R_a<10^{9} \qquad (13)$$

3. Mc. Adams Relation [19]:

$$N_{u\ th}=0.59\ .(R_a)^{1/4} \qquad (14)$$

## IV. RESULTS AND DISCUSSION

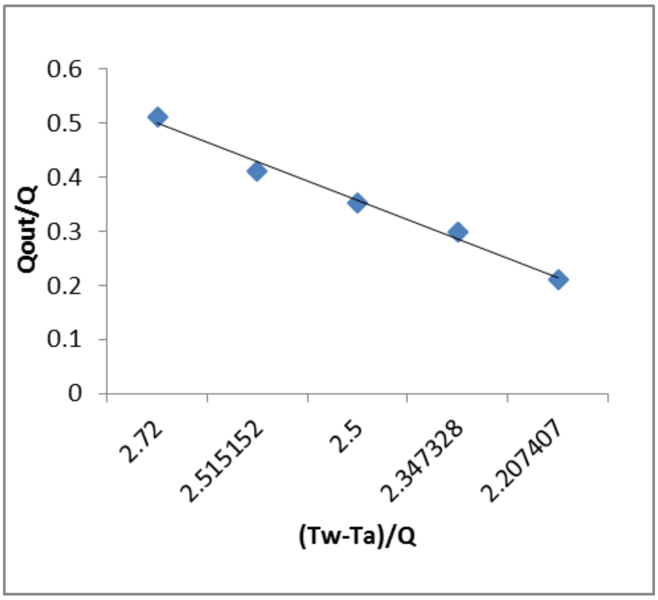


Fig 3.1 Calibration Curve for Set-up

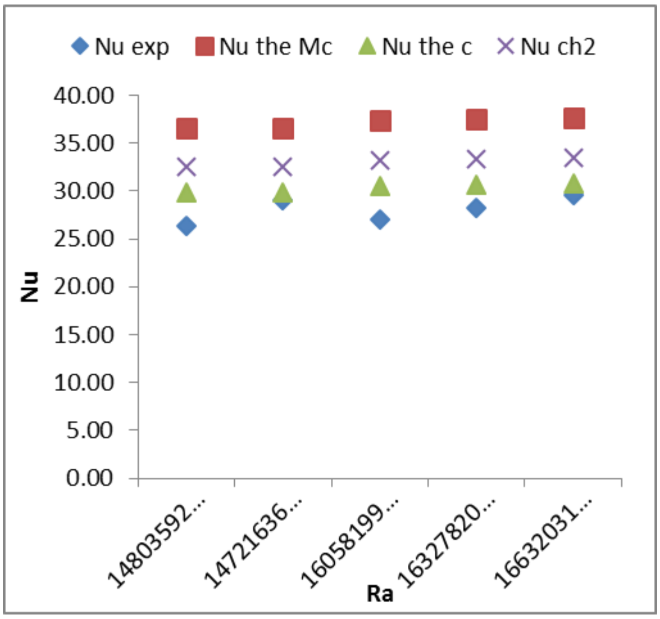


Fig. 3.2 Comparison of Experimental and Theoretical Nusselt Numbers

**Examination of Figure reveals that the experimental data are in a good agreement with the correlations. The average relative errors are less than 20 % for Churchill and Chu's and McAdams' correlations. These results indicate the validity of the experimental set-ups, the experimental procedure and the calibration method.**

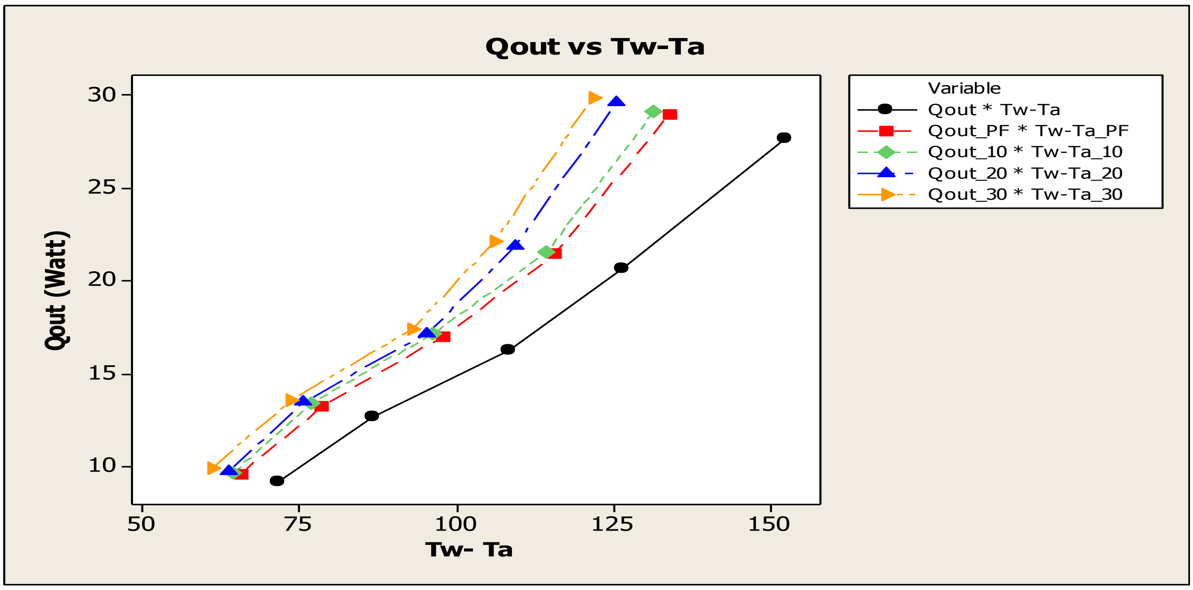


Fig 3.3Variation of Total Heat Transfer with Base-to-Ambient Temperature Difference

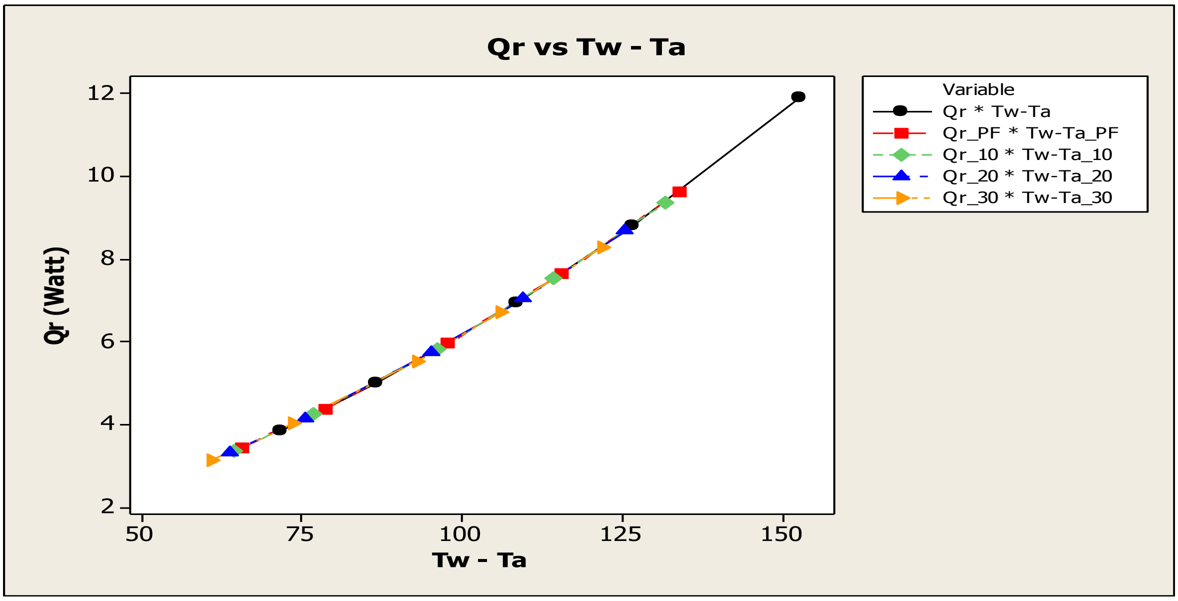


Fig 3.4Variation of Radiation Heat Transfer Rate with Base-to-Ambient Temperature Difference

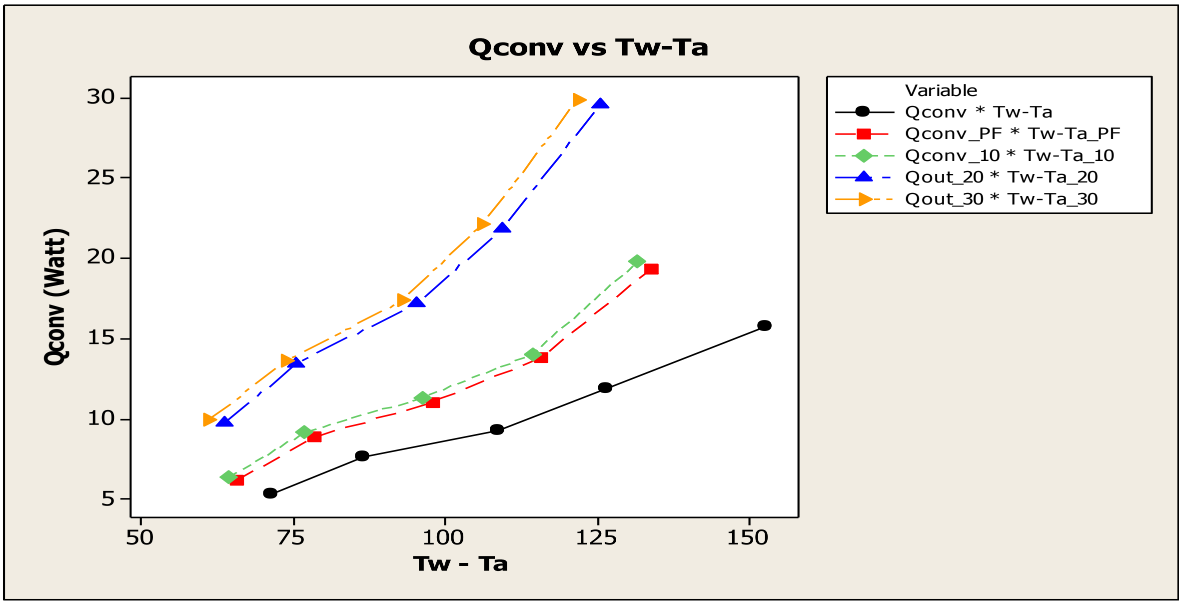


Fig 3.5 Variation of Convection Heat Transfer Rate with Base-to-Ambient Temperature Difference

## V. CONCLUSIONS

This study shows that for certain percentage of perforation, the perforated fin enhances the heat transfer. Total heat transfer rate from fin arrays depends on base-to-ambient temperature difference and percentage of perforation. As this temperature difference increases, total heat transfer increases. For the same base-to-ambient temperature difference, total heat transfer is Minimum for vertical flat plate while it increases for plane finned surface and goes on increasing as percentage of perforation increases. The total heat transfer is Maximum for 30% perforated fin array

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