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Experimental Investigation of Natural Convection Heat Transfer from Perforated Vertical Plate

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Abstract

Because of the capital and operating cost of the enhancement devices the active heat transfer enhancement techniques fails to found commercial interest. Most of the passive techniques employ special surface geometry or additives for enhancement which are fluid i.e No direct application of external power. The rate of heat transfer is difficult to increase by increasing heat transfer coefficient or by rising the temperature between the surface and surrounding fluid, the fins are commonly used.

One of the way to improve the effectiveness is to make the perforation in the plate or fins. By reducing the thermal boundary layer thickness continue researches are going on to improve the heat transfer rate. Heat sink on which perforation is done is one way to improve the heat transfer rate. An experimental presentation is done to investigate by natural convection heat transfer rate in perforated vertical plate. Shape modifications by removing some parts from fins to form cavities, slots, groves, holes and channels to rise heat transfer areas through the fin body and heat transfer coefficient. This experimental workconcentrates on vertical plate which is perforated with circular diameter. The results of perforated plates arecompare with plain vertical heated plate.

Keyword—*Vertical perforated plate, Natural Convection, Design and Development of Experimental Setup, Heat transfer enhancement techniques.*

1. Introduction

To avoid the damaging effects of burning or overheating the removal of excessive heat is essential. In thermal engineering enhancement of heat transfer is an important part. By increasing the heat transfer area of the surface heat transfer from surfaces may in general enhance by rising the heat transfer coefficient between a surface and surrounding.

Generally, in the base of the plate, fins are attached in the form of extended surfaces.It has been quite common to use fins as heat transfer enhancement devices. As the extended surface technology grows continuously, including fins made of various materials the new design ideas have emerged,. Due to increase demand for lightweight, economical fins,compact fin and the optimization of the fin size is of great importance. Therefore, fins have to be designed to achieve higher heat removal rate with minimum material expenditure, taken into account, and also with the ease ofthe fin shape manufacturing. Number of studies have been conducted on fin shapeoptimization. Other studies have introduced shape modifications by removing some materials from fins to make holes, slot, grooves, cavities or the channels through the fin body to increase heat transfer surface area and the heat transfer coefficient factor. The use of the rough or interrupted surfaces of different configurations is one of the popular heat transfer augmentation technique.

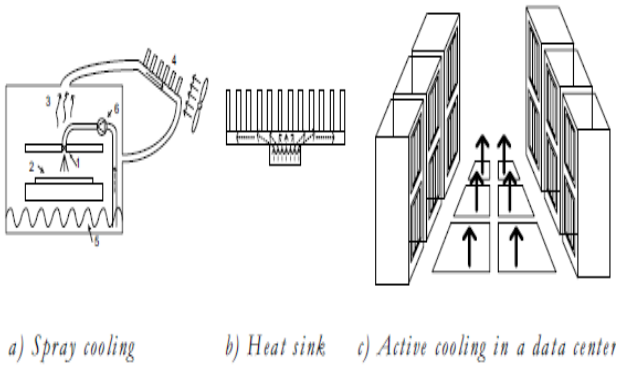

Fig1. Several ways to increase the cooling in an electronic system.

In natural convection, the concept of heat transfer through perforated material is one of the method of improving the heat transfer characteristic. Large number of research is going on to improve its effectiveness by reducing the thermal boundary layer thickness and increasing the heat transfer surface area.One of the cheapest and easiest way to dissipate unwanted heat and it has been commonly used for many engineering applications successfully is to use fins. Because of low production costs and high effectiveness rectangular fins are the most popular fin type. Although rectangular fins can be used in two seperate orientations as vertical and horizontal, vertical orientation since it is more effective than the horizontal one is use more widely. Two modes of heat transfer that takes place while dissipating heat from fins to surrounding are convection and radiation heat transfers. Since all of

the considered fin configurations are made of material such as aluminum alloys, which has low emissivity values, lower radiation heat transfer values. Therefore, the dominant heat transfer is convection heat transfer mode while dissipating heat from fins. Rate of heat dissipation from a fin configuration by convection heat transfer depends on the heat transfer coefficient and the surface area of the fins. It is also possible to rise the heat transfer coefficient, h by forcing the fluid to flow over the fins by using fans. But this is expensive and also requires more volume for the fans to operate. Due to which, sometimes the designer has to depend on natural convection heat transfer for dissipating unwanted heat from the fins. The surface area of the fins can also be increased by installing more fins to the base material in order to increase the total heat transfer from the fins. But the number of the fins should be optimized because it should be noted that installing more fins also reduces the distance between the adjacent fins. This may cause opposing to the air flow and boundary layer interference which in return reduces the heat transfer coefficient.

Although their thermal effectiveness and their rectangular fin geometries are investigated extensively in literature, most of the research are carried for limited range of fin configurations. Moreover, even though there are large number of the experimental studies on performance of fin configurations, the number of numerical studies are lacking. In this study, with the help of a commercial CFD program the heat transfer performances of wide range of vertical fin configurations are investigated. To further examine the performance of vertical fin arrays two experimental studies from the literature are selected for investigation in order to compare the results. On computer configuration the runtimes of the analyses iscarried on which varies from one fin configuration to another. The average run time is 25 minutes for the first computer configuration and 40 minutes for the second computer configuration. Over vertical flat plate the results are first checked with theoretical results of natural convection and then in order to guarantee the validity of the solution procedure with experimental studies.To determine the performance of fin configuration, is the main objective of this work to demonstrate a convenient CFD based solution. Since, on every possible fin it is not possible to perform an experiment, to predict the effectiveness a CFD solution can be used.

2. Literature review

Extensively both analytically and preliminary natural convection from finned surfaces has been investigated in literature. Numerical studies were also carried out to find a convenient model for the phenomena. Furthermore, to find the optimum fin structure for maximum heat transfer rate in all of the previous studies, different geometries and configurations were studied to find the optimum fin structure for maximum heat transfer rate.

Jong et. al. [2016] had done advanced experimental studies regarding the natural convection heat transfer for cooling the vertical cylinders with inclined plate fins. The optimal thermal resistance of cylinders with inclined plate fins and convectional radial plate fins are compared. It is found that the optimal therml resistance of the cylinder with inclined fins is 30% lower than that of the cylinder with radial plate fins. Therefore, inclined plate fins have potential for use in cooling equipment in various thermal systems.

Nazia et. al. [2015] have conducted one of the earliest studies about natural convection heat transfer from vertical plate. It is found that the perforated surface has a significant effect on the enhancement of heat transfer and Nusselt Number. The heat transfer rate is higher as perforation of plate is increases. For a given heat input Nusselt number and heat flux, the average plate temperatures at higher perforated diameter are higher than those from lower perforated diameter. However, heat transfer rate is increases as the heat supplied to plate is increases Therefore, the surface temperature of plate tends to increase as the heat supplied to plate increases. The surface temperature continues to increase with Nusselt number. The measured heat transfer coefficient obtained from the perforated surface at higher diameter is 1.66 times higher than those from the plane surfaces.

Hagote et. al. [2014]carriedout numerically by means of CFD.They used CFD and measurement technique such as LDA (Laser Doppler Anemometry). They performed an investigal study on heat transfer from perpendicularly placed fin arrays produced from an aluminium alloy. Observation was that for different configurations the highest heat transfer rate from the fin arrays was regulated out at the fin spacing value of 10 mm.Filtzroy carried on a investigation in manner to find the optimum fin type for the supreme rate of natural convective heat transfer from erect fins in the laminar flow regime. The following correlation which narrates the ratio of average heat transfer coefficient based on fin type to up-down heat transfer heat coefficient was recommended:

$$\frac{h_s}{h_b} = \frac{1.68}{24}.[\frac{S}{H}.(Gr_s.P_r)^{0.25}]^3.\left[1\exp(\frac{-24}{1.68*[\frac{S}{H}.(Gr_s.P_r)^{0.25}]^3})\right]$$

where $h_s$stands for average heat transfer coefficient based on fin spacing andh_b stands for upward heat transfer heat coefficient.

Abd Y.H. [2007] has done intellectual study for a three dimensional natural convection heat transfer from an isothermal straight and inclined heated

square plates (with and without circular hole) which involved the mathematical solution of the transient Navier-Stokes and energy equations by implementing Finite Different Method (F.D.M.)

Shafei et. al. [2010] has done developmental study conducted on optimizing fin shapes. Various studies have introduced shape conditioning by removing some material from fins to make cavities, holes, slots, grooves, or channels through the fin body to increase the heat transfer area and/or the heat transfer coefficient.

Dr. Aziz et. al. [2008] studied the natural convection heat transfer from perforated fins. The temperature dissipating was examined for an array of rectangular fins (15 fins) with uniform cross-sectional area (100 x 270 mm) plunge with different vertical body perforations that extend through the fin thickness. The patterns of perforations include 18 circular perforations (holes). Experiments were carried out in an experimental facility that was specifically design and constructed for this purpose. The heat transfer rate and the coefficient of heat transfer increases with perforation diameter increased.

Al Essa et.al. [2009] experimentally investigated the enhancement of natural convection heat transfer from a horizontal rectangular fin embedded with rectangular perforations of aspect ratio of two. Results of perforated fin related with its comparable solid one. An experimental study was carried out for geometrical dimensions of the fin and the perforations. The study investigated the gain in fin area and of heat transfer coefficients due to perforations. They concluded that, values of rectangular perforation dimension, the perforated fin enhances heat transfer. The magnitude of enhancement is proportional to the fin thickness and its thermal conductivity.

Shaeri et. al. [2009] studied the turbulent fluid flow and convection heat transfer around an array of rectangular solid with different number of perforation and different size. Experiments were conducted for the range of Reynolds no. from 2000 to 5000 based on fin thickness and pr= 0.71. This Study is aimed mainly at examining the extent transfer enhancement from vertical rectangular fins under natural convection. Conditions as a result of introducing body modification (perforations) to the fin body. Perforated fins can be used to increase the heat transfer coefficient and effective heat transfer area. The change in magnitude of the surface area depends on the geometry of the perforations.

Abdullah et. al. [2009] studied the natural convection heat transfer from a fin by triangular perforations of bases parallel and toward its base. 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.

Jassem et. al. [2013] performed an experimental study submitted to investigate the heat transfer by natural convection in a rectangular perforated fin plates. Five fins used in this work first fin non-perforated and others fins perforated by different shapes these fins perforation by different shapes (circle, square, triangle, and hexagon) but these perforations have the same cross section area (113 mm^2). These perforations distributed on 3 columns and 6 rows. Experiments produced through in an experimental facility that was specifically design and constructed for this purpose. The results show that the drop in the temperature of the non-perforated fin from 72 to 57^oC while the temperature drop in perforated fins, at the same power supplied (126W) was (72-52 ^oC), (72-51.5^oC), (72-50^oC) and (72-48^oC) for shapes (hexagonal, square, circular and triangular) respectively. The largest value of RAF at triangular perforation and the smaller value occurred in circular perforation. Also, triangular perforation gives best values of heat transfer coefficient and then the circular, square, hexagonal, and non-perforation respectively.

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.

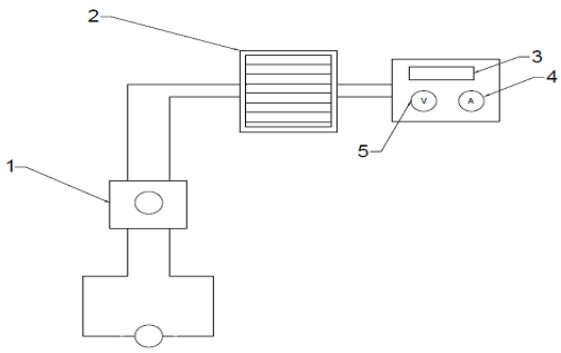

Fig2. Experimental Setup

1.Dimmerstat

2. Fin Array

3. Temperature Indicator

4. Ammeter

5. Voltmeter

Jonas and Smith conducted an experimental study in order to find the optimum fin spacing for horizontally placed rectangular fin arrays. An interferometer was used to measure local temperature gradients. The measuredtemperature gradients were used to determine the corresponding heat transfer coefficients. It was inferred from the results that the fin spacing, S has a significant effect on the heat transfer coefficients. The following correlations were also obtained:

$$Nu_s = 6.7 * 10^{-4} * Gr_s * Pr * \left[1 - \exp\left(\frac{7460}{Gr_s * Pr}\right)^{0.44}\right]^{1.7} \quad forS < 2\ inches$$

$$Nu_s = 0.54 * (Gr_s * \mathrm{Pr})^{0.25} forS > 2\ inches$$

where Pr stands for Prandtl number, Nu_s stands for Nusselt number and Gr_s stands for Grashof number based on fin spacing.

3. Numerical Analysis:

For the free convection the model of Churchill and Usagi provides good representation. The essential components of heat transfer by convection mechanism are given in Newton's law of cooling:

$$Q = h.A.\Delta T \qquad (1)$$

where, Q is the heat transfer rate between the exposed surface A of the vertical plate and the fluid. ΔT is the difference between the temperature of plate and surrounding. h is the convection coefficient and also the quantity we want to calculate.

Here, we know the weight and the specific heat of aluminium plate. So, we can use the following relation of the heat transfer rate with mass and specific heat.

$$Q = m.\mathrm{C}s.\Delta T$$

where, m is the mass and $C_s$is the specific heat.

The occurrence of turbulent flow is usually correlated with Rayleigh number, which is simply the product of the Grashof and Prandtl numbers:

$$Ra = Gr.Pr \qquad (2)$$

Grashof number is represented by the expression in (3)

$$Gr = \frac{L^3.\mathrm{g}.\beta.\rho.\Delta T}{\mu^2} = \frac{L^3.\mathrm{g}.\beta.\Delta T}{v^2} \qquad (3)$$

where L is the characteristic length, g is gravity, $\boldsymbol{\rho}$ is the density of the fluid, $\boldsymbol{\beta}$ is the thermal expansion coefficient, $\boldsymbol{\mu}$ is the dynamic viscosity, v is the kinematic viscosity.

All characteristics of fluid are evaluated at film temperature

$$\mathrm{T} = \frac{\mathrm{Tw+T\infty}}{2} \qquad (4)$$

For ideal gases ($\boldsymbol{\beta}$=1/T_f), while non-ideal gases the the expansion property to coefficient must be obtain from appropriate property tables.

Prandtl number is the ratio of molecular transport properties, the kinematic viscosity v which affects the velocity profile and the thermal diffusivity α, which affects the temperature profile.

$$\alpha = \frac{\mathrm{k}}{\rho.\mathrm{Cp}}$$

$$v = \frac{\mu}{\rho}$$

$$\mathrm{Pr} = \frac{\mathrm{v}}{\alpha} = \frac{\mu.\mathrm{Cp}}{k}$$

where k(W/mK) is the fluid thermal conductivity, C_p (J/KgK) is the fluid specific heat, $\boldsymbol{\mu}$(Pa.s) is the dynamic viscosity.

Nusselt number is used directly to evaluate the convection coefficient according to (5):

$$Nu = \frac{\mathrm{h.L}}{\mathrm{k}}$$

In equation (5) the bar above quantities signifies an average value.

A few comments regarding the characteristic length are necessary. For a vertical plate the characteristic length is the height H of the plate (for natural convection).

Natural convection originates from a thermal instability: warmer air moves upward while the cooler, heavier air moves in the downward direction. A different type of instabilities may also arise on the vertical plate: hydrodynamic instabilities that relate to a transition between a laminar flow to a turbulent flow. Since, we have to way to characterize the hydrodynamic character of the flow in order to choose the most appropriate formulas for the calculation of the heat transfer coefficients.

The coefficient of the convection is obtain from a relation of the form (6):

$$Nu = \frac{\mathrm{h.L}}{\mathrm{k}} = \mathrm{C.Ra}^{\mathrm{n}}$$

where Ra is the Rayleigh number, C and n are coefficients. Typically n=1/4 for laminar flow and n=1/3 for turbulent flow. For turbulent flow Ra is greater than 10^9, for laminar flow Ra is less than 10^9.

The alternate formula that can be used for the entire range of Rayleigh number is presented in equation (7).

Eq(7):

$$Nu = \left\{ 0.825 + \frac{0.387.\mathrm{Ra}^{\frac{1}{6}}}{\left[1+\left(\frac{0.492}{Pr}\right)^{\frac{9}{16}}\right]^{\frac{8}{27}}} \right\}^2$$

For our case no turbulent flow will take place since the plates are separate so we will consider laminar flow equation (7), however a better accuracy is obtained from (8).

Eq(8):

$$Nu = 0.68 + \frac{0.67.\mathrm{Ra}^{\frac{1}{6}}}{\left[1+\left(\frac{0.492}{\mathrm{Pr}}\right)^{\frac{9}{16}}\right]^{\frac{4}{9}}} \qquad \mathrm{Ra} \leq 10^9$$

4. Design And Development Of Experimental Setup

A schematic diagram of the experimental setup is as shown in Fig. The experimental setup consists of the components such as RTD's, Temperature Indicator, Heating element, an arrangement to vary the heat input, Duct, Stand and Hanger etc.

The experimental setup consists of the components such as RTD's,Temperature Indicator, Heating element, an arrangement to vary the heat input, Duct, Stand and Hanger etc.The experimental facility shown in Fig.2 and Fig.3 is to be designed and constructed to investigate the Natural convection heat transfer from perforated heated vertical plate.

Fig3: Actual photograph of experimental setup.

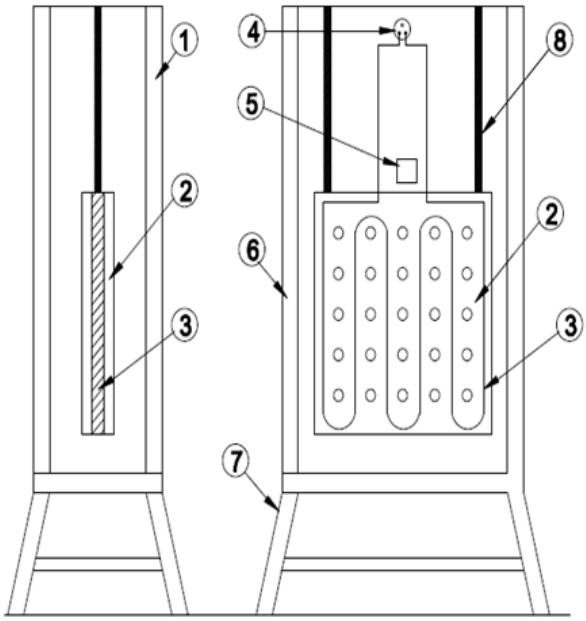

Fig4. Schematic layout experimental setup.

1)Duct, 2) Heated Plate, 3) Heater,
4) Heater Socket,5) ThermocoupleSocket,
6) Acrylic Sheet,7) Stand,8)Hanger

Experimental system to investigate the performance of perforated heated vertical plate under natural convection consist of following components..

- Heating element:-The heat is generated within the heat sink by means of one coil type heating element power of 750W.
- Transformer:- A variable transformer of type 50B with input 220 V and 50-60 Hz and output 0-240 V, 5A and 1 kVA were used to regulate the voltage supplied to the heating elements.
- RTD:- PT 100 Sensor with Temperature Range -20 to 150 ^{o}C.
- Duct:- Plywood duct of size 40 cm x 40 cm x 80 cm.
- Temperature Indicator: Twelve channel temperature indicator compatible with RTD.

5. Observation and experimental readings:

Table 1: Reading of Plane Plate vs Perforated Plate of Φ=5mm.

Sr. No.	Time Interval (minutes)	Plain Plate (^{0}C)	Perforated Plate(Φ=5) (^{0}C)
1	0	120	120
2	5	106.3	104.7
3	10	92.6	90.2
4	15	78.8	76.5
5	20	65.1	63.9
6	25	52	50.2
7	30	36.7	34.6

8	35	27.2	25.9
9	40	24	24

Table 2: Readings of Plane Plate vs Perforated Plate of Φ=10mm.

Sr. No.	Time Interval (minutes)	Plain Plate (^{0}C)	Perforated Plate(Φ=10) (^{0}C)
1	0	120	120
2	5	107.2	102.4
3	10	93.5	89.5
4	15	79	75.1
5	20	66.2	62.2
6	25	53.7	49
7	30	37.6	33.5
8	35	28.1	24.5
9	40	24	24

Table 3: Temperature readings of Plane Plate vs Perforated Plate Φ=15mm.

Sr. No.	Time Interval (minutes)	Plain Plate (^{0}C)	Perforated Plate(Φ=15) (^{0}C)
1	0	120	120
2	5	107.5	100.5
3	10	92.5	87.8
4	15	78.8	73.9
5	20	65	61.1
6	25	52.3	48
7	30	37	31.8
8	35	26.9	24
9	40	24	24

Table 4: Combining all the readings of Plane plate and all perforated plates.

Sr. No.	Time Interval (min)	Plain Plate (^{0}C)	Perforated Plate (Φ=5) (^{0}C)	Perforated Plate (Φ=10) (^{0}C)	Perforated Plate (Φ=15) (^{0}C)
1	0	120	120	120	120
2	5	106.3	104.7	102.4	100.5
3	10	92.6	90.2	89.5	87.8
4	15	78.8	76.5	75.1	73.9
5	20	65.1	63.9	62.2	61.1
6	25	52	50.2	49	48
7	30	36.7	34.6	33.5	31.8
8	35	27.2	25.9	24.5	24
9	40	24	24	24	24

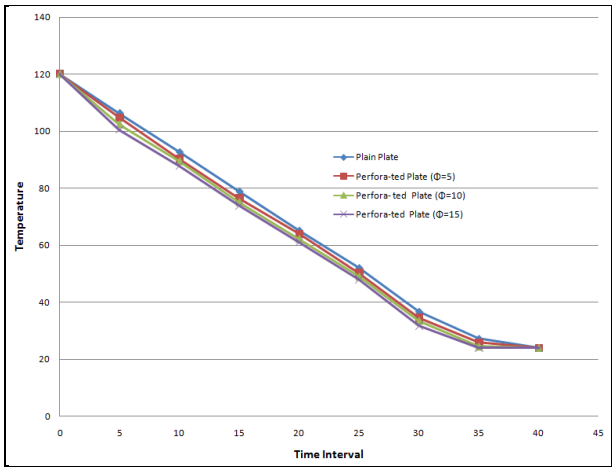

Fig5: Relation between the temperature(^{o}C) and time(minutes)

6. Experimental Results And Discussions

6.1Values calculated from the experimentation carried on various power supplies.

Table 5: For plane vertical plate:

P (W)	Q (J)	h (W/m^2-K)	Nu	ΔT (^{o}C)	β (X10^{-3})
750	2675.2	1238.5	7145.0	96	2.8985
650	2318.5	1157.8	6679.6	89	2.9282
550	1961.8	1038.1	5988.5	84	2.9498
450	1605.1	914.60	5276.5	78	2.9760
350	1248.4	760.08	4385.1	73	2.9985
250	891.74	591.53	3412.7	67	3.0257

Table 6: For perforated vertical plate(Φ=5mm):

P (W)	Q (J)	h (W/m^2-K)	Nu	ΔT (^{o}C)	β (X10^{-3})
750	2360.5	1028.5	5933.9	102	2.8735
650	2045.8	957.08	5521.6	95	2.9027
550	1731.1	854.83	4931.7	90	2.9231
450	1416.3	740.55	4272.4	85	2.9454
350	1101.5	619.72	3575.3	79	2.9713
250	786.83	472.57	2726.3	74	2.9942

Table 7: For perforated vertical plate(Φ=10mm):

P (W)	Q (J)	h (W/m^2-K)	Nu	ΔT (^{o}C)	β (X10^{-3})
750	2136.4	818.55	4722.4	116	2.8161
650	1851.5	761.96	4395.9	108	2.8492
550	1566.7	696.31	4017.2	100	2.8818
450	1281.8	599.69	3459.7	95	2.9027
350	996.99	515.24	2972.5	86	2.9411
250	712.14	411.04	2371.4	77	2.9806

Table 8: For perforated vertical plate(Φ=15mm):

P (W)	Q (J)	h (W/m²-K)	Nu	ΔT (°C)	β (X10⁻³)
750	1711.5	608.56	3510.9	125	2.7812
650	1483.3	573.27	3307.3	115	2.8208
550	1255.1	526.26	3036.1	106	2.8573
450	1026.9	470.53	2714.6	97	2.8943
350	798.73	417.63	2409.4	85	2.9455
250	570.52	320.96	1851.7	79	2.9712

1. Total (6x4=24) solutions has been carried out in above tables to carry out the appropriate values for Q, h, Nu, ΔT, β.
2. This values can be easily understood with the graphical representation.
3. So, we will compare the results graphically between (Q vs Nu), (Q vs h), (Q vs ΔT) and (ΔT vs Nu).

6.2 Graphical representation for the results.

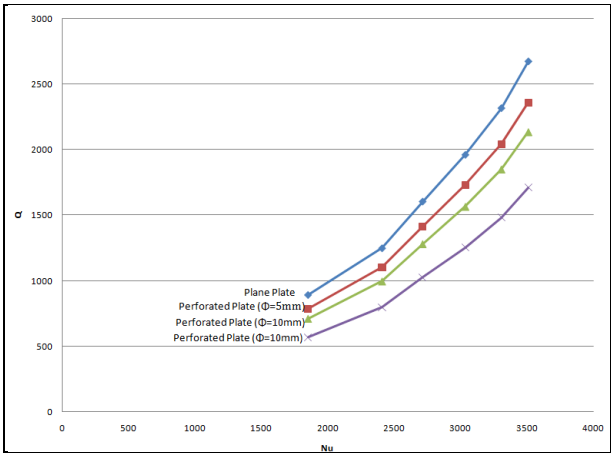

Fig6: Relation between the rate of heat transfer[Q,(J)] and Nusselt number(Nu).

1. The upper blue curve shows the values for the plane plate and the red curve denotes the perforated plate of 5mm diameter holes.
2. The green line represents the perforated plate with diameter 10mm and similarly at the most lower region the violet curve represents the perforated plate with 15mm diameter holes.

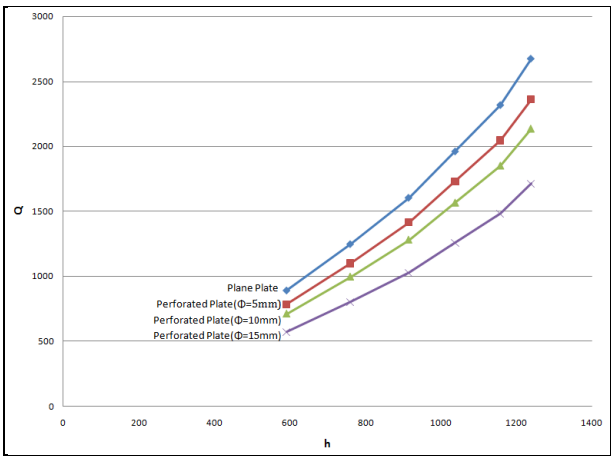

Fig7: Relation between the rate of heat transfer[Q,(J)] and heat transfer coefficient[h,(W/m²-K)].

1. Here we can see the simultaneous rise in the rate of heat transfer with respect to the heat transfer coefficient.
2. The smooth curve for the perforated plate with diameter holes of 15mm are showing certain change in path due to its greater efficiency then that of other plates.

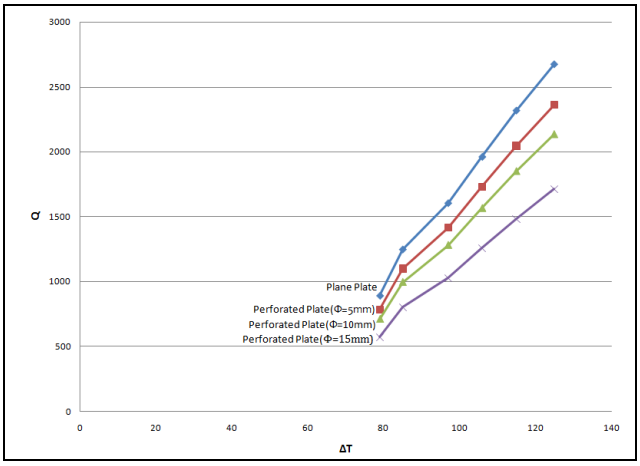

Fig8: Relation between the rate of heat transfer[Q,(J)] and temperature difference[ΔT,(°C)].

1. In the beginning there is the rise in the rate of heat transfer with the slightly rise in the difference in temperature.
2. Here the heat transfer rate is at the peek at the maximum temperature difference.

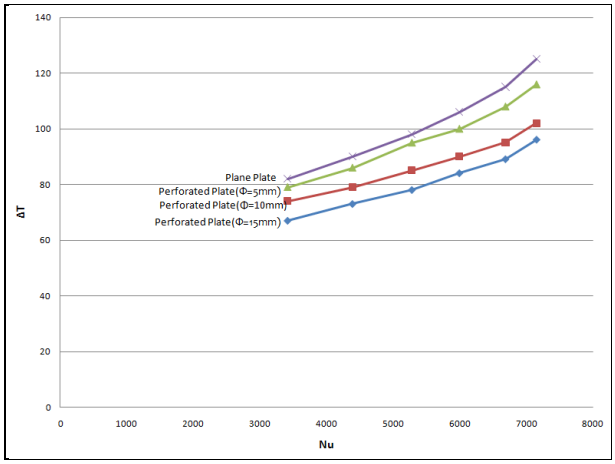

Fig9: Relation between the difference between temperature[ΔT,(°C)] and Nusselt number(Nu).

1. There is the slightly inclination in the curves showing the highlighted regions.
2. The rise in the temperature difference increases the values of the Nusselt number.

6.3. Discussion on experimental results.

1. In order to have a basis for the evaluation of the effects of the perforations, some experiments are carried out for plane plate(without any perforation) to compare with the perforated plates of various diameters.

2. The data is collected on six loads for four plates. One plane plate and other three with varying diameters of 5mm, 10mm, 15mm.

3. With the help of numerical analytical methods the overall 24 solutions has been carried out to get the comparative values of Q, *h,* Nu, β, ΔT.

4. The performance of heat transfer rate(Q) by varying heat transfer coefficient(*h*), Nusselt number(Nu), temperature difference(ΔT) is shown in figure 6, 7, 8. Also the relation between the ΔT and Nusselt number is shown in figure 9.

5. It is seen that the heat transfer rate through the graphical representation is more efficient in the perforated plate curve with diameter 15mm.

7.Conclusion

Experimental study is carried out based on perforated plate on experiential investigation following conclusion can be drawn:

1. Heat transfer from heated plate is strongly affected by getting disturb, refuses its thickness and coefficient of heat transfer enhancement.

2. To study the perforated surface has a significant outcome on the enhancement of heat transfer and Nusselt Number.

3. At 5% perforated area heat transfer coefficient can be increased up to 20%. And weight of plate significantly increases. The removal of excessive heat from system components is necessary to avoid the damaging effects of burning or overheating.

4. It is determine that for perforated fins, h is enhanced by about 15% as compared to plain horizontal fins. Thisshows that perforations created at the centre of fins helps toimprove heat transfer. From this analogy, it is clearedthat the results for present study are satisfactory.

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