Experimental Analysis between Rectangular Solid Fins with Different Circular Perforated Rectangular Fins under Natural Convection

Shivasheesh Kaushik[1], Vinay Sati [2], Dr. Anirudh Gupta[3], Kavita Puri[4]
[1&2]M.Tech [4th] Semester Student of ME Thermal, BTKIT Dwarahat Uttarakhand, India.
[2]Assocaite Professor, Mechanical BTKIT Dwarahat, Uttarakhand, India.
[3]Assisitant Professor, Mechanical BTKIT Dwarahat, Uttarakhand, India.

***Abstract*: The heat transfer rate from a solid horizontalrectangular fin and fins with of same geometrical dimension embedded with different number of same circular perforations under natural convection is numerically investigated. The parameters considered in this investigation are the geometrical dimension of fins, perforationdimension of fins with different number of same circular perforation. A comparison between heat transfer rates of the solid fin with different number of the circular perforated finsis presented. It is found that the heat transfer rate of solid horizontal rectangular fin is low as compared to different number of circular perforated fin and the lateral spacing between perforation decreases which increases heat transfer coefficient. Here in this experimental analysis of fins we compute the three different value of heat transfer coefficient lies on three different places first one is on flat perforated surface h_{ps},second one is on fin tip h_t, thirdly at inner perforated surface of perforation h_{pc}, here ($h_{ps} \neq h_{pc} \neq h_t$) for computing the value of total heat transfer rate of each fin to get the overall effectiveness of the fin. The problem of this study was numerically solved and results are plotted through mat lab software.**

Keywords:

NOMENCLATURE

A: cross sectional area of the fin
Ac : cross sectional area of the perforation
Bi : Biot number
b: Circular perforation dimension
g: gravity acceleration, g = 9.81m/sec2.
h: heat transfer coefficient
k: thermal conductivity of fin material
L: fin length
l: unit vector
c L : characteristic length
N: number of perforations
Nu: average Nusselt number, h.Lc / k
Nuc : average Nusselt number of the inner perforation surface
OA: open area of the perforated surface
Q: heat transfer rate
Ra: Rayleigh number, g.β. (Tm T).Lc4/(t.ν.μ) −∞
R*a*c : Rayleigh number of the perforation inner lining surface
ROA: ratio of open area,
RQF: Ratio of heat dissipation/transfer rate of the perforated fin to that of the non-perforated one, Qpf / Qsf
RWF: ratio of the perforated fin weight to that of the solid fin (perforated fin weight ratio), *W*pf /Wsf .
S: perforation spacing
T: temperature
t: fin thickness
W: fin width
Wpf : perforated fin weight
Wsf : solid fin weight

Subscripts and superscripts
b: Fin base
l: Lower surface of the fin
m: Mean value
p_c: Perforation inner surface (within the perforation)
p_f: Perforated fin
p_s: perforated surface or the remaining solid portion of the perforated fin
s_f: solid (non-perforated) fin
s_s: solid surface
t: Fin tip
u: Upper surface of fin
x: Longitudinal direction or coordinate
y: Transverse (lateral) direction with the fin width or coordinate
z: Transverse (lateral) direction with the fin thickness or coordinate
∞: Ambient

I.INTRODUCTION

Fins as heat transfer enhancement devices have beenquite common. As the extended surface technology continuesto grow, new design ideas emerge including finsmade of an isotropic composite, porous media, and interrupted plates [1, 2, 3].For a good thermal system it is required that the design and size of a thermal system must be appropriate which transmit, or dissipate the appropriate amount ofunwarranted heat with the required demand. The successfuland safe operation of thermal units based ondifferent needs including cooling and heating ofspecific component parts or partition walls of suchsystems. The cooling of these parts can be doneby removing heat continuously with adequate amount fromthem. In electric and electronic systems, the generatedheat may cause burning or overheating problems thatlead to system failure and costly damage. In most cases,imperfect designed thermal systems are associated withoverheated surfaces that are not able to transfer the accurate amount of undesired heat. The fin industry has been engagedwith regular search to reduce size of the fin, weightof fin and cost of the fin. The reduction in fin size and cost is achieved byincreasing the heat transfer carried out by the fins.This increment can be completed by differentmethod such as [4, 5]. Firstly, by increasing the ratio of the heattransfer surface area of the fin tothe volume of the fin. Secondly by producingfins from materials having high thermal conductivity,and increasing the heat transfer coefficient betweenthe fin and its surroundings.Several investigations have been conducted to find theoptimum shape of fins (rectangular, square,triangular, pin,wavy, serrated, and slotted). Some of these studies are depend on splitting a certain dimension of the fin in anoptimal way provided that the total volume of the finmaterial is fixed. Others have introduced some shapemodifications by cutting some material from the fin tomake cavities, holes, slots, grooves, or perforationsthrough the fin body in order to increase the effectiveheat transfer surface area and/or the heat transfercoefficient [5,6,7]. Present market trend is based on best optimized quality parameters with low quantity, so market demands economical, compact, lightweight and good effective fins. The optimization of size of fin is of bigger importance. Therefore, fin must be designed to achieve maximum amount of heat removal with low material expenditure.There is one popular heat transfer augmentation technique which states that the use of rough surfaces of different configurations which increases surface turbulence. Here the aim of surface roughness is to provide surface turbulence which automatically increases the heat transfer coefficient rather than the surface area. It is reported that the natural convection coefficient of non flat surface is lied between 50%-100% which is higher than those of flat surfaces. [2]. Further in the other researches researchers reported a similar trend for perforated fins in which the improvement is more than enough in heat transfer coefficient and carried out by restarting the thermal boundary layer after each interruption (perforation) [2, 3,8].Perforated plates (fins) represent an example of surface roughness, surface interruption [2, 9] and are widely used in different electronic industries, automobile companies, and use this technology in heatexchanger, film cooling, and solar collectorapplications for enhancement of efficiency of the system [5].Despite the fact that correlations for the convectioncoefficient within cavities and over the surfaces of nonperforatedplates are readily available [1,2], literature research indicated a lack of such relations for the perforatedsurfaces under natural convection. So, the three different surfacecoefficients located at three different locations were estimated through the concept of augmentation ratio [2] and open area of the perforated surface.This study aims mainly at examining the extent ofheat transfer enhancement from a different circular perforated and non perforated horizontal rectangularfin under natural convection conditions as a resultof introducing surface modifications (perforation) tothe fin. The modifications in this work are HorizontalCircular perforations made through the fin thicknesswith different number of perforation. The study investigates theinfluence of circular perforation and lateral spacingon heattransfer ratio, heat transfer rate, heat transfer coefficient of perforated surface (h_{ps}) , heat transfer within the perforation (h_{pc}), heat transfer coefficient from tip of the fin. The heat dissipation of the solidfin is compared with that ofthe fins with different number of parallel perforation. The studyeventually attempts to make the best use of the materialand size of a given fin, which involves some sort ofoptimization.The overall objective of this study was to evaluate thepotential of heat transfer enhancement when body perforationsof circular cross section are introduced to a horizontalrectangular plate (fin) under natural convectionconditions. The specific objectives of the work may besummarized as follows:

1. Determine some structural values and temperature ranges of parameters through experimental setup of the different fins.
2. Compare the heat transfer rate of the solid fins withdifferent number of perforated fins using heat transfer coefficient of the solid and perforated surface that will be calculated through different mathematical and heat transfer expression.
3. Calculating the fin weight reduction ratio and Q(conduction) for each fin for determining the whole process.
4. Determine the effect of lateral spacing on heat transfer coefficientas the number of perforation increases on the different fins of same length and width.
5. In this experimental analysis of fins we compute the three different value of heat transfer coefficient lies on three different places first one is on flat perforated surface h_{ps},second one is on fin tip h_t ,thirdly at inner perforated surface of perforation h_{pc} ,here ($h_{ps} \neq h_{pc} \neq h_t$) and we computing these values of heat transfer coefficient for determining the total heat transfer of each fin for getting the actual values of heat transfer rate and for calculate the overall effectiveness of the fin apparatus.

II. STRUCTURAL ANALYSIS

In this project, the number of perforations *N*x in the *x*-direction (*L*) and Ny in the *y*- direction (*W*) are assumed. The perforation cross sectional area (*Ac*) is assumed and then the dimension of any perforation is calculated.The heat transfer surface area including the tip of the uniform longitudinal rectangular perforated fin with circular perforations is expressed as

$A_{fp}=A_{ps}+A_{ps}+A_{pc}N_c$

$A_{fp}=(2W.L-2N_c.A_c)+(W_c.A_{pc})$

$A_{fp}=A_f+N_c(A_{pc}-2A_c)$

Equation 16. can be written as

$A_{fp}=A_f+[N_x.\ N_y.\Pi.b(t-(b/2)]$ (2)

In order to compare the heat transfer $_{\text{surface}}$ area of the perforated fin (Afp) to that of the conventional one (A_f), the fin surface area ratio (RAF) is introduced and is given by Eq.(3)

$RAF = A_{fp}/ A_f(17)$

$RAF = 1 + [N_x.\ N_y(A_{pc}- 2A_c)] / A_f(18)$

$RAF = 1 + (N_x.\ N_y.\Pi.b(t-(b/2)] / 2(WL) + Wt$ (3)

Where, $A_f = 2(WL) + Wt$

The material volume of the perforated fin is compared with the volume of non perforated fin by volume reduction ratio (*RVF*) which is expressed as Eq. (5):

$RVF = V_{fp} / V_f = (L.W.t-N_x.N_y.A_c.t) / (L.W.t)(20)$

$$RVF = 1 - (N_x . Ny.Ac) / W.L \quad (4)$$

$$RVF = 1 - (N_x . Ny.(\Pi/4.b^2)) / W.L \quad (5)$$

Similarly, the perforated fin has less weight than that of equivalent non- perforated one. This aspect is expressed by the fin weight reduction ratio (RWF) defined as Eq. (7):

$$RWF = W_{fp}/W_f = (W_f - N_x.N_y.A_{c.}.t.\rho) / W_f \quad (6)$$

$$RWF = W_{fp}/W_f = (1 - N_x.N_y.A_{c.}.t.\rho) / W.L \quad (7)$$

According to the perforation shape and dimension that is cut out from the fin body, the fin with the circular perforation pattern is studied. The number of perforation in longitudinal direction Nx, in the transverse direction Ny and the perforation diameter is b. The direction perforation spacing S_x and S_y:

$$W = N_X b + (N_y + 1)S_y (25)$$

$$L = N_X b + (N_x + 1)S_y (26)$$

$$S_x = (L - N_x.b) / (N_x + 1) \quad (9)$$

$$S_y = (W - N_y.b) / (N_y + 1) (28)$$

III. EXPERIMENTAL DETAILS

The experimental setup includes a heat sink supplied with heating elements and data acquisition system. The heat is generated within the heat sink by means of one heating element power of 670 W. All the experimental data are recorded by the data acquisition system. The heat sink chosen for experiments are aluminum cylinder of 60 mm diameter and 200 mm length. One hole was drilled in the cylinder in which one heating element was pressed. The power supplied by heating element was 670 W. Five aluminum straight fins were fitted racially. The fins are 100 mm long, 200 mm wide and 2 mm thick. There is one non-perforated fin and four perforated fins. These fins were divided into four groups as:

S.No	Perforated fins diameter (mm)	Number of perforations per fin
1	10	24
2	10	32
3	10	40
4	10	48

A variable transformer of type 2P1 with input 240 V and 50 Hz and output 0-240 V, 20 A and 7.5 kVA were used to regulate the voltage supplied to the heating elements. The experimental data were measured by a hand held, battery operated digital temperature sensor. Temperature was recorded on the surface of the test fins at equal spacing of 20 mm located along the length of fin. The apparatus was allowed to run until steady state was achieved. Recording of temperature was done after steady state was reached.

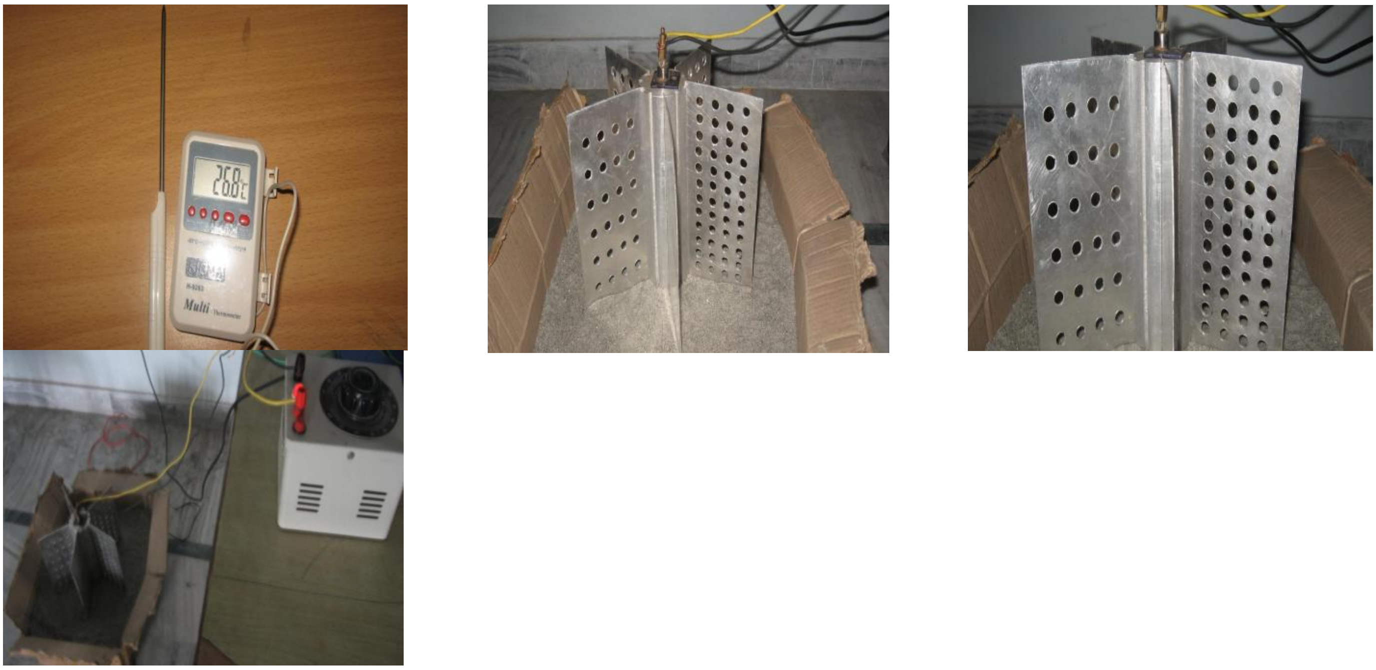


Fig.1 PHOTOGRAPH VIEW OF THE EXPERIMENTAL SET UP

IV. RESULT AND DISCUSSIONS

In this study investigated perforation shape geometry indicates that the increase or decrease in the surface area of the perforated fin with respect to that of the non-perforated one depends on the following parameters: the fin thickness, the total number of perforation, Nc and the perforation diameter, b. However, Afp is greater or smaller than Af depends on the fin thickness and perforation diameter. The calculation show that the heat transfer surface area of the perforated fin is a function of the fin dimensions and the perforation shape geometry. The temperature distribution of the perforated fins and the non-perforated along x-direction is plotted in figure 5. As shown in figure, it is obvious that the temperatures along the non-perforated fin are higher than those of the perforatedone in most cases. It is also indicated the temperature drop between the fin base and tip increases as the number of perforation are increased. This is because thermal resistance of the perforated fin decreases as the perforation diameter is increased.

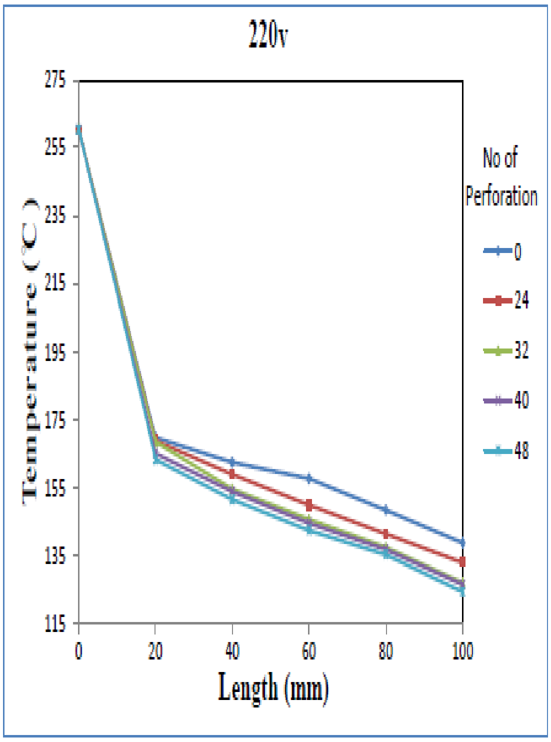


Fig. 5(a) Temperature distribution of each fin at 220V

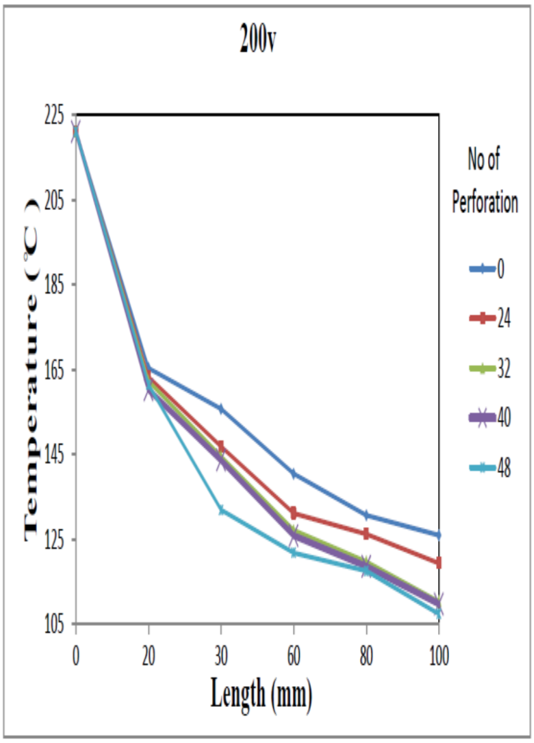
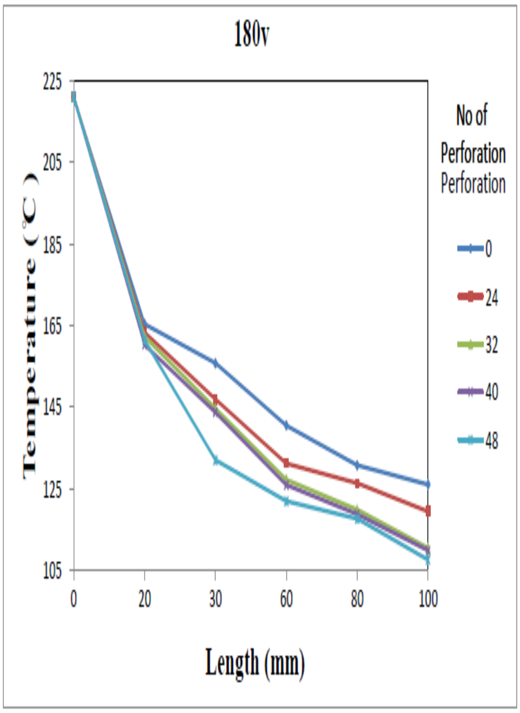


Fig. 5(b)Temperature distribution of of each fin at 200V



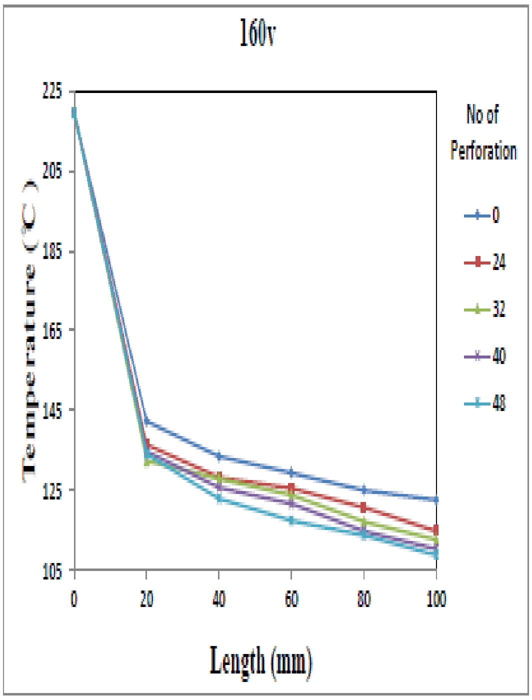


Fig. 5(c)Temperature distribution of each fin at 180 V

Fig. 5(d)Temperature distribution of each fin at 160

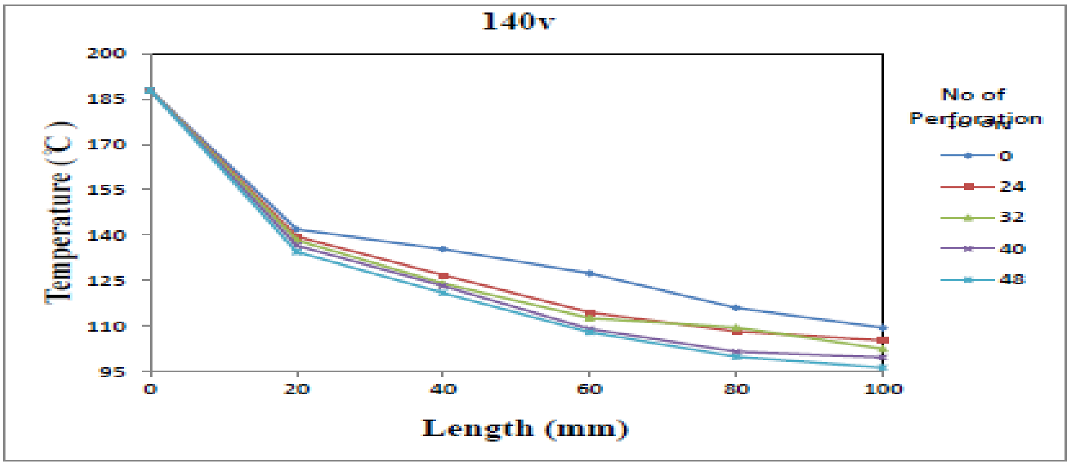
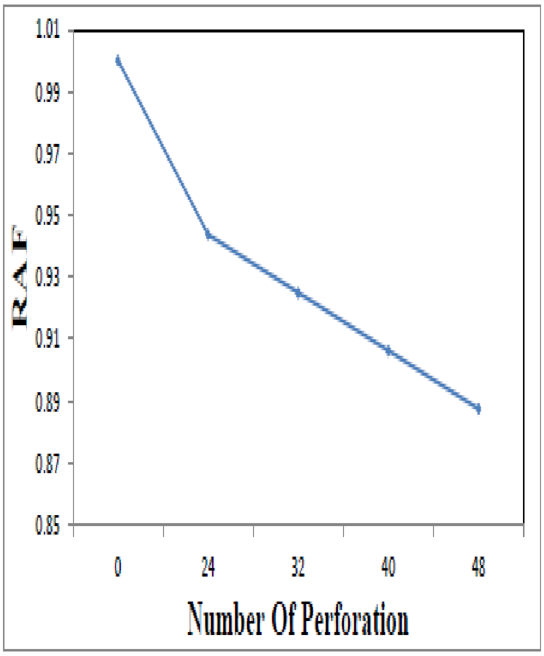


Fig 5(e)Temperature distribution of each fin at 140V

It indicated that RAF is weak function of the fin length and width. This is because the effect of the fin tip area which is smaller surface compared to that of the fin surface area and can be neglected. The temperature distribution along the fin has important effect on the fin performance. Higher fin temperatures exist as the fin thermal resistance is decreased.



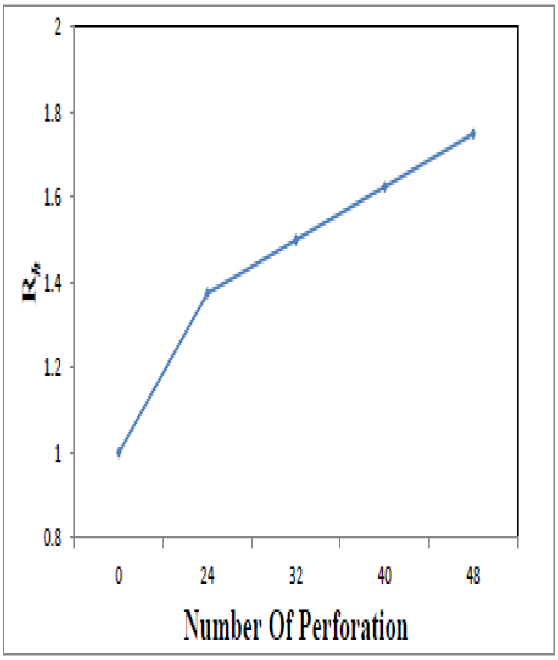


Fig 5(f)Relationship between fin area Ratio with No. of perforation

Fig 5(g)Relationship between augmentation ratio with No. of perforation

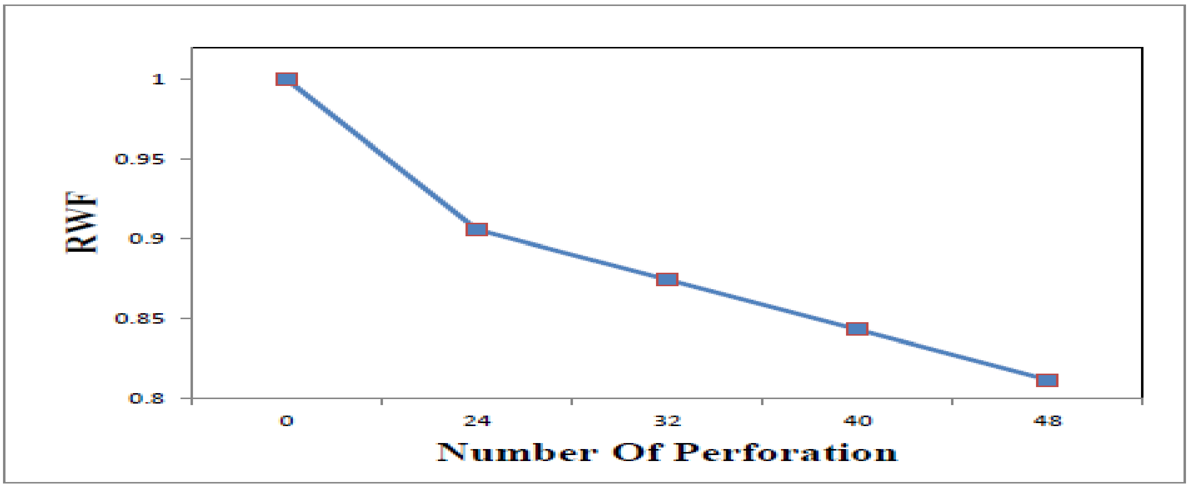


Fig 5(h)Relationship between weight reduction ratios with no. of perforation

Figure 5(f) shows the relation between RAF and number of perforations. This figure shows that RAF is smaller than unity. Heat dissipation rate of the perforated fin depends on the heat transfer coefficient and fin area. In this study, all the film heat transfer coefficient are assumed to be unity and increasing up to the upper limit 1.25 as the number of perforation increased, however, decreasing down to the lower limit of 1.1. The calculation of R_h was plotted against the number of perforation in Figure 5(g). The ratio RWF is plotted as functionof the number of perforation in Figure 5(h). The figure shows that the weight reduction ratio of the perforated fin continues to decrease as number of perforation is increased. the fig5(i)(j)(k)(l) indicates that as the perforations on the fins increases heat transfer rate ,heat transfer ratio and heat transfer coefficient of the perforated surface will also increases and the lateral spacing between the perforation decreases with increase in perforations and heat dissipation rate.

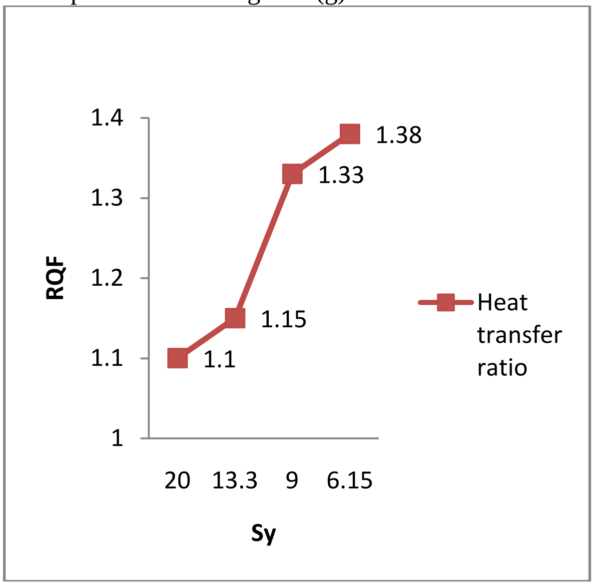


Fig5(i) Relationship between Heat Dissipation Rate with Lateral Spacing

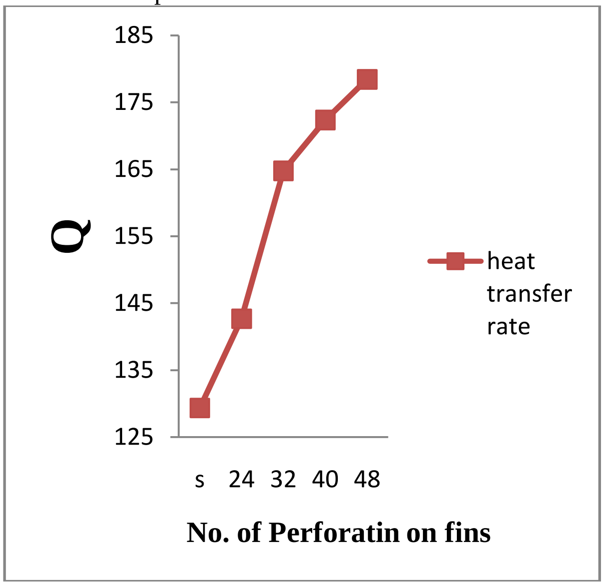


Fig5(j) Relationship between Heat Transfer Rate with No. of Perforation

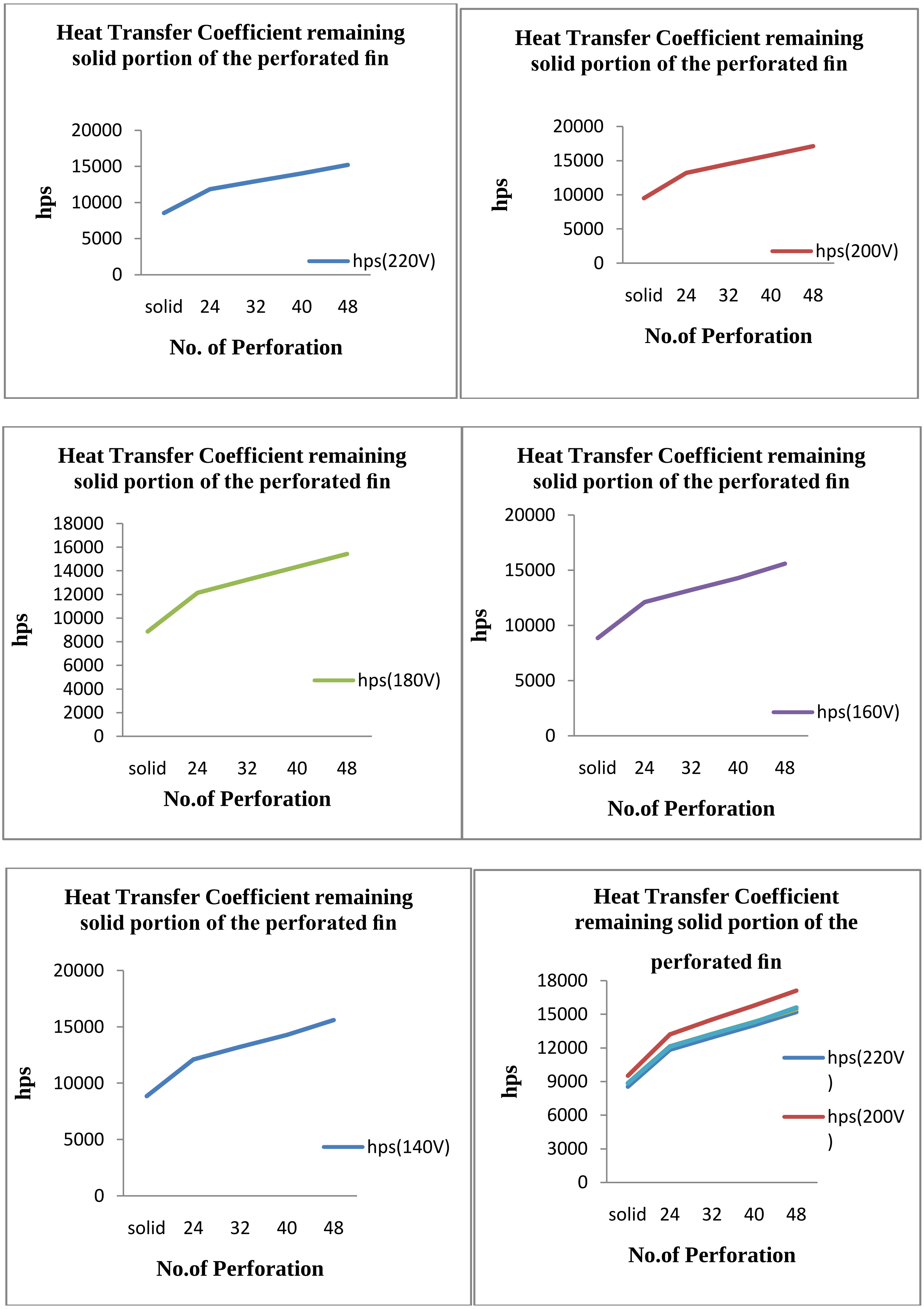


Fig5(k) Relationship between Heat Transfer Coefficient remaining solid portion of the perforated fin with No. of Perforations

The effect of lateral spacing (Sy) on the perforated fin performance is elucidated in Figure 6(k)(l), such that the variation of (RQF) with (Sy) and (RQF) with different number of perforation is studied for value of the fin thickness 2mm and its thermal conductivity 200W/m^{2}K. It is clear that RQF is increasing under low values of (Sy) then tends to decline thereafter. However, low values of (Sy) means more perforation area and less solid material, which increase the fin thermal resistance, which causes a reduction in RQF. Moreover, it can be said that the conflicting effects of fin thermal resistance and number of perforations are responsible for this style of fin thermal behavior. Figure 6(k) shows that (RQF) severely depends on the spacing (Sy) and as the perforation increases the lateral spacing(Sy) decreases which increases heat transfer rate of the different perforated fins.

The overall effectiveness of the fin apparatus:-

$$\varepsilon_f = \frac{Q(fin)}{Q(withoutfin)}$$

$$Q(fin) = 823.05(W)$$

$$Q(withoutfin) = 136.61(W)$$

$$\varepsilon_f = 6.02485$$

CONCLUSION:-

The temperature drop along the perforated fin length is consistently higher than that for the equivalent non-perforated fin. As the number of perforation increases on the fin weight reduction ratio also decreases, higher perforation on the fin, lower the weight of the fin with high heat transfer coefficient. The gain in heat dissipation rate for the perforated fin is a strong function of the perforation dimension and lateral spacing. Decreasing the perforation dimension reduces the rate of temperature drop along the perforated fin. Heat transfer coefficient for perforated fin that contained a larger number of perforations higher than the perforated fin that contained a small number of perforations.

REFERENCES

(1) A.E. Bergles, "Technique to Augment Heat Transfer", in Handbook of Heat Transfer Applications, Second Edition,Ch. 3, eds. Werren M. Rohsenow, James P. Hartnett, and Ejup N. Ganic, NY: McGraw-Hill.

(2) BayramSahin and AlparslanDemir, "Performance Analysis of a Heat Exchanger Having Perforated Square Fins",Applied Thermal Engineering, 6(2008), pp. 621–632.

(3) BayramSahin and AlparslanDemir, "Thermal Performance Analysis and Optimum Design Parameters of HeatExchanger Having Perforated Pin Fins", Energy Conversion and Management, available online 4 January 2008,doi:10.1016/j.enconman. 2007.

(4) A. H. Al-Essa and F. M. S. Al-Hussien, "The Effect of Orientation of Square Perforations on the Heat TransferEnhancement from a Fin Subjected to Natural Convection", Heat and Mass Transfer, 40(2004), pp. 509–515.

(5) R. Mullisen and R. Loehrke, "A Study of Flow Mechanisms Responsible for Heat Transfer Enhancement inInterrupted-Plate Heat Exchangers", Journal of Heat Transfer (Transactions of the ASME), 108(1986), pp. 377–385.

(6) C. F. Kutscher, "Heat Exchange Effectiveness and Pressure Drop for Air Flow Through Perforated Plates With andWithout Crosswind", Journal of Heat Transfer, 116(1994), pp.391–399.

(7) B. V. S. S. S. Prasad and A. V. S. S. K. S. Gupta, "Note on the Performance of an Optimal Straight Rectangular FinWith a Semicircular Cut at the Tip", Heat Transfer Engineering, 14(1998).

(8) B. T. F. Chung and J. R. Iyer, "Optimum Design of Longitudinal Rectangular Fins and Cylindrical Spines withVariable Heat Transfer Coefficient", Heat Transfer Engineering, 14(1993), pp. 31–42.

(9) S. Kakac, A. E. Bergles, and F. Mayinger, Heat Exchangers, Thermal-Hydraulic Fundamentals and Design.Hemisphere Publishing Corporation, 1981.

(10) A. H. Al-Essa, "Enhancement of Thermal Performance of Fins Subjected to Natural Convection Through BodyPerforation", Ph.D. thesis, Department of Mechanical Engineering, University of Baghdad, Iraq, and University ofScience and Technology, Jordan, 2000.

(11) A. Aziz and V. Lunadini, "Multidimensional Steady Conduction in Convicting, Radiating, and Convictingadiating Fins and Fin Assemblies", Heat Transfer Engineering, 16(1995), pp. 32–64.

(12) P. Razelos and E. Georgiou, "Two–Dimensional Effects and Design Criteria for Convective Extended Surfaces",Heat Transfer Engineering 13(3)(1992), pp. 38–48.

(13) S. S. Rao, "The Finite Element Method in Engineering", Second Edition, Elmsford, NY: Pergamon, 1989.

(14) Abdullah H. M. AlEssa, "One-Dimensional Finite Element Heat Transfer Solution of a Fin withTriangular Perforations of Bases Parallel and Towered Its Base", Archive of Applied Mechanics,Accepted: 17 June 2008, DOI 10.1007/s00419-008-0250-5).

(15) G. D. Raithby and K. G. T. Holands, "Natural Convection", in Handbook of Heat Transfer Applications, SecondEdition, Ch. 6, eds. W. Rohsenow, J. Hartnett, and E. Ganic , New York: McGraw-Hill, 1984.

(16) Frank P. Incropera and David P. Dewitt, "Fundamentals of Heat and Mass Transfer", Fourth Edition, NewYork: John Wiley and Sons, 1996, p. 110.

(17) Abdullah H. AlEssa and Mohammed Q. Al-Odat the Arabian Journal for Science and Engineering, Volume 34, Number 2B October 544 2009

(18) C P Kothandaraman,SSubramanian,HEAT AND MASS TRANSFER DATA BOOK