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Effect of the Circular Perforations on the Heat Transfer Enhancement by the Forced Convection from the Rectangular Fins

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Abstract

This study aims to investigate the effect of the circular perforation of the rectangular fin on the enhancement of the heat transfer by forced convection. The solid rectangular fin considered as a reference for comparison purpose with the perforated fin. The parameters taken into consideration are thermal properties and geometrical dimensions of the fin and its perforations. The area and heat transfer gain of the perforations fins were considered being the main parameters in this study. The results of this study showed that the heat dissipation was improved when used the perforation fins compared with the equivalent solid fin. The enhancement quantity of the heat dissipation from the fin depends on the thermal conductivity, the perforation dimension, thickness, longitudinal and lateral spacing. Finally, the perforating of the fins enhances the rate of heat dissipation as well as decreases the weight of the fin.

Keywords: Forced Convection, Heat transfer, perforated fin.
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1.Introduction

The heat transfer enhancements are a significant topic of thermal engineering applications. The heat removal from the system components is essential to avoid the damaging effects of the overheating or burning. In general, the enhancement of the heat transfer can be achieved by increasing the surface area of the heat transfer as well as increasing the coefficient of heat transfer. In the more important applications, the extended surfaces were used to increases the surface area of heat transfer by attached the fins to walls and surfaces of the heat transfer surface. In the industry, there are many uses for the fins due to their high performance of heat transfer e.g. heat exchangers, electronic component cooling, cooling of the blade of the gas turbine, etc. The industry was investigating different ways to reduce the size of fins as well as the cost of the manufacturing of the fins. Comparison to the surface with no fins, both fins arrangements (in-line and staggered) meaningfully argument the heat transfer through the surface. The extended surface (fins) technique represented the optimal solution of the effectiveness of the heat exchange which used to increase the transfer of the heat between two fluids. The fins heat exchanges were used in several types, these fins were added for heat exchange with different shapes such as square, rectangular, cylindrical, tapered or pin fins and annular, to a combination of different geometries, have been used.
A large number of studies have presented the modifications of the shape by cutting the fins material to make cavities, slots, holes, grooves, or channels to increase the area of the heat transfer and the coefficients of heat transfer:

P. Singh, et.al. (2014) [1], they have designed and analyzed the heat transfer through fin extension in plate fins. They studied about various geometries such as rectangular, trapezium, triangular, and circular extensions in plate fins. The results showed that plate fin with extensions provided 5% to 13% more heat transfer than fin without extensions. The effectiveness of rectangular extension plate fin is more than the other types of extension.

M. Reddy and G. Shivashankaran, (2014) [2], they have done a numerical simulation to enhance the heat transfer by forced convection for the rectangular channel based on porous pin fin. They had studied about circular, long elliptical and short elliptical pin fin heat sink by varying inlet velocities i.e. 0.5m/s, 1m/s, 1.5m/s and 2m/s using ANSYS CFD Fluent software. The result showed that the heat transfer efficiencies in porous pin fin are around 50% higher than solid pin fin. K. Dhanawade, V. Sunnapwar,(2014) [3], they have done the thermal analysis by forced convection of circular and square perforated fins array. They have varied the size of perforation for the analysis i.e. 10mm square, 8mm square, and 6mm square and for circular perforation 10mm, 8mm, 6mm diameter. The result obtained showed that the Nusselt numbers increased with increase in values of the Reynolds number, thermal friction increased with increase in perforation and use of perforated fin increase the heat transfer and also there is a reduction in weight, saving of material that ultimately decreases the expenditure on fin material. M. Ehteshum, et.al. (2014) [4], they performed hydraulic and thermal analysis of different number and size of the perforation in the rectangular fin arrays. They have done experiment study by taking base area 1088 mm. They varied perforation from 0 to 2, and varied perforation diameter from 0 to 3mm. The results showed that pressure drop and heat transfer increased with increase in the values of the Reynolds number for all fins. With experiments, it was found that with more or larger perforations the efficiency and effectiveness increased, whereas the thermal resistance and pressure drop decreased. Kavita H. Dhanawade et.al. (2014) [5], They investigated the enhancement of the heat transfer by the forced convection experimentally for the rectangular fin arrays attached to the flat horizontal surface. The attached rectangular fin was with lateral circular and square perforation. The dimensions of the rectangular plate were 200 mm x80 mm. The experimental data was used in the analysis of the heat transfer performance for fin arrays of material aluminum, by varying geometry and size of perforation as well as by varying Reynolds number from $21x10^4$ to $8.7x10^4$. The results of this study show that the Nusselt number was effective by the version of the perforation size and Reynolds number. K. Kumar, et.al. (2013) [6], they performed thermal and structural analysis of tree-shaped fin array. They had taken tree shaped fin with slots and tree shaped fin without slots for their analysis. They also studied the effect of the material on the results for the same geometries by taking aluminum alloy, structural steel and copper alloy for the same. The results obtained showed that the capabilities of the slotted tree fins are better than without slotted tree fins. According to the material, the copper fins with slots were best for heat transfer among all the fins. The aluminum slotted fin was found most effective as it has effective heat transfer without deformation among all the fins taken for the study. Wadhah Hussein.(2011) [7], conducted an experimental study to investigate the free convection heat transfer in heat sinks as fins plates rectangular shape with circular perforations. The perforations pattern was included 24 circular perforations for the first fin, and for each fin, the perforations were increased as 8 until reach to 56 in the fifth fin. He distributed the perforations in 6-14 rows and 4 columns. Also, he observed that the temperature along the non-perforated fins was from 30 to 23.7°C at lower power 6 W. Furthermore, the temperature drop between the base and tip of the fin were increased as the perforations diameter increased. At the maximum power 220 W the drop of the temperature was for non-perforated fins from 250 to 49°C. He concluded that the rate of heat transfer and the heat transfer coefficient were increased with increased perforations number. Abdullah H. Al Essa, et.al. (2009) [8], They enhanced the heat transfer by the natural convection from a horizontal rectangular fin embedded with rectangular perforations of the aspect ratio of two. In this study, they compared the equivalent solid fins with the perforated one. An experimental study was carried out for geometrical dimensions of the fin and the perforations, then, investigate the effect of the heat transfer coefficient of the fins based on its perforations. They note that the rectangular perforation was enhancing the heat transfer of the fins. The thermal conductivity of the fins and it is thickness were represented the key parameters of the magnitude of enhancement of the heat transfer. Bayram Sahin, an Alparslan Demir,(2008) [9], They performed an experimental study to show the effect of design parameters on the friction factor and the heat transfer coefficient for the heat exchanger. The heat exchanger was equipped perforated pin fins as square and circular rectangular channel. The results showed that, with used the perforation fins the Nusselt number enhanced and this reflected the improvement in the heat transfer coefficient. O.N.Sara, et al,(2001,2000) [10 & 11], They considered the thermal performance of perforated and solid rectangular blocks which was attached to a flat surface inside a rectangular duct. In the results, they show that the loss in the net energy was reduced and affected by the flow conditions and geometry of the perforated fins. The energy gain was about 20%. Furthermore, the use of the perforation blocks enhanced the heat transfer coefficients.

The aim of this study was an enhancement of the heat transfer by the forced convection from a rectangular fin based on body modifications such as; perforation which leads to interruptions to the fin. The circular perforations which made through the thickness of the rectangular fins represent the main modification. After the modification on the fin, can call as a perforated fin, but the equivalent solid one can call as non-perforated fins.

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2. **Experimentation Methodology:**

The heat transfer rate of is obtained based on the energy balance calculated by using a relation:

$$Q_{Electric\ power} = Q_{Convection} + Q_{Conduction} + Q_{Radiation} \quad \text{...........} \quad (1)$$

We rearranging the above equation get:

$$Q_{Convection} = Q_{Electric\ power} - Q_{Conduction} - Q_{Radiation} \quad \text{...........} \quad (2)$$

The heat generated (Electric power) in terms of voltage and current is given by:

$$Q_{Electric\ power} = V \times I \quad \text{............} \quad (3)$$

Where; $Q_{Electric\ power}$: represents is electrical heat generated in the primary surface

V : is a voltage in Volts supplied to the heating unit.

I: is the current in amperes.

In similar studies, the heat transfer due to radiation is 0.5 % of the total heat supplied from electric power input and hence can be neglected. The heat losses due to side, bottom and top walls of the test section were assumed to be neglected since the side walls are insulated. Thus the heat transfer due to convection is equal to net rate of electrical heat generated in the primary surface.

The heat transfer by convection can be expressed as:

$$Q_{Convection} = h_{av.} A_T \left[T_s \left(\frac{T_{out}+T_{in}}{2} \right) \right] \quad \text{.......} \quad (4)$$

Where; $h_{av.}$: the average convective heat transfer coefficient

A_T: Fin surface area.

T_s: Fin surface temperature.

T_{out} ,T_{in}: represents the duct inlet and outlet temperatures of the ambient air.

Hence average coefficient of the heat transfer by convective $h_{av.}$can be found out as:

$$h_{av.} = \frac{Q_{Convection}}{A_T \left[T_s - \left(\frac{T_{out}+T_{in}}{2} \right) \right]} \quad \text{................} \quad (5)$$

The Nusselt number can be calculated as follows

$$Nu = \frac{h_{av.} D_h}{k_{air}} \quad \text{..............} \quad (6)$$

Nusselt number (Nu) based on the projected area will show the affecting of the heat transfer by the difference in the surface area and disturbances in the flow due to fins. However, Nu based on the total area will appear the effect of the flow disturbances only. In this study, the enhancement characteristics of the heat transfer were calculated by using Nu-based projected area, while optimization was made by using Nu based total area.

$$f = \frac{\Delta p}{\left(\frac{L}{D_h} \right) \rho \frac{V^2}{2}} \quad \text{.................} \quad (7)$$

The relations to determine Reynolds number depend on the values of the hydraulic diameter and the averaged velocity is given by:

$$Re = \frac{V D_h}{\upsilon} \quad \text{..................} \quad (8)$$

In Equations (7) and (8) V is the entrance averaged velocity of the test section and Δp measured from inclined manometer, and D_h is the channel hydraulic diameter.

In all calculations, the values of the air thermophysical properties were obtained at the bulk mean temperature, which is:

$$T_m = \frac{T_{in}+T_{out}}{2} \quad \text{.................} \quad (9)$$

Moreover, the surface area of the heat transfer can be replaced with the projected area or the total area of the test surface area in the calculations.

The sum of the projected area is equal to the total area of the fins. These two types of the areas can be related to each other by:

Total area = Projected area + Total surface area contribution from the blocks:

$$A_{fp} = A_{ps} + A_t + N_c A_p \quad \text{................} \quad (10)$$

$$= (2W \cdot L - 2N_c \cdot A_c) + (W \cdot t) + (N_c \cdot A_c)$$

$$A_{fp} = A_f + N_c (A_p - 2A_c) \quad \text{............} \quad (11)$$

Where: $N_c = N_x \cdot N_y$

Equation (11) can be written as:

$$A_{fp} = A_f + \pi \cdot N_c \cdot b \left(t - \frac{b}{2} \right) \quad \text{...........} \quad (12)$$

The ratio of the fin surface area (RAF) is represented the function which shows the ratio of the heat transfer area of the perforated fin (A_{fp}) to that of the conventional one (A_f) and is given by:

$$RAF = \frac{A_{fp}}{A_f} \quad \text{………………..} (13)$$

$$RAF = 1 + \pi \cdot b \cdot N_x \cdot N_y \left(t - \frac{b}{2}\right) / (2W \cdot L + W \cdot t) \quad \text{……. …………} (14)$$

Where: $A_f = 2(W \cdot L) + (W \cdot t)$

Similarly, the fin weight reduction ratio (RWF) represents the ratio between the weight of the perforated fins and the non-perforated one. The RWF can be expressed by the following equation;

$$RWF = \frac{W_{fp}}{W_f} \quad \text{…………….} (15)$$

$$RWF = \frac{1-(N_x \cdot N_y \cdot A_c \cdot t \cdot \rho)}{L \cdot W \cdot t \cdot \rho} \quad \text{……………...} (16)$$

$$RWF = \frac{N_c \cdot A_c}{L \cdot W} \quad \text{……………...} (17)$$

The fin circular perforation is shown in Figure 1. The perforated fin with the circular perforation pattern is studied according to the shape of the perforation and the cut area out from the fin body. The number of perforation in the transverse direction N_y, in longitudinal direction N_x, and the b is the perforation diameter. The direction perforation spacing S_x and S_y:

$$L = N_x \cdot b + (N_x + 1) S_x$$

$$S_x = \frac{L - N_x \cdot b}{N_x + 1} \quad \text{And} \quad W = N_y \cdot b + (N_y + 1) S_y$$

$$S_y = \frac{W - N_y \cdot b}{N_y + 1}$$

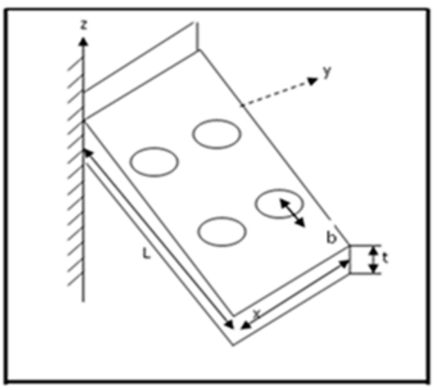

Fig. (1): fin with circular perforations

3.Experimental Set up:

The experimental test rig consists rectangular duct from aluminum, a heat sink supplied with heating elements, control panel, measuring instruments, manometer, temperature indicator, regulator, an air blower. The wind tunnel can supply a constant and variable air velocity. An air blower is used to blow air through the tunnel in which the finned surface heat sink is mounted. The heat generated within the heat sink by means of four heating elements each of 650 W powers. A variable transformer of type 50B with input 240V and 50-60Hz and output 0-270V, 25A, and 7.5KVA was used to regulate the voltage supplied to the heating elements all the experimental data are recorded by the data acquisition system. A schematic of this setup is shown in Figure 2.The heat sink chosen for experiments Figure 3 is Aluminum cylinder of 100 mm diameter and 270 mm length. Four holes were drilled in the cylinder in which four heating elements were pressed. The power supplied by each element was 650 W. Fifteen Aluminum Straight fins were fitted radially. The fins are 100 mm long, 270 mm wide and 2 mm thick. These fins were divided into five groups equally, All groups include three fins and we take the middle of them as a test fin. as Table 1.

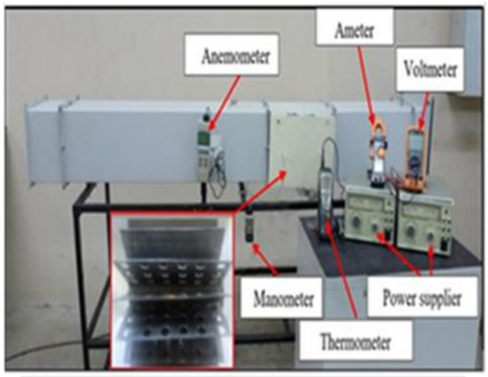

Fig. (2): View of the experimental Apparatus

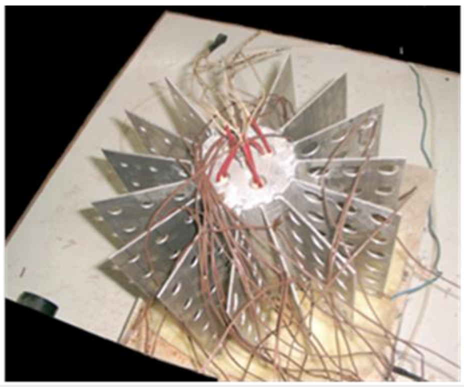

Fig. (3): View of the heat sink (test section)

Table 1 groups of fins.

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Size of perforation	Non perf.fin	8 mm	12 mm	16 mm	20 mm
Number. of perforation	-	18	18	18	18

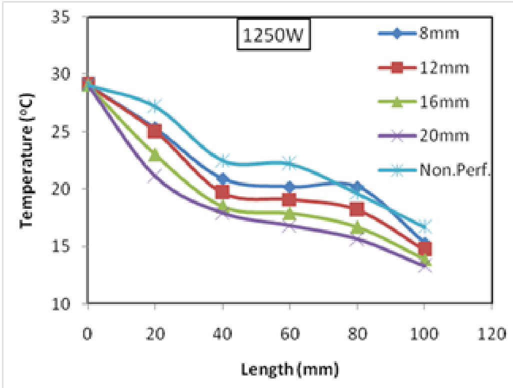
The temperatures at many locations of the fins were collected from the 27 calibrated thermocouples of type-K for all fins in the experimental work. The thermocouples were divided into five groups based on the fins types and each fin contains 5 thermocouples. Meanwhile, the distance separated between two thermocouples was (20mm) along the length of the fin. To measure the based temperature of the fins and the surface temperature of the aluminum shaft one thermocouple was fixed on the outside diameter of the aluminum shaft. The apparatus was running about 70 minutes to arrive the steady state; after reached to the steady state condition the temperature for 27 points can be collected.

4.Results and Discussion:

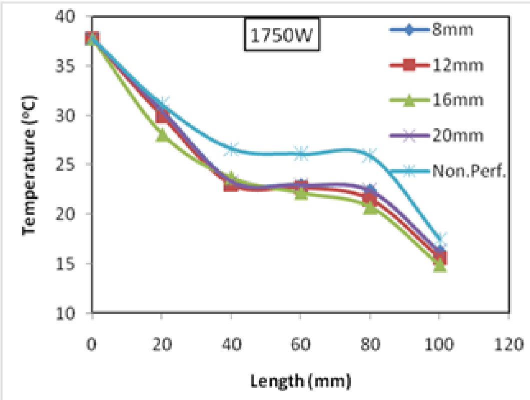
This study investigated the geometry shape of the perforation as an indicator to show the decreases or increases of the surface area for perforated fin compared with the non-perforated one. The affective parameters were considered in this study as the total number of perforation N_c, the fin thickness, and the diameter of the perforation (b). Conversely, A_{fp} is greater or smaller than A_f depends on the diameter of the perforation and fin thickness. The calculations show that, the perforation shape geometry and fin dimensions the key parameters to control the surface area of heat transfer for the perforated fin.

4.1 Temperature Distribution of the perforated Fin

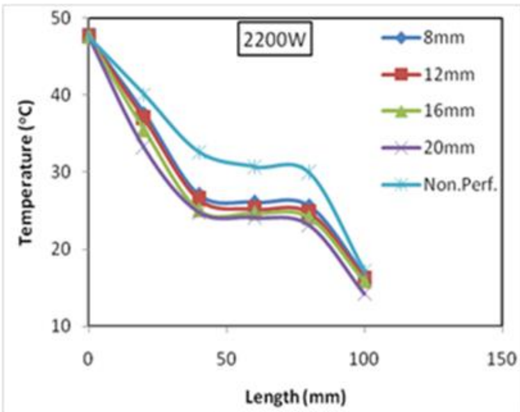
The fin performance was affected by the distribution of the temperatures along the fin length. The reduction in the thermal resistance of the perforated fin led to maximum temperatures of the fin. The distribution of perforated fin temperature and that of the conventional one along the x-coordinate are plotted in Figure 4 through (a,b,c,d,e). As shown in the figures, it is obvious that the temperatures along the perforated fins were lower than those of the non-perforated fin in most cases. Figures show the temperature distribution for the fin of circular perforations with the non-perforated fin. The temperature distributions were plotted for various amount of heat supplied by heating elements for circular perforations.



(a): 1250W



(b): 1750W



(c): 2200W

Diyala Journal of Engineering Sciences, Vol. 11, No. 3, September 2018, pages 62-70 ISSN 1999-8716
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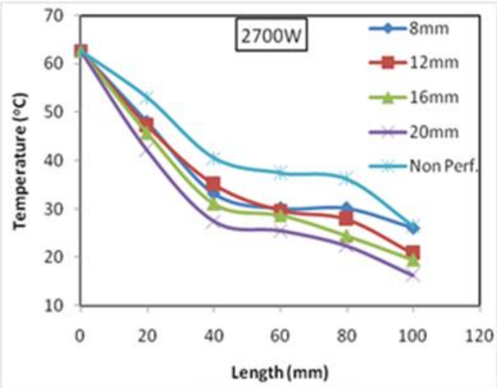
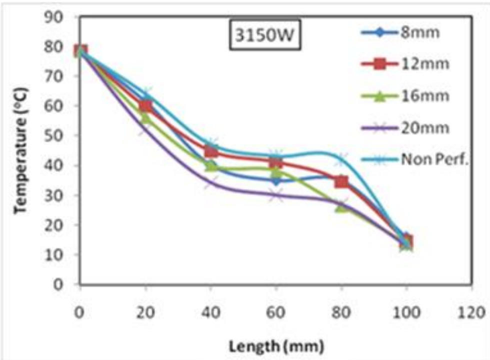


(d): 2700W



(e): 3150W

Fig.(4): The temperature distribution of the perforated and non- perforated along x-direction

4.2 Effect of perforation Size

The effect of perforation dimension of the fin was experimentally studied. Figures 5 show the effect of the perforated dimension on temperature distribution, for a fin with circular perforations of different diameters. This figure indicates that when the perforation dimension increased the temperature gradient between the base and tip of the fin was increased. This is because increased of the perforation dimension of perforation fins led to decrease the thermal resistance.

4.3 Geometrical Analysis of the perforated Fin

The results were indicated that RAF is a weak function of the fin length and width. It is because the effect of the tip area of the fin which is smaller surface area compared to that of the surface area of the fin and can be neglected. The temperature distribution along the fin has an important effect on the fin performance. The reduction of the thermal resistance led to high fin temperatures exist. Figure 6 shows the relation between

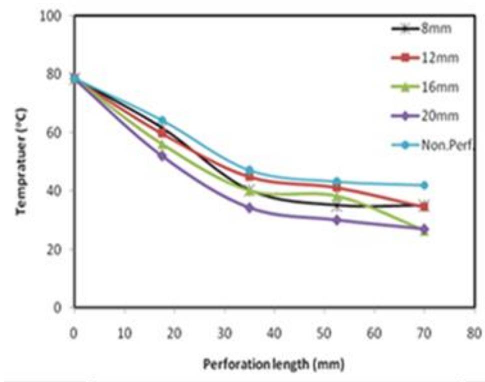


Fig. (5): Temperature distribution with perforation fin

perforation diameter (b) and RAF. The RAF in this figure was appearing less the unity.

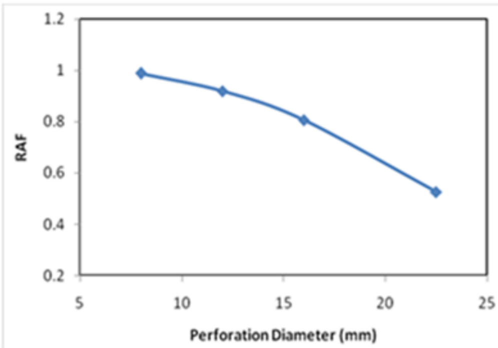


Fig.(6): The relation between RAF

4.4 The perforated Fin Weight Reduction

The perforated fin weight reduction ratio (RWF) is plotted as a function of the perforation dimension b (diameter) in Figure 7. the figure shows that the weight reduction ratio of the perforated fin continues to decrease as b is increased. This means that the perforated fins can improve heat transfer with a reduction in weight and material expenditure in manufacturing them.

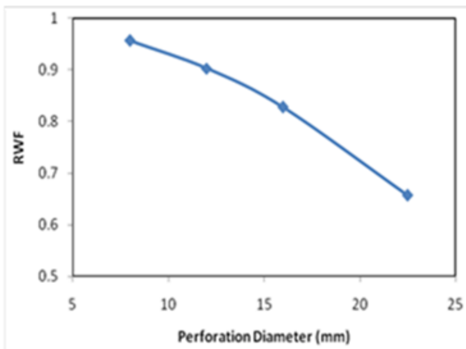


Fig.(7): The relation between RWF and perforation diameter

Diyala Journal of Engineering Sciences, Vol. 11, No. 3, September 2018, pages 62-70 ISSN 1999-8716
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4.5. Heat Transfer Coefficient for the perforated Fin

The rate of heat dissipation was strongly effective by the coefficient of heat transfer and surface area of the perforated fin. In this study, all film coefficients of heat transfer are assumed to be uniform and equal. It was mentioned before that (Rh) is always greater than unity and increasing up to the upper limit of 1.75 as the dimension b is increased, but decreasing down to the lower limit of 1, as shown plotted in Figure 8.

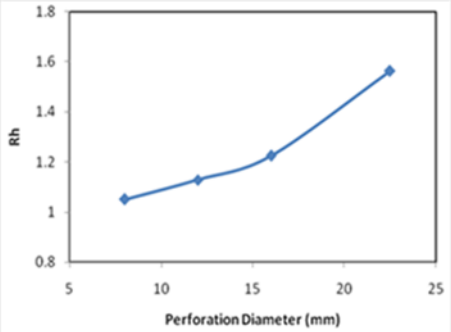

Fig.(8): The relation between heat transfer coefficient ratio Rh and perforation diameter.

4.6. Heat Transfer performance parameters

The dynamics of flow and convection heat transfer are strongly affected by three-dimensional analysis and the average convection heat transfer coefficient as well as are closely related to the fin spacing and Reynolds number. Heat transfer from the plate depends on the temperature field around the plate and variation of temperature over the plate surfaces. Figure 9 represent the temperature distribution along non-perforated fin with various air flow at constant heat flux of 3150 W. It is observed in Figure 9 that for high Reynolds number, the temperature variation over the plate surface is not significant, except at the edges, and recirculation produces a nearly uniform temperature in the wake and strong variation in the other regions. Far from the blocks, the effect of the heating element on the fluid temperature reduces.

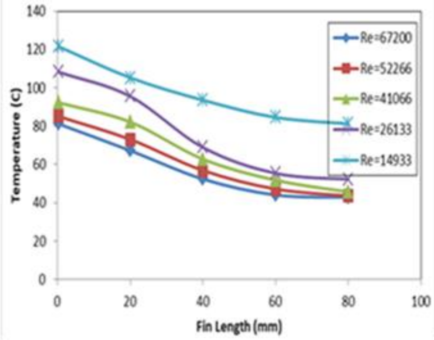

Fig.(9): The relation between Temperature and Fin length at various Reynolds number for solid fin with 3150W

This section discusses the results of forced convection heat transfer of non-perforated (solid) fins and various types of size perforated fins circular which are investigated for Reynolds number (Re) of (14,933 to 67,200).Perforated fins have different Nusselt number values at different Reynolds number. Figure 10 describe the effect size of perforation on the Nusselt number. Figure 10 illustrate the Nusselt number of solid fins and perforated fins with different perforation size of (8, 12, 16 and 20 mm) at various Reynolds number. It can be noted from the figure that the non-perforated fins have the highest and with increase the size of perforation the values of Nusselt number decrease as well, where the perforated fin with 20mm diameter perforated shows the lowest Nusselt number values at all Reynolds numbers in the range considered here. It is because due to perforation the air velocity decreases inside the perforation and some flows are also confined inside the perforation. It is also seen that the value of Nusselt number is little higher for the fins having 8mm diameter perforations in comparison with the fins having 12mm, 16mm and 20mm perforations respectively. This reduction in Nusselt number maybe due to flow confined inside the perforation holes and also reduction of velocity inside them.

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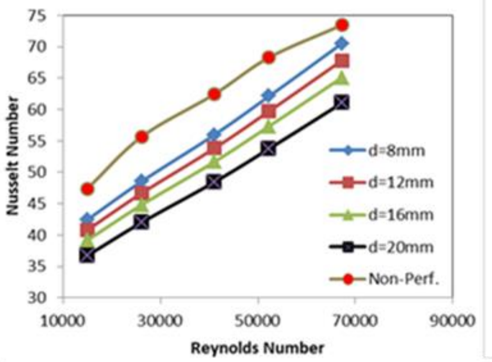

Fig.(10): The relation between Nusselt Number and Reynolds Number at different

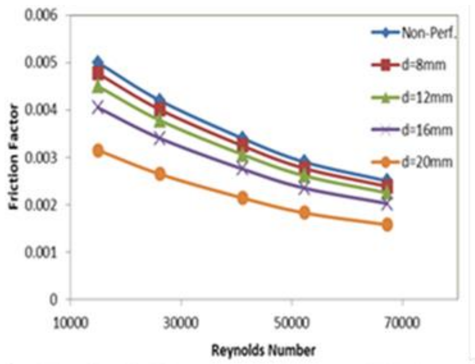

Fig.(11): The relation between Friction factor and Reynolds Number at different

Friction coefficient is calculated over the fin surfaces including perforations. It is seen that the friction coefficient values decrease with increase of Reynolds number. Solid fins have the largest friction coefficient value. Total drag force has two components, one is due to surface shear stress which is called friction drag and another is called form or pressure drag. Due to lateral perforation, flow velocity decreases and flow circulates inside the perforations. For this reason, the skin friction coefficient is smaller for fins with perforations. But in the case of solid fin, fluid interacts with larger surface area than the perforated ones. Figure 11 shows variation of average friction coefficient over the faces of different size of circular perforation at various Reynolds number. Furthermore, Figure illustrates the effect size of perforations on the friction coefficient. In order to do comparison among the perforation diameter at 8mm diameter perforated with respect the effect of friction coefficient, at all cases the pattern of friction coefficient decreased with increase Reynolds number. It can be concluded that the small perforations have the lowest value of friction coefficient followed big perforations. However, the friction coefficient decreased as Reynolds number increase and as the size of perforation increase.

5. Conclusions:

The conclusions of this study were as follow:

• The drop of the temperature on the length of the perforated fin is systematically greater than that for the equivalent non-perforated fin. This is due to the decrease in the cross-sectional area of the fin for heat conduction

• For the perforated fin, the perforation dimension and lateral spacing were effective parameters for the rate of heat dissipation.

• Heat transfer coefficient for circular perforated fin is greater than the non-perforated fin.

• The fin with perforations shows a greater heat transfer enhancement (11% of non–perforated).

• With decrease the size of perforations flow becomes complicated, average friction coefficient decreases and solid fin has the highest value of friction coefficient.

• Average Nusselt number decreases by decrease size of perforations. Solid fin has the largest average Nusselt number for each Reynolds number.

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