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Free convection heat transfer from radial heat sink with wing fins shape: An experimental investigation

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Abstract: Although cylindrical heat sinks increase the surface area of heat transfer, they are frequently used as enhancers of heat transfer. Base on free convection heat transfer, fins with novel designs are used to improve thermal heat performance. The current experimental investigation examines the influence of a new-fin shape effect on the thermal behavior of a circular-base radial heat sink. Heat-supplied rates of, 136, 260,375, 589 and 808 W were the subjects of the investigation. Parametric models were used to compare the impacts of fin shape, fin number, and heat flow on the thermal resistance and total heat transfer coefficient. The results show the Nusselt increase with 22% in the 16th fins case and 88% in the 24th fins case compared with ideal case 8th fins, respectively at high input heat rate. In addition, the heat transfer occurred under free convection conditions, which is reason it was discovered that the higher fins number were more efficient in removing heat compared to those with smaller fins.
Keywords: *Fins number, Free convection, New design fins, Thermal performance, Vertical heat sink.*

1. Introduction

The development of high-efficiency electric systems is increasingly needed for decreasing greenhouse gas emissions [1-3]. Although lighting uses about 20% of the world's electric energy, high-efficiency lights must take the place of conventional lights [4,5]. Due to improvements in LED luminous efficiency, the market for light-emitting diodes (LEDs) has increased recently. However, a thermal problem develops since an LED light uses approximately 70 percent of its energy to generate heat. An LED light's lifetime and performance decrease if this heat is not effectively removed. Therefore, before LED lights are put on for sale, the heat dissipation problem needs to be fixed. Natural convection heat sinks are acceptable given all the advantages of LED lighting [6-8]. Several investigations have focused on using free convection heat transfer to cool fin-equipped cylindrical heat sinks. In addition, books by Martynenko and Khramtsov [9] and Raithby and Hollands [10] have condensed it. It is suggested to cool LED lights using natural convection because no additional equipment is required [11,12]. If heat sinks are not properly constructed, LED lights will overheat and lose some of their lifespan [13]. Additionally, LED lighting is used in an array of settings, and the installation angle can vary based on the application and location. The installation angle is one of the most important factors for LED lighting's natural convection cooling. Therefore, it is important to evaluating how the installation angle in connection with inclination influences the cooling performance of heat sinks [14]. The naphthalene sublimation technique was used in an experiment by Sparrow and Bahrami [15] to measure the rate of heat transfer from vertical square fins attached on a horizontal tube. In both research and industry, the natural convection heat transfer of heated finned and unfinned cylinders has long been significant. Churchill and Chu [16] developed a basic empirical relationship for a cylinder's mean Nusselt number. They therefore examined data from many earlier studies to confirm that their correlation stayed valid over a wide range of Prandtl and Rayleigh numbers. Wall temperatures and uniform heating can be obtained with this the one being used. According to Morgan [17], who

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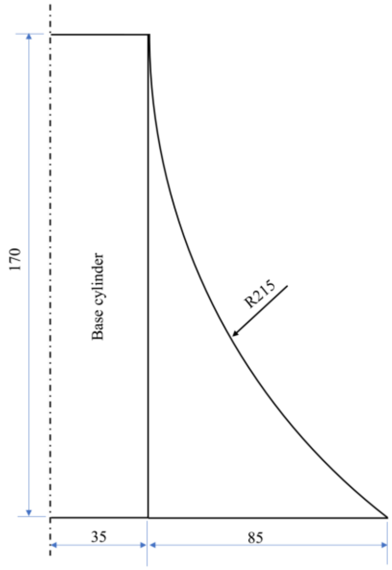
additionally examined at an extensive range of experimental data, there may be a correlation between the Nusselt and Rayleigh values for smooth horizontal cylinders when air is supposed to be the working fluid. Given the fin shapes of the radial heat sinks, wing fins have received fewer investigations than plate or pin fin

In order to study the phenomenon of natural convection heat transfer from vertical cylinders with branched pin fins, Kim and Kim [18] performed experiments. Experiments are conducted to determine the heat transfer rates from finned cylinders to the surrounding air at different cylinder wall temperatures, fin pitch angles, and fin heights. According to the results, typical vertical cylinders with plate fins have a 20% higher thermal resistance than vertical cylinders with branched pin fins. Comparing the former to conventional finned cylinders, the former's heat resistance is 40% lower per unit fin volume. Moreover, a correlation between the estimated thermal resistance values and the Nusselt number is discovered. less than 20% of the correlation-based error between the experimental and predicted data is Experimental research has been conducted on the thermal performance of radial heat sinks with triangular fins in Ref. [19] and semicircular fins in Ref. [20].

In summary, a few studies have been conducted on the heat characteristics of heat sink shapes that are compatible with most common LED light bulbs, such as the vertical cylinder base and wing fin heat sink indicated in Figure 1. The current work aims to investigate the thermal performance of a radial heat sink with wing fins on a circular base at various input variable input heat flow and fin number employing the new fin shapes.

2. Experiment

Figure 1 show the new design of fins with radial heat sink. An experimental laboratory constructed especially for these purposes served as the setting of the experiments. Standardized and calibrated experimental conditions were used to investigate heat transfer by free convection. Figure 2 shows the setup schematic diagram. This experimental setup includes of an aluminum cylinder, aluminum fins, a heat sink with heating elements, and a data collection apparatus. Five heating elements with rated powers ranging from 50W to 850W provide the heat within the base vertical cylinder. The data acquisition system recodes each temperature location within the experimental data. The photographs of this test setup are shown in Figure 3. The heat sinks are made of an aluminum alloy with a thermal conductivity of 167 W/(m K). A solid aluminum cylinder was employed combined with fabricated wing aluminum plate fins for the experimental testing. Heat sinks are made by overlaying fins and bases. The cylinder is 170 mm long and 70 mm in diameter on the outside. Figure 4 displays a cutaway design of the cylinder along with all of its measurements. Five holes, each measuring 8 mm in diameter and 160 mm in depth, were drilled into the cylinder's top surface. The wing fins in the numbers eight, sixteen and twenty-four obtained for this study. A wing with a radius of 215 mm, a height equivalent to the cylinder's length, and a thickness of 3 mm for each fin is used for fixing fins on base heat sink (cylinder). The assembly fins with base cylinder are shown in Figure 1(b).



(a) Wing fin

(b) A photograph of fins with base cylinder.

Figure 1.
The heat sink.

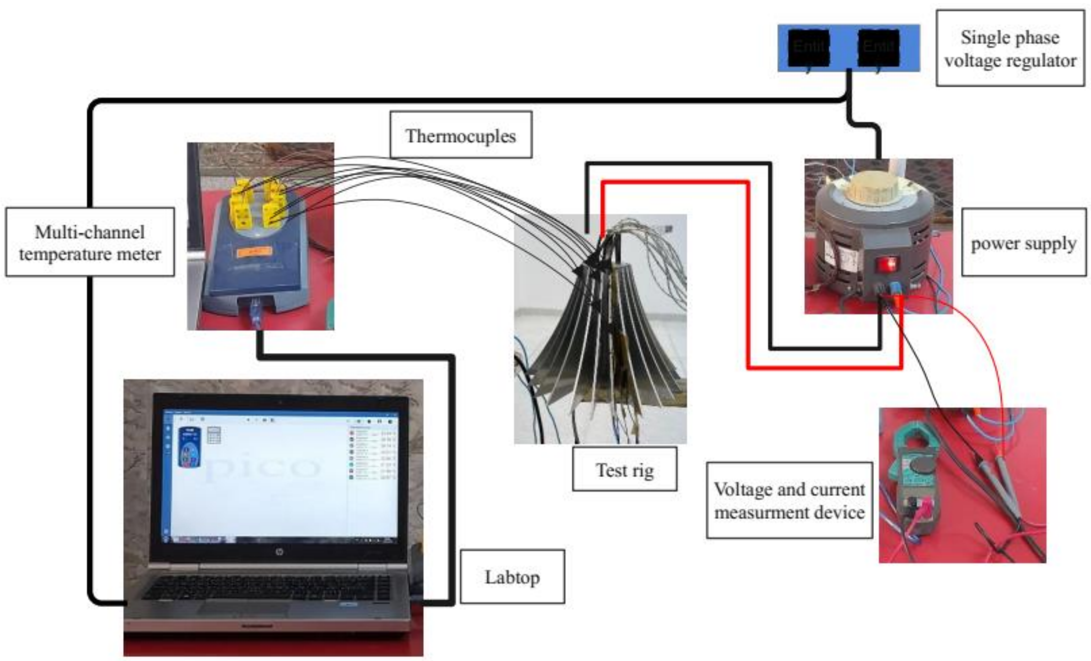
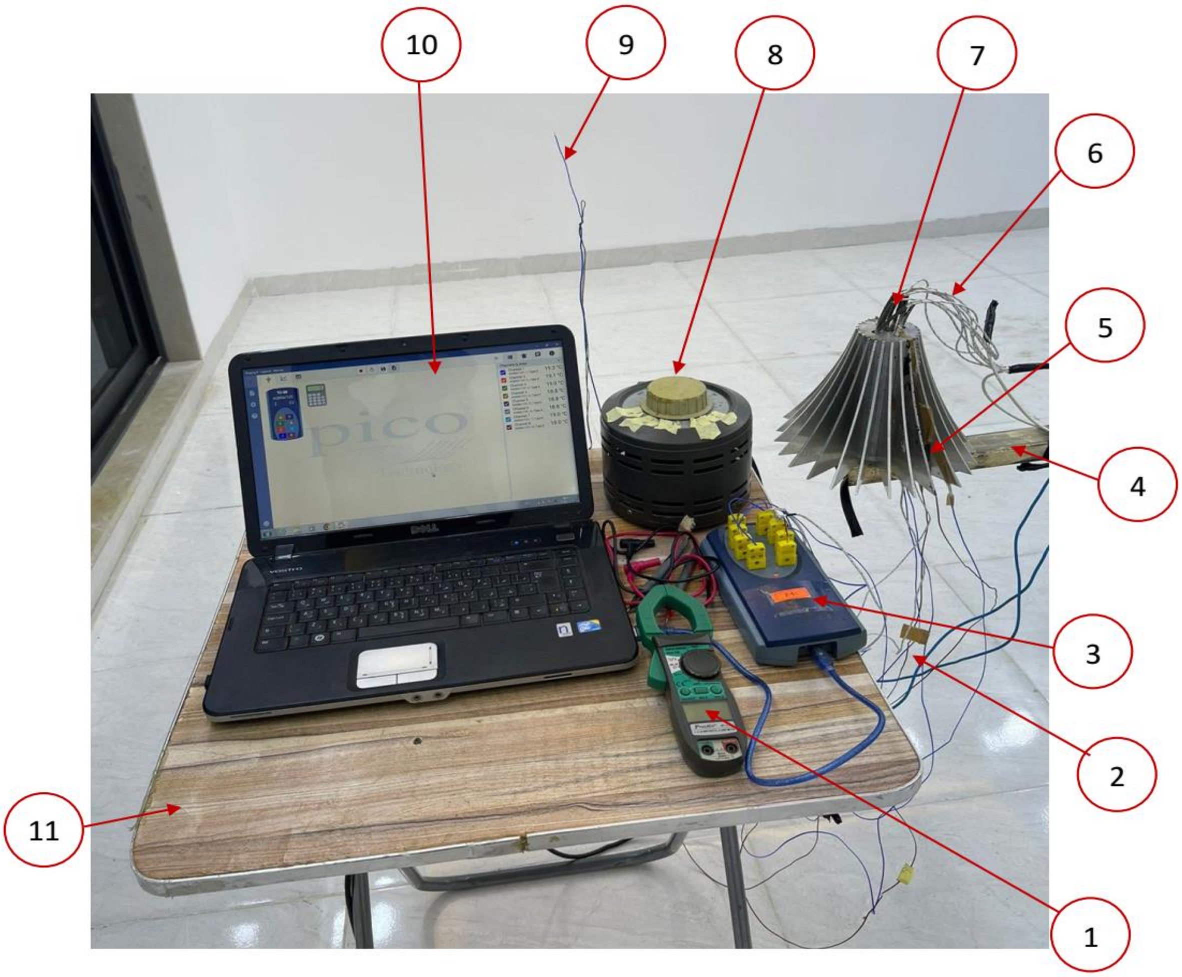


Figure 2.
The experimental setup for the wing radial heat sink analysis: A schematic diagram.



No.	Part	No.	Part	No.	Part
1	Current & voltage measu.	5	Test rig	9	Air temp. thermocuple
2	Thermocouples	6	Electrical wire	10	Labtop
3	Data logger	7	Cartridge heater	11	Table
4	Support	8	Power supply		

Figure 3.
Photograph of the experimental test facility.

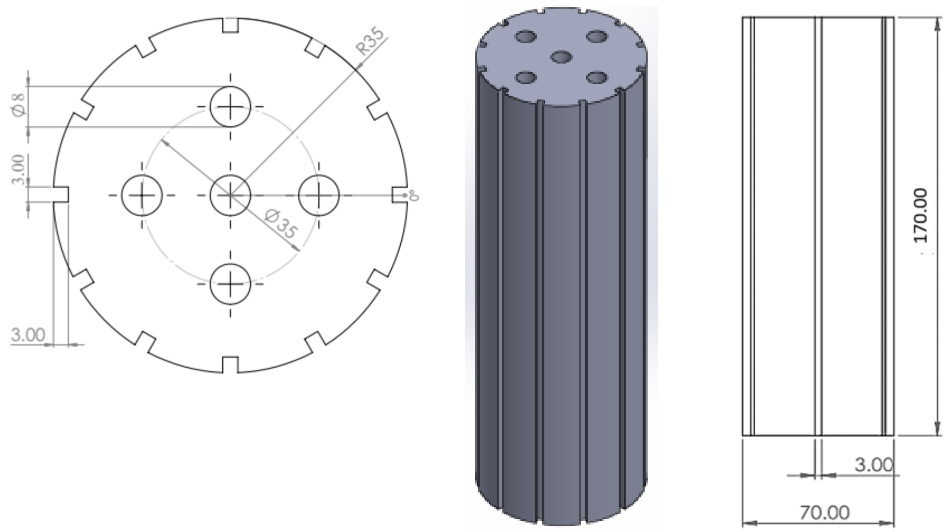


Figure 4.
The base cylinder diagram (in mm).

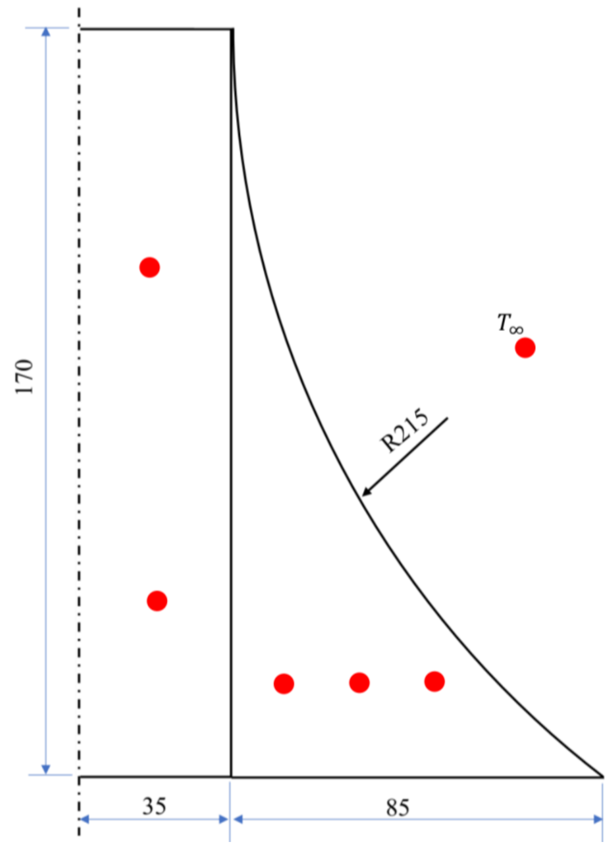


Figure 5.
Heat senser (thermocouples) locations (all dimensions in mm).

The HSN 0103 type electric power source voltage has a maximum electric current of 5 A and an AC voltage range of 0 to 250 V. The voltage measurement range of the voltmeter is AC 2 volts to 750 volts. The Pro'sKit MT-3102 clamp meter was used to measure the current, which may be as high as 400 A.

The electric resistors in this study are circular electric heaters with a double step of 1000 Ω that can dissipate up to 850 W of power at 220 V. Electric resistors are 200 mm long and include a small outside diameter (8 mm) that allows the installation of a cast iron. Temperature measurement has been carried out with K-type thermocouples. Because of its small size, the sensor has little effect on testing and is made to identify changes in resistivity with temperature. Eight thermocouples are employed to measure temperature: one to the measure of laboratory temperature, five on the surface of fins in various places, and two on the base cylinder surface. A schematic diagram of the thermocouples' location as is illustrated in Figure 5. Multi-channel data acquisition was used to collect temperature data. The Pico Technology USB TC-08 is an eight-channel temperature recorder meter with an LCD display that uses thermocouple data recording and has a temperature range of -270 to +1820°C (see Figure 3 component-3). Every experiment is repeated three times. To determine the error in the experimental results, an uncertainty study is conducted [21].

The following equations can be used to determine the electrical heat gain rate and uniform heat flux from the outer cylinder base surface:

$$\psi_{in} = \Phi \times I \tag{1}$$

The electrical heat evaluates surface's steady state heat balance can be expressed as follows:

$$\psi_{\text{convection}} = \psi_{in} - \psi_{\text{radiation}} \tag{2}$$

Calculating the average cylinder surface temperature may be conducted by:

$$T_s = \frac{1}{2}\sum_{i=1}^{3} T_{s,\mathrm{i}} \tag{3}$$

For a commercial aluminum tube with an emissivity of 0.028 and a form factor (F = 0.21) [22]. the experimental error at base temperatures below 0.2°C on average. The rate of heat transfer through radiation is lowered by around 5% due to emissivity loss, then $\psi_{\text{radiation}} = 0$.

The following formula defines the heat flux as follows:

$$\psi''_{\text{net}} = \frac{\psi_{\text{convection}}}{A_b} \tag{4}$$

where (A_b) is the base heat sink area is defined as:

$$A_b = 2\left(\frac{\pi}{4}D^2\right) + \pi DL - NLt - 5\left(\frac{\pi}{4}d_{\text{heater}}^2\right) \tag{5}$$

where (d_{heater}) diameter of electric heater.

Figure 1(a) shows the fin area, which may be calculated as follows:

$$A = 2\int_0^{85} (170 - 17.287x^{0.669} + 1.9727x)\,dx + R\left(\theta\frac{\pi}{180}\right)t \tag{6}$$

where θ is the centeral angle equal is 53^{o}

Therefore, we can determine the net heat transfer area by:

$$A_{\text{net}} = A_b + NA_{\text{fin}} \tag{7}$$

The air properties founded by commercial programming software engineering equation solver (EES).

The following expression was calculated for the average free heat transfer coefficient [22]:

$$h = \frac{\psi_{\text{net}}}{A_{\text{net}}(T_s - T_\infty)} \tag{8}$$

The Nusselt number (Nu) as follows.

$$Nu = \frac{hL}{k_{\text{air}}} \tag{9}$$

We define the thermal resistance based on the experimental data as [23]:

$$R_{th} = \frac{(T_s - T_\infty)}{\psi_{\text{net}}} \tag{10}$$

The study of uncertainty is carried out in order to estimate an error in the experimental results. They consist of measuring errors, bias, and accuracy. The accuracy error of Π_T temperature measurement is calculated by.

$$\Pi_T = t_{(N_{\text{data}}-1),95\%} \times \frac{\sigma_T}{\sqrt{N_{\text{data}}}} \tag{11}$$

where $t95\%, N_{\text{data}}$ and $\sigma_t$are the t distribution for a confidence level of 95, number of data and standard deviation of the temperature, respectively [24,25]. Based on data from the manufacturer, the instrument's temperature measurement error for the Φ_T bias error is as follows.

$$\Phi_T = 0.26℃$$

The following is a definition of thermocouple uncertainty:

$$\mathrm{X}_T = \pm\sqrt{{\Phi_T}^2 + {\Pi_T}^2} \tag{12}$$

The uncertainty of the temperature measurement is computed. This expression indicates the X_{Nu} uncertainty in the Nusselt number:

$$\frac{\mathrm{X}_{Nu}}{Nu} = \pm\sqrt{\left(\frac{X_T}{T_s}\right)^2 + \left(\frac{X_T}{T_\infty}\right)^2 + \left(\frac{X_k}{k}\right)^2 + \left(\frac{X_I}{I}\right)^2 + \left(\frac{X_\phi}{\phi}\right)^2} \tag{13}$$

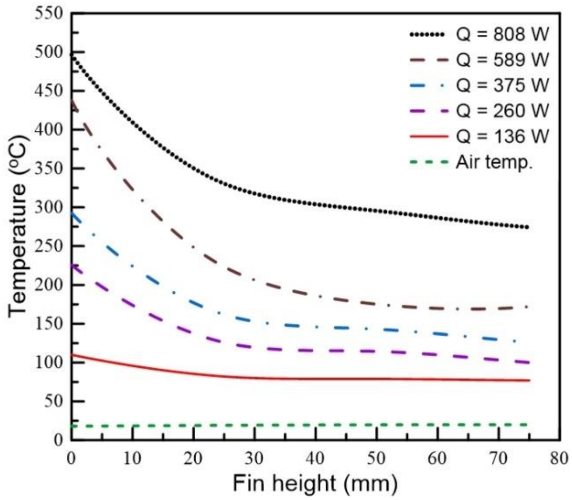
Thermocouples (0.06°C) and a data collection equipment (0.2%+1°C) are employed in this study to measure the temperatures at every location.

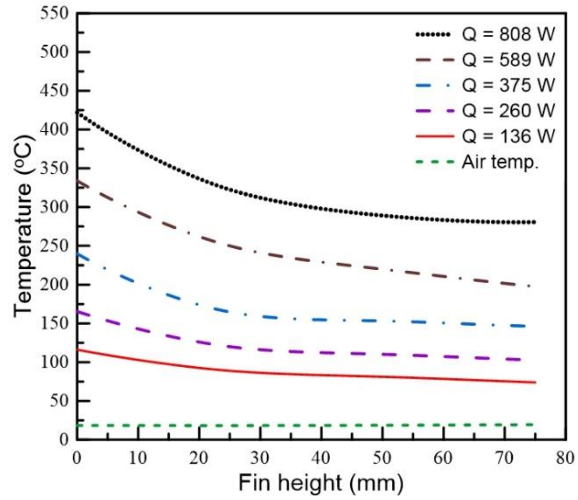
Therefore, the temperature measurement has an estimate of error of ±0.08°C. Hence, for Nu, I, ϕ, R_{th}, and ψ_{net}, the maximum amount of uncertainty is approximately within ±10.1%, ±3.35%, ±0.52%, ±1.34%, and ±3.12%, respectively.

3. Results and Discussion

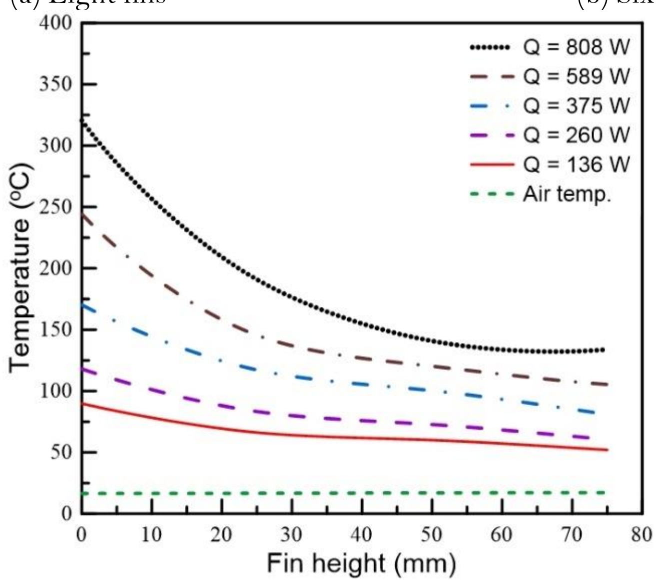
The thermal performance of radial wing fins is examined in this work. Five input heat rates 136, 260,375, 589 and 808 W with fin numbers of eight, sixteen and twenty-four were tested in the experimental investigation. Rectangular fins have been the focus of several experimental studies, as indicated in the previous paragraph. The results were typically displayed as graphs that showed the relationship between fin height, thermal resistance, Nusselt number and temperature distribution. To determine the fin base temperature distribution, several studies were conducted.

Figure 6 illustrates that fin height effects on heat sink temperature for different heat supplies using the several fin number. The figures show the temperature of the laboratory air compared to various surfaces. Generally, as fin height increases, the temperature decreases. The examined cases for free air flow around and over 8th fins with a different input heat rate are shown graphically on Figure 6(a). It is observed that as the number of fins increased, the temperature distribution decreased for all examined fin numbers. A fin prevents the predicted heating of the air at constant input heat rates by increasing the air around and through it. This maintains the temperature difference between the air and fin at a lower enough level to optimize the fins heat-removal effectiveness. The another cases are 2nd and 3rd of fins number are depicted in Figures 6(b) and 7(c).





(a) Eight fins (b) Sixteen fins



(c) Twenty-four fins

Figure 6.
Comparison of the fins number with temperature distribution various heights of the heat sinks.

Figure 7 illustrates the input heat flow influences on the root temperature of the heat sink compared to the ambient temperature for each of the fin number investigated cases. Temperature distributions from the heating elements were shown for different input heat rates. It should be noted that the temperature gradient increases with increasing heat input. However, the illustration's upper curve shows a drop in temperature in the laminar flow region. Because the second curve is near the transitional area, it behaves differently from other curves due to a strong drop. The high temperature is seen in the low fins number. The eight fins have the highest temperature because of the air layers that come into contact with several fin surfaces. It is apparent that the typical fins (wing) in cases 16th and 24th fins have differing maximum temperatures. The greater area of the fins provides this typical. Another significant element that was examined in this study was fin height.

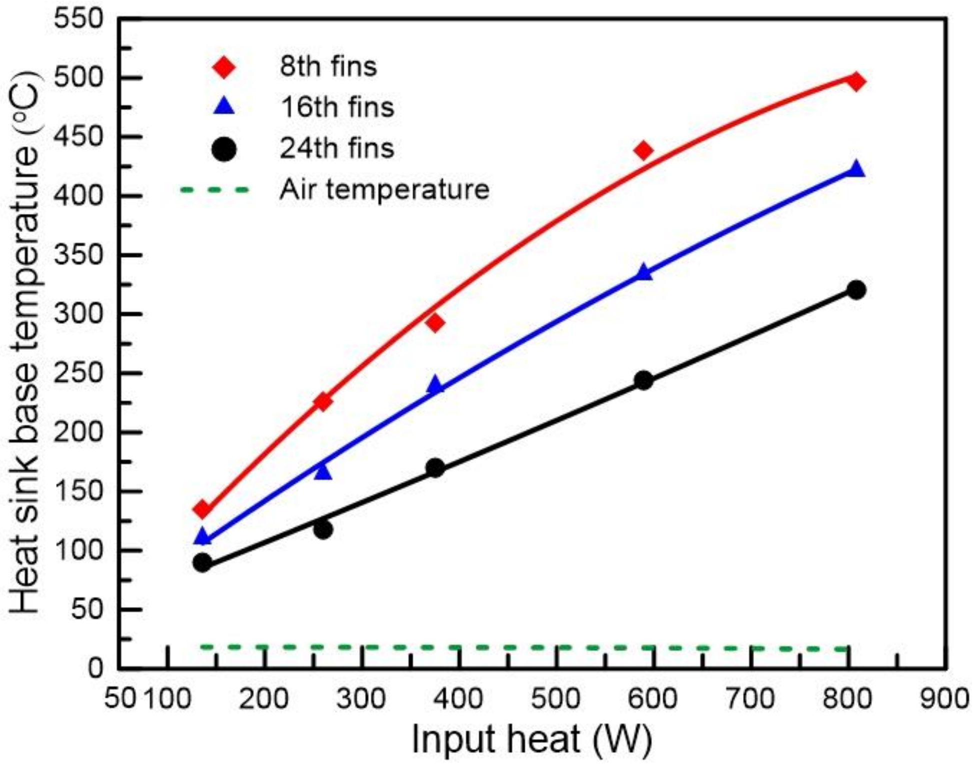


Figure 7.
The base temperature plotted against the input heat for different fins number of the heat sinks.

Figure 8 exhibits the typical Nu number vs. Ra number at three case of fins number. Additionally, but it also shows that in each case, the Nu number increases with increased Ra number. Moreover, it can be noticed that the 24th fins have a peak Nusselt number because to the maximum temperature differential between the ambient and bass heat sink and the heat transfer via free convection. The Nusselt number's peak value in the 24th case makes this clear from the numbers as well. The Nusselt increase with 22% in the 16th fins case and 88% in the 24th fins case compared with ideal case 8th fins, respectively at high input heat rate.

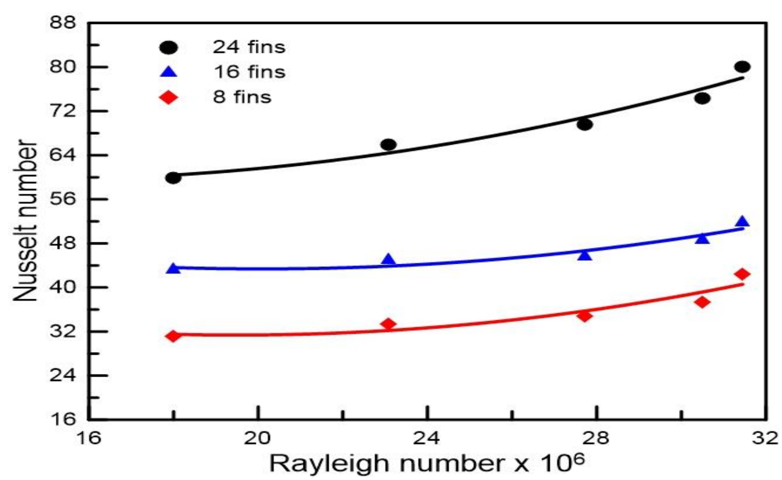


Figure 8.
Nusselt number plotted against Rayleigh number for different fins number.

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The reader must note that the abscissa in Figure 9 a fins number. it is evident that for a given the effect of fins number on the overall Nu number. The results for different input heat were comparable. The findings in Figure 9 proved that the input heat rate continuously increased with more fins number increases. The increasing surface area available for heat transfer caused the increase in coefficient heat transfer. However, because more fins increased the flow resistance, the increase in Nu numbers occurred at an increase of the induced mass flow rates. In summary, the value of the overall Nu number increases with increasing number of fins and increasing input heat flow.

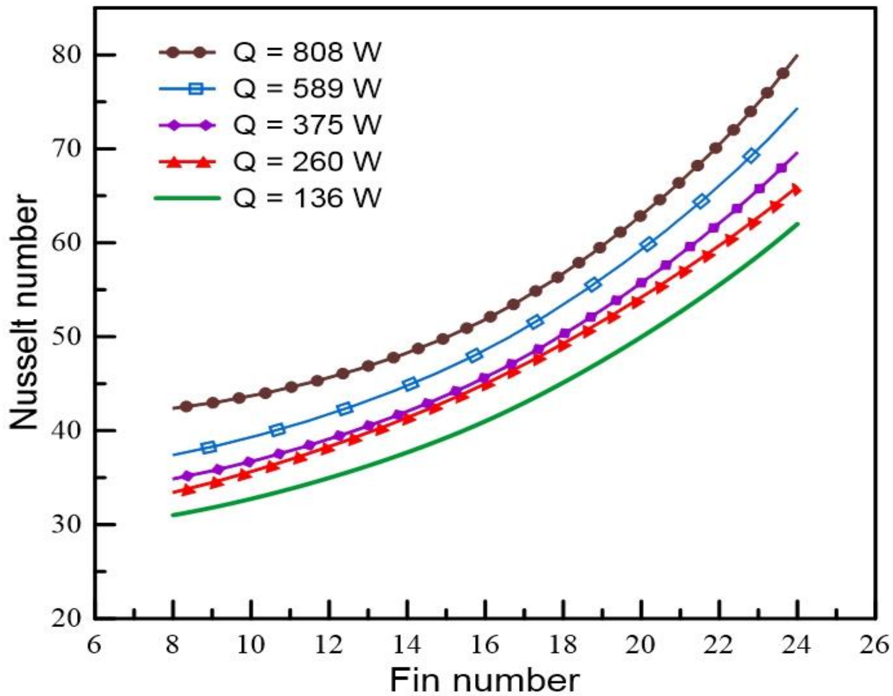

Figure 9.
Nusselt number plotted against fin number at various inputs heat rate.

It is also worthy of note in Figure 10, the effects of Ra number on thermal resistance for different fin numbers. The figure indicates that the thermal resistance decreases as the heat input increases. The figure illustrates that the thermal resistance decreases nonlinearly as the heat input increase. This is explained by the increased fluid flow. Adding more fins increased the rate of heat transfer and reduced the thermal resistance of the heat sink by increasing the effective area of heat transfer. The results show that thermal resistance significantly decreases as fin number increases. Because of boundary layer connect, the effective surface area grows as the overall surface area increases. As a result, the low value of thermal resistance is caused by high input heat.

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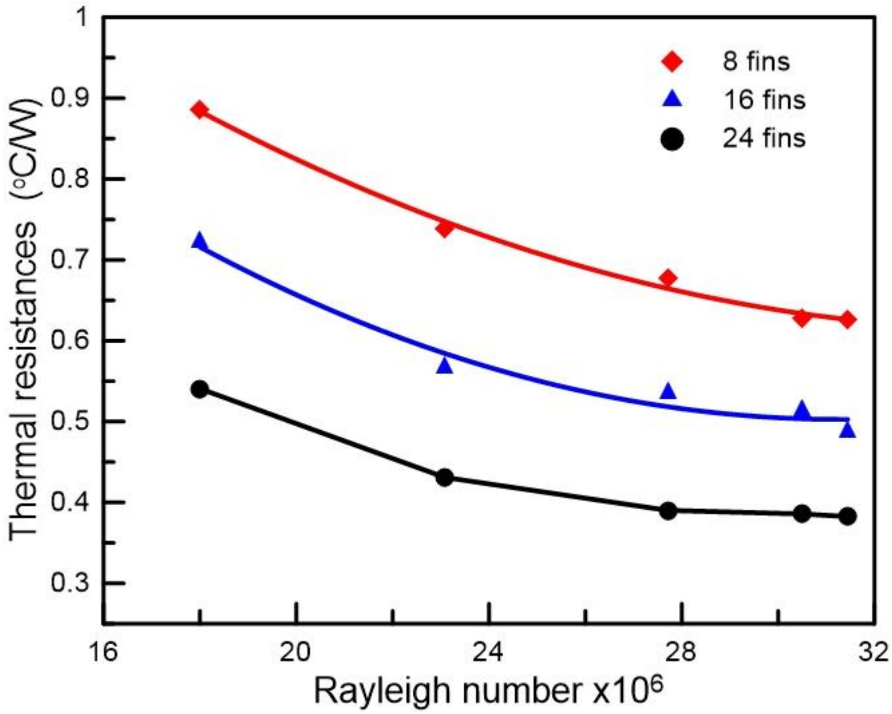

Figure 10.
Thermal resistance of heatsinks at different Rayleigh numbers.

4. Conclusions

The heat transfer in radial heat sinks with wing fins by natural convection at several heat rate input and fin numbers was thoroughly investigated. The heat input has been investigated, and the thermal performance was impacted by the fin numbers (8th, 16th, and 24th). According to the outcome, as input heat increases, it also increases the air coefficient of heat transfer. In each case of fins number, the Nu number increases as input heat increases. Increasing fin height always results in a higher temperature drop than increasing fin number. The continuously drops of thermal resistance as the Ra number increases. Finally, this study clearly shows that a radial heat sink's wing fin number is a significant variable in thermal performance.

Nomenclature

A	surface area, m^2
A_b	base surface area, m^2
cp	air specific heat capacity, J/(kg.℃)
D	base heat sink diameter, m
g	gravitational acceleration, $\mathrm{m/s^2}$
h	coefficient of heat transfer, W/(m^2℃)
I	electrical current, A
k	thermal conductivity, W/(m ℃)
L	height, m
N	number of fins
Nu	Nusselt number
R	radius of fin, m
Ra	Rayleigh number
R_{th}	thermal resistance, ℃/W

t thickness, m
T temperature, ℃
T_s base temperature, ℃
T_∞ amibent temperature, ℃
Greek symbols
α thermal diffusivity, m^2/s
μ dynamic viscosity, kg/(m s)
ρ air density, kg/m
Φ voltage, V
ψ_{in} input heat, W

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