EXPERIMENTAL STUDY FOR OPTIMUM FIN SPACING OF RECTANGULAR FIN ARRANGEMENTS UNDER THE INFLUENCES OF FREE CONVECTION

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Abstract

Heat transfer enhancement systems are widely used in various industries, such as thermal power plants, heating and air conditioning systems, and electronic equipment. In this research, the thermal performance of a vertical heat sink with rectangular fins geometry was investigated experimentally. The present study mainly aims to find the optimum fin spacing that provides the highest heat transfer rate and reduces the heat sink base temperature. An Aluminum heat sink of six 300 mm length fins was considered in the test. The fin thickness and height were kept constant at 4 mm and 45 mm, respectively. In the analysis, a range of fins spacing (i.e., 10, 11, 12, 13, and 15 mm) was used in the experiments, and a variety of Rayleigh number (i.e., from 7.6*10^6 to 1.48*10^8) was considered. The difference of heat sink and surrounding temperatures were obtained at various heat input and all the different parameters above. These temperature differences are then used for the estimation of the heat transfer rate of the fin array. The results showed that the optimum fin spacing was S = 12 mm, which gives the highest heat transfer rate. Besides, a reduction of 18 % of the heat sink weight is achieved due to the use of the optimum fin spacing compared to the weight of using 15 mm as a fin spacing. The reduction of heat sink weight and size can no doubt lead to a decrease in manufacturing cost, which consequently lead to the spread of heat sink applications. Furthermore, the decline of the heat sink base temperature will improve the working life span of the electronic devices that use the heat sink. The results of the present study also show the development of a new empirical equation predicting the Nusselt number in terms of Rayleigh number and the fin spacing to fin height ratio. The developed equation shows the high accuracy of prediction (e.g. about 90%).

Keywords: Fins, Fin spacing, Heat sink, Heat transfer rate, Natural convection.
1. Introduction

Natural convection phenomena in enclosures are necessary for an excellent performance of high-power density electrics. Buoyancy drove flows have many applications in widely preferred phenomena, such as air-cooled car engines, cooling of generators, motors, refrigerators, transformers, and cooling of computer processors. Natural convection is a method of heat dissipation that is relatively inexpensive, discreet, and most dependable. The heat generated by electronic devices can be controlled using fins. In the design of an efficient cooling system, generally, 55% of failure mechanisms in electronic devices are related to thermal effects Pascoe [1]. However, many parameters, such as fins height and fins spacing, affect the rate of convection heat dissipation. These parameters should be considered when the fins best design or the enhancement of heat dissipation rate is targeted.

The subject of investigation of the thermal performance of finned systems still attracts the interest of many researchers. A considerable number of researches have been published. Part of these publications focuses on the development of mathematical formulas that considered the fin arrangement specification. Some other parts of the papers focus on the investigation on fins system arrangements such as fins number, shapes, dimension, and fin spacing. For examples of the former part of studies above (i.e., studies that develop mathematical correlation formula), Elenbaas [2] studied the effect of small gap width between parallel plates on heat dissipation under natural convection condition experimentally.

The results developed a proportional relation between the Nusselt number and the Rayleigh number. Yazicioğlu and Yüncü [3] investigated the thermal performance of the vertical heat sink of rectangular fins under different parameters. The fin base–to-surrounding temperature difference ranges from 30 to 150 K. The fin length is taken as 250 mm and 340mm. The space between the fins and the height of the fin was from 4.5 to 85.5 mm and 5 to 25 mm, respectively. The authors reported that the optimum value of fin spacing varies with the height of the fin, which was from 6.1 to 11.9 mm. Besides, the rate of heat transfer enhanced depends on the average base temperature, fin length, and fin spacing.

The influence of fin height, fin spacing, and fin orientation on the natural heat transfer coefficient were studied experimentally by Walunj, et al. [4]. Bar-Cohen and Rohsenow [5] conducted an analytical investigation for the natural convective heat transfer from two parallel plates. Their work includes the development of a correlation formula describing Nusselt number in terms of Rayleigh number for isothermal and isoflux plates. The optimum fin spacing was also reported in the same study.

Bodoia and Osterle [6] developed a numerical technique for the investigation of flow in channels and heat transfer between symmetrically heated, isothermal plate to predict the channel length required to achieve fully developed flow as a function of the channel width and wall temperature. The heat transfer in inclined interrupted fin channels was investigated in Fujii [7]. A mathematical correlation was developed for fitting the experimental results. As was mentioned previously, some publications investigated the influence of some fin geometrical properties such as fins number, fins shapes, fins dimension, and fin spacing. Experimental studies on vertical rectangular fin arrays ware carried out by Güvenç and Yüncü [8] and Leung and Probert [9].
Numerical studies on vertical rectangular fin arrays were investigated by Fitzroy [10], Saikhedkar and Sukhatme [11]. Yüncü and Anbar [12] performed experiments on the free convection heat transfer of the horizontal rectangular fin. Fin thickness and fin length were kept constant at 3 and 100 mm, respectively, whereas the other variables such as fin spacing and fin height varied at from 6 mm to 26 mm and 6.2 to 83 mm respectively. The results showed that the fin spacing to fin height ratio was the strongest influence on the free convection heat transfer coefficient. However, the temperature difference decreases with the increase of fin height. The optimum fin spacing was 11.6 and 10.4 mm, when fin height was 16 and 26 mm, respectively.

Yazıcıoğlu and Yüncu [13] developed a new expression to predict the optimum fin spacing of vertical rectangular fins under natural convection condition. The study tested different parameters of an extended area such as fin spacing and fin height. The results indicated that the rate of heat transfer enhanced when the fin length, fin height, and fin base-to-surrounding temperature difference increased. Kumar, et al. [14] investigated the performance of a triangular fin model within a vertical orientation and with several influencing parameters such as the fin spacing. The results developed an empirical correlation connecting the Nusselt number to these parameters. It was found that the fin spacing has a significant impact on all geometric parameters of the heat sink. Also, the relation between fin height and fin spacing was reversed.

Shehab [15] investigated experimentally the effect of the horizontal heat sink that has various fin spacing and number of fins on the heat transfer coefficient and Nusselt number under natural convection condition. The results showed that the fin spacing has a significant effect on the average Nusselt number. Karami, et al. [16] examined three types of finned tube exchangers with square fins and different spacing (5, 9, and 14) mm experimentally. The Nusselt number and heat transfer coefficient enhanced with the increase of the fin spacing. The performance of the vertical heat sink under natural convection is examined by Al-Jewaree [17]. Different materials and fin spacing are used in this study. The results showed that the heat transfer rate improved when the fin spacing decreased and, the maximum improvement was 22%.

An experimental based study has been conducted by Al-Jessani and Al-Bugharbee [18] to examine the effect of circular perforations on the weight of the heat sink. The results found that the optimum number of perforations per fin was 10. Also, the heat sink weight was reduced by 7%. Leung, et al. [19-22], Leung and Probert [23, 24] and Van de Pol and Tierney [25] are some examples, which were mostly focused on the effects of varying fin geometric parameters, the array, and base plate orientation.

The heat transfer under natural convection and radiation of twelve vertical fins system are investigated in Chaddock [26]. The fin width plate kept constant during the experiments while the fin spacing and fin height is varied. The latter showed a significant influence on the radiation of heat transfer.

Recently, the natural convection in porous fins and enclosures with a porous medium is investigated in Ahmed, et al. [27], Ahmed, et al. [28], Hoseinzadeh, et al. [29] and Hoseinzadeh, et al. [30]. The present study focuses on investigating the effect of space between the fins on the rate of heat transfer, heat transfer coefficient and fin base-to-surrounding temperature difference. The present study offers an investigation for finding the optimum fin spacing, which provides best
heat dissipation of the heat sink. Consequently, this optimum spacing will no doubt reduce the weight and manufacturing cost of the heat sink. In addition, an empirical equation was developed for describing the relation between Nusselt number, Rayleigh number, and dimensionless parameter of fins spacing and fins height.

2. Experimental Test Rig

An experimental device was designed and made from several parts, which are assembled to install the test section unit. The instrument has been used for measuring the base-to-surrounding temperature difference from a vertical rectangular fin configuration to calculate heat transfer parameters. The tested frame was manufactured from square cast iron material of a height of 320 mm and a cross-sectional area (10*10) mm². The structure is designed with rectangular shapes and fastened on four stands. The frame was covered with sheet aluminium of 0.9 mm thickness, which was insulated with a stratum of wool thermal of 15 mm thickness. The unique plate material is used as a reflector of heat with a thickness of 0.5 mm. Three heaters of 600 W are used in the test section. The heaters were covered by a glass tube with a diameter of 20 mm. The fins are installed on the test section by a sliding channel and are controlled by locking screws.

The fins have been manufactured with aluminium metal due to its high thermal conductivity and low emissivity at 20 °C. These fins were formed by making several longitudinal channels along the top face of the rectangular bar. Fins length L and the fin height H are kept as 300 mm and 45 mm respectively during all the experiment work. Fins arrangement width W are varied as 95, 100, 105, 110, and 120 mm. Different fin spacing S is used, namely 10, 11, 12, 13, and 15 mm. Fins were kept integral with 7 mm thickness of the base plate, the fin thickness is remained fixed at t = 4 mm, and the number of fins is n = 6. The geometry of in-line continuous rectangular fins shown in Fig. 1. The base plate is kept at constant heat flux, and the upper surface of the fins is at the environment temperature.

![Heat sink dimension and locations of thermocouples](image)

**Fig. 1. Heat sink dimension (left) Locations of thermocouples (right).**

3. Methodology

The methodology contains constructing a test rig of the finned heat sink for conducting several experiments to obtain an average base and ambient temperature
at different heat inputs. The heat input can be determined by knowing the supplied electrical voltage and current provided to the electrical heater. Then, the convection heat transfer rate can be determined by subtracting the radiation and loss heat rates from the generated heat transfer (e.g., input heat). Different convection heat transfer coefficients can then be obtained at different input voltage and current, different heat sink fins spacing. Eventually, an optimum fin spacing can be found as that fin spacing providing heights heat transfer coefficients. Besides, an empirical equation predicting the relation of the Nusselt number in terms of Rayleigh number and the fin spacing to fin height ratio. The reduction of heat sink weight and size can no doubt lead to a decrease in manufacturing cost, which consequently lead to the spread of heat sink applications.

4. Experimental Procedure

The experiments were conducted at various heat flux. This variation is made through controlling the heater voltage by a voltage regulator. The provided voltage to the heater is varied from 50 to 150 V by 25 V step. The surrounding temperature was measured via a thermometer. The fin base plate temperature was acquired at eight positions to calculate the average temperature along with the base plate. The temperature was measured using a temperature scanner thermometer (8 channel thermocouple) manufactured by Altop Industries Ltd, Gujarat, India (SI. 1005164. Model ADT 5003) with a resolution of 0.1. The average of these eight positions represents the fin base temperature. Figure 2 shows the experimental rig with the measuring instrument actions. The predefined heater inputs were modified with the assistance of dimmer stat. The temperature of an aggregated fin heat sink array of various locations and surrounding temperatures was registered during time intervals of 30 min. It requires close to four hours to reach a steady condition. Consequently, when the temperature reading does not vary by 0.3°C, in this case, the system reaches a steady-state condition. The experimental tested was carry out in the Fluid Dynamics Laboratory of the mechanical engineering department/Wasit University.

Fig. 2. Photograph of the experimental rig with the measuring instruments.

5. Analytical Analysis

The heat input of the heating base was dissipated based on two approaches, primarily through the heat sink and, slightly by the insulation fins and heat sink
structure. A voltage $V$ is provided through the voltage regulator to the heater, which is fixed on the heat sink base. The temperature from the eight positions (predefined in the previous section) as well as the ambient temperature is acquired. The provided voltage and the received temperature are used for the calculation of $Q_{\text{conv}}$, as in the following equations Jubear [31].

$$Q_{\text{gen.}} = Q_{\text{conv.}} + Q_{\text{rad.}} + Q_{\text{loss}}$$  \hspace{1cm} (1)

where

$$Q_{\text{gen.}} = V \times I$$  \hspace{1cm} (2)

And the heat transfer by radiation was:

$$Q_{\text{rad.}} = \sigma \times \varepsilon \times S_{\text{sur.}} \times A_l(T_{\text{av.}}^4 - T_{\text{air}}^4)$$  \hspace{1cm} (3)

The loss is estimated according to the temperature difference between the heat sink base and ambient, as follow Feng, et al. [32]:

$$Q_{\text{loss}} = -9 \times 10^{-5} \times \Delta T^2 + 0.0202 \times \Delta T$$  \hspace{1cm} (4)

In the present study, the heat loss for all tested was less than 4% of the overall power input. Consequently, for these ranges of heat input, the heat loss is ignored. Therefore, the convection heat transfer was calculated as follows:

$$Q_{\text{conv.}} = Q_{\text{gen.}} - Q_{\text{rad.}} - Q_{\text{loss}}$$  \hspace{1cm} (5)

Hence, the heat transfer coefficient in free convection was calculated from the equation which, is known as Newton’s cooling law, as follows:

$$h = \frac{Q_{\text{conv.}}}{A_l(T_{\text{av.}} - T_{\text{air}})}$$  \hspace{1cm} (6)

Totally exposed area

$$A_l = n \times A_f + L \times s \times (n - 1)$$  \hspace{1cm} (7)

The ratio of the convection heat flux to the conduction heat flux is known as the Nusselt Number and written as follows:

$$Nu = \frac{h L}{K_f}$$  \hspace{1cm} (8)

The Rayleigh Number was defined as the ratio between buoyancy and viscosity and written as follows:

$$Ra = \frac{l^3 g \beta (T_{\text{ave.}} - T_{\text{air}})}{\nu^2 Pr}$$  \hspace{1cm} (9)

6. Results and Discussion

The experimental study used the vertical rectangular fin, based on six fins model with different configurations of fin spacing ($S = 10, 12$ and $15$ mm), length $L = 300$ mm, height $H = 45$ mm and thickness $t = 4$ mm, respectively, under a different heat flux ($78, 176, 317, 496$ and $715$ W). The experiments were performed using the Rayleigh Number ($7.6 \times 10^6$ – $1.48 \times 10^8$). The results show the effect of spacing among the fins on the heat transfer coefficient, Nusselt number ($Nu$), and the Rayleigh Number ($Ra$). This parameter can be discussed, given the following.
6.1. Model validation

The present experimental results have been compared with Yazicioğlu and Yüncü [3], Tari and Mehrdash [33], Tari and Mehrtash [34] and Mehrtash and Tari [35]. The values of optimum fin spacing were 11.9 and 11.75 mm. It was noted there is a good agreement between the present work and the other research, as shown in Table 1, and the error is no more than 2%.

<table>
<thead>
<tr>
<th>Reference No.</th>
<th>Fin Length ((L))</th>
<th>Base width ((W))</th>
<th>Fin height ((H))</th>
<th>Fin Thickness ((t))</th>
<th>Fin Spacing ((S))</th>
<th>Optimum Fin Spacing</th>
</tr>
</thead>
<tbody>
<tr>
<td>3</td>
<td>250,340</td>
<td>180</td>
<td>5,15,25</td>
<td>3</td>
<td>5.8-85.5</td>
<td>10.4-11.9</td>
</tr>
<tr>
<td>33,34,35</td>
<td>250,340</td>
<td>180</td>
<td>5-25</td>
<td>3</td>
<td>5-85.5</td>
<td>11.75</td>
</tr>
<tr>
<td>Present Work</td>
<td>300</td>
<td>100-150</td>
<td>45</td>
<td>4</td>
<td>10-15</td>
<td>12</td>
</tr>
</tbody>
</table>

6.2. Temperature distribution

To clarify the experimental results by a more reliable technique the Fluke Ti32 Thermal Imager (hereafter “the Imager”) was used to show the temperature distribution on the fin surface from different sides (top and side view). The Thermal Imager uses various colours or shades of grey to show the temperature distribution on the fin surface over the Imager’s domain of view, as shown in Fig. 3.

Fig. 3. Snapshots of temperature distribution imager.
Figure 3 is presented for further visualisation purpose as several considerations should be taken for achieving accurate readings. This is because both local emissivity and the background temperature affect the IR system. The accuracy of the IR reading is also influenced by several factors such as air temperature and wind speed.

6.3. Temperature difference variation

The influence of the fin’s base-to-surrounding temperature difference on the coefficient of heat transfer, the rate of heat transfer, and Nusselt number according to different fins spacing of ($S = 10, 11, 12, 13, \text{ and } 15\text{mm}$) are presented in Figs. 4 to 6. These parameters possibly increase, according to increasing of the fin base-to-surrounding temperature difference for all fin spacing values. The behaviour of the heat transfer coefficient with the fin base-to-surrounding temperature difference is shown in Fig. 4. The coefficient of heat transfer improves with the increase of the temperature difference between the average base temperature of the heat sink and the surrounding temperature. Besides, the heat transfer coefficient has a maximum value when the fin spacing is 12 mm; despite that, the temperature difference has been reduced. The figure also shows that the benefits of the heat transfer coefficient become closer when the temperature difference is slight. In other words, the heat flux on the base of the heat sink is weak, while it diverged as a result of the high range of the heat flux. Finally, the enhancement of the fin base-to-surrounding temperature difference (reduction of the heat sink base) at 12 mm fin spacing was 25% compared with 15 mm.

![Fig. 4. The influence of temperature difference on heat transfer coefficient.](image-url)

Figure 5 shows the effect of fin base-to-surrounding temperature difference on convective heat transfer. The result illustrates that the rate of heat transfer of the heat sink is improved when the temperature difference increased for all values of fin spacing. Also, the space between the fins and the fin base-to-surrounding temperature difference parameters have strongly influenced on the heat transfer rate of the heat sink. The rate of heat transfer is enhanced dramatically, according to the fin base-to-surrounding temperature difference. The behaviour of the Nusselt
number with the fin base-to-surrounding temperature difference is similar to the action of heat transfer coefficient conduct, as shown in Fig. 6. A straight line cannot represent the result, and as a result, an exponential equation has been used.

6.4. Heat transfer coefficient variation

The relation between heat transfer coefficient and several variables such as rate of convection heat transfer, and Nusselt number with different fin spacing ($S = 10, 11, 12, 13,$ and $15$ mm) can be seen in Figs. 7 and 8. The proportional relation between the heat transfer coefficient and the heat transfer rate is shown in Fig. 7. This figure exhibited, the convection heat transfer rate is decreased when space between fins was increased, reached a minimum value at fin spacing 12 mm, and then grew and reached a maximum value at fin spacing 15 mm. It can be observed that the same behaviour was correct in each of the previous action for the Nusselt number, as shown in Fig. 8.

6.5. Fin spacing variation

The influence of fin spacing on the fin base-to-surrounding temperature difference, free convection heat transfer coefficient and Nusselt number can be observed in
Figs. 9 to 11. The effect of heat input on the fin base-to-surrounding temperature difference can be seen more clearly in Fig. 9. In this figure, the fin base-to-surrounding temperature difference is plotted as a function of fin spacing. It can be deduced that the fin base-to-surrounding temperature difference from a fin array decreases with the space of fins, and reach to a minimum level when the fin spacing was 12 mm, and then increasing. The corresponding fin spacing value of the minimum base-to-ambient temperature difference is named the optimum fin spacing. It was noted that the optimum fin spacing of this study was $S = 12$ mm. This result reveals that the optimum value of fin spacing was hypersensitive to the variations in heat generated and temperature difference parameters.

The values of the natural convection heat transfer coefficient obtained for different heat inputs are shown in Fig. 10. It can be seen that when the space between two fins (i.e., gap) increases from 10 to 12 mm, the free convection heat transfer coefficient ultimately increases. Besides, a further increase in the fin spacing leads to a decrease in this coefficient. If the gaps between the fins are evenly matched or small, the free convection heat transfer coefficient is minimized as mixing of the thermal boundary layer occurs. Figure 10 clearly shows that the heat transfer coefficient decreases as the gap between fins decreases. However, if the fins are closely spaced, there is also a more dissipating surface area (more fins for a given volume). The additional surface area can counteract the reduced heat transfer coefficient.

More precisely, a variation of the convective heat transfer coefficient with different heat input is studied in this figure. Initially, the heat transfer coefficient enhanced when the space among the fins increases. It reaches a maximum value at fin spacing ($S = 12$ mm), and with further additions of fin spacing, it begins to decrease. The space between the fins is called optimum value when the convection heat transfer coefficient has maximized. Inspection of Fig. 10, reveals that the optimum fin spacing value was $(S = 12$ mm) at the different heat input. One of the essential parameters in designing a heat sink is the fin spacing, $S$. The enhancement
of the heat transfer coefficient due to optimum fin spacing was 27%. The same behaviour is shown in Fig. 11.

Fig. 10. The effect of fin spacing on heat transfer coefficient of various heat input.

Fig. 11. The effect of fin spacing on Nusselt number of various heat input.

6.6. Nusselt number variation

Figures 12 and 13 show the variations of Rayleigh number and fin heat transfer rate with Nusselt number for all the fin spacing. Figure 12 shows the continuous increase in Nusselt number with a rate of heat transfer as reported for all the fin spacing. A straight line cannot represent the relation between the Nusselt number and the heat transfer rate. As a result, an exponential equation has been used. The variations of the Nusselt number with fin spacing have been illustrated in Fig. 13, at a particular fin height value. The Nusselt number increases continuously with an increase in fin spacing for all the Rayleigh number values. It reaches a maximum value at the fin spacing 12 mm. An increase in the Nusselt number immediately indicates an enhanced convection heat transfer rate.

Fig. 12. The relation between heat transfer rate and Nusselt number.

Fig. 13. The relation between Nusselt number and Rayleigh number.

It is essential to mention that 18% of heat sink weight is achieved due to the use of the optimum fin spacing compared to use the 15 mm. The reduction percentage can be calculated either by 1) weighing the heat sink at using fin spacing
of 15 mm and optimum fin spacing (e.g., 12 mm) or 2) calculating the heat sink volume and multiplying by the material density to fins the mass.

6.7. Correlation equation

In the current study, different correlation formulas were investigated to find the best representation of the experimental readings. The developed empirical equation was in the form of a nonlinear equation, which is a general form as in Eq. 10.

\[
N_u = C_1 + C_2 \left( \frac{L_c}{H} \right)^{C_3} + C_4 \times Ra^{C_5}
\]  

(10)

where \( C_1 = -1095 \), \( C_2 = -389 \), \( C_3 = 222.8 \), \( C_4 = 0.8 \) and \( C_5 = 0.1 \)

Figure 14 compares the predicted and experimental Nusselt number readings. In this figure, the x-axis represents the data points of the experimental test; the y-axis represents the Nusselt number. The data points refer to the number of recorded points. It can be seen from this figure that both predicted and experimental results have the same trend and fluctuation.

![Fig. 14. The relation between experimental and predicted data.](image)

The accuracy of the empirical equation can be statistically assessed using the following measures Zubaidi, et al. [36]:

\[
%RE = \frac{\sum_{m=1}^{N} \left( y_o - y_p \right)^2}{\sum_{m=1}^{N} y_o^2} \times 100
\]  

(11)

\[
NRMSE = 1 - \frac{\sum_{m=1}^{N} \left( y_o - y_p \right)^2}{\sum_{m=1}^{N} y_o^2} \times \frac{N}{\sum_{m=1}^{N} (y_o - \bar{y}_o)} \times \frac{N}{\sum_{m=1}^{N} (y_p - \bar{y}_p)}
\]  

(12)

\[
R = \frac{\sum_{m=1}^{N} \left( y_o - \bar{y}_o \right) \left( y_p - \bar{y}_p \right)}{\sqrt{\sum_{m=1}^{N} \left( y_o - \bar{y}_o \right)^2 \sum_{m=1}^{N} \left( y_p - \bar{y}_p \right)^2}}
\]  

(13)

where:

- \( RE \) Relative error.
- \( RMSE \) Root mean square error
- \( NRMSE \) Normalized root mean square error
Figure 15 shows the residuals of the predicted and experimental Nusselt number. It is clear from the figure that the residuals fluctuating around zero and has randomly distributed.

![Residuals graph]

**Fig. 15.** The residuals of the experimental and predicted data.

The statistical measures of the predicted and experimental results show that $\%RE = 10$, NRMSE= 0.43 and $R= 0.84$.

For the present work, the uncertainty of the experimental results is obtained using mathematical formulas in Moffat [37]. The uncertainty parameter for the measured base temperature was $\pm 1.2$°C. The correlation coefficient of the innovative and predicted Nusselt values are computed and found 0.82.

**7. Conclusions**

In this study, the effect of different fin spacing, heat flux, and Rayleigh number (Ra) on the performance of vertical rectangular fins under free convection heat transfer was investigated experimentally. The heat transfer coefficient and base temperature are obtained analytically. From the different results, it was found that the optimum fin spacing is $S=12$ mm. This optimum $S$ was selected in terms of providing maximum heat transfer coefficient, maximum Nusselt number, and heat transfer rate. Additionally, the fin base-surrounding temperature increases by 25% at this fin spacing. The selection of optimum $S$ also leads to a18% of fin weight reduction. In the present study, a precise empirical correlation equation was also developed for describing the Nusselt in terms of Rayleigh number and the fin spacing to fin height ($S/H$). The maximum error of the equation prediction was 10 %. The studies on the selection of fin spacing are still of great of importance as it provides practical guidelines for manufacturing heat sinks with lighter weights, less expensive cost and smaller size.
Nomenclatures

\( A_f \) Exposed fin area, \( m^2 \)
\( A_t \) Total exposed area, \( m^2 \)
\( g \) Gravitational acceleration, \( m/s^2 \)
\( h \) Convection heat transfer coefficient, \( W/m^2 \degree C \)
\( I \) Electrical current, Amper
\( K_f \) Thermal conductivity, \( W/m \degree C \)
\( L \) Fin length, \( m \)
\( n \) Number of fins
\( \text{Nu} \) Nusselt number
\( Pr \) Prandtl number
\( Q_{\text{conv.}} \) Convection heat transfer, \( W \)
\( Q_{\text{gen}} \) Heat generated, \( W \)
\( Q_{\text{rad}} \) Radiation heat transfer, \( W \)
\( Ra \) Rayleigh number
\( s \) Fin spacing, \( m \)
\( S_{\text{sur}} \) The shape factor
\( T_{\text{air}} \) Air temperature, \( \degree C \)
\( T_{\text{av.}} \) Average temperature, \( \degree C \)
\( V \) Electrical voltage, volt

Greek Symbols

\( \sigma \) Steven-Boltzmann, \( W/m^2 \degree K^4 \)
\( \beta \) The volume coefficient of expansion, \( \degree K^{-1} \)
\( \varepsilon \) Emissivity
\( \nu \) Kinematic viscosity, \( m^2/s \)

References


