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Original Research Article

# Natural convection from perforated vertical fins with different hole diameters



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## ABSTRACT

Natural convection is a physical mechanism that is mostly benefited in cooling of electronics. Due to the variety of the geometrical and operational parameters, industrial and scientific studies are continuing for better performance. One of the focuses is on perforations of heat sink fins as a passive flow control technique. This work experimentally investigates a sinusoidal wavy fin heat sink after fins were perforated with two different hole diameters. Heat sink was heated by using an electrical resistance for six different heating powers. The temperature at the heater-heat sink interface was measured with the aid of four thermocouples. Transient and steady temperature values were measured and then recorded by means of a data-logger. The details of the experimental setup are given alongside of visuals. It is desired to state some assessment and evaluations about the experimental setup. Related literature studies are also summarized in the introduction section. Heat transfer, Rayleigh and Nusselt numbers were calculated and compared with each other and parameters by means of 2D plot graphics. The time for reaching steady state is changing between 1.5 to 2 hours. Average wall temperature changes linearly with heating power. Average wall temperature values are between 300 and 350 Kelvin. Nusselt number decreases with increasing Rayleigh number. While Rayleigh number changes between 500,000 to 2,000,000, Nusselt number decreases from 425 to 250. It is concluded that increasing perforation hole diameter slightly increases convection heat transfer. Some remarks for the future work is given in conclusion section.

*Keywords:* Heat sink, Heat transfer, Natural convection, Perforated fin, Sinusoidal wavy fin

## 1. Introduction

The methods that are being used for cool electronic devices vary greatly depending on the applications and required cooling capacity. The fins are widely used in the cooling of computer processors, aircraft engines, air-cooled automobile engines, generators, motors, transformers, refrigerators and other electronic devices. It is a very important element in the geometry of heat sinks. Many studies have been conducted to determine the performance of heat transfer surfaces. These studies can be exemplified as; different geometry designs, the arrangement of fins, different position angles, holes on the fins, hole geometries. One of the methods of improving the natural convection heat transfer using the heat sinks is to drill holes. Because boundary layer formed on the fins is broken by the opening holes and possibility of better heat transfer is achieved. Before giving detailed information about the study, review of the related literature is given below.

Harahap and McManus [1] worked on rectangular fins on a horizontal plane. Heat sinks consist of eight different lengths and ranges of rectangular shaped fins. In their work, the flow area is examined. They have seen that air flows through the open ends goes up from the middle of the fin for the long fins. They stated that the fin length is the most important geometric parameter. Liang and Yang [2] investigated the effects of friction loss performance and perforation geometry on heat transfer of compact heat exchangers with perforated rectangular plate surfaces. Patankar et al. [3] report that perforated blades can be used for effective heat transfer area and/or heat transfer coefficient. According to the authors, the change in the size of the surface area depends on the geometry of the holes. In 2008, Jasim et al. [4], tested the natural convection of a perforated fin. The temperature distribution was examined for a series of rectangular fins (15 fins). Cross-sectional area (100 x 270 mm) placed on different vertical bodies. Molds of the holes comprise 18 circular holes. Experiments were carried out in a pilot facility. They showed that the heat transfer rate and the heat transfer coefficient increase when the diameters of the holes are increased. Shaeri et al. [5] investigated turbulent fluid flow and convection heat transfer around a different number of perforations and a series of rectangular solids of different sizes. Results are based on the fin thickness from 2000 to 5000 and pr = 0.71. The aim of their study was to investigate the mechanism of heat transfer mechanism by changing surface geometry and heat exchanger configuration. They stated that increasing the number of holes, the temperature at the base of the fins and the tip of the fin becomes greater, which means that the efficiency of heat transfer between the surface and the flow of fluids with convection is higher. Masao et al. [6] report a work in which the surface has many perforations and is bent to form a trapezoidal shape. Dimensionless correlations on heat transfer and pressure drop are presented. Kumbhar et al. [7] tested the different shape of holes that give different effects to heat transfer coefficient, low pressure and heat distribution. The heat transfer rate is different for different materials. Their experiment was carried out triangular shape of the hole at the tip of the fin. Similar result was found with triangular and elliptic perforation for the circular shape of perforation in the paper of Qui et al. [8]. The transfer coefficient increases, the pressure drop decreases and the temperature distribution throughout the fins increases. The number of perforations also becomes the main feature in increasing the perforation to the plate fin. Al-Essa and Al-Odat [9] examined the effect of triangular perforations on the heat transfer performance of a plate fin exposed to natural convection. They confirmed that they had optimum drilling size and optimum gap values. It is claimed that it can give the maximum heat dissipation rate through the perforated fin. Kim [10] compared the thermal performance of pin fin heat sink and plate fin heat sink for various channel lengths and blade heights.

This work distinguishes from the literature by its specific sinusoidal wavy fins with a certain amplitude on a base plate as a heat sink for natural convection heat transfer. Also 3 mm and 6 mm inline symmetrical holes were drilled on two identical heat sinks in order to show their effect on the heat transfer. An experimental setup and procedure were utilized which is also explained in the following sections. This experimental data is contributing the literature as a validation tool for Computational Fluid Dynamics (CFD) and possible empirical correlations. The data also can be used directly for industrial purposes since the operational conditions are very similar to cooling of electronics.

## 2. Material and Methods

This section contains two subsections. The first one gives information about the heat sinks that are investigated in respect of the natural convection heat transfer and the experimental equipment that were used for the investigation. The second subsection presents the equations that are used for data reduction.

### 2.1. Fins and equipment

Technical drawings of the heat sinks are given in Figure 1 and 2, respectively for 3 mm and 6 mm hole dimensions.

These two heat sinks were placed on a heater in an insulated box in order to be ensure one dimensional conduction towards heat sinks. Heat sinks that are located on the heater and insulation box are given in Figure 3. Complete test setup is given in Figure 4. The power of the electrical heater is arranged by a dimmer. The power of the electrical heater is read from wattmeter screen. Yet, it is also verified by calculating the power using the ampere value of the circuit using a digital multi-meter (VOM) and multiplying it with the known grid voltage which is 220 V. Temperatures were recorded by a 10 channel data-logger. Only five thermocouples were used during experiments. One of them was for ambient temperature and remaining four were for the wall temperature of the heat sink. Four thermocouples for the wall temperature were embedded in the surface of the heat sink that was contacting to the heater. Their average was used during the calculations.

Figure 5 is given as an example of the thermocouple measurements by means of the data-logger. The image is from the interface of the data-logger. As seen in the figure, temperature resolution is 0.1 Kelvin or Degrees Celsius. Experiments were conducted for time periods between 4000 to 8000 seconds. Because, it was desired to have the steady state operation conditions.

The experimental setup is an improved version of a previous setup that was used in a MSc study [11].

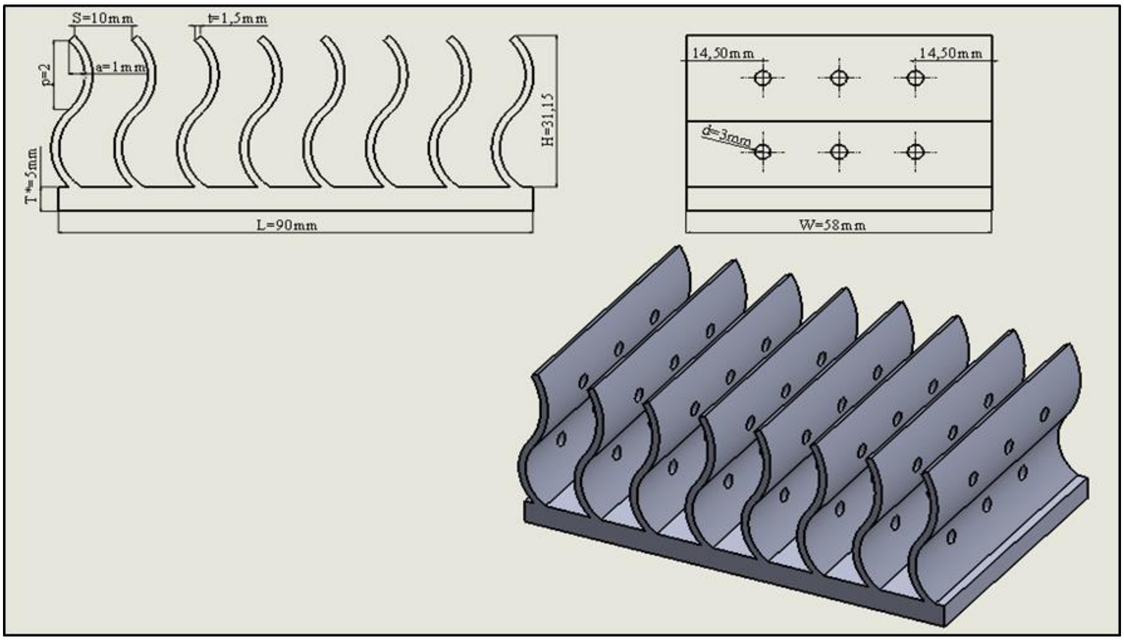



**Figure 1.** Sinusoidal wavy fin heat sink with 3 mm holes

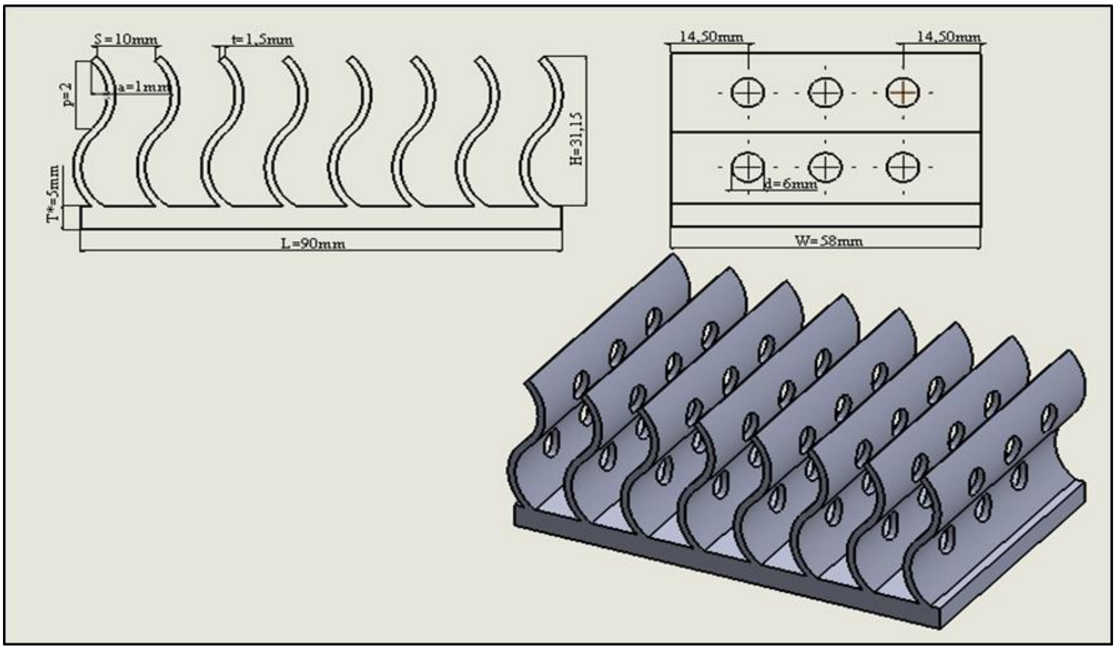
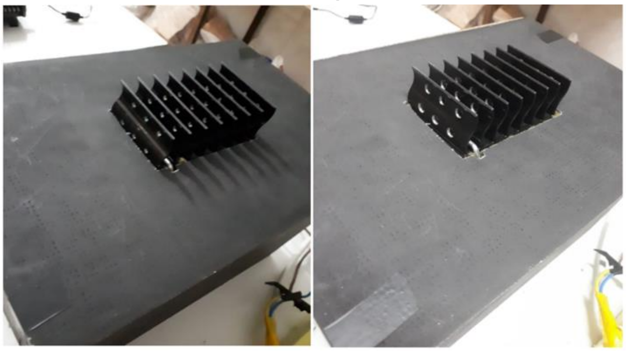



**Figure 2.** Sinusoidal wavy fin heat sink with 6 mm holes



**a**- 3 mm perforations **b**- 6 mm perforations

**Figure 3.** Heat sinks on experimental stand


**Figure 4.** Experimental setup

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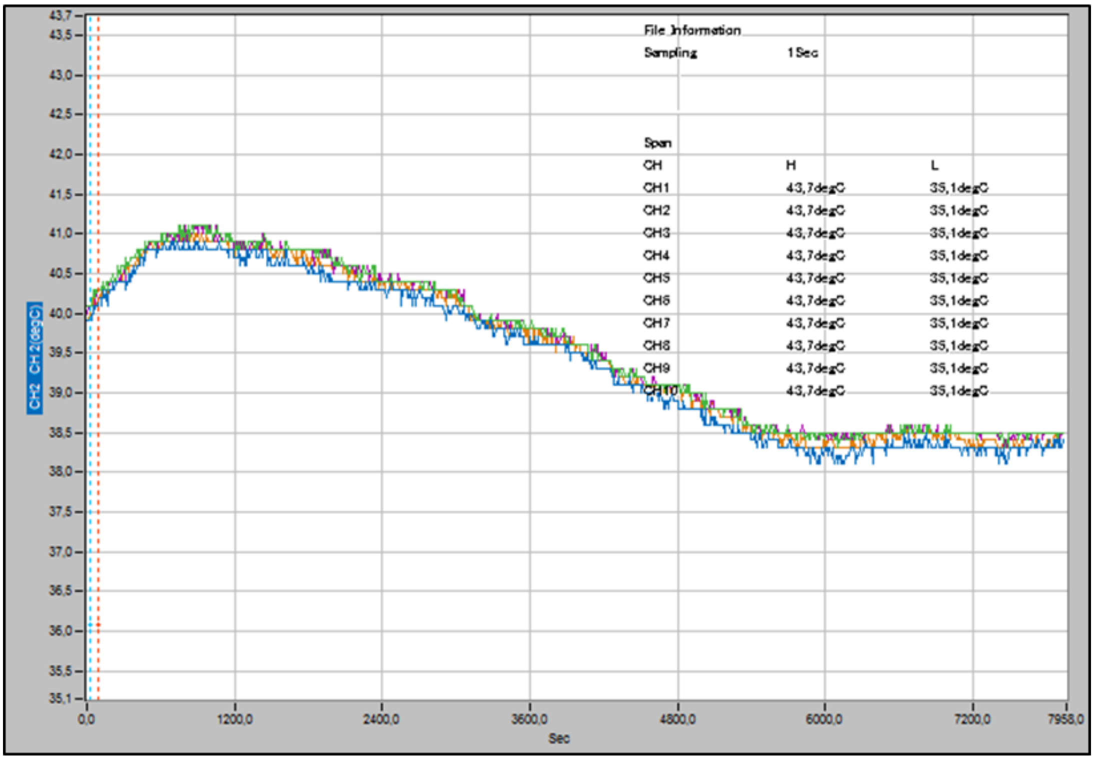


**Figure 5.** Interface of data-logger showing the measurements of the thermocouples

### 2.2. Data reduction

Data reduction equations and flow are also the same of a previously conducted MSc work and a report about it can be found in a conference proceeding [11].

For the steady state condition, total heat input to the system is equal to heat transfer from the system. This applies to all test sets. The following formulas are used for data reduction. Heat sink material is assumed isotropic and has constant thermal conduction.

All symbols and abbreviations are explained in the Nomenclature section. Rayleigh number is given in (1). Volumetric expansion coefficient is given in (2).

$$R_a = \frac{g\,.\,\beta\,.\,\Delta T\,.\,L^3}{\alpha\,.\,v} \quad (1)$$

$$\beta = \frac{1}{T_f} \quad (2)$$

For determining the film temperature in the volumetric expansion coefficient, (3) is given as the following. In a similar way, temperature difference for the heat transfer is given in (4).

$$T_f = \frac{T_w + T_\infty}{2} \quad (3)$$

$$\Delta T = T_w - T_\infty \quad (4)$$

Nusselt number and coefficient of convection in this number is given in (5) and (6) respectively.

$$N_u = \frac{h\,.\,L}{k} \quad (5)$$

$$h = \frac{Q}{A\,.\,\Delta T} \quad (6)$$

The surface areas in the existence of the holes are calculated by (7).

$$A = W\,.H - n\,.\pi\,.r \quad (7)$$

Electrical power is equal to the heating power and it is given in (8).

$$Q = P = V\,.I \quad (8)$$

### 3. Results and Discussion

The first two figures in this section are given for the transition from transient to steady state of experiments in Figure 6 and 7, respectively for 3 mm hole diameter and 6 mm hole diameter. Both figures show that experiments continued sufficiently long to observe the temperature change of the heat sink versus time. Looking to the final parts of the trends, approximately 5 W heating power increase leads to about 10 K increase in averaged wall temperature. Some instant changes are visible in some of the trends. These are due to the rearrangement of the heating power since the changing temperature of the heater also changes the resistance value of it and hence the heating power changes. For those instances, heating power was rearranged. The rearrangement does not increase the time to reach steady state too much, because the temperature difference between heat sink and ambient is already close to its final value. Nevertheless, recording heating power would lead to better flexibility in calculations and instantaneous heat transfer values. Also, a control device or system would be good for controlling the power of the heat source since the resistance of the heater changes as the heat sink temperature increases. Another fact about final temperatures of the heat sink according to the heating power is the ambient temperature. When ambient temperature is relatively high, heat sink final temperature becomes high in order to maintain the heat transfer rate of the heat source. Final temperatures for heating powers are given in Figure 8.

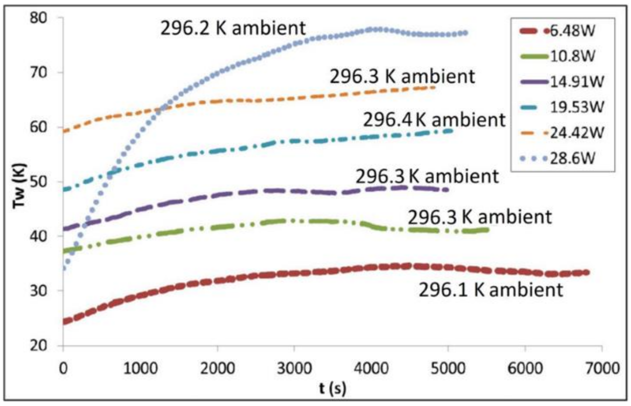


**Figure 6.** Changes of fin temperature versus time for 3 mm case

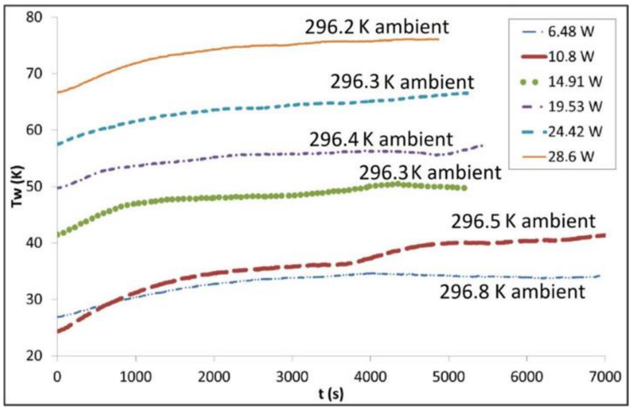


**Figure 7.** Changes of fin temperature versus time for 6 mm case

There are six power values for the experiments and the first three do not show a significant difference between 3 mm perforations and 6 mm perforations. For the remaining three power values, which change from 20 W to 30 W, hole diameters distinguish from each other and 3 mm hole diameter exhibits higher final temperatures implying that 6 mm hole diameter resulted better convection. This inferring comes from the fact that the trends in Figure 8 are for the same heating powers yet 6 mm hole diameters achieved the same heat transfer rate with lesser temperature difference values. However, considering the experimental uncertainties and ambient air temperatures, this difference cannot be used to compare the convection of heat sinks. Therefore, dimensionless comparison should be used. Figure 9 is given for Nusselt numbers calculated for measured temperatures of the heat sinks. Of course, ambient temperature values are also used in calculation of Nusselt numbers. Thus, Figure 9 is the final comparison for evaluating heat sinks comparatively.

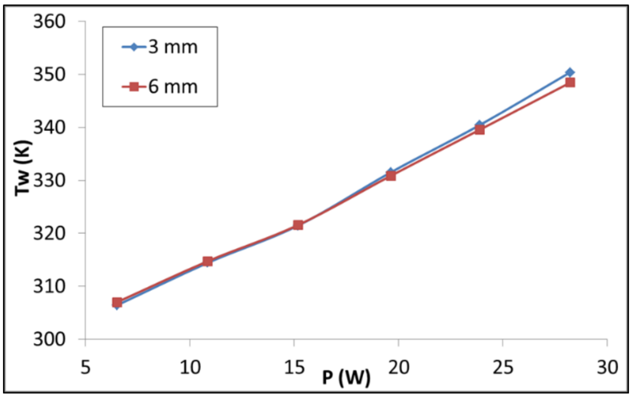


**Figure 8.** Average fin temperatures for tested heating powers and hole diameters

Figure 9 obviously shows that 6 mm hole diameters on the same fin pattern exhibits better heat transfer. Since almost all parameters for the heat sinks are same except the hole diameters, it can be concluded that 6 mm holes lead to better flow mixing and convection. It should also be mentioned that Nusselt number is drawn depending to Rayleigh number. Because, Rayleigh number is an indicator of natural convection flow. It is interesting to see that both heat sinks yield an almost constant Nusselt number between 1,000,000 and 1,400,000 Rayleigh number. Actually, the trends of two heat sinks are almost same or parallel. Only magnitudes are higher for 6 mm hole diameter. As a conclusion, perforation of the heat sink fins and making them bigger for natural convection seem favorable in respect of heat transfer.

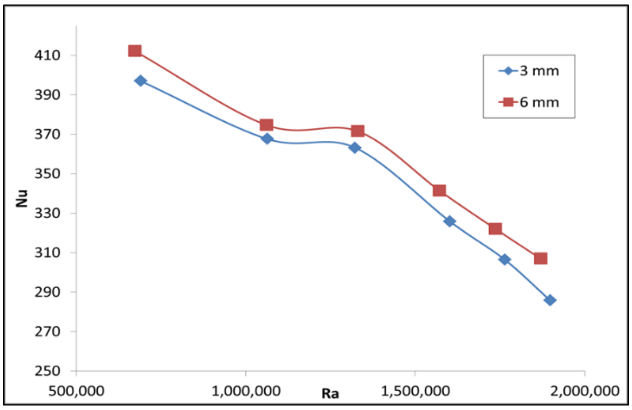


**Figure 9.** Calculated Nusselt numbers versus Rayleigh numbers for two different hole diameters

## 4. Conclusion

Two identical heat sinks with sinusoidal wavy fins were perforated with two different hole diameters, namely 3 mm and 6 mm. Heat sinks over an electrical heater were heated with six different heating powers. Heat sink temperatures were recorded for transient and steady state operations. Nusselt and Rayleigh numbers were calculated for comparing natural convection heat transfer of the two heat sinks. Following conclusions and remarks for future studies are drawn.

Conclusions:

- An increase of 5 W in electrical heater changes the heat sink temperature about 10 K.
- Temperature values of the heat sinks are increasing linearly with the heating power implying that natural convection is negatively effected by the temperature difference and therefore wall temperature increases. Nusselt number in Figure 9 justifies this finding. The reason is thought to be the constant gap between fins, constituting a flow resistance for higher heat transfer rates as the Rayleigh number increases.
- Although the wall temperatures are nearly same for two heat sinks, 6 mm hole diameter heat sink yielded better heat transfer considering the Nusselt numbers graphic. This is due to the less heat transfer area due to the 6 mm hole gaps for the same heat transfer rate.
- Both heat sinks have almost same or parallel trends for Nusselt numbers versus Rayleigh numbers.
- Between $1x10^6$ and $1.4x10^6$ Rayleigh numbers, the Nusselt number becomes almost fixed.
- Perforation of the heat sink fins and making them bigger for natural convection seem favorable

Remarks for future work:

- Heating power should be controlled by means of an electronic control system such as proportional–integral–derivative controller (PID) in order to provide more stable heat transfer rate.
- Heating power should be recorded depending on time for the dimensionless parameter calculations.
- Thermal camera can be used not only for measuring temperature distribution on fins but also distinguishing the thermal radiation.
- In the absence of thermal camera, fin temperature distribution can be measured with several thermocouples. Fin efficiencies can be calculated accordingly.
- A vacuum chamber can be used in order to determine the radiation heat transfer rate for certain temperatures in order to distinguish convection heat transfer from total heat transfer.
- A correlation expression can be derived including hole diameters and heating power as variables alongside with the experimental and correlation uncertainties.
- Computational Fluid Dynamics can be used in order to highlight the flow mechanics leading to heat transfer differences.

Considering the flow patterns, different hole configurations and dimensions can be tested for optimization purposes considering a certain operational condition. Several operational conditions can also be investigated.

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## Nomenclature

| | | | |
|---|---|---|---|
| A | : Plate surface area ($m^2$) | Ra | : Rayleigh number |
| h | : Heat transfer coefficient ($W/m^2K$) | $T_f$ | : Film temperature (K) |
| k | : Thermal conductivity (W/mK) | $T_W$ | : Plate surface temperature (K) |
| H | : High of fins (mm) | $T_\infty$ | : Ambient temperature (K) |
| n | : Number of fins | V | : Voltage (V) |
| Nu | : Nusselt number | W | : Width of fins (mm) |
| P | : Power (W) | β | : Volumetric coefficient of expansion (1/K) |
| $Q_{loss}$ | : Heat transfer loss (W) | ΔT | : Temperature difference (K) |
| $Q_{radiation}$ | : Heat transfer by radiation (W) | | |
| $Q_{convection}$ | : Heat transfer by convection | | |

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