Enhancing thermal performance of vertical fin array heat sink using mist assisted evaporative cooling

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Abstract. In this paper, natural convection heat transfer of parallel-plate rectangular fins heat sink positioned at a certain angle on the vertical substrate was experimentally investigated. Experiments were done with an aim to examine and evaluate the enhancement of heat transfer by deploying fine water mist into ambient air under a range of input power. To empower this investigation, a special purpose experimental set-up was constructed and developed so as to perform experiments on both air and water mist cooling scheme. The water mist concentration influences on the heat transfer enhancement was examined for (m_w=90 – 430 mL/h), while the surface temperature varies from 29 to 110 °C. Experimental results revealed that the heat transfer enhancement increase and reached its maximum value under m_w= 430 mL/h. The average heat transfer enhancement increases by about 1.28%, 1.81%, 2.37%, and 2.83% when compared against traditional air cooling under (m_w=90 – 430 mL/h). In addition, influence of water mist on the fin spacing, fins number, and dissipation area was also examined.

1. Introduction

Due to the development of electronic industry and emergence of modern specialized devices that need to processing data at high speed, reliable and efficient cooling techniques were required. The central processing unit of electronic devices assembled with millions of microscopic transistors to be able to process more data speedily. As a result of passes an electrical signal, the temperature and amount of heat generation in the central processing unit will be increased, making it less reliable and increase the chance of breakdown of data processing in a few years. Dissipation heat it’s a very important case to ensure dependable continuous operation for electronic devices and that can be done by several ways using heat sink [1], heat pipes [2], jet impingement [3] and piezoelectrically-driven droplets [4], etc. Altun et al. [5] have experimentally investigated the impact of sinusoidal wavy fins on the natural convection heat transfer augmentation with different amplitudes. They found an appreciable augmentation in the heat transfer with wavy fins than among the rectangular fins. Also, they found that increasing wave amplitude led to a negative effect on the heat transfer augmentation due to the air motion blockage. Numerical and experimental investigation of natural convection heat transfer mechanism was performed by Zhang et al. [6] with a vertical W-type finned heat sink. In comparison with parallel-plate finned heat sink, they pointed out that the average temperature decrease by about 2.9 °C
while the maximum temperature decrease by about 4.6 °C. In addition, they found that the heat dissipation area reduced by 10%. Another experimental investigation of rectangular fin array heat sink positioned on the horizontal substrate was performed by Taji et al. [7] under a natural and assisting mode of mixed convection. They showed that the average heat transfer coefficient of mixed convection has increased by 69.46% at low energy input from 0.3 to 0.65 W. Güvenç et al. [8] examined the impacts of fin spacing and fin height on the heat transfer coefficient. They concluded that the essential geometric parameter is fin spacing, and it should be selected as characteristic dimension. Using water mist technique as a cooling medium has many exceptional features and has been effectively implemented in many industrial and engineering applications [9-12]. Compared with air cooling, the essential characteristics of water mist can be illustrated as follows: absorbs a great deal of energy in the form of latent heat during the water mist evaporation process by direct contact of the water droplets with the heated surface, increases the humidity, and specific heat of the mixture [13]. For example, a number of numerical and experimental studies found in the literature will be presented, but not limited. Numerical simulation and parametric analyses [14] conducted to investigate the effect of water mist concentration and water droplet diameter on the fin array heat sink heat transfer augmentation under forced flow. They reported that the heat transfer rate obtained with mist flow increases dramatically with increased concentration of water mist and highlighted the possibility to reduce thermal resistance of heat sink up to 97%. Bahadur et al. [15] suggested a new cooling scheme in which mist injected directly into the air cooling, then the evaporated mist can be considered and re-circulated back to their inlets. The results illustrated that the proposed mist cooling scheme provides a significant promise for giving high-flux heat removal methods for telecommunications center. In the present work, the experimental investigation was performed taking into account both natural convection and radiation to analyze and understand the water mist influences on the thermal performance and heat dissipation area of a parallel-plate rectangular fins heat sink. The influence of the different working conditions such as water mist concentration and input power on heat transfer characteristics was examined.

2. Experimental setup and instrumentation
The main goal of the experimental study is to examine and evaluate the water mist influences on the natural convection heat transfer from a finned heat sink. To empower this investigation, the experimental set-up was constructed and developed so as to perform experiments on both air and water mist cooling scheme. The experimental setup has primarily consisted of a parallel-plate finned heat sink, electrical heater, insulated cover, supported frame, mist generator, and data acquisition system. The schematic layout of experimental set-up and geometry of the finned heat sink is given in figure 1. The heat sink comprising rectangular fins was made of the aluminum alloy due to high-emissivity and high thermal conductivity and positioned at a certain angle on the vertical substrate. A.C. silicon band drum heater whose dimension are 150mm-high and 100mm-width was used as a heat source to heat the heat sink. The maximum power and working temperature of heater are 50W and 160 °C respectively. The input power has been calculated by multiplying the measured voltage and current using a digital multimeter type APPA 79 with (±0.06%) in accuracy. A voltage regulator type (EQ-VT250) was used to modify and control heater input power so as to achieve the required surface temperature. To avoid any thermal resistance between heater and heat sink, a thin layer of high thermal conductivity paste was applied. The insulated cover was made of a wooden box that has dimensions of 260, 180, and 30mm for high, width, and depth respectively. In order to minimize the heat loss from the rear side of heater and maintain the insulation of the four lateral surfaces of heat sink, the insulated cover was loaded with 30mm thickness fibreglass wool insulated material. The surface temperatures of base plate are gauged using calibrated thermocouples (type-K) that are implanted in drilled holes at 6 different locations as presented in figure 1. The ambient temperature and temperature over the heat sink for two cases (air and water mist) were also measured. Simultaneously, the humidity over the heat sink was measured using humidity meter type (GM1362) with accuracy (±4.5%). All thermocouples are connected to (MSD200) type digital data logger with accuracy (±1.0%). Micro-sized water mists (2–10μm) were deployed in the air using an ultrasonic atomizer consisting of a piezoelectric transducer.
working with high-frequency oscillation of 1.7MHz. This high-frequency oscillation led to create a natural floating water mist without any mechanical agent or heating sources.

Figure 1. Schematic layout of experimental set-up and geometry dimensions of the finned heat sink.

3. Data processing and uncertainty analysis
The experiments were done for a finned heat sink with and without water mist for various input power. Initially, the results for the finned heat sink without water mist was obtained and regarded as reference data for the other experiments. For experiments without water mist, the heat transfer coefficient is estimated from the following equations [16].

\[
Q_{\text{conv.}} = h \cdot A \cdot (T_s - T_\infty);
\]

where:
\[ T_s - T_\infty \] – temperature difference,
\[ Q_{\text{conv.}} \] – heat transfer rate by convection which was evaluated as:

\[
Q_{\text{conv.}} = Q_p - Q_{\text{rad}};
\]

where:
\[ Q_p \] – total input power,
\[ Q_{\text{rad.}} \] – heat transfer rate by radiation which was evaluated as:
\begin{equation}
Q_{rad} = A \cdot \varepsilon \cdot \sigma \cdot (T_f^4 - T_\infty^4);
\end{equation}

Rayleigh number (Ra) can be defined as:
\begin{equation}
Ra = Gr \cdot Pr = \frac{\Delta T \cdot \rho^2 \cdot g \cdot \beta \cdot L^3}{\mu^2 \cdot \frac{\mu}{k}};
\end{equation}

Herein, the thermal expansion coefficient can be obtained as:
\begin{equation}
\beta = \frac{1}{T_f};
\end{equation}

where \( T_f \) represent the film temperature at which all thermo-physical properties of air were estimated, it is can be defined as:
\begin{equation}
T_f = \frac{T_s + T_\infty}{2};
\end{equation}

For experiments with water mist, the heat transfer rate is estimated from the following equations [17].
\begin{equation}
h_m \cdot A \cdot (T_s - T_\infty) + k_c \cdot h_{fg} \cdot (x_s - x_{sat}) = Q;
\end{equation}

where:
- \( h_m \) – heat transfer coefficient by water mist,
- \( A \) – total surface area,
- \( k_c \) – mass transfer coefficient,
- \( h_{fg} \) – the latent heat of evaporation,
- \( x_s \) – moisture content of saturated air by surface temperature,
- \( x_{sat} \) – moisture content of saturated air by ambient temperature.

When water is exposed to the adequately ultrasonic field, separation of water begins to form water mist and they are expelled from water interface into the ambient air. For obtaining the average diameter of water mist, Lang [18] formulated an equation depending on the ultrasonic frequency as follow:
\begin{equation}
d_p = C \cdot \left( \frac{8 \pi \sigma}{\rho f^2} \right)^{0.333};
\end{equation}

where \( C \) – constant fraction of the capillary wavelength equal to 0.34, and \( f \) - the ultrasonic mist generator frequency. The Weber number (We) can be defined as:
\begin{equation}
We = \frac{m_m \cdot d_p}{2A^2 \rho \sigma};
\end{equation}

For a vertical heat sink, the optimum fin spacing is obtained as follow [19]:
\begin{equation}
S_{opt} = 2.714 \cdot \frac{L}{Ra^{0.25}};
\end{equation}

The fins number of the optimum fin spacing case was calculated as follow:
\begin{equation}
n = \frac{W}{S + t};
\end{equation}

The uncertainty of experimental data was calculated with an aim to evaluate the experimental facilities reliability. Uncertainties of the heat transfer coefficient and Rayleigh number can be estimated as follow [20]:
\[
\left( \frac{\Delta h}{h} \right)^2 = \frac{1}{\alpha} \left[ \left( \frac{\partial}{\partial Q} (h) \Delta Q \right)^2 + \left( \frac{\partial}{\partial T_r} (h) \Delta T_r \right)^2 + \left( \frac{\partial}{\partial T_\infty} (h) \Delta T_\infty \right)^2 \right]
\]
(12)

\[
\left( \frac{\Delta Ra}{Ra} \right)^2 = \frac{1}{Ra} \left[ \left( \frac{\partial}{\partial \Delta T} (Ra) \Delta T \right)^2 + \left( \frac{\partial}{\partial \rho} (Ra) \Delta \rho \right)^2 + \left( \frac{\partial}{\partial \mu} (Ra) \Delta \mu \right)^2 + \left( \frac{\partial}{\partial L} (Ra) \Delta L \right)^2 \right]
\]
(13)

The maximum uncertainties of heat transfer coefficient and Rayleigh number were found to be 4.1%, and 5.3%, respectively.

4. Results and discussion

Herein, the experimental results of heat transfer characteristics, thermal performance, fin spacing, and the overall area of heat sink associated with the uses of water mist are presented. Before start experiments, a trial analysis were performed to evaluate the reliability of the experimental setup and experimental procedures employed. The present results were validated and compared with the previous experimental and numerical data of Harahap F et al. [21] and Senol Baskaya et al. [22] as depicted in figure 2. As observed, the present heat transfer coefficients are in good agreement with previous studies which illustrates that the experimental setup and experimental procedures in the present work are satisfactory and reliable.

![Figure 2. Validation of the heat transfer coefficient: present work compared with experimental results by Harahap F et al. [21] and CFD results by Baskaya S et al. [22]:](image)

(Figures 3, 4 gives the variations of average surface temperature (base plate temperature) and heat transfer coefficient with input power with and without water mist. It can be seen from figure 3 that the surface temperature increases as input power increase for all cases. For air cooling case, cold air enters the space between fins generally from the heat sink bottom side and leaves from the top side after exchanging heat with the heat sink fins. During this time, the air is significantly heated and resulting in increasing its temperature along the heat sink length. In the water mist case, the influence of water evaporation was clearly observed and it can be noticed that the average surface temperature decreases as water mist concentration increases. The average surface temperature decreases by about 19.1%, 34%, 43.5%, and 48.2% in comparison with traditional air cooling under a range of water mist concentration (\(m_w = 90 - 430 \text{ mL/h}\)). Depending on equation (7) the heat transfer rate by water mist is equal to the sum of heat dissipated by water mist evaporation for wetted regions and normally heat transfer by...
convection. In view of this, the heat transfer coefficient increases by about 27.5%, 79.1%, 135.4%, and 281% in comparison with traditional air cooling under \( m_w = 90 \text{ – 430 mL/h} \) as shown in figure 4.

At the same time as heat transfer measurement, the relative humidity over the heat sink for both cases was also measured with a 60 sec time interval as presented in figure 5. Initially, the air humidity was equal to the ambient, and then as the temperature of the heat sink became warmer and thus air temperature due the convection flow, the air humidity began to decrease gradually. For the water mist case, an opposite pattern was found, the relative humidity measured drastically increased with time and reached up to 100% after 300 sec.

**Figure 3.** Variation of the average surface temperature: (1) only air, (2) \( M_w = 90 \text{ mL/h} \), (3) \( M_w = 180 \text{ mL/h} \), (4) \( M_w = 340 \text{ mL/h} \), (5) \( M_w = 430 \text{ mL/h} \) for all cases of the input power.

**Figure 4.** Variation of the average heat transfer coefficient: (1) only air, (2) \( m_w = 90 \text{ mL/h} \), (3) \( m_w = 180 \text{ mL/h} \), (4) \( m_w = 340 \text{ mL/h} \), (5) \( m_w = 430 \text{ mL/h} \) for all cases of the input power.

**Figure 5.** Relative humidity over the heat sink in the time function: (1) only air, (2) with water mist.

**Figure 6.** Variations of heat transfer enhancement: (1) \( m_w = 90 \text{ mL/h} \), (2) \( m_w = 180 \text{ mL/h} \), (3) \( m_w = 340 \text{ mL/h} \), (4) \( m_w = 430 \text{ mL/h} \) for all cases of the input power.
The ratio of heat transfer coefficient for the finned heat sink with water mist to that when water mist was not deployed at an equal input power can indeed be characterized as the enhancement rate in heat transfer. Figure 6 gives variations of heat transfer enhancement as a function of input power under \((m_w=90 – 430 \text{ mL/h})\). It can be viewed that the heat transfer enhancement for all water mist concentration was greater than unity and that refers to the important role of water mist on heat transfer process. As the water mist increases, the heat transfer enhancement increase and reached its maximum value under \(m_w= 430 \text{ mL/h}\). The average heat transfer enhancement increases by about 1.28%, 1.81%, 2.37%, and 2.83% in comparison with traditional air cooling. The approximation of the experimental data by the least-squares method made it possible to obtain an empirical correlation between heat transfer enhancement and the regime parameters of Rayleigh and Weber numbers as follow:

\[
\frac{h_w}{h_l} = 1 + \left(Gr \cdot Pr\right)^{0.19} We^{0.38},
\]

(14)

With respect to fin spacing, fins number, and dissipation area, one of the important features of water mist in parallel with heat transfer enhancement it’s the reduction of total heat dissipation area. As mentioned above, using water mist as a working medium of heat sink cooling led to decrease the surface temperature by about 43.5%, which contributes to the reduction of the dimensionless Rayleigh number and hence increase the fin spacing and decrease the fin numbers according to equations (10). It is experimentally confirmed that the fin spacing increased as water mist concentration increases, while the fins number was decreased as water mist concentration increases, which ultimately led to a decrease in the dissipation area as shown in figure 7.

![Figure 7](image.png)

**Figure 7.** Fin spacing and fin number versus water mist for \(P=65.44 \text{ W}\).

5. **Conclusions**

The water mist influences on the natural convection heat transfer of a finned heat sink is experimentally investigated. Thermal performances of the finned heat sink with and without water mist were compared. Based on the obtained experimental results, it can summarize the following:

- Water mist decreased the surface temperature substantially over the use of air alone. The average surface temperature decreases by about 19.1%, 34%, 43.5%, and 48.2% in comparison with traditional air cooling under \((m_w=90 – 430 \text{ mL/h})\).
The heat transfer coefficient increases by about 27.5%, 79.1%, 135.4%, and 281% in comparison with traditional air cooling.

The heat transfer enhancement increase and reached its maximum value under mw= 430 mL/h. The average heat transfer enhancement increases by about 1.28%, 1.81%, 2.37%, and 2.83% in comparison with traditional air cooling.

The fin spacing increased as water mist concentration increases, while the fins number was decreased as water mist concentration increases.

Empirical heat transfer enhancement correlation based on the experimental data obtained from the related experiment was developed and introduced for practical use.

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