



A novel heat dissipation fin design applicable for natural convection augmentation[☆]

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ABSTRACT

In this study, a comparative study of heat sink having various fin assembly under natural convection is investigated. The fin pattern includes a rectangular, a trapezoidal and an inverted trapezoidal configuration. Tests were performed in a well controlled environmental chamber having a heat load ranging from 3 to 20 W. From the test results, the heat transfer coefficient of the conventional rectangular fins is higher than that of the trapezoidal fins while the heat transfer coefficient of the inverted trapezoidal fins is higher than the trapezoidal one by approximately 25%, and it exceeds that of convectional rectangular fin by about 10%. The heat transfer improvements of the inverted trapezoidal fin are mainly associated with a larger temperature difference and inducing more air flow into the heat sink.

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1. Introduction

Air-cooling is still the most widely used methods for heat dissipation in electronic applications. This is because air cooling is reliable and easy to implement. However, because the considerably low thermal conductivity of air heat sink normally incorporated with substantial fin surfaces to reimburse the poor heat transfer performance of air for effective heat removal. Yet the added surfaces may occupy a lot of space and weight especially when heat sinks were operated under natural convection. Hence it is imperative to reduce the size/volume as much as possible provided the junction temperature is below the threshold requirement.

Many researchers had devoted efforts to fin design for natural convection condition and Table 1 tabulated some related studies for the past decade. Among them, Bar-Cohen et al. [1] demonstrated that there is a least-material optimization for the vertical rectangular longitudinal plate fin arrays in natural convective heat transfer. Mokheimer [2] considered the effect of the variation of local heat transfer coefficient alongside the fin surface, and reported a considerable deviation of the conventional constant heat transfer coefficient assumption that may result in a significant underestimation of the fin efficiency. Khan et al. [3] examined some selected fin geometries subject to influences of axis ratio, aspect ratio, and Reynolds number. Their results clearly indicated that the preferred profile is very dependent on these parameters. Huang et al. [4] concluded that the optimal porosity of a plate heat sink is around 83% for the upward arrangement and is around 91% for the side-ward arrangement. Goshayeshi and Ampofo [5] found that vertical plate with vertical fins gives the best performance in natural convection.

Suryawanshi and Sane [6] investigated the performance between plate and inverted notched fin array. They reported that the average heat transfer coefficient for the inverted notched fin arrays is nearly 30–40% higher than that of conventional plate fin array.

Zhang and Liu [7] investigated the optimal spacing between isothermal laminar natural convection analytically and numerically. They showed that the optimal plate spacing depends on the outlet velocity of the heat sink. By inserting multi-scale plates in the boundary layer, the heat transfer enhancement could be achieved effectively. Fahiminia et al. [8] stated that a maximum heat transfer rate is attainable for an optimum fin spacing. Similar results were reported by Goshayeshi et al. [9] who concluded that at a given fin height and a temperature difference, the heat transfer rate may first increase with fin spacing and it reaches a maximum, followed by a noted decline. Torabi et al. [10] highlighted that the concave parabolic fin yields the optimum utilization of material, giving the highest heat transfer rate, fin efficiency, and fin effectiveness. For all three geometries, the effect of the temperature-dependent heat transfer coefficient is to decrease the fin heat transfer rate as compared to those with a constant heat transfer coefficient. However, the fin efficiency is higher when the convective heat transfer coefficient is temperature-dependent than when it is a constant. Tari and Mehrtash [11] reported effectiveness of the selected fin design with various fin thickness by visualizing temperature distributions. The effect of tilt angle is thoroughly investigated and their results suggested that within some small inclinations from the vertical in both directions, the inclination does not reduce the convection heat transfer rate and the heat transfer rate can even slightly increase at a very small downward inclination due to the thinner boundary layer. Mehrtash and Tari [12] investigated the optimum fin spacing for all inclinations varying from the downward facing horizontal to the upward horizontal arrangement, the optimum tilt angle is vertical arrangement. They concluded that

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Nomenclature

A	cross sectional area of the fins, m ²
A _b	cross sectional area of the Bakelite, m ²
h	heat transfer coefficient, W m ⁻² K ⁻¹
k _b	thermal conductivity of the Bakelite, W m ⁻¹ K ⁻¹
k	thermal conductivity of fluid, W m ⁻¹ K ⁻¹
Nu	Nusselt number, dimensionless
Pr	Prandtl number, dimensionless
Q	Rate of heat transfer, W
Ra	Rayleigh number, dimensionless
T _f	fin temperature, °C
T _a	ambient temperature, °C
T _b & T _w	base plate temperature, °C
T _s	surface temperature of the heat sink, °C
ΔT	effective temperature difference, °C
x	thickness of the Bakelite, m
θ	inclined angle, degree

Subscripts

f	plate-fin
b	base
c	convection
i	input
t	total
l	loss

optimum fin spacing for the vertical orientation is most favorable. Tari and Mehrtash [13] later investigated the previously uncovered inclination angle for the inclined cases and developed a set of correlations. These correlations were shown to be very accurate in predicting heat transfer rates using the available horizontal data from the literature. Kim et al. [14] showed that the thermal resistance revealed an optimal value at a specific fin number. However, the thermal resistance decreases continuously without reaching an optimal value as the fin height is increased. Naserian et al. [15] concluded that by increasing the fin number and the fin spacing, the ratio of natural convection heat transfer coefficient of various fin configurations to the corresponding vertical plate is increased.

Based on the foregoing discussion, apparently the fin design casts significant influences on the natural convection. Normally, the easiest way is just to impose more surface area for effective reduction of thermal resistance. However, as aforementioned from previous studies, the induced air flow under natural convection is normally quite restricted and can be significantly impaired by dense fin design, jeopardizing the heat transfer accordingly. In this regard, the principal objective of this study is to propose a novel fin design applicable for natural convection.

2. Experimental setup

The designated concept of the fin pattern is simple. Instead of increasing the surface area, the basic idea is to increase the effective temperature difference, especially at the rear part of fin surface. Fig. 1 illustrates the proposed concept in association with the conventional design. Fig. 1(a) is the convective rectangular fin with identical cross-sectional surface area at the fin base and fin tip, Fig. 1(b) is the trapezoidal fin whose cross-sectional area at the fin base is larger than that in the fin tip, and Fig. 1(c) is the proposed inverted trapezoid fin in which the cross-sectional area at the fin tip is larger than that at the fin base. Note that the effective surface areas amid these three designs are the same, and their detailed dimensions are shown in Fig. 1(d). Experiments are performed in an environmental chamber whose volume

is 900 mm × 900 mm × 1240 mm. The environmental chamber can provide a temperature condition in the range of 20–50 °C with a controlled resolution of 0.2 °C. In this study, the ambient temperature is fixed at 25 °C. To simulate the natural flow condition, the air ventilator is turned off inside the test chamber when the ambient temperature reaches 25 °C. In particular, the air conditioner outside the test chamber continues to operate to maintain the room temperature at 25 °C. The test facility is inside the test chamber which consists of a heat sink, a heater, and insulation box, and a tilting mechanism as illustrated in Fig. 2(a).

The heat sinks are made of aluminum alloy 5083 with a thermal conductivity 121 W m⁻¹ K⁻¹. Five pin fin heat sinks are made via CNC machining with a manufacturing precision of 0.03 mm. The heat sinks are operated with the power inputs from 3 to 20 W. Detailed dimensions of the test samples are also shown in Fig. 1. A Kapton heater with identical size as the base plate of the heat sink is used to eliminate the spreading resistance. An insulation box made of bakelite with a low thermal conductivity of 0.233 W m⁻¹ K⁻¹ is placed beneath the heater to reduce the heat loss. In addition, a high thermal conductivity grease (k = 2.1 W m⁻¹ K⁻¹) is used to connect the heat sink and the heater for further minimization of the contact resistance. The heater is powered by a DC power supply.

As seen in Fig. 2, a total of five T-type thermocouples which are equally divided and located at beneath the base plate are used to obtain the mean temperature of the base plate (T_b) of the heat sink. In addition, a total of 10 T-type thermocouples are installed inside the insulation box at two cross positions to calculate the heat loss from the bottom of the Kapton heater. Each cross-section is equally instrumented with five T-type thermocouples to obtain the mean temperature of that cross section. The average temperature is then used to estimate the heat loss via Fourier's law of conduction. The thermocouples were pre-calibrated with an accuracy of 0.1 °C. The exact total heat transfer rate by natural convection supply (Q_t) is then obtained by subtracting the estimated loss from the power input. The signals from thermocouples are then transmitted to a data acquisition system for further data reductions. Usually, each test run needs approximate 2.5 h to reach equilibrium when the power is turned on.

3. Data reduction

In the present study, the ambient air temperature is always controlled at 25 °C and the thermophysical properties are evaluated at the film temperature, i.e.

$$T_f = 1/2(T_a + T_b). \quad (1)$$

The actual heat transfer rate, Q_t, is determined by subtraction the heat loss Q_l from the measured heat input Q_i of the Kapton heater:

$$Q_t = Q_i - Q_l \quad (2)$$

$$Q_l = \frac{k_b A_b (T_b - T_a)}{x}. \quad (3)$$

The average heat coefficient can be calculated from the following:

$$h = \frac{Q_t}{A(T_b - T_a)}. \quad (4)$$

The experimental uncertainty is estimated using the uncertainty propagation equation proposed by Kline and McClintock [16]. The maximum measured uncertainties of the heat transfer coefficient are about 11%, occurring at the lowest input power of 3 W. In particular, this uncertainty drastically decreases to less than 3% when the power input is larger than 5 W.

Table 1
Summary of the recent efforts of fin design subject to natural convection.

Authors	Year	Geometry	Orientation	Experimental/numerical	Range	Conclusion
Bar-Cohen et al. [1]	2003	Rectangular fins	Vertical	Numerically	N/A	Existence of a least-material optimization for vertical rectangular longitudinal plate fin.
Mokheimer [2]	2003	Straight and pin fin with rectangular, convex parabolic, triangular and concave parabolic, and radial fins	Horizontal	Experimentally	N/A	The local heat transfer coefficient as function of the local temperature.
Khan et al. [3]	2006	Circular, elliptical, square and rectangular fins	Horizontal	Numerically	$Nu = 0.75Re^{1/2}Pr^{1/3}$	Temperature profile depends on various parameters (e.g. Reynolds number...).
Huang et al. [4]	2008	Flat and square pin fin heat sinks	Upward, sideward and downward	Experimentally	$Ra \leq 4 \times 10^7$	The optimal heat sink porosity is around 83% for the upward arrangement, and 91% for the sideward arrangement.
Goshayeshi and Ampofo [5]	2009	Rectangular fins	Vertical & horizontal	Numerically	N/A	Vertical plate with vertical fins gives the best performance for natural cooling.
Suryawanshi and Sane [6]	2009	Rectangular fins	Horizontal	Experimentally & numerically	$Pr = 0.7$	The average heat transfer coefficient for inverted notched fin arrays is nearly 30–40% higher as compared to normal array.
Zhang and Liu [7]	2010	Rectangular fins	Vertical	Numerically & analytically	$Pr = 1$	The heat transfer enhancement could be conducted effectively and the intervals of the multi-scale plates depend on the height of the plates.
Fahiminia et al. [8]	2011	Rectangular fins	Vertical	Numerically	$Ra = 10^9$	The convection rates increase with increasing fin spacing and it reaches a maximum at a certain fin spacing, followed by a decline of heat transfer rate.
Goshayeshi et al. [9]	2011	Rectangular fins	Vertical	Experimentally	$Ra = 10^9$	At a given fin height and temperature difference, the convection heat transfer rates increase with increasing fin spacing and reach a maximum.
Torabi et al. [10]	2013	Rectangular, trapezoidal and concave parabolic	Horizontal	Numerically & analytically	N/A	The fin efficiency is higher when the convective heat transfer coefficient is temperature-dependent than when it is a constant.
Tari and mehrtash [11]	2013	Rectangular fins	Inclined $\pm 4^\circ \leq \theta \leq \pm 90^\circ$	Numerically & experimentally	$250 < GrPr < 10^4$	The inclination does not reduce the convection heat transfer rate.
Mehrtash and Tari [12]	2013	Rectangular fins	Inclined $\pm 4^\circ \leq \theta \leq \pm 90^\circ$	Numerically & experimentally	$0 < Ra < 2 \cdot 10^8$	The optimum fin spacing for the vertical orientation is obtainable for a certain inclination of the heat sink.
Tari and Mehrtash [13]	2013	Rectangular fins	Slightly inclined: $\theta = \pm 60^\circ$ to $\pm 90^\circ$	Experimentally & numerically	N/A	The proposed correlations are shown to be very accurate in predicting heat transfer.
Kim et al. [14]	2013	Cylindrical heat sink	Horizontal	Experimentally	$Ra = 300,000-1,000,000$	The thermal resistance has an optimal value at a specific fin number. However, the thermal resistance decreases continuously without reaching an optimal value as the fin height increases.
Naserian et al. [15]	2013	V-type fin configurations	Vertical	Experimentally & numerically	$Nu = 0.59 (Ra)^{1/4}$ $10^4 < Ra < 10^9$	An optimum fin shape occurs on a vertical base arrangement.

4. Results and discussion

This experimental result in terms of heat transfer coefficients for all the test fin configurations, namely rectangular, trapezoidal and inverted trapezoidal (so-called dovetail fin) is shown in Fig. 3. In order to simulate the heat sink of the different fin profiles for different inputs, the experiment were carried out with various input powers of 3 W, 5 W, 10 W, 15 W and 20 W, respectively. As shown in Fig. 3, with the same surface area, the heat transfer coefficient (HTC) for the proposed inverted trapezoidal fin configuration appreciably outperforms those of rectangular and trapezoidal designs. For a supplied heat flux of 600 W m^{-2} , the HTC for inverted trapezoidal configuration is about 10% and 25% higher than that of rectangular and trapezoidal configuration, respectively. The difference prevails even when the heat flux is less than 200 W m^{-2} . The augmentation of the proposed inverted

trapezoidal fin configuration is mainly coming from making use of the effective temperature difference ΔT . Note that the entrained air flow primarily comes from the bottom of the two entrances of the heat sink (fin gap). The air flow is heated up and flows toward the center of the heat sink, yet it departs the heat sink upwardly. As expected, the temperature difference amid the air flow and fin surface at the entrance is the largest and is reduced accordingly when the air flows toward the center of the heat sink, thereby leading to a tremendous deterioration of heat transfer performance at the exit of the heat sink. The results imply that the surface area near the exit of air flow is comparatively futile due to its lower temperature difference.

To tailor this problem, with the proposed inverted trapezoidal configuration, more fin surface was accommodated to have a larger temperature difference. This can be made clear from the inverted protruded fin area at the entrance portion of the entrained air flow.

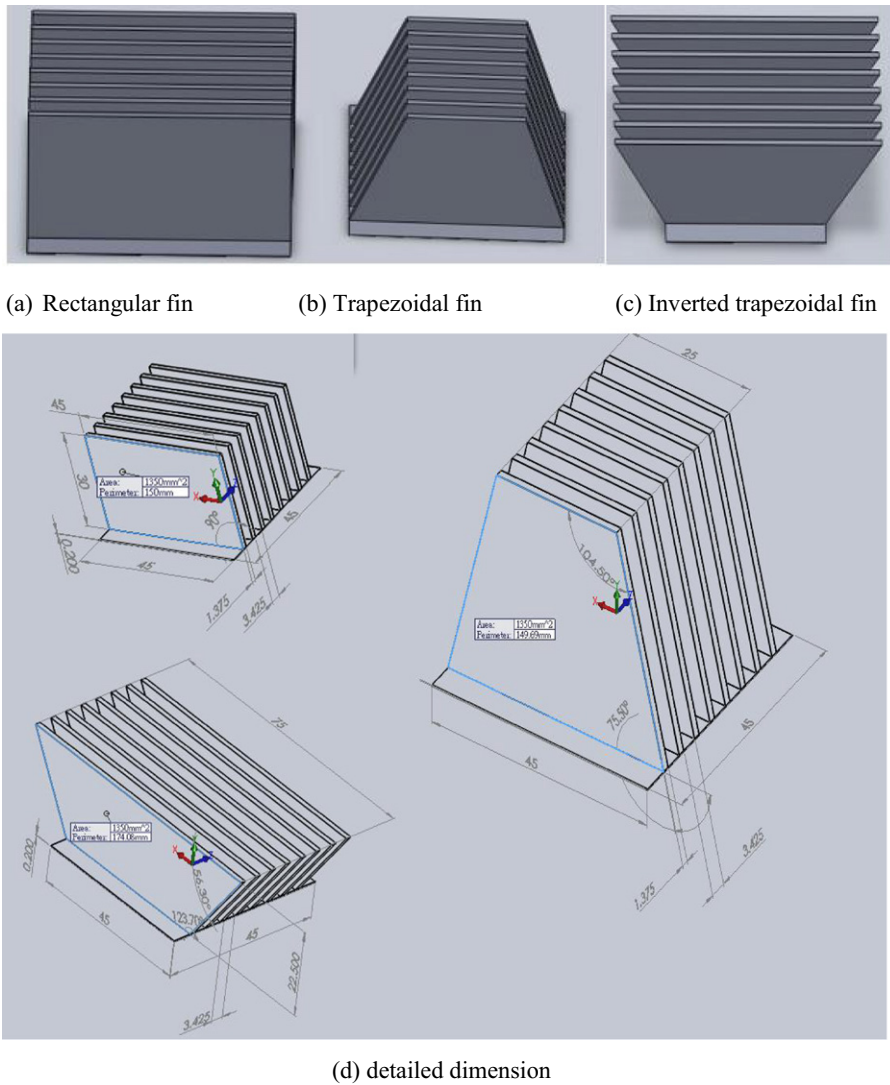


Fig. 1. Rectangular fins, trapezoidal fins and inverted trapezoidal fins (with and without dimensions).

Secondly, the slightly lower fin height profile of the inverted trapezoidal fin geometry also help to reduce the lower temperature difference portion at the exit of the air flow. In addition, the lower profile of inverted fin geometry also contributes a smaller viscous drag of the airflow across the fin pattern, thereby leading to a higher velocity. This can also be made clear from a numerical simulation carried out by a commercially available software – SolidWorks Flow Simulation with a grid number of 5,963,636 cells. The trend of the calculated results is in line with the measurements where the proposed inverted trapezoidal configuration outperforms rectangular and trapezoidal configuration by approximately 10% and 25%, respectively as also depicted in Fig. 3. The simulation is carried out by omission of radiation contribution, hence the calculated HTC values are smaller than those of measurements but the trend of the calculations is analogous to the measurements. Moreover, note that the induced maximum air flow velocity at the exit of fin array from the numerical simulation for the inverted trapezoidal, rectangular, and trapezoidal fin geometry is 0.071, 0.06, and 0.05 m s^{-1} , respectively at a supplied heat flux of 600 W m^{-2} . The calculated higher velocity of the inverted trapezoidal configuration agrees with previous inference. Hence, in summary of these three effects, with an appreciable rise of effective temperature difference and a moderate rise of entrained air flow may bring about a higher heat transfer coefficient of the

inverted trapezoidal configuration. However, it should be emphasized that the present inverted trapezoidal configuration may slightly offset the effective fin efficiency if the protrusion part is too much.

5. Conclusion

The heat transfer coefficient characteristics of rectangular, trapezoidal and inverted trapezoidal pin fin heat sinks subject to horizontal arrangement are examined under natural convection in the present study. The experimental results showed that the heat transfer coefficient (HTC) for the proposed inverted trapezoidal fin configuration appreciably outperforms those of rectangular and trapezoidal design. For a supplied heat flux of 600 W m^{-2} , the HTC for inverted trapezoidal configuration is about 10% and 25% higher than that of rectangular and trapezoidal configuration. The difference prevails even when the heat flux is less than 200 W m^{-2} . A numerical simulation for the three surfaces also reveals a similar trend. The proposed inverted trapezoidal configuration features a larger temperature difference at the entrance of the heat sink and its slightly lower fin height profile also increases the temperature difference at the exit of air flow. In the meantime, the lower profile of inverted fin geometry also contributes to a smaller viscous drag of the airflow across the fin pattern, thereby leading to a higher velocity. In

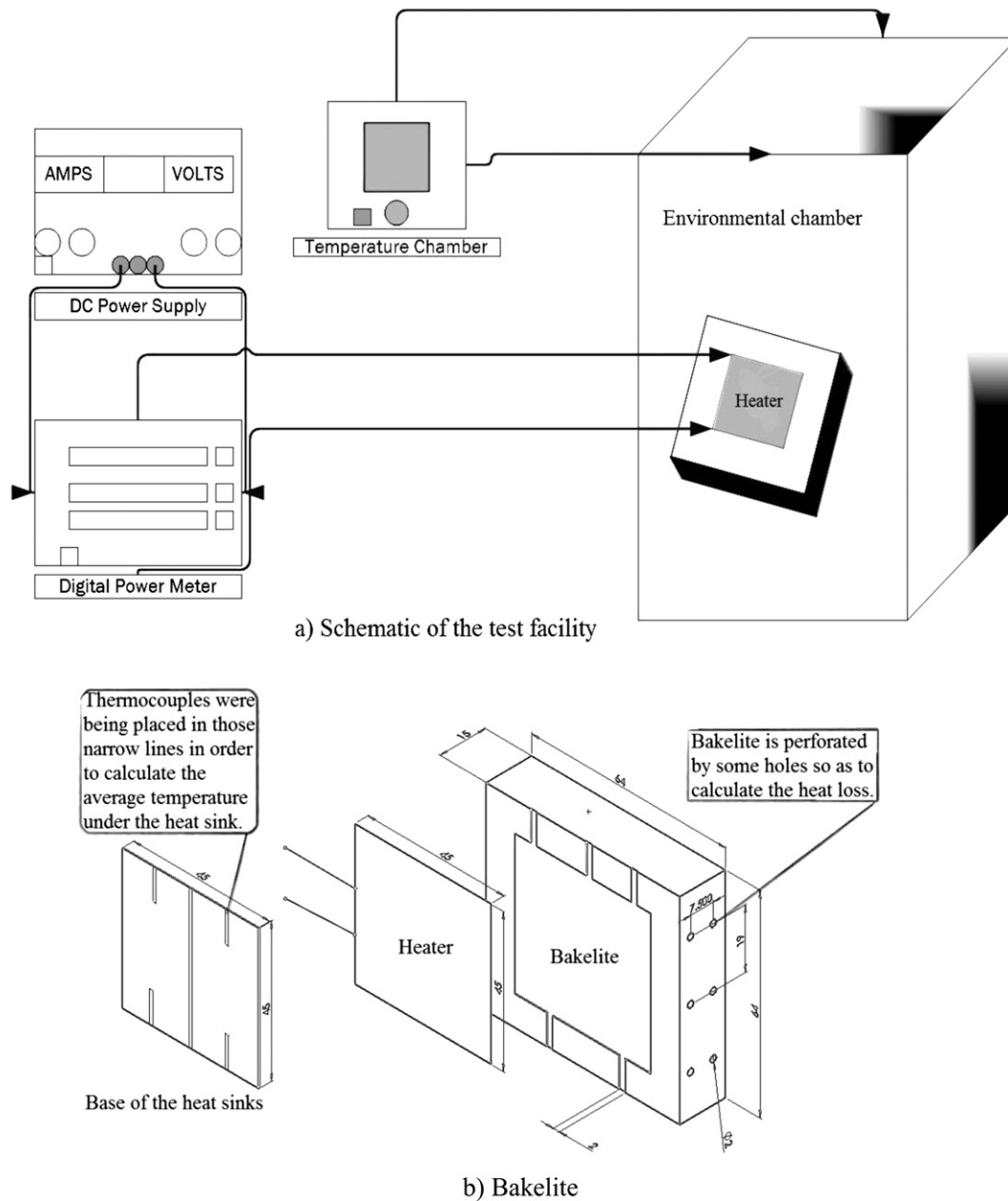


Fig. 2. Schematic of the test facility and the Bakelite.

summary of these three effects bring about a much better heat transfer characteristics of the inverted trapezoidal fin geometry.

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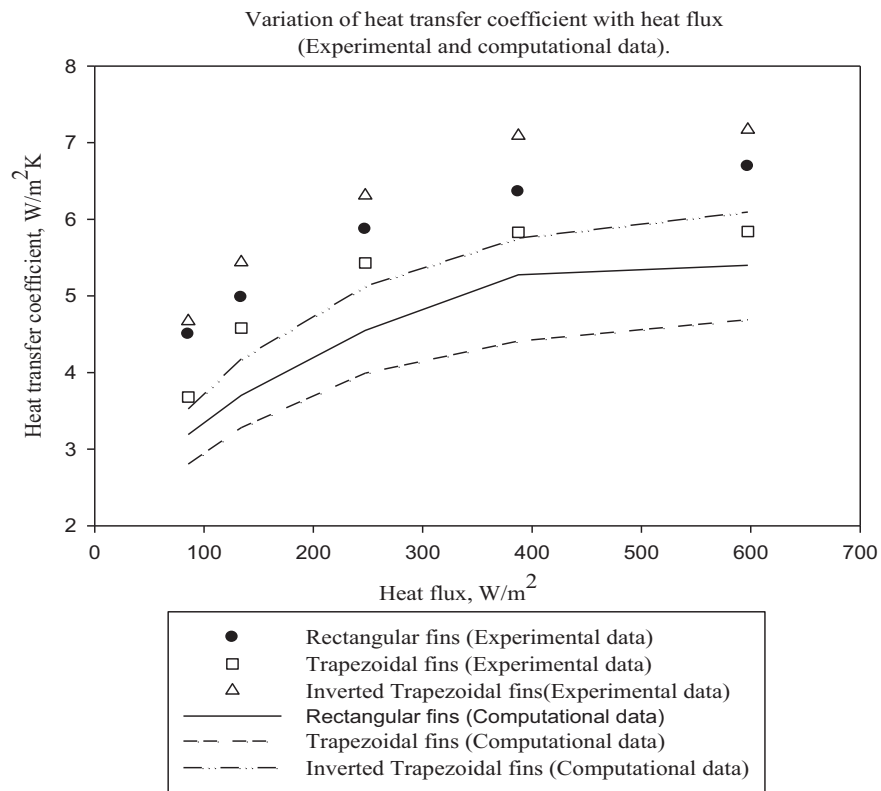


Fig. 3. Performance of the heat transfer coefficient compared to the heat flux.

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