CFD analysis of the thermal performance improvement in heat sinks with corrugated plate-fin

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Abstract
A 3-dimensional numerical analysis is performed to study the thermal performance in three new designs of the corrugated plate-fin subjected to impinging airflow for Reynolds number range from 1600 to 7700. The three designs of plate-fin heat sink include a triangular (TRI-PFHS), a rectangular (REC-PFHS), and a semi-circular (SCR-PFHS). A comparative evaluation has been performed on the thermal performance of the suggested designs with the traditional type (T-PFHS). The results revealed that the thermal performance of new designs is higher than that of the traditional heat sink. The new designs of corrugated plate-fin heat sink have proved the ability to improve heat transfer due to the larger area offered by corrugation and high flow disruptions caused by the presence of a semi-confined channel. The performance of the new designs is assessed in terms of Nusselt number, pressure drop, and performance factor. The overall performance factor (Pf) for SCR-PFHS is predominant over all other designs which achieved a highest value of 1.76 at Re=7700.

1. Introduction

The tremendous development in electronics is facing a serious challenge of finding an effective cooling method that can meet the particular demands of thermal management in terms of high efficiency and low cost. The air-cooling using plate-fin heat sink has received considerable attention in the past several decades due to good performance, low-cost, and easy fabrication. To keep pace with ascending demand for a more efficient design of plate-fin heat sink, researchers have focused on utilizing passive heat transfer enhancement techniques to acquire more efficient design. The alter of the geometric
configuration for fins shape is one of heat transfer augmentation method which aimed to increase the heat exchange surface area as large as possible such as using wavy, perforated, slit, louvered, and composite fins.

Various studies have been conducted on heat sink design that aimed to augment the cooling of electronic components including numerical, analytical, and experimental approaches. Li et al.[1] studied numerically and experimentally the effect of Reynolds number, fin dimensions, and impingement distance on the hydrothermal performance of plate-fin heat sink (PFHS). The outcomes showed that thermal resistance increase with both of fin width and impingement distance. Yang and Peng[2] performed a numerical study of hydrothermal performance for PFHS with non-uniform fin height with a confined impingement. They showed that the base temperature can be decreased by increasing the fin height within the centre zone of the heat sink. In the same context, Huang and Chen[3, 4] investigated numerically the influences of two parameters namely fin height and width on the performance of PFHS. The study aimed to find the optimal fin height and width under impingement flow. Their findings revealed that the Nusselt number and enhancement coefficient could be improved by 3.20% and 3.20% respectively contrasted with standard heat sink. Kim et al.[5] conducted a comparative experimental study to assess two kinds of heat sinks in terms of hydrothermal performance: plate-fin and pin-fin heat sinks. The study includes several values of fin widths and flow rates. The results illustrated that fin-pin heat sink have lower thermal resistances than plate-fin heat sinks. Wiriyasart and Naphon[6, 7] performed numerically and experimentally the influences of three various fin designs namely rectangular, conical, and circular. The outcomes asserted that circular fin shape manifested a higher thermal performance compared with other fin shapes. Tang et al.[8] conducted experimentally the impact of conical fin-shaped on the hydrothermal performance of the heat sink. The study includes the effect of conical geometrical parameters such as cone angle, cone bottom diameter on the Nusselt number. The results emphasized that the hydrothermal performance of the cone heat sink is superior to that of conventional smooth plate heat sink. Freegah et al.[9] investigated numerically the effect of integrating of a half-round pin to the plate-fins in parallel and vertical arrangements. These pins are made of the solid material extracted from the fillets which configurated on the fin base. The study
proved that heat sink with half-round hollow pins arranged vertically which subjected to impinging flow has the best thermal performance compared with other configurations. Wong and Indran [10] studied numerically the impact of fillet profiles that configured at the bottom of the plate-fin heat sink. Their results revealed that the fillet profile could augment the thermal performance of the heat sink around 13% more than the traditional plate-fin heat sink.

According to the aforementioned, it can be seen that most research findings proved that the pin-fin heat sink showed more heat transfer augmentation than plate-fin heat sinks. So, most researchers trended to enhance heat transfer by either using pin-fin with different shapes to find the optimal geometrical parameters or trying to incorporate the pin fins with plate fins in one configuration for more heat transfer augmentation. The present study aims to investigate the influence of plate-fin corrugation with three shapes including rectangular, triangular, and semi-circular on the thermal performance of heat sink with flow impingement. These corrugations are formed on both sides of plate-fin in a periodic arrangement. This design offers more heat transfer exchanging areas without adding any mass to the original design of the plate-fin heat sink.

2. Mathematical modelling of plate-fin heat sink

In the present paper, four geometries of plate-fin heat sink are studied such as; traditional (T-PFHS), rectangular corrugation (REC-PFHS), triangular corrugation (TRI-PFHS), and semi-circular corrugation (SCR-PFHS). Fig. (1) illustrates the main dimensions of T-PFHS namely:

1- The base dimensions are: 40 mm × 39.7 mm, and thickness of (5 mm).

2- Channel width is 3.3 mm, and fin thickness is 1 mm.

3- The height of fin is 25 mm.

Fig. (2) and fig.(3) show the cross-sections of PFHS for the four configurations.
3. Numerical method

In the current paper, numerical solutions have been conducted to solve the hydrothermal performance of the PFHS. The governing equations are solved applying the finite volume approach. Furthermore, a second-order upwind scheme is utilized to solve the terms of
the Navier stokes equation. The SIMPLE algorithm is applied to solve and handle the velocity - pressure coupling in the momentum equation. The iterations are carried up to the residual target at $10^{-6}$ for both continuity, momentum and $10^{-9}$ for energy equation. $k-\varepsilon$ model is adopted in the current study because it has reasonable accuracy in minimum time-consuming compared with other models. The governing equations are solved using (CFD) software (FLUENT 16.1) according to the following assumptions:

1- Incompressible fluid flow.
2- Fluid is Newtonian
3- The flow is turbulent
4- The solution is considered in steady-state.
5- The thermal and physical properties are constant.

T-PFHS

TRI-PFHS
3.1 Governing equations

According to the previous, the governing equations can be written as follows [11, 12]:

\[
\frac{\partial}{\partial x_k} (\rho u_k) = 0
\]  

(1)

\[
\frac{\partial}{\partial x_k} \left( \rho u_i u_k \right) = -\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_k} \left( \mu + \mu_t \right) \left( \frac{\partial u_i}{\partial x_i} + \frac{\partial u_k}{\partial x_k} \right)
\]  

(2)

\[
\frac{\partial}{\partial x_k} (u_k T) = \frac{\partial}{\partial x_k} \left( \left( \frac{\mu}{Pr} + \frac{\mu_t}{Pr_t} \right) \frac{\partial T}{\partial x_k} \right)
\]  

(3)

where, \( \rho \) is (the density), \( u_i \) is (velocity components), \( P \) is (pressure), \( \mu_t \) is (turbulent viscosity), \( T \) is (temperature), \( Pr_t \) is (turbulent Prandtl number).

The equation of turbulence dissipation rate \( \varepsilon \) and turbulence kinetic energy \( K \) are:

\[
\rho \frac{\partial}{\partial x_k} (u_k \varepsilon) = \frac{\partial}{\partial x_k} \left( \left( \mu + \frac{\mu_t}{\sigma_{\varepsilon}} \right) \frac{\partial \varepsilon}{\partial x_k} \right) + \Gamma - \rho \varepsilon
\]  

(4)

\[
\rho \frac{\partial}{\partial x_k} (u_k K) = \frac{\partial}{\partial x_k} \left( \left( \mu + \frac{\mu_t}{\sigma_K} \right) \frac{\partial K}{\partial x_k} \right) + C_1 \Gamma \varepsilon - C_2 \frac{\varepsilon^2}{K + \sqrt{\varepsilon N}}
\]  

(5)
\( \Gamma \) is the rate of production of kinetic energy (K) and can be expressed as:

\[
\Gamma = \frac{\mu}{\rho} \left( \frac{\partial u_i}{\partial x_k} + \frac{\partial u_k}{\partial x_i} \right) \frac{\partial u_i}{\partial x_k}
\]  

(6)

\[ \mu_t = \mu_t' \frac{\rho K^2}{c} \]

\[ C_\mu = 0.09, C_1 = \max \left[ 0.43, \frac{\mu_t}{\mu_t + S} \right], C_2 = 1 \]

3.2 Boundary condition

The boundary conditions that applied on the proposed design are shown in fig.4 and as following:

1- channels inlet at
\( v = v_f, u = 0, w = 0 \)

In fluid \( T_f = 300, T_{in} = 300K \) (where \( T_f \) is fluid temperature)

In solid \(-k_s \left( \frac{\partial T_s}{\partial y} \right) = 0\) (where \( k_s \) is thermal conductivity)

2- channels outlet
\( p_f = 1\, atm, p_{out} = 1\, atm \)

In fluid \(-k_f \left( \frac{\partial T_f}{\partial z} \right) = 0\)

In solid \(-k_f \left( \frac{\partial T_s}{\partial z} \right) = 0\)

3- The outer wall at
\( u = 0, v = 0, w = 0 \)

\(-k_s \left( \frac{\partial T_s}{\partial x} \right) = 0\)

4- Interior walls of plate-fin
\( u = 0, v = 0, w = 0 \)
\[-k_s \frac{\partial T_s}{\partial n} = -k_f \frac{\partial T_f}{\partial n}\] Where (n) is represent the normal coordinate to the wall.

5- The substrate
\[-k_s \frac{\partial T_s}{\partial y} = q = 18750 \text{ W/m}^2\]

Figure 4. Boundary condition

3.3 Meshing

The quality of the mesh is vital to the achievement of accurate, thus a grid independence check is implemented to explore the optimal mesh composition that achieves the best accuracy in a minimum time manner as in table (1). The mesh type that used for the proposed PFHS geometries are non-structural mesh except for T-PFHS which meshed a structural mesh as shown in Fig.(5).

Table 1. Mesh independent test for T-PFHS, TRI-PFHS, REC-PFHS and SCR-PFHS

| Heat sink type | Number of grid element | Nusselt number | Difference ratio % |
|---------------|------------------------|----------------|--------------------|

3.4 Data reduction

The relevant terms that utilized to determine heat transfer and fluid flow in PFHS are:

1- The Reynolds number (Re) equation is written as follow

\[
Re = \frac{\rho u_m D_h}{\mu} \tag{7}
\]

Where \( D_h \) represents the hydraulic diameter

Where \( \rho, \ u_m, \ \mu \), are liquid density, average velocity, and dynamic viscosity respectively.
2- The pressure drop is given by

$$\Delta P = P_{in} - P_{out}$$

(8)

Where $\Delta P$ is the (pressure drop).

3- The average heat transfer coefficient is given by

$$h_{ave} = \frac{d_w A_{film}}{A_{con}(T_{W,ave} - T_{f,ave})}$$

(9)

Where

$d_w$, $A_{film}$, $A_{con}$, $T_{W,ave}$ and $T_{f,ave}$ are the heat flux, film area, convective heat transfer area, average temperature of microchannel wall and average fluid temperature.
4. Average Nu is written as flowing:

\[ \text{Nu}_{\text{ave}} = \frac{h_{\text{ave}} D_h}{k_f} \]  \hspace{1cm} (10)

Where \( k_f \) is the thermal conductivity of the fluid \( f \).

4. Result and discussion

4.1 Validation of numerical simulation

To validate the numerical results of the current study, the pressure drop and thermal resistance of T-PFHS are compared with experimental study of Ref [5]. The comparison shows a maximum disagreement of 12.4% for pressure drop and 8.8% for thermal resistance which considered a good agreement.

![Figure 6](a) pressure drop comparison with Ref.[11], (b) thermal resistance comparison with Ref.[5]

4.2 Characteristics of temperature distribution

Fig.7 depicts the temperature contours of various designs subject to impinging flow at a constant mass flow rate i.e. 0.0092 kg/s which equivalent to Reynolds number of 7700. In general, the impinging flow leads to form two distinct regions of temperature
distribution; the cold zone concentrated in the upper part of the fins, and the hot zone is occupying the bottom of the heat sink. The temperature contours of all proposed configurations have revealed a noticeable reduction in the temperature distributions compared to T-PFHS. Besides, SCR-PFHS shows the lowest temperature distribution among the other designs. The maximum temperature of these configurations attained about 334 K, 329 K, 324 K, 322 K for T-PHFS, TRI-PHFS, REC-PHFS, and SCR-PFHS respectively. The reduction in temperatures of new designs can be attributed to the existence of the additional surface area which provided due to the corrugation of the plate-fin. Besides, the corrugated plate is forming semi-confined channels such as rectangular channels, triangular channels, and semi-circular channels. These channels are accelerating the flow and consequently increase turbulence intensity inside the channel. Furthermore, the impinging flow which targeting the corrugated parts induces flow recirculation which increases the turbulence. This phenomenon is aiding to enhance the convective heat transfer coefficients in the wall-jet zone.

(a) T-PFHS  
(b) SCR-PFHS  
(c) TRI-PFHS  
(d) REC-PFHS
4.3 Characteristics of velocity distribution

Figs. (a), (b), (c) display the streamlines of impinging flow through the new configurations. As mentioned previously, the corrugated plate is forming semi-confined channels that accelerate the flow between plate fins and increase turbulence intensity inside these channels. The maximum velocity in these configurations (at Re=7700) attained about 1.38, 6.6, 6.23 and 6.84 m/s for T-PFHS, TRI-PFHS, REC-PFHS, and SCR-PFHS respectively.

Figure 8 Streamlines distributions in (a) T-PFHS, (b) SCR-PFHS, (c) TRI-PFHS, and (d) REC-PFHS

4.4 Characteristics of pressure drop distribution
The variation of pressure drop with the Reynolds number is illustrated in fig.(9) for the proposed designs. The results indicate that the pressure drop for all designs is directly proportional to Reynolds number. Besides, the results show a significant increase in pressure drop through the new designs compared with traditional heat sink T-PFHS. This can be attributed to the effect of surface corrugation which causes a considerable increase in the friction factor. Meanwhile, the fig.(9) shows that the highest pressure drop occurs in SCR-PFHS.

4.5 Performance analysis

To investigate the thermal efficiency of the different plate-fin heat sinks designs, the Nusselt number, thermal resistance, and changes in the base temperature are compared at various Reynolds numbers in Figs. 8a, b, and c respectively. Figs.10b and 10c, show that both base temperature, and thermal resistance decrease as the Reynolds numbers increases, while Fig.10a indicates that the Nusselt number increases with the Reynolds numbers. Besides, it can be noticed a noticeable enhancement in heat transfer for new designs compared with the traditional plate-fin heat sink(T-PFHS). At the same time, the SCR-PFHS design exhibited the highest Nusselt number value which attains about 138 at Reynolds number of 7700, while it revealed the lowest value of thermal resistance and base temperature among the other designs.

The performance evaluation of the new designs is based on the comparison with the traditional plate-fin heat sink(T-PFHS). The Nusselt number ratio over the pressure drop ratio is the distinct factor utilized to recognize the characteristics of overall performance.

The nondimensional formula of the performance factor is according to the following

$$Pf = \frac{Nu_{corr.}/Nu_{smooth}}{(P_{corr.}/P_{smooth})^{1/3}}$$

(11)

Where the Nusselt number ratio represents its value for the new design over that for the traditional heat sink, while the pressure ratio is the ratio of pressure drop through new design over that for the traditional heat sink.
Fig. 9 Variation of pressure drop in T-PFHS, TRI-PFHS, REC-PFHS, and SCR-PFHS
Figure 10 (a) variation of Nusselt number versus Reynolds No., (b) variation of thermal resistance versus Reynolds No., and (c) variation of average base temperature versus Reynolds No.

Figs. 11a, 11b illustrate the Nusselt number ratio and pressure drop ratio respectively. As it is evident, the heat transfer augmentation is associated with a high-pressure loss. The plate corrugation leads to an increase in surface friction which consequently increases the pressure drop. In contrast with other designs, the Nusselt number ratio of SCR-PFHS design is predominant over the pressure drop ratio which gives it the priority over other designs. Fig. (11c) shows the variation of performance factor ($P_f$) with Reynolds number. It can be seen that performance factor is proportional to Reynolds number. The highest performance factor attains about 1.76 in SCR-PFHS design.
5. Conclusions

In current study, the hydrothermal performance in three new designs of the corrugated plate-fin heat sink subjected to impinging airflow is studied numerically. The impacts of corrugations are evaluated according to the traditional plate-fin heat sink. The conclusions can be illustrated as follows:

1. The use of the corrugated plate-fin heat sink with impinging airflow is significantly enhanced the performance of heat sink compared with smooth plate-fin.
2. The use of the corrugated plate-fin contributed to accelerate flow and intensify the turbulence of the flow.
3. The current designs provide a larger convective heat transfer area than that of T-PFHS.
4. The SCR-PFHS achieved a best overall performance of $P_f = 1.76$ at $Re = 7700$.

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