Application of splitter plate on the modification of hydro-thermal behavior of PPFHS

S.E. Razavi, B. Osanloo*, R. Sajedi
Faculty of Mechanical Engineering, University of Tabriz, Tabriz 5166616471, Iran

HIGHLIGHTS

- This paper presents a new kind of heat sink using splitter behind pin fins.
- Splitter locating behind the pin-fin reduces thermal resistance and pressure drop.
- Splitters streamline the flow around the cylinder and decrease the pressure drop.

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ABSTRACT

This paper presents a Splitter Plate Pin-Fin Heat Sink (SPPFS) as a new kind of heat sink to enhance hydro-thermal performance of heat sinks. To reach this purpose, a thin plate known as a splitter is located behind the pin-fin. According to the numerical results, placing the splitter behind the pin-fin, reduces the thermal resistance and pressure drop on the heat sink. Splitter streamlines the flow around the cylinder and decreases the pressure drop around the pin-fins. Besides the pressure drop, increase of heat-transfer area dominates the heat transfer coefficient reduction and results to thermal resistance decrement. Furthermore, for considering the both thermal resistance and pressure drop effects; non-dimensional parameter of profit factor (J) is computed. Comparison between present SPPFS and Plate Pin Fin Heat Sink (PPFHS) confirms the ability of splitter in modification of hydro-thermal behavior of heat sinks by simultaneous reducing the pressure drop and thermal resistance.

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1. Introduction

Increasing the life and performance of electronic devices is largely associated with temperature control mechanisms, which are widely being developed. The methodology used for electronic device cooling can be generally divided into two categories, namely single and two-phase cooling. At first, efforts focused on the use of the air-cooled heat sinks. However, with the development of electronic equipment and the more application varieties, the generated heat fluxes have been increased, which leads to use of other dielectric fluids, such as water. But, the dielectric fluid is still used in the single-phase mode. With appearance of modern electronic devices such as electronic equipment used in military devices, microprocessors and laser diodes, the amount of heat fluxes are often more than 100 W/cm² [1]. Consequently, single-phase cooling techniques have reached their limits. Hence, attempts have been focused on the innovative methods, like two-phase cooling, which are known as effective ones. Different configurations in the two-phase cooling methods have been proposed, and the most efficient are categorized in the following three groups: Mini/micro-channels, jet impingements and spray. Note that depending on the applied conditions each of these methods has their preferences. Mini/micro channels have good performance in situations that require the large heat loss values in confined spaces [2–6]. Heat transfer coefficient in jet impingement is very high, especially in a jet incident zone. Nevertheless, to achieve a more uniform temperature on the heated surface the use of multiple jets is needed [7–11]. Besides this, spray evaporating is able enough to make a relatively uniform surface temperature in high-heat fluxes [12–16]. In this work, heat transfer enhancement and pumping power reduction are considered as interesting subjects in conjunction with increasing the range of heat sink application by single phase methods. One of the key techniques considered by researchers to reduce the both pumping power and thermal resistance is making

* Corresponding author. Tel.: +98 912 067 2210.
E-mail address: b_osanloo91@ms.tabrizu.ac.ir (B. Osanloo).

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the changes in their geometry, which increase the heat-transfer area and disturb the fluid flow. At the case of single phase cooling techniques, Plate Fin Heat Sinks (PFHSs) have been selected as an efficient alternative for enhancing heat transfer and reducing the thermal resistance. Investigations on the efficiency and optimized working conditions of PFHSs were published in some studies [17–19]. Jonsson and Moshfegh [20] presented an experimental study for plate fin, strip fin, and pin fin heat sinks. They reported their results by varying the fin height, fin-to-fin distance, and the tip and lateral clearance, and derived an empirical correlation to predict thermal and hydraulic behavior of heat sinks under above parameters. Khan et al. [21] studied thermodynamic loss in cylindrical PFHSs. For this reason, they applied entropy minimization method to consider heat transfer and pressure drop effects on the PFHS. They showed that all effective parameters such as geometric parameters, material properties, and flow conditions could be fixed in an optimized state. Yuan et al. [22] studied the effects of significant parameters on the PPFHS performance. They concluded that the thermal hydraulic performance of a PPFHS is mainly controlled by pin height and air velocity, and pins arrangements have fewer influences on thermal hydraulic behavior. Chen et al. [23] investigated the heat coefficient on the fin in the PFH for various fin species using inverse technic with corresponding experimental temperatures. Yu et al. [24] developed PFHSs by locating some columnar pins between plate fins, and introduced a new type of plate pin fin heat sink (PPFHS). Based on their numerical and experimental results, thermal resistance of PPFHSs was reduced by 30% compared to the PFHSs. Although the existence of cylindrical pins between the plate fins magnifies the heat transfer from a heated surface, but it will be modified using splitters behind the pins. This modification arises from the fact that the splitters streamline the flow around the cylinder and accordingly decrease the pressure drop, despite heat loss increment. Tiwari et al. [25] presented a 3D numerical investigation on the hydro-thermal flow fields in a circular tube with and without splitters. Existence of a splitter spreads streamlines behind the tube and descends the vertex length in comparison to tubes without splitter. Shrivastava et al. [27] studied the heat transfer and fluid flow through a tube, and tube with triangular and rectangular splitters. Flow analysis was done using FLUENT software in the laminar flow regime for different L/D. Since the separated flow on the cylinder surface will reach again at the top of the splitter, it was seen that at the back of the cylinder with small moving splitter, the flow pattern has no considerable difference with simple cylinder. They have reported that the average pressure in the cylinder in this case is more than a simple cylinder, and drag coefficient and oscillating lift coefficients are 30% and 90% less than the simple cylindrical, respectively.

In this work, the idea of locating the splitters behind the pin-fins is investigated. It is expected that the hydro-thermal behavior of the flow inside the heat sinks with splitter pin fins will be considerably affected. To show these effects on the flow and temperature fields, pressure drop, thermal resistance, average temperature of base plate and profit factor (J) are numerically presented and compared with previous PFHSs and PPFHSs. Results confirm and justify the application of splitters in the heat sink modification.

2. Problem definition

Plate Fin Heat Sinks (PFHSs) are designed and manufactured for increasing heat transfer from heat source plates as used in electronic devices. Fig. 1a shows a simple PFHs type which is usually used in common applications. Yu et al. [24] modified this heat sink by locating cylindrical pins between plates and introduced Plate Pin Fin Heat Sinks (PPFHSs) (see Fig. 1b). As it is asserted in Ref. [24] locating the cylindrical pins between the plate fins reduces the thermal resistance of PPFHSs by 30% compared to the PFHSs. Considering the PPFHS with extruded pins across the streamlines, will ascend the pressure drop which dramatically affects the application aspects. To overcome this drawback and modify the PPFHS performance, a thin splitter will be placed behind the cylinder pins. Locating the splitter behind the pins streamlines the flow around the cylinder and reduces the pressure drop. Also, surface increment caused by splitter amplifies the amount of heat transfer which
additionally improves the PPFHS performance. A Splitter PPFHS (SPPFHS) with all geometrical parameters is shown in Fig. 2, and detailed information is reported in Table 1. According to periodic structure of the heat sink and reducing the computational costs, only one passage flow is investigated as a single duct. Fig. 3 illustrates the computational domain with extruded pins and corresponding splitters. It is assumed that the heat sink is made of aluminum with conductivity of 202.4 W/m·K, and located on a heated surface. For verification of numerical procedure, the heat flux of surface is set to 3665 W/m² corresponding for heating power of 10 W.

3. Numerical approach and validation

3.1. Governing equations

Momentum and energy conservation equations along with continuity should be solved as governing equations to consider the conjugate heat transfer between fins and fluid flow. To reach the following governing equations, it is assumed that the flow is incompressible and viscous dissipations are also negligible. Due to the flow pattern and vortex generating condition inside the SPPFHS and previous numerical recommends [24], air flow between the plate fins are assumed to be turbulence. Decomposing the velocities to mean and fluctuating parts and taking the mean from the governing equations will reach to:

Continuity:

\[ \frac{\partial \rho u_i}{\partial x_i} = 0 \]  

(1)

Momentum:

\[ \rho \frac{\partial u_i}{\partial t} + \rho u_j \frac{\partial u_i}{\partial x_j} = -\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j} \left[ \mu \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) - \rho \vec{u}_i \vec{u}_j \right] \]  

(2)

Energy:

\[ \rho \frac{\partial T}{\partial t} = \frac{\partial}{\partial x_j} \left[ \frac{\mu}{\sigma_f} + \mu_t \right] \frac{\partial T}{\partial x_j} \]  

(3)

where, \( u \) expresses the velocity, \( \mu \) is kinematic viscosity and superscript \( \overline{\cdot} \) stands for mean values. For solid parts, Eq. (3) simplified to:

\[ \frac{\partial T}{\partial t} = \frac{\partial}{\partial x_j} \left[ \frac{\mu}{\sigma_f} \frac{\partial T}{\partial x_j} \right] \]  

Table 1

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fin length, ( l ) (mm)</td>
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</tr>
<tr>
<td>Fin width, ( w ) (mm)</td>
<td>53.5</td>
</tr>
<tr>
<td>Fin height, ( H ) (mm)</td>
<td>11.5</td>
</tr>
<tr>
<td>Fin thickness, ( t ) (mm)</td>
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</tr>
<tr>
<td>Fin spacing, ( \delta ) (mm)</td>
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</tr>
<tr>
<td>Fin number, ( N )</td>
<td>10</td>
</tr>
<tr>
<td>Pin height, ( h ) (mm)</td>
<td>2</td>
</tr>
<tr>
<td>Pin diameter, ( D ) (mm)</td>
<td>2</td>
</tr>
<tr>
<td>Pin spacing, ( S ) (mm)</td>
<td>12.75</td>
</tr>
<tr>
<td>Pin number, ( n )</td>
<td>24</td>
</tr>
<tr>
<td>Splitter plate length, ( L ) (mm)</td>
<td>1D, 2D and 3D</td>
</tr>
<tr>
<td>Cross section area, ( A ) (mm²)</td>
<td>50</td>
</tr>
</tbody>
</table>

Fig. 1. Schematic diagrams of PFHS and PPFHS: (a) Plate fin heat sink (b) Plate-pin fin heat sink.

Fig. 2. Schematic diagrams of SPPFHS.

Fig. 3. Computational domain.
where \( k_s \) is solid conductivity.

To continue and close the above equations, \( k-\varepsilon \) turbulence model has been used, and results obtained based on this model. The transport equations for \( k \) and \( \varepsilon \) are given as follows:

Transport equation for \( k \)

\[
\rho \frac{\partial k}{\partial t} + \rho u_j \frac{\partial k}{\partial x_j} = \frac{\partial}{\partial x_j} \left[ \mu + \frac{\mu_t}{\sigma_k} \frac{\partial k}{\partial x_j} \right] + \frac{\partial}{\partial x_j} \left( k \frac{\partial \theta}{\partial x_j} \right) - \rho E \tag{5}
\]
Fig. 8. Comparison of temperature contours for $Q = 50$ W and $U = 10$ m/s: a) PFHS, b) PPFHS, c) SPPFHS ($L = 2D$).
Fig. 9. Comparison of temperature contours for $Q = 100$ W and $U = 10$ m/s: a) PFHS, b) PPFHS, c) SPPFHS ($L = 2D$).
Transport equation for \( e \)

\[
\rho \frac{\partial e}{\partial t} = \frac{\partial}{\partial x_j} \left[ \left( \frac{\mu_t}{\sigma_t} + \frac{\mu}{\sigma} \right) \frac{\partial e}{\partial x_j} \right] + C_1 \mu_t \frac{e}{\rho K} \left( \frac{\partial u}{\partial x_j} + \frac{\partial v}{\partial x_j} \right) \frac{\partial u}{\partial x_j} + C_2 p \frac{e^2}{K}
\]

where \( \mu_t = \rho C_v k^2 / e \) and closures coefficients are set to: \( C_1 = 1.44, C_2 = 1.92, C_1 = 0.09, \sigma_k = 1, \sigma_e = 1.3 \).

SIMPLE algorithm was imposed to segregate the pressure–velocity coupled equations. The power-law discretization scheme interpolates the face value of momentum, \( k \), \( e \) and second order discretization scheme is applied for pressure values. To measure the flow and temperature characteristic, pressure drop (\( \Delta p \)) and thermal resistance \( R_{th} \) are evaluated as:

\[
\Delta = p_{in} - p_{out}
\]

\[
R_{th} = \frac{\Delta T}{Q}
\]

where \( p_{in} \) and \( p_{out} \) express the inlet and outlet pressures of the air in a single duct, \( \Delta T \) stands for the difference between the highest temperature on the fin base and the ambient air temperature, and \( Q \) is the constant heating power which is applied on the base plate of the heat sink.

### 3.2. Boundary conditions

Eqs. (1)–(3), should be solved considering accurate boundary conditions. Fig. 3 shows boundary conditions in the computational domain, containing a part of the heat sink and the outlet computational section. Note that the outlet section is considered to prevent the back flow at the outlet. Also, as it is seen from this figure, in order to reduce the computational domain, only one passage flow is investigated as a single duct. This simplifying assumption facilitates the problem to consider the symmetric boundary conditions at the outlet section of the computational domain. Note that because of the turbulent flow regime, assuming the symmetric boundary condition may be defected. Hence, to confirm the accuracy of symmetric boundary condition, we compared thermal resistance of one-passage flow assumption in a SPFPHS with two-passage flow in a SPFPHS (Fig. 4), which preterms symmetric BC at the outlet section. This is presented in Fig. 5 and indicates that considering the symmetric BC at the outlet section does not effectively affect the problem. Beside our demonstration, the similar assumption has been taken at the outlet section by previous authors [22,24]. Also, to cover the energy equation, and velocity field, adiabatic walls and no-slip BC are imposed on the fin’s wall. Furthermore, uniform inlet velocity and outlet pressure are considered with known inlet temperature as depicted in Fig. 3.

### 3.3. Numerical validation

Five inlet velocities of 4.5, 6.5, 8, 10 and 12.2 m/s are corresponded for Reynolds numbers of 2053, 2967, 3651, 4563 and 5567, respectively, where the Reynolds number is evaluated based on the hydraulic diameter. In order to validate the numerical simulations, available experimental data of Yu et al. [24] in a PFHS are used and corresponding parameters such as pressure drop and thermal resistance are compared with the current numerical results. Fig. 6a shows the pressure drop variation with inlet velocity. From this figure, it is seen that pressure drop is approximately a linear function of inlet velocity and current numerical results are also in good agreement with experimental data. Similar consequent arises from Fig. 6b, which shows the thermal resistance reduction with inlet velocity increment.

Hexagonal type of meshes was constructed in the fluid and solid zones (Fig. 7a). Also to ensure the numerical accuracy, mesh independence was performed using convective heat transfer coefficient of \( h \) in a heat sink and finally 587866 meshes were chosen for the case of \( L = 2D \) (Fig. 7b). This procedure was repeated for other simulations, and proper mesh numbers was also obtained for the cases of \( L/D = 1 \) and \( L/D = 3 \).

### 4. Results and discussion

Fig. 8 (a–c), and Fig. 9 (a–c) display the temperature contours for three types of heat sinks and two heat fluxes of \( Q = 50 \) W and \( Q = 100 \) W, respectively. As it is shown, the maximum temperature value reduces by locating the pin fins on the heat sink base plate, and it will also decrease with placing the splitter on the back of the pins. This behavior is strongly repeated for \( Q = 100 \) W (Fig. 9), where the maximum temperature gradient between PPFHS and SPFPFS is increased from 4 to 8 by increasing the heat flux. Also, it is seen that the temperature values increase from inlet to outlet. This temperature increment occurs to supply the constant heat flux to the heat sink bottom which arises from weakening the \( h \) values caused by boundary-layer development. Since the computation of surface parameters such as \( h \), directly depends on the velocity and temperature distribution near the surface, any cause, which disturbs temperature and velocity distributions, could directly affect the \( h \) values. Based on this fact, it could be concluded that locating the pin on the base plate and additionally placing the splitter on the
back of the pins, would be a cause that changes or amplifies the heat transfer coefficient. To make it clearer and understandable, $h$ values are plotted as a function of $x$ at the place between pins and plate fin (see Fig. 10) for three different cases. Fig. 11 shows that the use of pins magnifies the $h$ values, which confirms the pins effect on the boundary layer resistance. Results for $h$ in a PFHS and SPPFHS indicate that reducing the boundary layer resistance leads to more heat transfer, especially in presence of pins. Another important point in Fig. 11, expresses that the placing the splitter on the back of the pins, reduces the $h$ values compared to PPFHS. This reduction is also related to splitter role on the flow stabilization.

Figs. 12–14 shows the variations of pressure drop, thermal resistance and profit factor as a function of inlet velocity for different heat fluxes. In all these figures results of PFHS, PPFHS and SPPFHS with three splitter lengths are presented. Fig. 12 plot the pressure drop variation inside the heat sink. It is clear that the minimum pressure drop is related to PFHS, which behaves as a simple duct. Locating the pin-fin inside the PFHS grows the
resistance against the flow, and leads to pressure drop increment. This resistance increment would be reduced by adding the splitters behind the cylindrical pins. Fig. 12 specifies that the splitter with longer length of $L$, has fewer pressure drop. Also remember that the pressure drop arises from two sources; separation and viscosity effects. Hence, increasing the splitter’s length may ascend the viscosity pressure drops, but weakening of the vortex behind the pin, will reduce the total pressure drop. Thermal resistance variation as an important design parameter is depicted in Fig. 13. High boundary-layer resistance in the PFHS, owing to regular layer motion of fluid makes it to have high thermal resistance compare to PPFHS and splitter PPFHS. Existence of pin-fin inside the heat sink disturbs the regular motion of layers, and weakens the boundary-layer resistance and reduces the thermal resistance. Fig. 13 shows that locating the splitter behind the pin-fin modifies the heat sink performance by reducing the thermal resistance. To combine the thermal resistance and pressure drop in a single parameter and compare the results in various heat sinks, non-dimensional parameter of profit factor is defined as follows:

$$J = \frac{Q}{E}$$

$$E = UA\Delta p$$

where $E$ expresses the pumping power, $U$ is the free stream velocity and $A$ indicates the cross section area. Comparison of profit factor $J$ values in the present SPPFHS and PPFHS confirms the splitter’s efficiency on performance of a heat sink by simultaneous reduction of thermal resistance and pressure drop. As it is shown in Fig. 14 in a same inlet temperature and velocity, $J$ values increase by locating the splitters behind the fin-pins. Since the large axis ranges in Fig. 14 does not clearly display the $J$ factors differences, Fig. 15 displays the velocity effect on the $J$ factor in SPPFHS with $L = 3D$, and PPFHS in two heat fluxes of 25 W and 100 W. For this reason, the parameter of “$J$ growth increment” is defined as follows:

$$J \text{ growth increment} (\%) = \frac{J_{\text{SPPFHS}} - J_{\text{PPFHS}}}{J_{\text{PPFHS}}} \times 100$$

$$U = 4.5 \text{ (m/s)} \quad U = 6.5 \text{ (m/s)} \quad U = 8 \text{ (m/s)} \quad U = 10 \text{ (m/s)} \quad U = 12.2 \text{ (m/s)}$$

Fig. 15. The velocity effect on the $J$ factor in SPPFHS with $L = 3D$, and PPFHS in two heat fluxes of 25 W and 100 W.
Based on Fig. 15, at the case of $Q = 100\ W$, $J$ growth increment increases from 13% to 23%, and at the case of $Q = 25\ W$ it increases 8% – 15%, by increasing the free stream velocity from 4.5 m/s to 12.2 m/s. As another important point, it is found that by increasing the heat flux, the difference between $J$ factors increases. This fact causes more peak temperature reduction in high-heat fluxes compare to PPFHS. Average temperature of base plate on the heat sink is shown in Fig. 16. As it is expected, average temperature on the heat sink descends by heat convection enhancement arises from inlet velocity increment. Despite the reduction in the average temperature, $J$ factor decreases, but this $J$ reduction also occurs in PPFHS and does not disaf firm the logic of splitters application. To describe the effect of heat flux at the corresponding parameters, the variation of thermal resistance, $J$ factor and the average temperature of the base plate are presented in Figs. 17–19. Figs. 17 and 18 display that with an increment in the heat flux, the thermal resistance is reduced and the profit factor parameter is increased, respectively. Also confirm the splitter efficiency at the large L/D values. Finally, average temperature of the base plate is plotted in Fig. 19 as a function of the heat flux for different heat sinks. It is

![Fig. 16. Comparison of average temperature of base plate: a) $Q = 10(W)$, b) $Q = 50(W)$, c) $Q = 75(W)$, d) $Q = 100(W)$.](image)

![Fig. 17. Thermal resistance variation as a function of heating power for $V = 10\ m/s$.](image)

![Fig. 18. Profit factor variation as a function of heating power for $V = 10\ m/s$.](image)
clearly shown that the use of fin at the PFHS will reduce the average temperature of the plate, and placing a splitter on the back of the pin, will additionally decrease the average temperature, especially at high L/D values.

Fig. 20 indicates that the average temperatures of base plate are all limited under 358 K in the range of $Q = 10 – 75 \, \text{W/cm}^2$, and this condition will be kept with free stream velocity larger than 6.5 m/s in the heat flux of 100 W/cm². Finally, a comparison between SPPFHS (L = 3D) and PPFHS is presented in Table 2. For this reason, two parameters of percentage of pressure drop reduction (PPDR) and percentage of thermal resistance reduction (PTRR) are defined as follows:

$$\text{PPDR} = \frac{\Delta p_{\text{PFHS}} - \Delta p^{3D}_{\text{SPPFHS}}}{\Delta p_{\text{PFHS}}}$$

$$\text{PTRR} = \frac{R_{\text{th,PFHS}} - R^{3D}_{\text{th,SPPFHS}}}{R_{\text{th,PFHS}}}$$

Following parameters are computed in five heat fluxes and five free stream velocities. As it is inferred, in all conditions, splitter has a positive effect in both PPDR and PTRR, and would be more effective with heat flux enhancement.

5. Conclusion

Numerical procedure has been used to optimize PPFHSs. A thin plate has been located behind the cylindrical pin, and hydro-thermal behavior was studied computing pressure drop and thermal resistance coefficients. Also, to combine thermal resistance and pressure drop effects, non-dimensional parameter of profit factor (J) is computed. Comparing the results showed that the locating the splitters behind the pin, considerably reduces the pressure drop and thermal resistance and would be considered as an efficient alternative for heat sinks. According to numerical results, we can conclude that the SPPFHS usage is a practical idea in the electronic device cooling procedures which has been following advantageous:

1. Increasing the splitter length to pin diameter ratio L/D, reduces the both pressure drop and thermal resistance and ascends the profit factor.
2. The best performance of splitter plate occurs at high heat fluxes and high free stream velocities.
3. Maximum profit factor growth is found at the L/D = 3, and J growth increment is 25% more than PPFHS.
4. The use of SPPFHS guarantees the up limit temperature of 358K for electronic devices, in all heating powers below the 75 W, and in all free stream velocities. Also, for heating power of 100 W, the use of SPPFHS with stream velocities beyond the 6.5 m/s guarantees the up limit temperature of 358K.

References