Thermo-Fluid Characteristics of Microchannel Heat Sink With Multi-Configuration NACA 2412 Hydrofoil Ribs

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ABSTRACT In the present study, hydrofoil ribs made from NACA 2412 profile were mounted in a novel way at all walls of microchannel heat sink with various configurations to improve its thermo-hydraulic characteristics. The various configurations include comparison between staggered and aligned arrangements, and the variation of rib spacing between two consecutive hydrofoils (Sr = 0.2-0.6 mm) at a constant cord length (Cr = 0.4 mm). The study was performed with the help of commercially available computational fluid dynamics tool, ANSYS Fluent. The performance of various configurations were compared with each other and with smooth channel using parameters like friction factor (f), Nusselt number (Nu), and overall performance is determined by thermal enhancement factor (η). The results of this study showed that the best performance is achieved by the configuration where rib spacing becomes equal to cord length (Sr = 0.4 mm). Moreover, it has been observed in this study that staggered configuration performed better than aligned configuration.

INDEX TERMS Microchannel heat sink, NACA 2412 profile, hydrofoil ribs, aligned configuration, staggered configuration, thermal enhancement factor (η).

I. INTRODUCTION

In today’s electronic systems, the removal of dissipated heat from microelectronic systems is a big problem due to the decrease in the size of electronic products and devices. According to the international technology roadmap for semiconductors (ITRS), in 2018 the power density of a single chip package will raise to 108 W/cm2 [1]. Therefore, the conventional force convection cooling methods are not effective enough to remove the dissipated heat from these small compact systems and became obsolete by the requirement of very high flow velocity which arises the problem of vibration and noise [2]. As a result, rise in temperature and a hot spot is produced in the substrate which reduces the performance of ultra-large-scale integrated circuits. For this sake, the heat flux produced need to be removed from the hot spot; otherwise, it can result in the shortening of the life span of chip and can also cause permanent damage to the ultra-large-scale integrated circuit [3].

To resolve this challenging problem, in 1981, Tuckerman and Pease [4] proposed an alternative cooling technique known as a Microchannel heat sink (MCHS) which was capable enough of dissipating $7.9 \times 10^6$ W/m². Later on, scientists find out that the compact high-powered laser diode array stacks are cooled effectively through MCHS with ribs. Which shows promising results than conventional straight MCHS due to the increase in the effective cooling area and also the thermal properties improve with the increase in flow velocity [5]. To find out performance parameters and increase the efficiency of MCHS, different studies were carried out over the years by many researchers. As the introduction of ribs increases the thermal properties improves but at the same time, it also increases the frictional losses which increase power consumption. To properly evaluate
the performance enhancement of the MCHS, researchers established a performance enhancement criterion known as the thermal enhancement factor ($\eta$), which evaluates MCHS performance in terms of both thermal and frictional losses [6].

A study was carried out in which the researchers found that the conventional Navier-Stokes and energy equations can adequately predict the heat transfer and fluid flow characteristics across the MCHS by comparing experimental and computational data. A rectangular smooth MCHS of copper with a cover plate of polycarbonate plastic and deionized water with Re from 100 to 1600 was used as a coolant. Several microchannel heat sinks were stacked together with a height of $713 \mu m$ and width of $231 \mu m$. A heat flux of $100 \ W/cm^2$ and $200 \ W/cm^2$ were employed at the base of the stacked MCHS and it was found that there is no transition from laminar to turbulent flow for Reynolds number less than 1600 [7].

Different techniques were used for the improvement of cooling capability of MCHS which include the changing of cross section shapes like rectangular, circular, hexagonal, etc. of MCHS [8] and disturbing the flow of fluid through a change in the periphery of the channel by introducing different kinds of obstructions on walls such as, cavities, ribs dimples, and groves, etc. [9, 10]. Furthermore, many computational and experimental studies were conducted to determine the effects of ribs on the thermo-fluid properties of the modified MCHS. For example, Wang et al. [11] examined experimentally and numerically the performance of MCHS with different heights of ribs on its walls. Their study’s findings revealed that MCHS with higher rib heights has a higher Nusselt number due to more fluid mixing in MCH. However, due to a large pressure drop in comparison to the amount of heat extracted, the thermal enhancement factor was less than unity.

Similarly, Jasperson et al. [12] compared smooth MCHS and pin fin MCHS of copper with single-phase water flow both having the height of $670 \mu m$ and width of $200 \mu m$. They observed that for $Re < 2400$, the pin fins did not show significant thermal enhancement. However, for flows $Re > 2400$, pin fins showed good thermal performance results, but at the cost of pressure drop twice as large as that of the pressure drop of smooth MCHS and increased significantly with the increase in Reynolds number. Moreover, they also mentioned that micro end milling and micro EDM are ideal for prototyping MCHS. Furthermore, the effects of ribs on the thermo-fluid properties of MCHS different configurations of ribs (i.e. triangular semicircular, rectangular, triangular–circular, rectangular–semicircular, and rectangular–triangular) were studied numerically by Khan et al. [13]. They found that triangular ribs achieved best performance with maximum friction factor ($f$) of 3.7 and higher Nusselt number (Nu) ratio of 1.25 at Reynolds number 500.

Some researchers, such as Rehman et al. [14] used various MCHS surface enhancers like protrusions/dimples with different flow rates ranging from 100 to 1000 Reynolds numbers and studied the effects of geometric parameters on the thermal and hydraulic properties of the MCHS. They found that among different configurations of surface enhancements, all wall protrusions showed the highest thermal enhancement. Furthermore, another research was conducted by Abdollahi et al. [15] on diamond shape and elliptical ribs and it was shown that elliptical ribs have better performance than diamond ribs. Similarly, Abdoli et al. [16] performed a numerical study on MCHS with six different types of fins on the cooling of the single hot spot from high heat flux electronic chips. Among various types of ribs, hydrofoil shape fins showed a considerable decrease in pumping power of about 30.4% at the expense of a 3.2% increase in the convection load as compared to circular fins. Furthermore, Zhai et al. [17] carried out a numerical study of six designs of cavities and fins used as surface enhancers in MCHS. Their results revealed that in the range of (300-600) Reynolds number (Re), the triangular cavities and triangular ribs configuration performed best on the basis of exergy and entropy analysis.

According to the research of Xia et al. [18], adding cavities on the walls of MCHS improves fluid mixing by disrupting the boundary layer and forming vertices through the introduction of complex geometries. Moreover, it is being recorded by Li et al. [19] that the geometry of fins has a greater effect than cavity on the performance of MCHS. Additional flow disturbances can be introduced by different configurations of ribs and cavities, dimples, and protrusions on the walls of channel which will result in improvement in the heat transfer but the pressure drop will increase dramatically. This pressure drop can be reduced by introducing surface enhancement in the direction of flow [9]. Furthermore, some researchers have combined the ribs and cavities and numerically studied their effects on the thermal and hydraulic properties and thermal enhancement factors. For example, Ghani et al. [20] carried out a study by combining rectangular ribs and sinusoidal cavities. They discovered that the presence of cavities compensating for the pressure drop, therefore, the overall thermal enhancement factor increases. Furthermore, researchers conducted additional research on the combination of different ribs and cavities and discovered that the influence of ribs shape is greater than that of cavities on the thermal and hydraulic properties of MCHS [21–23].

The present study is based on the numerical analysis to compare smooth channel with MCHS having ribs mounted on all walls (all walls ribs MCHS) with aligned and staggered configuration and with the variation of stream-wise spacing (Sr) between the ribs. In addition, its effects on the thermal and hydraulic properties of MCHS in terms of Nusselt Number (Nu), friction factor ($f$), and thermal enhancement factor ($\eta$) will be investigated.

II. MODEL DESCRIPTION AND METHODOLOGY

A. COMPUTATIONAL DOMAIN

In this study, a rectangular channel is used as the computational domain because previous research studies have shown...
that rectangular channel has lower thermal resistance than triangular and trapezoidal channels. [8]. The geometry of the computational domain is given in the Figure 1(a), where $H$ is the height of MCHS, $H_c$ is the channel height, $W$ is the width of MCHS, $W_c$ is the channel width, and $L$ is the total length of MCHS. Moreover, Figure 1(b) shows the section view of aligned configuration of all wall ribbed channel, where $Sr$ is the distance between the leading and trailing edge of two adjacent hydrofoils. Moreover, the staggered configuration was made with the same dimensions but the ribs on opposite sides were not aligned with each other, as done by Rehman et al. [14].

The hydrofoil ribs used in this study are designed using the dimensions of upper surface of asymmetrical NACA 2412 profile and revolve it along the centerline, thus making it a symmetrical shape hydrofoil rib. Hydrofoils were placed at the centerline of each wall of MCHS as shown in Fig. 1. (b) To neglect the effects of entrance length on pressure drop, the first rib is kept at 0.3 mm from the entrance. The spacing between the consecutive hydrofoils varies from 0.2 to 0.6 mm. Moreover, Table 1 shows the geometrical dimensions of the MCHS and Table 2 shows the thermophysical properties of the materials used i.e. water and copper.

### B. GOVERNING EQUATIONS

Based on previous research by various scientists, the following assumptions were made to build a conjugate model of 3D flow and heat transfer:

1. The fluid flow in the MCHS is classified as laminar.
2. Fluid is incompressible and Newtonian.
3. Heat transfer to the fluid and fluid flow is assumed to be steady.
4. The solid domain’s thermophysical properties are assumed to be constant.
5. Because of the low Reynolds number, viscous dissipation effects are ignored in the energy equation.

#### TABLE 1. MCHS geometric dimensions.

| Geometric Parameters          | Symbols | Values (mm) |
|------------------------------|---------|--------------|
| Length of the MCHS           | $L$     | 10           |
| Height of the MCHS           | $H$     | 1.5          |
| Width of the MCHS            | $W$     | 0.6          |
| Height of the channel        | $H_c$   | 0.5          |
| Width of the channel         | $W_c$   | 0.35         |
| Cord length of hydrofoil rib | $Cr$    | 0.4          |
| Height of hydrofoil rib      | $Dr$    | 0.08         |
| Spacing between consecutive ribs | $Sr$ | 0.2, 0.4, 0.6 |

#### TABLE 2. Thermo physical properties of materials.

| Properties                  | Symbols | Water | Copper |
|-----------------------------|---------|-------|--------|
| Density                     | $\rho_l$ (kg/m$^3$) | 998.2 | 8978   |
| Dynamic viscosity           | $\mu$ (kg/m/s)      | 0.001003 | -      |
| Specific heat               | $C_p$ (J/K-kg)       | 4184  | 386    |
| Thermal conductivity        | $k$ (W/m.k)          | 0.6   | 387.6  |

Taking the preceding assumptions into account, the following governing equations were used to construct the mathematical computational model.

Continuity Equation:

\[
\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0 \quad (1)
\]

where $u$, $v$, and $w$ are the components of velocity in x, y, and z-direction.

Momentum equations:

In x-direction

\[
\frac{\partial u}{\partial x} + \frac{\partial u}{\partial y} + \frac{\partial u}{\partial z} = -\frac{1}{\rho_f} \frac{\partial p}{\partial x} + \frac{\mu}{\rho_f} \left( \frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2} \right) \quad (2)
\]
In y-direction
\[
\frac{\partial v}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial v}{\partial z} = -\frac{1}{\rho_f} \frac{\partial p}{\partial y} + \frac{\mu}{\rho_f} \left( \frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2} \right)
\]
(3)

In z-direction
\[
\frac{\partial w}{\partial x} + \frac{\partial w}{\partial y} + \frac{\partial w}{\partial z} = -\frac{1}{\rho_f} \frac{\partial p}{\partial z} + \frac{\mu}{\rho_f} \left( \frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2} \right)
\]
(4)

where \(\rho_f\) is the density, \(\mu\) is the dynamic viscosity of the fluid and \(p\) is the pressure.

Energy equations:
For solid domain
\[
k_s \left( \frac{\partial^2 T_s}{\partial x^2} + \frac{\partial^2 T_s}{\partial y^2} + \frac{\partial^2 T_s}{\partial z^2} \right) = 0
\]
(5)

For fluid domain
\[
\frac{\partial T_f}{\partial x} + \frac{\partial T_f}{\partial y} + \frac{\partial T_f}{\partial z} = \frac{K_f}{\rho_f C_p f} \left( \frac{\partial^2 T_f}{\partial x^2} + \frac{\partial^2 T_f}{\partial y^2} + \frac{\partial^2 T_f}{\partial z^2} \right)
\]
(6)

where \(T\) is the temperature and \(k\) is the thermal conductivity and \(C_p\) is the specific heat for solid (copper) and fluid (fluid) domains, respectively.

**C. BOUNDARY CONDITIONS**

The following conditions were used to solve the governing equations:

The hydraulic diameter of the microchannel is large enough, therefore, no-slip condition is applied to the inner surface of the wall. i.e.
\[
u = v = w = 0
\]
(7)

At the inlet of the channel, i.e. at \(x = 0\) mm.
\[
u = u_{in}; T_f = T_{in} = 293.15K
\]
(8)

where \(u_{in}\) is the uniform velocity applied ranging from 0.243 m/s to 2.428 m/s which is calculated from Reynolds number and inlet temperature is taken as \(T_{in} = 293.15\) K in this study.

At the outlet of the channel i.e. at \(x = 10\) mm, the pressure is atmospheric i.e.
\[
p = p_{out} = 1
\]
(9)

At the bottom surface of the channel a constant heat flux \(q_w\) of 100 W/cm\(^2\) is applied.
\[
-k_s \frac{\partial T_s}{\partial x} = q_w = 100W/cm^2
\]
(10)

whereas all the remaining surfaces are considered as adiabatic.
\[
\frac{\partial T_f}{\partial x} = \frac{\partial T_s}{\partial x} = 0
\]
(11)

**D. DATA REDUCTION**

The Reynolds number is defined as
\[
Re = \frac{\rho f u_{in} D_h}{\mu_f}
\]
(12)

where \(Re\) is the Reynolds number, \(u_{in}\) is the mean velocity, and \(D_h\) is the hydraulic diameter. This is expressed for a rectangular channel with height \(H_c\) and width \(W_c\) as:
\[
D_h = \frac{2HcWc}{H_c + W_c}
\]
(13)

The average apparent fraction factor \(f_{app}\) is expressed as:
\[
f_{app} = \frac{2D_h \Delta p}{L \rho_f u_{in}}
\]
(14)

whereas \(\Delta p\) is the change in pressure between the inlet and outlet of the channel and \(L\) is the total length of the channel.

The average Nusselt number can be calculated as:
\[
Nu = \frac{hD_h}{k_f}
\]
(15)

where \(h\) is the average heat transfer coefficient which can be expressed as:
\[
h_{avg} = \frac{q_w A_w}{2(Hc + Wc) L \Delta T}
\]
(16)

where \(q_w\) and \(A_w\) are the heat flux applied and area of the base side of the microchannel, respectively. Whereas \(\Delta T\) is the temperature difference between mean fluid temperature \(T_f\) and channel average wall temperature \(T_w\) as shown below:
\[
\Delta T = T_w - T_f
\]
(17)

The thermal enhancement factor can be expressed as:
\[
\eta = \frac{Nu/Nu_o}{\sqrt{f/f_o}}
\]
(18)

where \(Nu\) and \(f\) is used for the enhanced surface microchannel and \(Nu_o\) and \(f_o\) is for the smooth surface microchannel.
III. NUMERICAL MODEL DESCRIPTION

A. NUMERICAL SOLUTION AND CONVERGENCE CRITERIA

The finite volume method is used in this study to discretize the mathematical domains of solids and fluids into finite-sized cells using the computational fluid dynamic (CFD) code ANSYS 15.0. In other words, the algebraic equations are derived from the governing equations via the control volume. For the current numerical model to achieve rapid convergence, the convective and diffusive parts of the momentum, continuity, and energy equations are solved using QUICK method and the first-order upwind interpolation scheme. For pressure velocity coupling, the SIMPLE Scheme is used. In addition, for the residuals of the continuity momentum and energy equations, convergence criteria of less than $10^{-6}$ are set.

B. MESH INDEPENDENCE

To divide the solid and fluid domains of MCHS, the mesh was created using hexahedral elements. To achieve accurate modeling results for heat transfer between solid and fluid, the conjugate boundary was refined further. The orthogonal quality was within the acceptable range. A mesh independence test was performed at 100 Reynolds numbers on all wall ribs MCHS with Sr = 0.2 mm and through 5 grid sizes with the number of elements ranging from 117571 to 474628 to determine whether the solution is independent of grid size.

The relative percentage error expression can be expressed as:

$$E\% = \left| \frac{J_f - J}{J_f} \right| \times 100$$

where $J_f$ represents any parameter of finest grid size and $J$ represents any grid size parameter such as pressure drop and Nusselt number, etc. Table 3 represents the relative error for pressure drop and Nusselt number and it is observed that as the number of elements increases above 0.3 million, the error drops to 1% for pressure drop and 2% for Nusselt number henceforth mesh no. 4 is used for analysis in the current study.
IV. RESULTS AND DISCUSSIONS

A. MODEL VALIDATION

To validate the computational results, the Nusselt number obtained from this model is compared with the experimental work of Wang et al [11] as shown in Fig. 2. This demonstrates that the results deviate from the experimental work by about 5% for Reynold Numbers 100 to 1000, thus the present model can be considered as suitable for predicting the thermo-fluid characteristics of MCHS.

B. PRESSURE AND VELOCITY DISTRIBUTION

The counters of pressure and velocity distribution across the length of MCHS are shown in this section. The pressure drop across the MCHS is relatively high as the spacing between the hydrofoils decreases from 0.6 mm to 0.2 mm, as shown in Fig. 3. This is due to an increase in the obstruction to fluid flow, which causes non-uniformities in fluid velocity, resulting in hydrodynamics boundary layer disruption.

Fig. 4 depicts the velocity distribution along the length of the channel. It can be observed from Fig. 4 that as the spacing between the ribs decreases, the fluid mixing increases, resulting in an increase in non-uniformities in the velocity which increases heat transfer. However, this increase in thermal performance because of the fluid mixing leads to increase in fractional loss increases, resulting in an increase in pressure drops and pumping power requirements.

C. OVERALL PERFORMANCE ANALYSIS

The temperature distribution of MCHS for the Reynolds number Re = 400, heat flux 100 W/cm² for different configuration and rib spacing is shown in Fig. 5. It is also compared with the smooth channel temperature distribution to see how the thermal performance of MCHS is improved. It can be seen from Fig. 5 that the temperature distribution in the straight channel is not uniform and a hot spot can be noticed at...
FIGURE 7. (a) (b) Effects of Reynolds number (Re) on fractional factor (F_{APP}) across MCHS with different RIBS configurations.

the exit of the channel. The hot spot produced at the end can ultimately damage the device and decrease its life span. After the addition of ribs, it can be noticed that the hot spot produced is normalized which can lead to reduce thermal stresses and enhance the performance of device. Moreover, the temperature decreases across the MCHS as the rib spacing decreases from 0.6 mm to 0.2 mm which can be attributed to the better mixing of fluid.

Fig. 6 shows that by adding ribs to the MCHS, the pressure drop across the MCHS increases significantly, necessitating the use of more electrical energy to pump the fluid back to the heat sink. The graph in Fig. 6 (a) shows that the pressure drop increases as the spacing between the MCHS hydrofoils decreases, which is due to an increase in the amount of obstruction in the channel. Moreover, the graph in Fig. 6 (b) shows that the pressure drop for both staggered and aligned hydrofoils is approximately the same for Re = 400; however, for Re > 400, the aligned hydrofoil has a greater pressure drop than the staggered one.

The variation of apparent frictional factor with Reynolds number for different configurations of MCHS is shown in Fig. 7. According to Fig. 7 (a), the straight channel with no ribs has the lowest friction factor because there are no obstacles in the flow direction. However, the MCHS with ribs has a higher f_{app} because the presence of ribs offers obstruction to flow. Furthermore, as the spacing between the ribs decreases, the f_{app} increases due to an to fluid flow, which causes non-uniformities in fluid velocity, resulting in hydrodynamics boundary layer disruption. increase in the amount of obstruction in the fluid flow direction, resulting in more nonuniformities in velocity and mixing. Moreover, the graph in Fig. 7 (b) shows that MCHS with staggered ribs produces less f_{app} than MCHS with aligned ribs for Reynolds numbers less than 400, but f_{app} for Reynolds numbers greater than 400.
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FIGURE 9. (a) (b) Effects of reynold number (Re) on thermal enhancement factor (η) across MCHS with different RIBS configurations.

V. CONCLUSION

The thermal and hydraulic characteristics of MCHS with different design configurations of hydrofoil rib has been investigated by conducting a numerical study. The performance enhancement with the addition of different ribs configurations to the MCHS was evaluated through the flow properties, thermal characteristics, and thermal enhancement factor. It was observed that the addition of ribs increases heat transfer rate as well as the pumping power; however, an overall performance investigation revealed that the addition of hydrofoil ribs is advantageous to MCHS. Moreover, the performance of different rib spacing was compared with each other and it was found that decreasing rib spacing is useful; however, after a certain limit it declines the overall performance. In addition to this, it was observed that the MCHS with rib spacing equal to cord length (0.4 mm) offers the highest thermal enhancement factor. Moreover, it was also found that the staggered combination of the hydrofoil ribs performed better than the aligned one.

NOMENCLATURE

MCHS Microchannel heat sink.
AWH All wall hydrofoil ribs.
Re Reynolds number.
Nu Nusselt number.
fapp Friction factor.
L Length of the MCHS [mm].
H Height of the MCHS [mm].
W Width of the MCHS [mm].
Hc Height of the channel [mm].
Wc Width of the channel [mm].
Cr Cord length of hydrofoil rib [mm].
Dr Height of hydrofoil rib [mm].
Sr Spacing between two consecutive ribs [mm].
Dh Hydraulic diameter [mm].
ΔT Temperature difference [K].
Δp Pressure drop [Pa].
**E** Relative error [%].

**u_m** Mean velocity [m/s].

**U, v, w** Components of velocity [m/s].

**qw** Wall flux [W/cm²].

**H** Heat transfer coefficient [W/m².K].

**C_p** Specific heat capacity at constant pressure [J/kg.K].

| GREEK LETTERS |
|----------------|
| μ Dynamic viscosity [kg/m.s]. |
| k Thermal conductivity [W/m.K]. |
| ρ Density [kg/m³]. |
| η Thermal enhancement factor. |

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