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# Thermal Performances of Multi-Layered Liquid Cold Plates

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***Abstract*** **-** Thermal performances of multi-layered cold plates (CPs) with varying numbers of channels are investigated through three-dimensional simulation of laminar flow and heat transfer. Thermal performances are characterized by the maximum temperature and temperature variation across the heating surface. The thermal performances are presented as functions of flow rates and pumping power to provide better insight on CP's practical applications. It was found that at both a given flow rate and pumping power, increasing the number of layers monotonically enhances the heat transfer rate.; however, the percentage of enhancement of heat transfer is reduced by increasing the number of layers beyond two due to additional thermal resistance experienced between the lower-level channels/layers and the heat source. The findings suggest the existence of a threshold number of layers such that beyond that threshold, the heat transfer is not enhanced.

***Keywords*****:** Thermal management; Liquid cold plate; Pumping power; Heat transfer enhancement; CFD.

## 1. Introduction

Liquid cold plates (CPs) have been widely used as effective thermal management systems (TMSs) in broad applications [1]. A CP consists of serpentine channels inside a metal plate in which liquid flows through the channels to remove heat from heat sources located on the CP [2]. Despite extensive research on cooling performances of CPs, available datasets are mostly applicable to the design parameters and operating conditions of their respective research; therefore, the performance of a customized CP needs to be addressed. In this study, the effects of varying the number of channels of a CP at different layers and at a fixed cross section on CP's hydrothermal characteristics are investigated. In addition to maintaining the maximum temperature of the device below a design limit and improving the temperature uniformity across the device, an efficient active TMS must operate with a low pumping power ($P_P$) to minimize required external power [3]. $P_P$ is a key design parameter of an active TMS such that a penalty in the $P_P$ may hinder the use of the TMS, regardless of its capability to improve thermal performances [4]. To provide better insight on the practical applications of the CPs in this study, their thermal performances are described as functions of $P_P$.

## 2. Computational Procedure

Fig. 1 illustrates the three CPs made of copper. The CPs include one, two, and three layers of serpentine channels, and are specified as N=1, N=2, and N=3, respectively, which N stands for the number of layers. It is assumed that the channels are grooved inside the copper plate; therefore, the channels have zero thickness. The cross section of the channels plus the distance between adjacent channels (Fig. 1(c)) is 6 mm × 6 mm; therefore, the channel height is 6, 2.5, and 1.33 mm in CPs with N=1, N=2, and N=3, respectively, while the channel width is 6 mm in all designs. Liquid water enters the inlet(s) at a temperature of 22 °C and exits from the outlet(s). The governing equations to simulate the flow and heat transfer by assuming a steady, laminar, and incompressible flow, and constant properties for the fluid and solid are as follows:

Continuity:
$$\nabla.\boldsymbol{u} = 0 \tag{1}$$
Momentum conservation:
$$(\boldsymbol{u}.\nabla)\rho\boldsymbol{u} = -\nabla p + \mu\nabla^2\boldsymbol{u} \tag{2}$$
Energy conservation (fluid):
$$\boldsymbol{u}.\nabla T_f = \frac{\lambda}{\rho c_p}\nabla^2 T_f \tag{3}$$
Energy conservation (solid):
$$\nabla^2 T_s = 0 \tag{4}$$

where $\rho$, $u$, $p$, $\mu$, $\lambda$, $c_p$, and $T_f$ are the fluid density, velocity, pressure, viscosity, thermal conductivity, specific heat, and temperature, respectively, and $T_s$ is the temperature of the solid. The simulations are performed for a wide range of flow rates within the laminar regime. Flow rates and temperature are set at the inlet, and zero axial gradients for all the variables are imposed at the outlet. The remaining surfaces are walls with a no-slip boundary condition. A constant heat flux equivalent to 120 W is applied at the heating surface located at the top of the CP (Fig. 1(a)). The remaining exterior surfaces are insulated. A grid structure with ~373,000 computational cells is selected through a grid independence test since increasing the number of cells beyond 373,000 keeps the magnitude of heat transfer coefficients and friction coefficients below 5% and 9%, respectively. The transport equations are solved using Ansys Fluent.

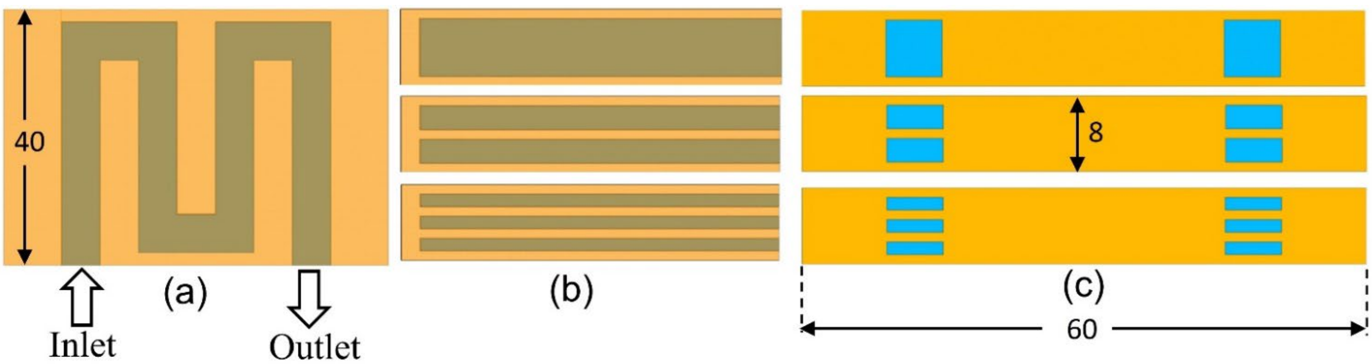


Fig. 1: (a) Top view of the CPs (heating surface); (b) Side view of the CPs with N=1, 2, and 3; (c) Front view of the CPs with N=1, 2, and 3. Dimensions are in mm.

## 3. Results

The flow remained within the laminar regime throughout the simulations since the maximum Reynolds number was ~1936. Fig. 2 illustrates the maximum temperature ($T_{\max}$) and temperature variation ($\Delta T$) across the heating surfaces at different flow rates. The flow rates are specified in LPM (liter per minute). For a given CP, a higher flow rate results in larger sensible heat and, in turn, a monotonic reduction in both $T_{\max}$ and $\Delta T$, in which the latter corresponds to greater temperature uniformity across the heating surface.

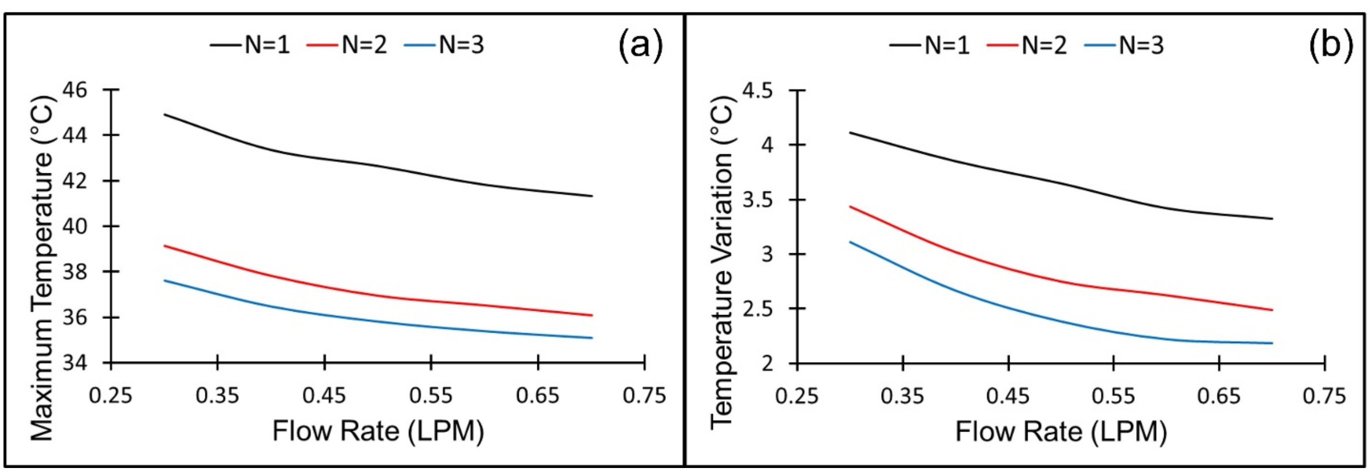


Fig. 2: (a) Maximum temperature, and (b) temperature variation over the heating surface at different flow rates.

At a fixed cross-sectional area, increasing the number of layers results in an increase in heat transfer area and, consequently, an enhancement in heat transfer. In this study, the effects of number of layers on the heat transfer enhancement is explained by the thermal efficiency ($\eta$), which specifies the percent difference between the $T_{\max}$ of $i$th CP and $T_{\max}$ of $j$th CP, as follows:

$$\eta = -\frac{T_{\max,j} - T_{\max,i}}{T_{\max,i}} \times 100 \tag{5}$$

A positive value of $\eta$ corresponds to an enhancement of heat transfer, which translates to a decrease in $T_{\max}$. Based on Fig. 2 (a), increasing the number of channels from one to two, and from two to three changes the range of $\eta$ from 12.7 to 13.6%, and 2.8 to 3.9%, respectively. The results indicate that the heat transfer enhancement is reduced when increasing the number of layers beyond two.

Increasing the number of channels at different layers results in additional thermal resistance between different layers and the heat source. Introducing one more layer in the CP results in additional thermal resistance due to convection ($R$) to the overall thermal resistance. Note that $R = 1/hA$, where $h$ and $A$ are the convection heat transfer at the upper-level channel, and the heat transfer area, respectively. At a given flow rate, an increase in the number of channels in a CP with a fixed cross-section results in smaller flow rates inside each individual channel since the total flow rate is divided between them. The lower flow rate results in a lower $h$ and, in turn, a larger $R$. Therefore, increasing the number of layers leads to a monotonic increase of the thermal resistance and reduction in heat transfer enhancement of the CP. This indicates that $T_{\max}$ remains unchanged (i.e., $\eta \sim 0$) beyond a threshold number of layers. Fig. 3, which compares the temperature distributions on the heating surfaces of the CPs at 0.7 LPM, exemplifies this concept.

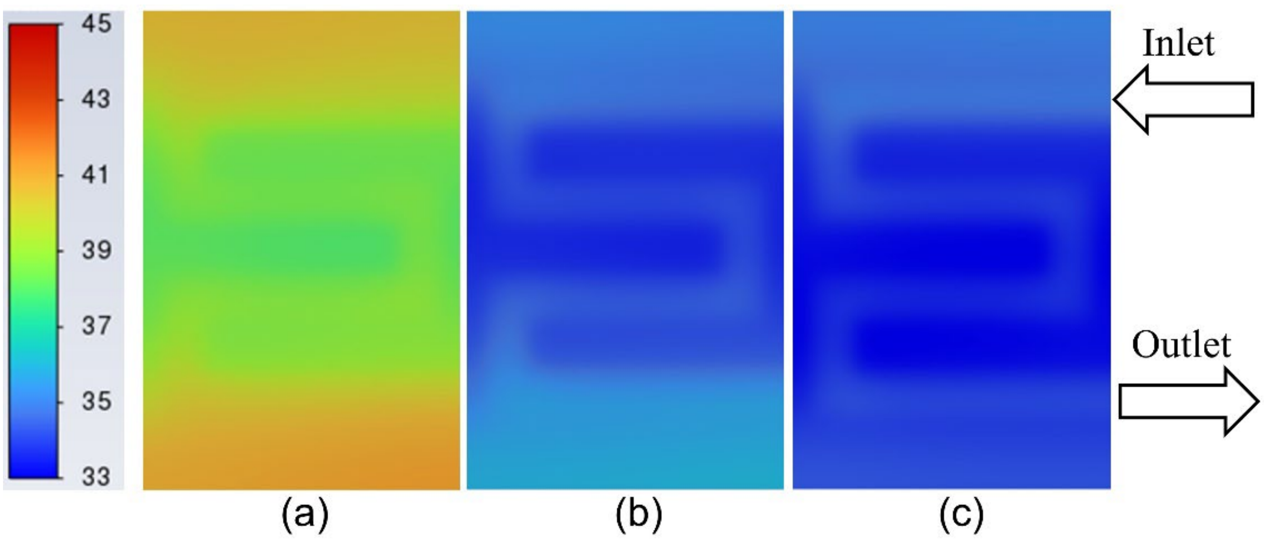


Fig. 3: Temperature distribution (in °C) over the heating surface of CPs with (a) N=1; (b) N=2; (c) N=3. Flow rate: 0.7 LPM.

Fig. 4 illustrates the thermal performances of CPs as functions of $P_P$, which is calculated as follows [4]:

$$P_P = \Delta p \times \dot{V} \tag{6}$$

where $\Delta p$ and $\dot{V}$ are the pressure drop across the channel and the total volume flow rate, respectively. The results in Fig. 4 cover different ranges of flow rates within the laminar regime. For a given CP, an increase in the $P_P$ that is caused by a higher flow rate leads to larger sensible heat, and as a result, a monotonic reduction in both $T_{\max}$ and $\Delta T$.

A smaller channel hydraulic diameter and, in turn, a larger $\Delta p$, are the consequences of increasing the number of layers. On the other hand, at a given flow rate, increasing the number of layers results in a lower flow rate inside each channel. This is due to the division of the total flow rate amongst more channels. Therefore, despite an increased $\Delta p$ that leads to a larger $P_P$, lower flow rates potentially reduce the $P_P$. An interesting conclusion for the practical applications of CPs in this study is a considerable reduction of $T_{\max}$ and $\Delta T$ at a given $P_P$ by increasing the number of layers from one to two, but negligible reduction in the $T_{\max}$ and $\Delta T$ by adding the number of layers from two to three. The reason behind the large and small drop in $T_{\max}$ experienced by the CPs when increasing the number of channels to two and three, respectively, is due to the additional thermal resistances, as explained earlier. The results from Fig. 4 are particularly important in practical applications because implementing extra layers leads to a more complex and costly fabrication process. The findings from this work indicate that there is an optimum number of layers to achieve the highest heat transfer enhancement. Finding this optimum number of layers reduces the complexity of the fabrication process. Determining the optimum number of layers for given ranges of design parameters should be further investigated in future research.

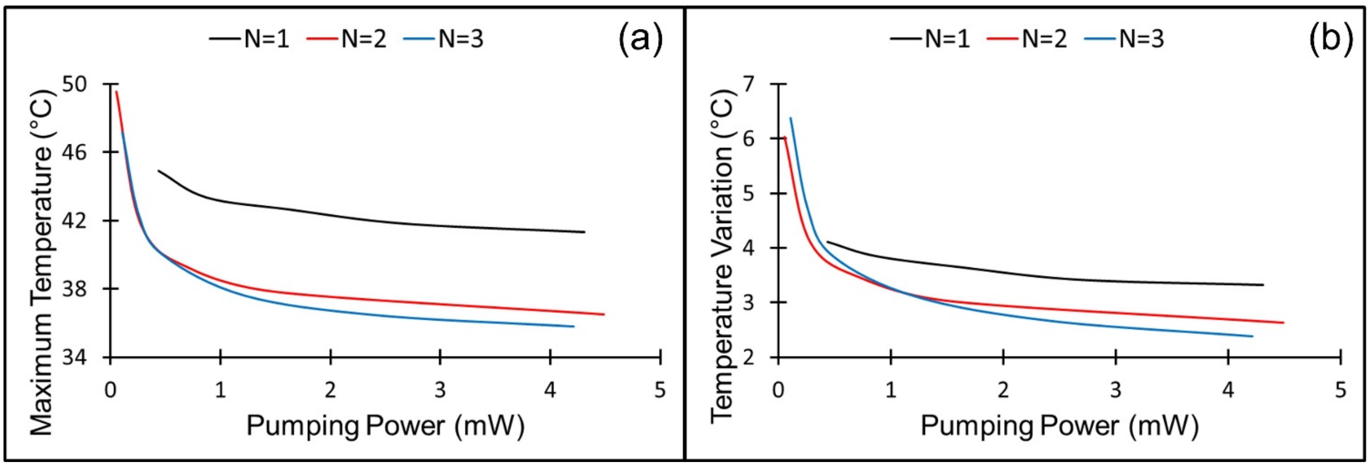

Fig. 4: (a) Maximum temperature, and (b) temperature variation over the heating surface at different pumping power.

## 4. Conclusion

Three-dimensional laminar flow and heat transfer in multi-layered CPs were investigated. It was found that an increased number of layers monotonically enhances the cooling performances at both a given flow rate and $P_P$. However, heat transfer enhancement is reduced when increasing the number of layers from two to three. The results indicate an existence of a threshold number of layers, in which beyond that threshold value, the heat transfer is not enhanced.


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