DESIGN AND ANALYSIS OF LIQUID COOLING PLATES FOR DIFFERENT FLOW CHANNEL CONFIGURATIONS

by

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A number of thermal management devices are used to actuate concentrated electronic appliances in an efficient way. A liquid cooling plate acts as a heat sink enclosed by materialized walls. This work aims to carry out design of liquid cooling plates such that the heat diffused by the electronic equipment is removed while their temperatures levels remain within safe limits. The liquid cooling plates expose "cold surfaces" to electronic appliances. The performance of a cooling plate is estimated depending upon heat carrying capacity, associated heat transfer rates and concentrated thermal regions on the plate surface. For this study, the design of liquid cooling plate was done with SOLIDWORKS. Pure water was used as a working fluid in test channels. A comparative analysis of flow distribution, temperature contours, pressure drop, and pumping power for different channel configurations was carried out with ANSYS. It was observed that a channel configuration is of key importance in liquid cooling plates. The findings from this study are beneficial for the optimum design of cooling systems for high heat flux applications, i.e., in electronic devices, computer processors and automotive engines.

Key words: *liquid cooling plate, channel configuration, flow distribution, heat transfer*

Introduction

Liquid cooling plates have vast applications in electronic devices, computer processors, automotive engines, *etc.* Amid dense processing units, temperature stability is of key importance in technological advancement. Multitask processors are widely used in this techno-era. Liquid cooling processes in the perspective of system space and working fluid may lead to an appropriate thermal environment.

A liquid cooling plate is used for the transmission of heat from hot surfaces of gadgets having a high load to the fluid circulating within a liquid plate system. A number of vari-

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ous heat sink plates have been introduced to manage high heat flux while considering the allowable temperature in electrical equipment [1]. The use of conventional air cooling systems is being outpaced to meet demands of modern high tech appliances [2, 3]. So, liquid cooling plates with micro-channel setups have been considered as a possible method to overcome high power concentrations [4] and hybrid micro-channel systems [5, 6] have been considered as a possible method to overcome high power concentrations. While micro-channel liquid plates may have relatively better thermal performance than their conventional counterparts, they do require high pumping power, and this is due to the enhanced compactness of the flow channels [7]. Consequently, liquid cooling plates having micro-channel setups should be designed carefully with reliable estimates of pressure drop to fulfill cooling demands of high heat flux applications.

High power electric equipment, fuel cell power bases and concentrated solar plates all require operational thermal stability to attain a harmless and better effective process. Heat bases containing comparative outsized smooth plate measurements stand commonly required for heat dissipation in fuel cell tools [8, 9]. In the latest practices of solar energy technology, concentrated solar panels also required flat plate dimensions [10, 11]. Presently, air convection heat bases used as a feasible way to thermally stabilize electric appliances, mainly addressing the little cost with great consistency [12]. Significant research on air-cooling heat exchangers is carried out in the last few years, and substantial perfections in heat exchanger models mainly accomplished centered with CFD studies [13, 14], corresponding with tentative evaluations [15, 16].

Owing to its low heat carrying capacity, convective cooling with air cannot fulfil the cooling demands of modern high heat-dissipating electronic devices [17]. Liquid cooling is thus becoming increasingly popular in improving the thermal efficiency of heat sinks [18, 19]. Liquid cooling is therefore, being considered a potential solution for concentrated solar panels and fuel cell stacks [10, 11]. Liquids generally have high thermal conductivity and heat capacity than commonly used gaseous mediums, and their utilization can eventually help sustain high heat flux with relatively low temperature lifts across the heat sink. Specific liquids (like dielectrics for electronics) are deployed to attain effective improvement in cooling heat sinks [20, 21].

Compact devices are now increasingly being employed in all spheres of life, and their utilization also helps in saving materials and charge inventories. Modern machining facilities now made it possible to machine fine compact channels for heat sinks. Heat sinks made with such compact channels ($d < 3$ mm), technically called minichannels, are supposed to have better thermal performance, therefore, they can withstand high heat fluxes. Micro-channel configurations of heat sinks are compatible for high heat flux transfer while considering the required performance of electric appliances [22]. Data centres of integrated artificial neural networks seek temperature optimization for best efficacy [23].

A remarkable research work has been carried out over the subject of cooling systems incorporated with micro-channels [24, 25], along with consideration of optimized characteristics of heat transfer. Collectively all these works pay inadequate attention regarding the issue of maldistribution of flow through channels in a flat liquid cooling system [26]. Different heat sinks were developed with micro-channels experiencing the parallel flow of fluid. These micro-channel heat sinks offer larger heat transfer area and also assist the fluid in making significant outer connections for effective flow distribution. A non-uniform distribution of flow drastically fluctuates the efficiency of the heat sink, causing significant pressure drop that end ups in increased pumping power. Flow maldistribution has been considered in studies with a focus on fluid mechanics [27, 28]. The subject of maldistribution in fluid mechanics has been addressed by considering different channel configuration, like *z*-type and *u*-type parallel channel configurations. However, eradicating non-uniformity of flow and accomplishing significant way for uniform distribution is still demanding further exploration. Topology orientation regarding heat sink optimization is being addressed as another factor of geometry that is effective in its layout approach [29].

Flow distribution is repeatedly a concern in heat transfer, and in this regard, a tree moulded pattern is examined in respective research [30, 31]. An analysis is carried out with assumptions of sub creeks in channels to optimize flow distribution and to improve the passage of flow in cascade regime [32]. However, the question is to model such a tree moulded pattern in the concerned grace. For uniform dispersal of specimen in fuel cells, a *T*-shaped configuration is introduced particularly [33]. Induction of *T*-shaped configuration theoretically enabled the sort to optimize heat transfer efficiency through designing of considered configuration in heat sinks.

The present work will sort out a distinctive design of liquid cooling plates with modified channel dimension for a conventional heat sink configuration. The CFD analysis was carried out for different channel configurations to identify the hotspot surfaces, flow distribution, pumping power consumption and pressure drop in a particular limit of boundary conditions for different heat sinks. The CAD model was prepared with SOLIDWOKS, whereas CFD analysis was done with ANSYS FLUENT.

Regarding correlations and governing equations, a considerable work has been presented. The common understanding lies in the range of millimeters while considering microchannel dimensions, to which particular correlation and governing equations are compatible [34, 35]. In this work, overall channel dimensions are not exceeding a few millimeters, and thus CFD analysis is used to determine the cooling plate's behavior.

Flow channel configurations

Modelling of cooling plates is carried out using SOLIDWORKS with four different configurations. Two channel configurations named Parallel-I and Parallel-II are designed that have aligned junction in a symmetric way with parallel channels as presented in figs. 1(a) and 1(b). These parallel configurations are actually effective in proper flow in horizontal and vertical directions. Another configuration has distribution through step level channels at inlet and outlet as shown in figs. 1(c) and 1(d). These steps are referred to as *bifurcation zone* a way to point out manifold slots [33]. Further, these step channels are characterized at both inlet and outlet. At the outlet, liquid carrying heat is filtered out. A considerable advantage of these types of designing containing parallel and transverse channel patterns is to achieve uniform flow throughout the fluid region. Tables 1-3 show the dimensions of liquid plates for all channel configurations.

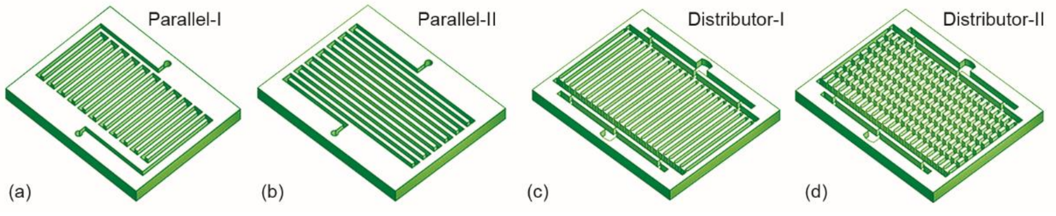

Figure 1. Designed heat sink channel configurations

The channels of the heat sink are modelled as with the material of aluminum. The liquid cooling plate is connected to heater to investigate the heat transfer effect. The conductive resistance of plate and of heater in combination with convective resistance define the overall heat transfer coefficient. The area of plate, except where the heater is attached, is considered as an adiabatic.

Table 1. Flow channel configuration of type Parallel-I

Parameters	Dimensions/Details
Diameter of inlet/outlet	2 mm
Width of channels	1 mm
Width of walls	1 mm
Depth of channels	1.5 mm
No. of channels	19

Table 2. Flow channel configuration of type Parallel-II

Parameters	Dimensions/Details
Diameter of inlet/outlet	2 mm
Width of channels	1 mm
Width of walls	1 mm
Depth of channels	1.5 mm
No. of channels	13

Table 3. Flow channel configurations of type Distributor-I and Distributor-II

Parameters	Length	Width
Main channel	39 mm	1.5 mm
1st level	47 mm	1.5 mm
Parallel channels (24)	47 mm	24 mm
Parallel walls (23)	47 mm	24 mm
Length of an element in Distributor 2	2 mm	–
Gap between elements in Distributor 2	1 mm	–
Diameter of inlet/outlet is 2 mm		
Depth of channels is 1.5 mm		

Conventional heat sinks, including both liquid and fluid zones, when analyzed with reference to examine their mesh statistics, they show some specific statistics near about 25000000. These standard estimations are evaluated through examining mesh statistics. Margining different level options of mesh quality are utilized to identify pressure variations and temperature uniformity at different regions on the cooling plate. Consequently, alteration of mesh statistics qualities with three different levels, such as coarse, medium and fin. Table 4 shows that meshing does have no significant impact on results. So, this work is carried out while focusing mesh size as *medium* for all type of heat sink channels.

Assessment of cooling plate

The liquid plate can be assessed using the thermal characterization of a flat surface and cooling medium. The cooling rate of the cooling medium is controlled to optimize the heat transfer rate from the liquid cooling plate. Another essential consideration is the cost of a liquid cooling plate, and it should be addressed, though it is not a part of this study.

In the quantitative evaluation of cooling performance, the deviation of temperature from mean to extreme value on plate surfaces is examined subject to pumping power availa-

Table 4. Mesh statistics (q = 200 W, Re_{in} = 400)

Flow channels	Mesh size	Elements	$\Delta P_{i\text{-}o}$ [Pa]	T_{max} [°C]
Parallel-I	Coarse Medium Fine	21500000 3360086 4100000	214205.7 227205.1 237205.1	34.8 34.6 34.5
Parallel-II	Coarse Medium Fine	2545514 3105072 4347101	154566 166582 175546	35.5 35.1 35.2

Figure 2. Meshed model of heat sink

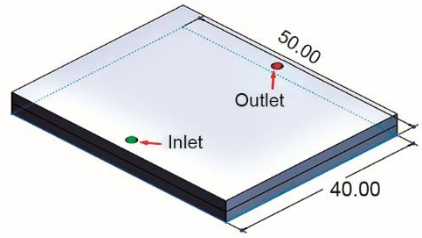

Figure 3. The 3-D sketch of the liquid cooling plate as computation domain

bility. The temperature difference on induction of heat flux can be assessed by relating the temperature difference of the plate surface to heat flux induced on the surface of the cooling plate [36], and is given in:

$$\theta_T = \frac{T_{max} - T_{min}}{q} \tag{1}$$

However, one may not merely use the temperature non-uniformity per unit heat flux to assess the distinction of a heat sink. An ordinary low temperature of the heating surface is also significant and should be supervised using:

$$R = \frac{\bar{T}_s - T_f^{in}}{q} \tag{2}$$

Thermal resistance, R, is correlated with temperature, as given in eq. (2). Heat transfer through the inner face of the cooling plate has a direct relation with thermal resistance, as a high heat transfer will result a low average plate surface temperature causing R to be minimum.

A meshed model of the heat sink is shown in fig. 2 while fig. 3 shows its computation domain.

A liquid cooling plate having efficient performance will have low values for both basic factors *i.e.* θ_T and R. So, this benchmark can be marked as for proficient liquid cooling plate. But, if a liquid cooling plate does not show such required behavior about θ_T and R then preference should be given to those heat base that fulfils the criterion. Similarly, liquid cooling plates with higher values of θ_T and R will indicate their poor performance and need for optimization.

In the sense of evaluating the performance of a liquid cooling plate, an important factor, *pumping power*, should be considered marginally. The consumption of pumping power can be described:

$$\dot{W}_{pump} = \Delta P \dot{V} \tag{3}$$

In this relation, ΔP is pressure drop across the liquid cooling plate, and volumetric flow is indicated by $\dot{V}$.

In the assessment of different flow channel configurations, this flow rate is observed that for dissimilar channels with the same working fluid may also fluctuate the pressure drop. Thus, performance evaluation of liquid cooling plates should be judged with considerations of θ_{T} and R. Conclusively, pressure drop and volumetric flow coherently define the pumping power consumption.

Numerical model

The ANSYS is used for analysis related to heat transfer and also activated for observation of different flow regimes. The software is used as tool for the numerical solution. Furthermore, some numerical points are also exemplified in this section.

The main governing equations are equations of continuity, conservation of momentum, and energy for different flow regimes. The flow is assumed as laminar, incompressible, steady-state with no radiation. The governing equations are:

$$\mathrm{div}\vec{\mathrm{V}} = 0 \tag{4}$$

$$\mathrm{div}(\rho_{\mathrm{f}} u_{\mathrm{j}} \vec{\mathrm{V}}) = -\frac{\partial P}{\partial x_j} + \mu_{\mathrm{f}} - \mathrm{div}(\mathrm{grad}\, u_j) \tag{5}$$

$$\rho C_P \mathrm{div}(\vec{\mathrm{V}} T) = K \,\mathrm{div}\; (\mathrm{grad} T) \tag{6}$$

where j indicates Cartesian directions x, y, z, respectively. The liquid cooling plate has a computational regime of both solid and fluid zones. Specifying solid zone, aluminum properties are endorsed, including its thermal conductivity. Here, for the solid zone, zero velocity along the walls is considered.

Fluid velocity is assumed at entry as the boundary condition. Reynolds number of fluid is:

$$\mathrm{Re}_{\mathrm{in}} = \frac{\rho_{\mathrm{f}} v_{\mathrm{in}} D_{\mathrm{h}}}{\mu_{\mathrm{f}}} \tag{7}$$

The fluid temperature at the inlet is 293.15 K. At the outlet, boundary conditions are applied by satisfying law of mass conservation $\dot{m}_{\mathrm{in}} = \dot{m}_{\mathrm{out}}$ [37, 38]. The liquid cooling plate from sides and top acts as an insulator thus adiabatically not exposed to heat flux. But the bottom surface act as a liquid cooling heat sink and thus, continual heat flux is attached with it.

A 200 W heat is the heat supplied for the assessment of cooling plate performance while marking the maximum allowable temperature as 70 °C. Consequently, examined that no hot spots arose even at above 70 °C. Thus, respective liquid cooling plates are even efficient for high processors that require an appropriate temperature for their efficient working.

Numerical methods

The ANSYS FLUENT®, is used in this evaluation. The flow regime is conducted as laminar. Firstly, a scheme of first order is employed then on later a 2$^{\mathrm{nd}}$ order upwind scheme is commanded to compute convergence. To link coherence of velocity and pressure, a simple algorithm is assigned-an appropriate value of different factors assigned to compute and find the convergence of temperature and velocity at exit.

Residual volumes restricted ranging in $1.0 \cdot 10^{-9}$ for mass conservation. For conclusive computations, residual volumes restricted to $1.0 \cdot 10^{-12}$.

Results and discussion

The thermal behavior of fluid at the outlet is assessed numerically, and the law of energy balance is also satisfied. This energy balance evaluation is:

$$\dot{Q} = \dot{m}_\mathrm{f} Cp_\mathrm{f} \Delta T_\mathrm{f} \tag{8}$$

An appropriate flow distribution across the liquid cooling plate is mainly investigated for assessing the heat transfer efficiency. An insufficient fluid flow can be the cause of hot spots on cooling plate surface. A quantitative flow distribution should be addressed by specific parameters as examined [39]. The velocity may fluctuate drastically in the channels of liquid cooling plate. The limitations are with the lowest, average and highest average velocity of channels are studied in detail.

Flow, temperature, and pressure distributions in fluid regions

Figures 4-6 show quantitative analysis of flow, pressure, and temperature distributions with different types of configuration, and it is found that flow in channel configuration of type Distributor-I and Distributor-II showed a better performance in terms of flow distribution at a given velocity.

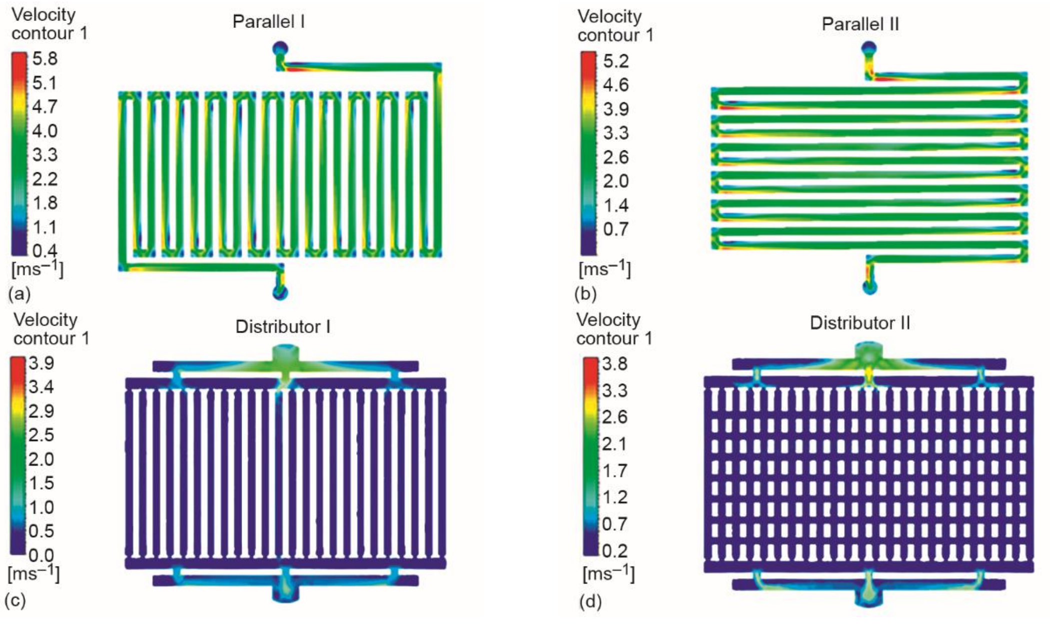

Figure 4. Channels' average velocities in different flow configurations
(*for color image see journal web site*)

Similarly, the Parallel-I and Parallel-II are analyzed and compared with Distributor-I and II patterns. Among the Parallel-I and Parallel-II configuration, Parallel-II showed more uniform distribution of flow at a given Reynolds number, as shown in fig. 4. Conclusively, this can be pointed out that a uniform flow distribution will lead to a uniform temperature distribution and hence, causing a better heat transfer.

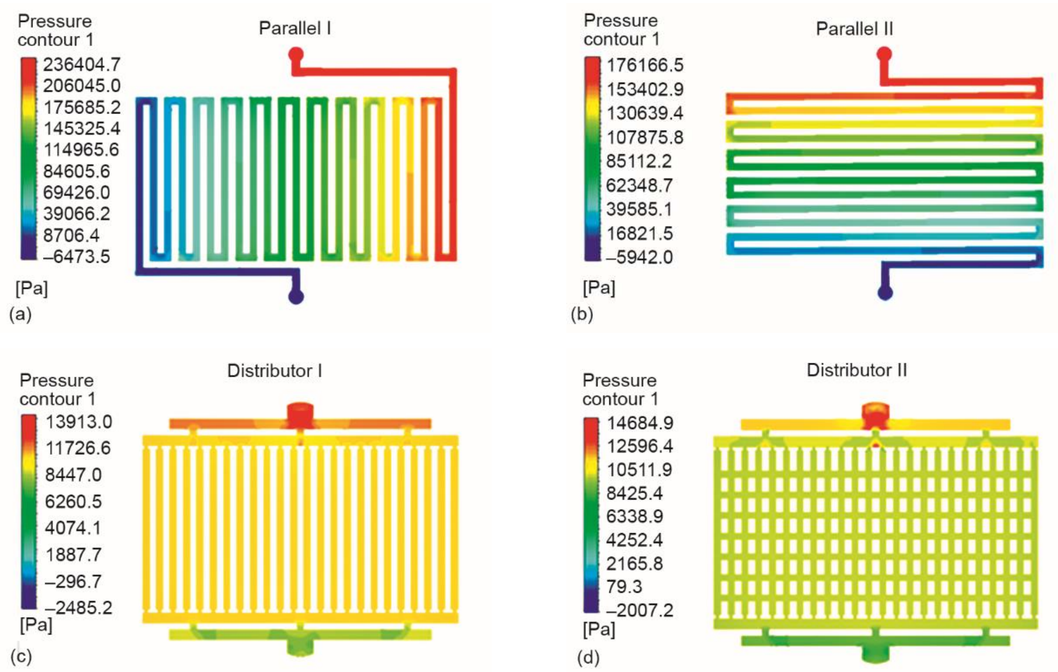

Figure 5. Pressure distribution of fluid in different flow configurations at 3 m/s
(*for color image see journal web site*)

Figure 6 shows the temperature of the cooling medium with different flow configurations. As the distribution of fluid is better in Distributor-I, a high amount of heat is transferred from hot surface to the liquid causing high temperature, hence causing a better heat transfer.

Temperature distribution on heat sink surface

The temperature fluctuations of the cooling medium caused the hot spots on the heat sink surfaces. Figure 7 presents temperature contours on the heat sink surface for different configurations. The temperature contours are displayed for all type of flow channel configurations at applied heat of 200 W at different flow velocities.

It is observed that for each type of flow channel configuration, the temperature contours are of different trend depending on flow distribution. Distributor-I shows a maximum temperature hot spot as it has the tendency to remove maximum heat due to the better flow rate of the fluid. In flow channel configuration Distributor-II, thermal lines pattern is also uniform as compared to other flow channel configurations Parallel-I and Parallel-II, which leads to relatively higher heat transfer. In the case of Parallel-I and Parallel-II, the temperature distribution is strongly related to flow patterns in the horizontal and vertical channels.

Overall, the configuration type Distributor-I and Distributor-II showed an effective temperature and flow distribution irrespective to Reynolds number and variation in heat flux. These configurations accomplished almost a uniform temperature distribution. When comparing the results with different flow velocities, it is observed that temperature distribution on the surface of the cooling plate depends on the flow rate of the cooling medium. The uniformity of temperature distribution is more obvious at low Reynolds number, as shown in fig. 7.

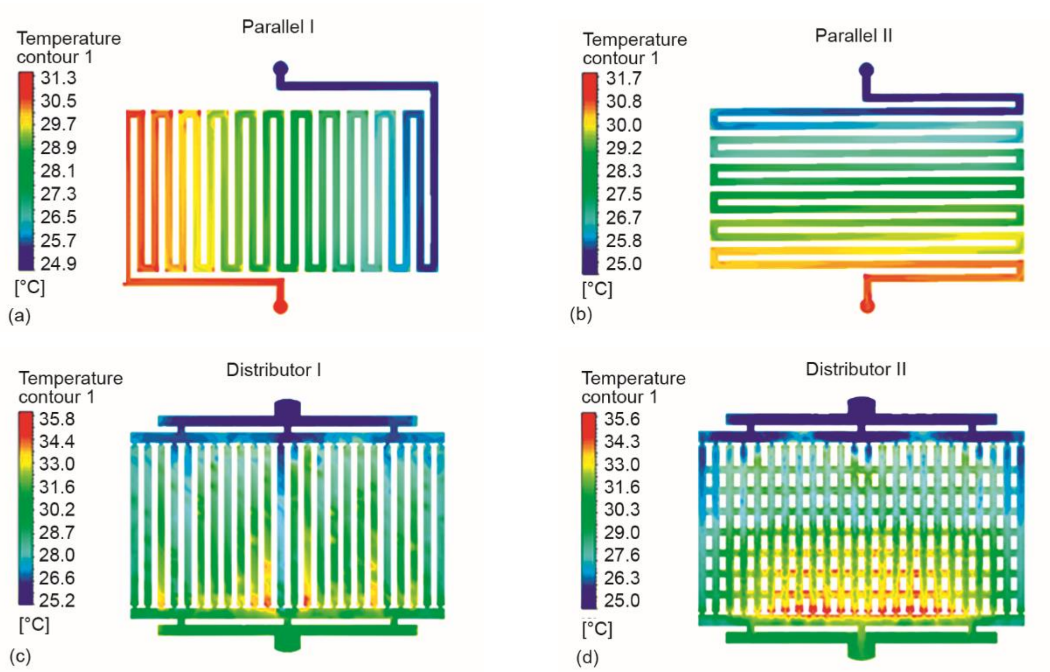

Figure 6. Temperature distribution of fluid in different flow configurations at 3 m/s
(*for color image see journal web site*)

Effect of flow rate on pressure drop and fluid temperature

For the comparison of pressure drop and temperature variations, comparative plots of all flow channel configurations are presented in figs. 8 and 9. The graphs relate the flow rate of fluid to pressure drop and maximum temperature of the fluid. Figure 8 shows that the maximum temperature of fluid decreases with the increasing value of flow rate for all the cooling plate configurations. Distributor-I revealed an extreme temperature as compared to other configurations due to the better flow rate through channels. Distributor-I has shown a better heat dissipation capacity. As θ_T and R are mainly function of surface temperatures and fluid temperature, therefore, it can be deduced that a less thermal resistance occurs in case of Distributor-I with a uniform temperature distribution.

Pressure drop increases with increasing value of flow rate, and system pressure increases which demand more pumping power to run the system. The flow distribution in Distributor-I and Distributor-II configurations are better than parallel configuration. These results in less pressure drop against varying values of flow rates of fluid. Figure 9 presents the highest pressure that occurs in Parallel-I configurations.

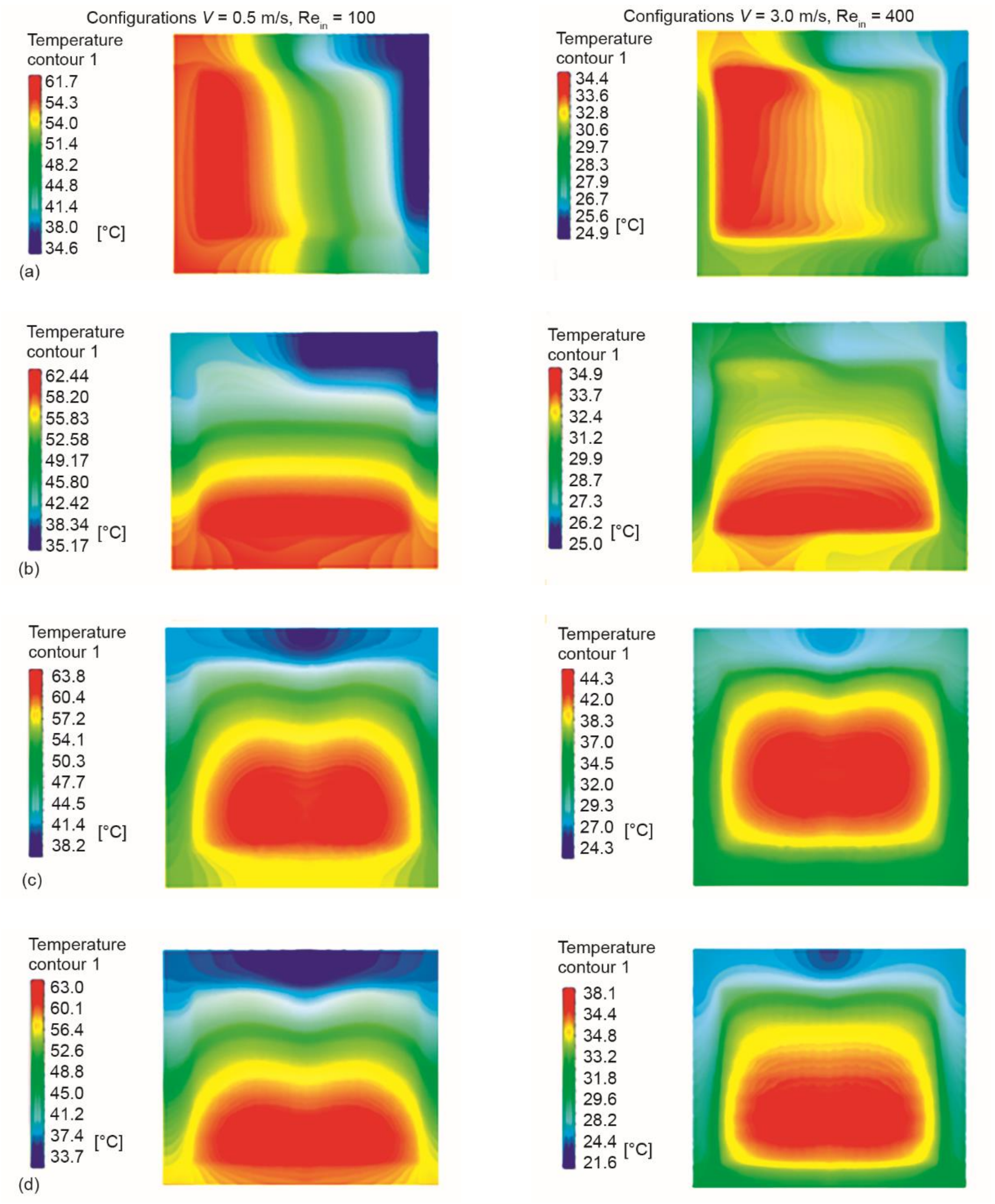

Figure 7. Temperature contours of hot surface of liquid cooling plate at different Reynolds numbers; (a) parallel-I, (b) parallel-II, (c) distributor-I, and (d) distributor-II (*for color image see journal web site*)

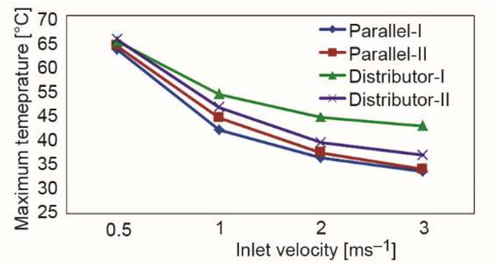

Figure 8. Maximum temperature for different channel configurations against velocity of fluid

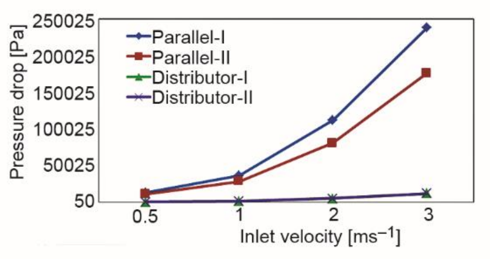

Figure 9. Pressure drop for different channel configurations against velocity of fluid

Conclusions

This work focuses on an appropriate design of a liquid cooling plate by evaluating different factors such as flow distribution, thermal hot spots on the plate surface, uniformity of temperature contours and pressure drop. The designing and assessment of cooling plate performance were related to heat supplied, heat transfer rate and hot spots on the plate surface.

A liquid cooling plate is optimized using different flow configurations and mass--flow rate. Different flow channel configurations have caused significantly different pressure drop with an inappropriate hot spot on plate surfaces. As compared to other configurations, Distributor-I channel configuration is considered as the most efficient design which spotted uniform flow distribution and regular hot spots location on flat plate surface corresponding to various mass flow rates at a given heat supplied. It is also observed that pressure drop across Distributor-I configurations is not very high, thus required very small pumping power input for flowing the fluid.

Acknowledgment

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Nomenclature

C_p – specific heat, [Jkg^{-1}K^{-1}]
K – thermal conductivity, [Wm^{-1}K^{-1}]
$\dot{m}$ – mass-flow rate, [kgs^{-1}]
P – pressure, [Pa]
$\dot{Q}$ – heat rate, [Ws^{-1}]
q – heat, [W]
Re – Reynolds number
T – temperature, [K]
$\vec{V}$ – velocity vector, [ms^{-1}]
$\dot{V}$ – volumetric flow rate, [m^{3}s^{-1}]
$\dot{W}$ – pumping power, [W]

Greek symbol

μ – dynamic viscosity, [kgm^{-1}s^{-1}]

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