Experimental Thermal and Fluid Science 67 (2015) 81–87

Contents lists available at ScienceDirect

Experimental Thermal and Fluid Science

journal homepage: www.elsevier.com/locate/etfs

Nozzle to plate optimization of the jet impingement inlet of a tailored-width microchannel heat exchanger

J. Barrau [a,*], S. Riera [a], E. Léveillé [b], L.G. Fréchette [b], J.I. Rosell [a]

[a] University of Lleida, Solar Energy Group, Edifici CREA, C/Pere de Cabrera, s/n, 25001 Lleida, Spain
[b] Université de Sherbrooke, Dépt Génie Mécanique, 2500 Boul. Université, QC J1K 2R1, Canada

ARTICLE INFO

Article history:
Received 31 July 2014
Received in revised form 17 November 2014
Accepted 22 November 2014
Available online 28 November 2014

Keywords:
Heat sink
Cooling device
Jet impingement
Nozzle geometry
Microchannel
Temperature uniformity

ABSTRACT

This work presents an experimental study on the effect of the nozzle geometry on the performance of a hybrid jet-impingement/microchannels cooling system. The proposed heat sink is designed so as to maintain the temperature of the cooled surface uniform, even with the coolant flow temperature increasing along the flow path. Previous results showed quite uniform temperature distributions with the exception of the zone located just below the jet impingement, where the temperature is higher. In the present work, several nozzle to plate spacings (z/b) are experimentally studied. The impact of nozzle to plate spacing on the stagnation point Nusselt number and the overall temperature distribution varies as a function of the coolant flow rate. A strong coupling between the slot jet geometry, the heat exchange in the varying width microchannel sections and the flow regime is demonstrated. The global thermal resistance, the temperature uniformity and the pressure drops show distinct behavior as a function of the nozzle to plate spacing, implying that the design procedure may weight up these parameters according to the cooling needs.

1. Introduction

The heat flux densities managed in microelectronics are rising rapidly. On the one hand it is necessary, for the proper working system of these devices, to make an active cooling of the elements that generates heat to reduce their temperature up to acceptable ranges. On the other hand, the temperature uniformity of the cooled object must be improved. In effect, this parameter affects the performance of electrical systems and reduces their reliability. Heat sinks currently achieve the first objective, but they do not offer definitive solution to the second one.

Jet impingement and microchannel are the most common cooling schemes used in microelectronics [1]. They both present high heat flux extraction capacities but still have serious drawbacks. Microchannels can only minimize the increase of temperature of the object to cool in the direction of the fluid flow by increasing the flow of refrigerant – a fact which entails the increase of the power of the circulation pump and, therefore, of the total system cost – but cannot eliminate this temperature gradient. For jet impingements, the only way to obtain a quite uniform temperature distribution is the use of matrix of jets, which implies the reduction of the heat exchange coefficients and a high complexity of the return architectures [2].

Along these lines, Barrau et al. [3] developed a hybrid jet-impingement/microchannels cooling device that improves the temperature uniformity of the cooled object. A stepwise varying width design counteracts the increase of the water temperature by increasing both the local heat transfer coefficient and the heat exchange area to maintain constant the heat flux removal capacity. Furthermore, numerical studies made on this cooling device [4] and other studies related to similar geometries [5,6] showed that this cooling scheme causes lower pressure drops than conventional microchannels. This characteristic is a key factor for the sizing of the cooling system, as the price of the pumps depends mainly of the pressure drop that they may overcome [7].

This cooling scheme has also been implemented at microscale [8,9]. The results demonstrate the validity of the proposed design, but also show that the highest temperature of the cooled object is located just below the jet impingement.

Many research works were dedicated to study the influence of the ratio of nozzle hydraulic diameter (d) to its spacing from the target surface (z) on the local heat transfer distribution. O'Donovan and Murray [10] showed that local distribution of the heat exchange varies in hugely as a function of the nozzle to plate spacing (z/d) and the Reynolds number. For low the nozzle to plate

* Corresponding author.
E-mail addresses: jerome.barrau@udl.cat (J. Barrau), sriera@macs.udl.cat (S. Riera), etienne.leveille@usherbrooke.ca (E. Léveillé), Luc.Frechette@usherbrooke.ca (L.G. Fréchette), rosell@macs.udl.cat (J.I. Rosell).

http://dx.doi.org/10.1016/j.expthermflusci.2014.11.012
0894-1777/

spacings ($z/d < 4$), the mean heat transfer distribution in the radial direction exhibits secondary peaks. Katti and Prabhu [11] confirmed these results and also stated that, for a given Reynolds number, the stagnation point Nusselt number increases with augments in z/d from 1.0 till around $z/d = 6.0$.

In the case of slot jets, the nozzle to plate spacing is expressed by the ratio z/b (b is the width of the slot jet). Previous studies on slot jet impingement cooling schemes [12,13] showed that the nozzle to plate spacing has a varying impact on the stagnation point Nusselt number as a function of the Reynolds number. Zukowski [14] also showed that, at relatively low Reynolds numbers ($Re < 2000$), the local Nusselt distribution (Nu_x) trend do not show huge differences for the nozzle to plate spacings studied. Nevertheless, the absolute Nu_x values and the influence zone of the slot jet impingement vary largely.

The purpose of the present work is to experimentally assess the impact of the nozzle to plate spacing (z/b) on the heat exchange performance of a hybrid jet impingement/microchannel cooling scheme. The sensitivities of the temperature distribution of the whole cooling device and the stagnation point Nusselt number to the nozzle to plate spacing are analyzed. Finally, the impact of this parameter on the global performance of the heat sink is studied, through the analysis of the thermal resistance coefficient, the temperature uniformity and the pressure drop.

2. Experimental mount

2.1. Test module description

The test module setup, with the coolant circuit and the equipments used for the measurements is represented in Fig. 1.

The coolant (water) is stored in a reservoir with a thermostatic bath to maintain a constant inlet temperature ($T_{in} = 20$ °C). The water circulates in the loop with the aid of a variable speed peristaltic pump (JP Selecta PERCOM N-M). As the pressure losses of the whole coolant circuit are large, the pump provides a constant flow rate (Q). This characteristic has been validated trough the use of an instantaneous flow meter. Then, the liquid passes through a 5-μm filter before entering the test module.

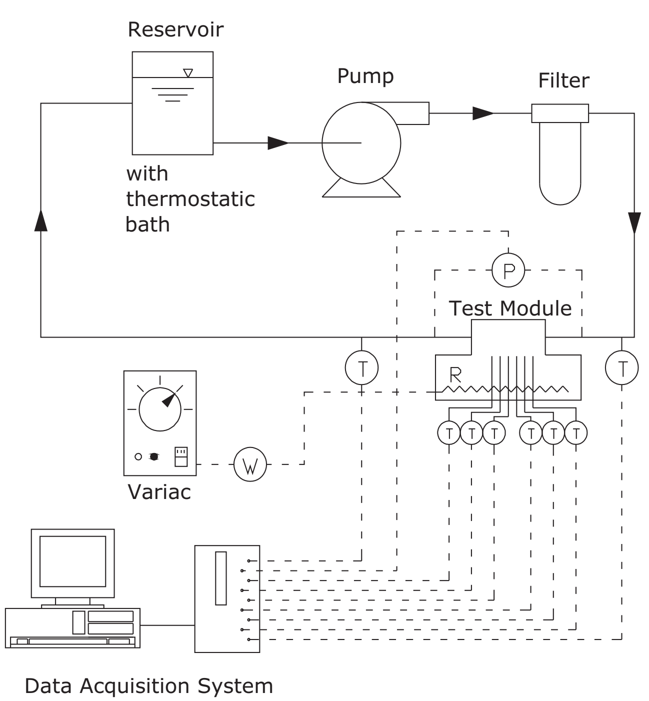

Fig. 1. Test module setup.

In this study, the heat flux (q''), generated by an advanced ceramic heater (Watlow Ultramic 600), is set to 50 W/cm^2.

The schematic design of the experimental test module used for this study is shown in Fig. 2.

The fluid is introduced to the heat sink through the central slot of the slot plate, located below the entry volume of the coolant distributor. After impacting on the middle zone of the device, the water flow is divided in two and passes through the stepwise-varying width microchannel sections until the two outlets of the heat sink. The lower face of the slot plate seals the top of the microchannels.

Type-K thermocouples are used to measure the water temperature at the inlet and outlet of the cooling device and the temperature distribution of the copper layer along the flow path, at different positions (x) along the centerline ($y = 0$ mm) of the cooling device. The pressure drop is also measured between the inlet and outlets of the coolant distributor. All this information is acquired by a datalogger (Campbell CR23X) and sent to a computer to be stored and analyzed.

The cooling device entails two well differentiated areas. The first corresponds to a slot jet impingement, while the second consists in a series of microchannel sections, with different widths (Fig. 3).

The jet impingement slot width can be modified from 50 μm to 1.5 mm. This is achieved by interchangeable nozzle plates to be sited just under the coolant distributor original slot. The widths of the microchannel sections decrease from 1.53 mm to 140 μm in order to increase, along the flow path, both the convective heat transfer coefficient and the heat exchange surface and, thereby, compensate the increase of the flow temperature (Fig. 4).

The depth of microchannel pattern, which is also the spacing between the bottom of the inlet and the target surface, is around $z = 300 \pm 20$ μm.

The symmetry plane ($x = 0$ mm, y, z) allows to reduce the temperature measurements to half of the cooling device length ($x \geqslant 0$ mm).

The microchannel heat exchanger is sealed on the copper layer of the test module through a thin layer of Thermal Interface Material (TIM). The thickness of the TIM layer has been measured with a microscope (100 μm, ±5 μm) and its thermal conductivity has been verified ($\lambda_{TIM} = 2.22$ W/m K).

The coolant (water) is pumped to the inlet, implemented through a slot jet impingement. The flow is then divided in two, following the microchannels path to both ends of the module. Two outlets collect the coolant after passing through the microchannels.

Type-K thermocouples are used to measure the water temperature at the inlet and outlet of the microchannel heat exchanger and the temperature distribution of the copper layer along the flow path, at different positions (T_x) of the centerline ($y = 0$ mm, $z = -1.3$ mm) of the microchannel heat exchanger. All this information is acquired by a datalogger Campbell CR23X and sent to a computer to be stored and analyzed. Finally the flow is channeled again to the thermostatic bath, closing the loop.

2.2. Fabrication

Most parts of the experimental setup were made using conventional machining processes.

The microchannels of the heat exchanger were etched into a generic 550 μm thick silicon wafer using the Deep Reactive-Ion Etching (DRIE) process. The etched pattern has been defined by a photolithographic process using AZ9245 photoresist and a low cost laser printed acetate mask.

The accuracy of the fabrication process has been verified by checking the test section widths and the depth of the microchannels through microscopy images (Fig. 5).