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Research paper

Performance improvement of high power liquid-cooled heat sink via non-uniform metal foam arrangement

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HIGHLIGHTS

- A novel concept for liquid cooling by employing non-uniform arrangements of metal foams is presented.
- Seven arrangements by varying pore size density (PPI) of metal foam alongside heat sink are examined.
- The non-linearity term in the Forchheimer equation rises significantly with the PPI.
- The non-uniform metal foam appreciably outperforms conventional multiple channel heat sink.
- The best arrangement is to place the largest PPI first, followed by smaller and least PPI metal foam.

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ABSTRACT

The present study proposes a novel concept by employing non-uniform arrangements of metal foams alongside the heat sink applicable for liquid cooling. A total of 7 arrangements by varying PPI (pores per inch) of metal foam are made and additional 4 multiple channel designs are fabricated for comparison. Test results indicate that non-linearity term in Forchheimer equation rises significantly with increase of PPI (or with the decrease of permeability). The results suggest that the transition from the linear Darcy regime to the quadratic Forchheimer regime depends on the metal foam structure. It is also found that metal foam appreciably outperforms conventional multiple channel heat sink. The thermal resistance can be reduced by more than 62% as compared to that of empty plate design. For a given pumping power, the non-uniform arrangement of metal foam shows the best performance by employing a larger PPI metal foam at the inlet, followed by a smaller PPI metal foam, and the least PPI metal foam should be placed at the rear part of the heat sink.

1. Introduction

The microelectronic industry has continued to follow the Moore's law of increasing transistor density on a single chip [1]. Hence, the heat dissipation of a typical chip rises dramatically since more and more transistors are being packed in a confined area. As a consequence, thermal management exploiting conventional air-cooling becomes insufficient to handle the increasing heat dissipation of the current and future microprocessors. In this regard, liquid cooling is recognized as an effective alternative especially for the high flux cooling applications. However, there are two major issues in association with liquid-cooled heat sink. The first is related to the coolant distribution, and it may become more conspicuous when employs a multiple channel design. The associated influence of maldistribution had been investigated and reported considerably, i.e. Chein and Chen [2], Lu and Wang [3], and Cho et al. [4,5].

Secondly, the heat sources on the electronic devices may be uneven distributed in practice and yield additional spreading resistance. For instance, the high power IGBT which switches electric power in many modern appliances such as variable-frequency drives (VFDs) and electric cars. The presence of non-uniform heating may substantially raise the temperature for its gigantic spreading resistance. To ease the problems caused by non-uniform heat sources and maldistribution, employing metal foam heat sinks design is especially promising for it provides effective heat removal within a limited space, and can well disperse the liquid in short distance. Hence using metal foam in liquid cooling

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had attracted much attention. Typical liquid cooling using metal foams were conducted by Hung et al. [6], Boomsma et al. [7,8], Wan et al. [9], Chein et al. [10], Singh et al. [11], Zehforoosh and Hossainpour [12], and Jeng et al. [13]. These studies examined the fluid flow and heat transfer characteristics of a porous heat sink either numerically or numerically. Their results showed that metal foam for liquid cooling is quite effective but it may suffer considerable pressure drop penalty in the same time. To tailor this shortcoming, the present study proposes a non-uniform metal foam arrangement through which the corresponding pumping power is appreciably reduced at no additional cost of deteriorating the heat transfer performance.

2. Experimental setup and test sections

The experimental apparatus and measurement equipment are similar to a previous study [14]. Those interested readers can refer to this study for further details. The photos of the test samples of the copper metal foam include three different pore density (10, 20, and 30 PPI) are shown in Fig. 1(a). A rectangular housing with internal dimension of 193 mm (L) × 52.8 mm (W) × 8 mm (H) is used to accommodate the copper foams whit dimension being 52 mm (L) × 52.8 mm (W) × 8 mm (H). Tests were performed with three individual metal foams which were placed consecutively alongside the rectangular housing with a 10 mm spacing. A total of 7 arrangements of metal foam in the housing were made, including (10, 10, 10), (20, 20, 20), (30, 30, 30), (30, 20, 10), (30, 10, 10), (10, 20, 30), and (20, 10, 10). For comparison purpose, additional 4 multiple channel designs were fabricated, including the conventional multiple channels (Fig. 1(b)), interrupted channels (Fig. 1(c)), block channels (Fig. 1(d)) and trapezoidal channels (Fig. 1(e)). The corresponding dimension of the multiple channel designs is also depicted in Fig. 1(b)–(e). Heat is supplied from top of rectangular housing. To simulate the concentrate heat sources of the IGBT board, eight tubular electric heaters were embedded in copper block as shown in Fig. 2(a). The generated heat then penetrated uniformly through eighteen rectangular bumps as shown in Fig. 2(b) to simulate the concentrated heat source from the IGBT board. Power supplies are used to control power input into eight tubular electric heaters with a maximum power of 4000 W. The actual power into the heat sink is calculated from the measured temperature difference:

$$Q = \dot{m}C_p(T_{out} - T_{in}) \tag{1}$$

For a more elaborate estimation of the performance of the tested heat sinks, the required pumping power may be more appropriate which is calculated as follows:

$$\dot{W} = \dot{V}\Delta P \tag{2}$$

The heat transfer performance of the heat sink is termed as thermal resistance. Normally the thermal resistance is defined as $\Delta T/Q$ where ΔT is the temperature difference between wall surface and the coolant. Yet the average temperature difference is adopted. However, from the viewpoint of electronic cooling, the maximum temperature may be more suitable than average temperature. Hence the thermal resistance in this study is defined as,

$$R = \frac{T_M - T_m}{Q} \tag{3}$$

The relevant uncertainties of the derived quantities, based on the single sample analysis [15], for the heat transfer rate, pumping power, and thermal resistance are in the ranges of 2.8%–5.1%, 0.5%–4.3%, and 3.2%–6.7%, respectively.

3. Results and discussion

Based on the Forchheimer's equation, the pressure drop for porous media has a quadratic relationship with the inlet velocity, i.e.,

$$\frac{\Delta P}{L_c} = \frac{\mu}{K}u + \rho C u^2 \tag{4}$$

where K is the permeability, u is the average velocity, and C is the form coefficient, and these values can be obtained from a separate isothermal test and the results are tabulated in Table 1. As clearly seen from the Table, the form coefficient which represents the non-linearity term in Eq. (4) rises significantly from 173 (10 PPI) to 569 (30 PPI). Correspondingly, the permeability also decreases appreciably when PPI is increased. The results suggest that the transition from the linear Darcy regime to the quadratic Forchheimer regime depends on the permeability and the metal foam structure. For the 10 PPI structure, the linear Darcy regime prevails when the water inlet velocity is less than 0.4 m s^{-1} but the linear regime is appreciably reduced to a lower velocity of approximately 0.2 m s^{-1} for the 30 PPI structure. The results suggest the influential role of permeability on the transition from Darcy to Forchheimer regime.

The associated maximum thermal resistances of the test metal foams subject to various arrangements are shown in Fig. 3. Tests are conducted with three separate metal foams and the inlet water temperature is fixed at 55 °C. The copper foams are placed consecutively alongside the rectangular housing with a supplied power of 3300 W. As aforementioned previously, seven distinctive metal foam arrangements are tested and compared. Note that the ordinate of Fig. 3(b) is the pumping power. For a given pumping power, the 30-20-10 PPI arrangement shows the lowest thermal resistance whereas the 30 PPI (30-30-30) configuration depicts the worst performance at a given pumping power. It is interesting to know that reversing the best arrangement (30-20-10) into 10-20-30 configuration brings about a rather low performance (comparable to 30-30-30). For instance, at a pumping power of 0.5 W, the thermal resistance of 10-20-30 arrangement exceeds that of 30-20-10 arrangement by more than 15%. The results indicate that the dispersion effect of higher porous metal foam structure may play some essential role in the cooling process. By placing the higher porous metal foam (30 PPI) at the entrance part, the liquid flow is comparatively well dispersed when the water enters the cold plate and yields a better uniform flow distribution accordingly. Conversely, placing a higher PPI metal foam at the rear part of the heat sink is less effective as that of at the entrance. This is because the water flow in the metal foam eventually needs to merge together before leaving the heat sink. Metal foam with good dispersion may impair the re-merge of the liquid flow. Hence it would be appropriate to use a lower PPI (10 PPI) metal foam at the rear part of the heat sink to ease the pressure drop while still retains the heat transfer performance.

Note that it is also interesting that the 20-10-10 arrangement moderately outperforms 30-10-10 arrangement by about 5%. The results imply that dispersion is not the sole concern when employing the non-uniform concept of metal foams. This is because more porous structure often accompanies with a higher pressure drop, yet this phenomenon becomes more pronounced with PPI. As aforementioned in Table 1, the corresponding form coefficient for 10, 20, and 30 PPI is 173, 213, and 569, respectively. This suggests an enormous pressure drop may develop with rising PPI as clearly seen from Fig. 3(a) where the relative pressure drops for 30 PPI to 20 PPI is much larger than that of 20 PPI to 10 PPI. On the other hand, the dispersion effect by 20 PPI is comparable to that of 30 PPI. In fact, at the same Reynolds number, the effective heat transfer