Numerical and experimental study of cellular structures as a heat dissipation media

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Abstract
High heat flux generation in electronic devices demands new modes, methods and structures to dissipate heat effectively. We investigate the thermal performance of cellular structures using computational fluid dynamics (CFD) and obtained an optimal cellular structure for effective heat dissipation. Then, we validate our numerical results with experimental results obtained using optimized cellular structure. We found the minimum base temperature for the optimized cellular structure to be 43.6 °C and 47.4 °C numerically and experimentally respectively at inlet velocity of 10 m/s. We carried out experiments and simulations at the heat flux of 35,503 W/m². We found a close agreement between numerical and experimental results with an error of 8.71% for the base temperature. Previously the best base temperatures were reported to be 55 °C and 40.5 °C using air and water respectively [1, 2].

Nomenclature

\[ A_b \] Frontal blocked area (m²)

\[ A_t \] Frontal total area (m²)

\[ \alpha_{sf} \] Surface area density (1/m)

\[ A_s \] Surface area (m²)

\[ b \] Length of square side (mm)

\[ \epsilon \] Porosity

\[ c \] Centre to centre distance (mm)

\[ \rho \] Density (kg/m³)

\[ C_p \] Specific heat (kJ/kgK)

\[ \eta \] Thermal management index

\[ d \] Bore diameter (mm)

\[ \mu \] Viscosity (kg/ms)

\[ d_h \] Hole diameter (mm)

\[ \mu_t \] Turbulence viscosity (kg/ms)

\[ H \] Height (mm)

\[ \lambda \] Thermal conductivity (W/m°C)

\[ K_{cell} \] Pressure loss coefficient

\[ \Delta P \] Pressure Difference (Pa).

\[ L \] Length (mm)

\[ p \] Pressure (Pa)

\[ m \] Mass flow rate (kg/s).

\[ R_{BR} \] Blockage ratio

\[ R_{OPEN} \] Open area ratio

\[ \Delta T \] Temperature (°C)

\[ T_{b} \] Base temperature (°C)

\[ T_{i} \] Inlet Temperature of air (°C)

\[ T_{o} \] Outlet Temperature of air (°C)

\[ t \] Wall thickness (mm)

\[ u \] Velocity in x (m/s)

\[ v_{i} \] Inlet velocity (m/s)

\[ v_{o} \] Outlet velocity (m/s)

\[ V \] Volume of the structure (m³)

\[ V_{b} \] Total volume of whole solid block (m³)

\[ W \] Width (mm)

1 Introduction
Electronic devices have become very essential part in our daily lives. The continuous generation of heat flux in the electronic devices should be removed continuously to
maintain the base temperature of these devices in a safe zone. With rapid increase in computing speed and reduction in chip sizes, the pin fin array heat exchange method is becoming insufficient to meet the high heat transfer requirements in electronic devices. Many researchers increased the heat transfer rate by increasing the surface to volume ratio of heat sink devices. This method provides higher surface area densities as compared to the conventional methods, which allows a dynamic mixing between the flowing fluid and the internal structure. As a result, the convection heat transfer between the solid and fluid is enhanced.

Various new techniques are under discussion for effective thermal management and to maintain the base temperature of the computer processors in a safe zone which is between 60° and 80 °C [3–5]. It was reported that the heat dissipation of desktop and mobile processors were 100 W and 30 W respectively [6]. The thermal performance of air cooled mini channel heat sinks with different configurations were investigated experimentally. It was observed that the heat transfer from a solid to air increased with increase in flow rate and base temperature. The base temperature was reported to be 55 °C at the inlet velocity of 11.1 m/s and heat generation of 160 W [1].

Cellular metal structures are ultra-light materials with high porosities. In past few years they have been emerged as a compact heat exchanger to dissipate heat in small spaces [7, 8]. It is evident from previous research work that the heat transfer performance of aluminium foam heat sinks is better than that of conventional pin finned and parallel plate heat sinks [9–13]. Cellular metallic foams with stochastic structure are used as a compact heat exchanger. Metal foam has the ability to endorse the eddies and mixing of the coolant fluid efficiently which dissipates the heat five times faster than the conventional pin fin array method and have three time lesser weight [14–19]. Metal foam offers unique thermal, mechanical and electrical properties at relatively low density. Several heat transfer characteristics of metal foams have been investigated including the effect of microstructure properties such as porosity, relative density, pore density, pore size, ligament diameter, and permeability [20–23].

Overall reliability and operating speed of the modern high-speed computers depends not only on the base temperature but also on the temperature uniformity over the surface. Cellular periodic structures are preferred over stochastic structures (metal foams) because they provide a uniform temperature distribution over the surface. Stochastic cellular structures are good in dissipating heat as compared to periodic cellular structures, but their load bearing capability is much inferior than periodic cellular structure [13, 24]. Xu et al. optimize the mini channel for the uniform temperature distribution over the surface [25]. It has been observed that the thermal efficiency of the brazed textile structures (cellular periodic structure) for forced air convection is approximately three times larger than that of open-celled metal foams (cellular stochastic structure). Metal foams offer higher surface to volume ratio and higher pressure drop as compared to periodic textile structure. In case of cellular periodic structure, heat conduction also plays an important role along with convection in dissipating the heat. A good heat exchanger dissipates the heat flux at higher rate with a very low pumping power which is required to overcome the pressure drop to move the coolant through the heat exchanger. Thermal management of any heat sink/heat exchanger can be estimated from the ratio of heat transfer rate to pressure drop [15]. Tang et al. investigate a novel structure to reduce pressure drop with uniform temperature distribution [26].

Recently, experiments have been performed to cool the microprocessor temperature using nano-fluids [3, 27] and stability of nano-fluids has been discussed [3]. The largest temperature drop have been observed from 49.4 °C to 43.9 °C using alumina nano-fluids for 1% of volumetric concentration at a flow rate of 1 L/min compared to the pure base fluid [27]. Five different geometries with different fin spacing have been investigated using water as a coolant. The base temperature of 40.5 °C has been reported for a heat sink with pin spacing of 0.2 mm at a heating power of 325 W [2].

In this research, we investigate the 2-D void arrangement which has square from one side and circular cross section from other side. To the best of our knowledge, such cellular periodic structure has not been addressed yet. Moreover, a comparison of cellular structures with 2-D void arrangement circular bore structures, 2-D and 3-D void arrangement square bore structures are discussed in detail. We performed numerical simulations for different cellular structures to find the optimized cellular structure and then numerical results for optimized structure are validated with experimental results.

2 Numerical analysis

We used computational fluid dynamics (CFD) approach to simulate the conjugate heat transfer problem. We analysed different cellular structure designs using commercial CFD software package. We used ANSYS design modeller for modelling, ANSYS meshing for CFD meshing and ANSYS fluent as a CFD solver. In order to examine the thermal and flow characteristics numerically, the governing equations include conservation of mass, momentum, energy, turbulence kinetic energy and turbulence dissipation rate. These equations are as follows:

$$\frac{\partial}{\partial x_i} (\rho u_i) = 0$$ (1)
\[
\frac{\partial}{\partial x_j} (\rho \mu u_j) = \frac{\partial p}{\partial x_j} + \frac{\partial}{\partial x_i} \left[ (\mu + \mu_t) \frac{\partial u_j}{\partial x_i} \right] + \frac{\partial}{\partial x_i} \left[ (\mu + \mu_t) \frac{\partial p}{\partial x_i} \right].
\]

(2)

\[
\frac{\partial}{\partial x_j} (\rho u_j T) = \frac{\partial}{\partial x_i} \left[ \left( \frac{\lambda}{C_p} + \mu_t \right) \frac{\partial T}{\partial x_i} \right]
\]

(3)

\[
\frac{\partial}{\partial x_j} (\rho k u_j) = \frac{\partial}{\partial x_i} \left[ \left( \mu + \mu_t \right) \frac{\partial k}{\partial x_i} \right] + G_k - \rho \varepsilon
\]

(4)

\[
\frac{\partial}{\partial x_j} (\rho \varepsilon u_j) = \frac{\partial}{\partial x_i} \left[ \left( \mu + \mu_t \right) \frac{\partial \varepsilon}{\partial x_i} \right] + \frac{\varepsilon}{k} (c_1 G_k - c_2 \rho \varepsilon)
\]

(5)

Where \( k \), \( \varepsilon \) and \( G_k \) represent turbulence kinetic energy, turbulence dissipation rate and generation of turbulence kinetic energy. We found an optimal cellular structure for heat dissipation on the basis of these simulations.

### 2.1 Boundary conditions

The following boundary conditions were imposed:-

- Heat supplied of 150 W at the bottom surface
- Inlet velocity of air was at 6 m/s
- Pressure outlet was fixed at 0 gauge pressure
- Insulated boundary conditions were imposed to the top and on two side walls
- Inlet air temperature was fixed at 25 °C

### 2.2 Data reduction

We adopted following procedure for data reduction for all the structures.

Heat removed by the air circulating in the structure can be calculated from (6) by applying conservation principle with one inlet and one outlet and assuming negligible changes in kinetic and potential energies as:-

\[
Q = \dot{m} C_p (T_o - T_i)
\]

(6)

Porosity can be calculated from (7).

\[
\epsilon = 1 - \left( \frac{V}{V_b} \right)
\]

(7)

Blockage ratio and open area ratio can be calculated from (8) and (9).

\[
R_{BR} = \frac{A_b}{A_t}
\]

(8)

\[
R_{OPEN} = (1 - R_{BR})
\]

(9)

Pressure loss coefficient can be calculated from (10).

\[
K_{cell} = \left( \frac{1 - R_{OPEN}}{R_{OPEN}} \right)^2
\]

(10)

Surface area density can be calculated from (11).

\[
\alpha_{sf} = \frac{A_s}{V}
\]

(11)

Thermal management index can be calculated from (12).

\[
\eta = \frac{Q}{\Delta P}
\]

(12)

### 2.3 Parametric study

A parametric study was carried out to find the optimized cellular structure. In this process, we varied only one parameter and keep all of the other parameters constant. We initially selected Copper as structure material. Following sequence was followed for the parametric study:-

- Effect of Bore Diameter
- Effect of Design
- Effect of Height
- Effect of Material

#### 2.3.1 Effect of bore diameter

We investigated the effects of bore diameter on the heat transfer for different structures. We provide details of these structures in Table 1. We found the optimum bore diameter by keeping all the other parameters (H, W, L, and t) constant. All the circular bore structures had void arrangement along x and z axis. We show one of the cellular structures in the Fig. 1.

We provide a comparison of results for three structures in Table 2.

In Table 2, it can be seen that the thermal management index (\( \eta \)) of the three structures is almost the same. It can also be seen from Table 2 that the circularbore1 shows lowest base temperature and maximum heat transfer as compared to other two structures. As the thermal management index is same for all structures and circularbore1 shows minimum base temperature, we select bore diameter of 3 mm for further investigation.

| Table 1 Cellular structures dimension (mm) |
|------------------------------------------|
| Circularbore1 | Circularbore2 | Circularbore3 |
| d | 3 | 4 | 5 |
| c | 4 | 5 | 6 |
| t | 1 | 1 | 1 |
| H | 48.5 | 48.5 | 49 |
| L | 69 | 69 | 67 |
| W | 69 | 69 | 67 |
2.3.2 Effect of design

In order to analyze the effect of different designs, we used the squarebore of dimensions 3 mm × 3 mm and compared the results with already selected 3 mm bore diameter of circularbore1 structure. As in previous section, d was selected as 3 mm, so b was also selected as 3 mm for this investigation. All the parameters (H, W, L, t, and b) were fixed, while structure designs were varied. Squarebore1 had square void arrangement along x,y,z axis, squarebore2 had square void arrangement along x and z axis while squarebore3, squarebore microhole1 and squarebore microhole2 had square void arrangement along y and z axis. We provide details of these structures in Table 3.

We show results for five different designs (squarebore1, squarebore2, squarebore3, squarebore microhole1 and squarebore microhole2) in Table 4. It can be seen that all the squarebore structures are offering higher efficiency as compared to circularbore structures which were discussed in last section. Squarebore1 and squarebore microhole2 structures are dissipating maximum heat among all structures but there is a less pressure drop in squarebore microhole2 structure. As squarebore microhole2 have the highest thermal management index as compared to other structures, we selected squarebore microhole2 structure as the final design for the further study.

2.3.3 Effect of height

In order to determine the optimum height of the selected design, we investigated the temperature variation with the height. We show a plot between temperature and height in Fig. 2. It can be seen that after a certain height, the temperature is not changing considerably and the curve becomes flat, so we selected 0.037 m as an optimum height of the structure.

2.3.4 Effect of material

We find that the squarebore microhole3 is the optimal structure from the above analyzed structures with height of 37 mm. We used Copper material for the cellular structures till now. The optimized structure is then tested with the Aluminum material. We provide a comparison between Copper and Aluminum in Table 5.

Thermal management index of structure using Copper is greater than Aluminum. Due to higher conductivity of Copper, heat removed by the structure is greater as compare to Aluminum. We selected Copper material by considering the results shown in Table 5.

We created a YZ plane in the middle of the solid-fluid domain to show the temperature distribution inside the structure as shown in Figs 3, 4, 5. It can be seen that temperature is higher at the base and then it gradually starts decreasing away from the base in the structure. We observed the maximum base temperature for circularbore1, circularbore2 and circularbore3 are 47.8 °C, 60.2 °C and 66.6 °C respectively as shown in the Fig. 3. We observed the maximum base temperature for squarebore1, squarebore2, squarebore3, squarebore microhole1 and squarebore microhole2 to be 62.1 °C, 53.4 °C, 46.4 °C, 46.5 °C and 45.8 °C respectively, as shown in the Fig. 4. We observed the maximum base temperature for squarebore microhole3 (Cu) and square microhole3 (Al) to be 49.5 °C and 53.4 °C respectively, as shown in the Fig. 5. The inlet face is directly facing the cool air coming from the fan and entering the structure, therefore temperature is lower at the inlet as compared to the outlet face.

| Table 2 | Effect of bore diameter |
|---------|-------------------------|
| Circularbore1 (Cu) | Circularbore2 (Cu) | Circularbore3 (Cu) |
| T_b (°C) | 47.8 | 60.2 | 66.6 |
| ΔP (Pa) | 445.6 | 305.4 | 196.1 |
| Q̇ (W) | 117.1 | 80.4 | 48.4 |
| ε | 0.58 | 0.63 | 0.67 |
| R_{BR} | 0.57 | 0.51 | 0.47 |
| R_{OPEN} | 0.43 | 0.49 | 0.53 |
| α_{sf} (1/m) | 1486.7 | 1404.2 | 1331 |
| η (m³/s) | 0.26 | 0.26 | 0.24 |

| Table 3 | Cellular structures dimension (mm) |
|---------|----------------------------------|
| Squarebore1 | Squarebore2 | Squarebore3 | Squarebore microhole1 | Squarebore microhole2 |
| b | 3 | 3 | 3 | 3 | 3 |
| t | 1 | 1 | 1 | 1 | 1 |
| d_s | – | – | – | 2 | 1.5 |
| c | – | – | – | 4 | 4 |
| H | 49 | 49 | 49 | 49 | 49 |
| L | 65 | 65 | 65 | 65 | 65 |
| W | 65 | 65 | 65 | 65 | 65 |

Fig. 1 Cellular structure; circularbore1
3 Experimental setup

From numerical investigations, we selected squarebore microhole3 (Cu) structure because we found it to be the optimal structure as heat dissipation media among all the structures discussed in this research work. We performed experiments on this selected structure and the obtained results were compared with numerical predictions. The optimized cellular structure was manufactured on the Wire Cut Electrical Discharge Machining (EDM). We show this structure in Fig. 6.

3.1 Test loop

A constant heat was supplied to Copper block by DC electric iron heater which was producing a heat flux of 35,503 W/m² continuously. A constant voltage of 210 V and current of 0.71 A was supplied to maintain the constant heat generation of 150 W. An electric iron heater was connected to the AC to DC converter because a DC iron heater was operated. Copper block was mounted on the top of cellular structure for uniform distribution of the heat flux. All exposed surfaces were insulated using fiber-glass wool. In order to measure the base temperature, a fine wire K-type thermocouple was attached with the temperature meter, which was inserted between the heated copper block and the cellular structure. The outlet air temperature was measured by another K-type thermocouple. Set of data is obtained when steady state condition was reached. Steady state was identified when the temperature measurements changed by no more than ±0.1 °C in 3 min. A DC fan with speed controller to supply inlet air at different velocities was attached to AC to DC converter. Anemometer was used to measure the inlet air velocity and a duct was used for the smooth passage of air flow. After the duct, cellular structure was placed. Schematic of experimental setup is shown in Fig. 7.

3.2 Uncertainty analysis

The uncertainty in the experimental data was estimated by the method developed by Kline and McClintock [28]. This method

Table 4 Effect of design

|                | Squarebore1 (Cu) | Squarebore2 (Cu) | Squarebore3 (Cu) | Squarebore microhole1 (Cu) | Squarebore microhole2 (Cu) |
|----------------|------------------|------------------|------------------|---------------------------|---------------------------|
| $T_b$ (°C)     | 62.1             | 53.4             | 46.4             | 46.5                      | 45.8                      |
| $\Delta P$ (Pa)| 360.8            | 256.1            | 254.1            | 175.1                     | 163.1                     |
| $Q$ (W)        | 112.8            | 83.2             | 94.1             | 98.7                      | 111                       |
| $\varepsilon$  | 0.80             | 0.67             | 0.67             | 0.59                      | 0.57                      |
| $R_{BB}$       | 0.45             | 0.45             | 0.45             | 0.45                      | 0.45                      |
| $R_{OPEN}$     | 0.55             | 0.55             | 0.55             | 0.55                      | 0.55                      |
| $\alpha_{sf}$ (1/−m) | 2914           | 2029             | 2018             | 1827                      | 1786                      |
| $\eta$ (m³/s) | 0.31             | 0.32             | 0.37             | 0.56                      | 0.66                      |

Table 5 Effect of material

|                | Squarebore microhole3 (Cu) | Squarebore microhole3 (Al) |
|----------------|----------------------------|---------------------------|
| $T_b$ (°C)     | 49.5                       | 53.4                      |
| $\Delta P$ (Pa)| 166.4                     | 163.5                     |
| $Q$ (W)        | 99.2                       | 88.4                      |
| $\varepsilon$  | 0.57                       | 0.57                      |
| $R_{BB}$       | 0.46                       | 0.46                      |
| $R_{OPEN}$     | 0.54                       | 0.54                      |
| $\alpha_{sf}$ (1/−m) | 1770                  | 1770                      |
| $\eta$ (m³/s) | 0.60                       | 0.54                      |
incorporates the estimated uncertainties in the experimental measurement of heat transfer rate. The uncertainty in the heat transfer rate was found to be 4.9%. The uncertainty for the temperature measurements was estimated to be ±0.1 °C.

## 4 Results and discussions

### 4.1 Effect of pressure drop and pressure loss coefficient with open area ratio

Open area ratio is basically the remaining area ratio of blockage ratio (1-R_{BR}). Blockage ratio is the ratio of frontal blocked area to the frontal total area. We show variation in pressure drop with open area ratio in Fig. 8. All three structures (squarebore microhole1, squarebore microhole2, squarebore microhole3) offered the minimum pressure drop with maximum open area ratio at the inlet velocity of 6 m/s. Maximum pressure drop is observed by circularbore1 structure with minimum open area ratio. Circularbore3 structure shows lower pressure drop because of larger bore diameter which makes the passage of air through the bore easier. It is also observed that circularbore1 offer 42.5% more pressure drop as compare to squarebore2 while the only difference between the two structures is of circular and square void cross-section, which indicated that the squarebore structure is better than circularbore structure. Indeed, the squarebore1 is a wire mesh structure, therefore, it shows greater pressure loss as compared to other squarebore structures.

We show variation in pressure loss coefficient with open area ratio in Fig. 9. We observed the similar trend in variation of pressure loss coefficient with open area ratio as reported by Tian et al. [29] for square and diamond shape cellular metal core panels in Fig. 7. The pressure loss across a unit cell can be described by a single parameter R_{open} (or ε). The curve shows that the frontal area blockage causes large pressure loss that is why circularbore structures has large pressure loss coefficient as compared to squarebore structures. Circularbore3 has larger bore diameter that is why it shows lesser pressure loss coefficient as compare to other circularbore structures.

### 4.2 Effect of pressure drop with porosity

The porosity can be obtained by subtracting the ratio of the volume of structure to total volume of the block from unity [1-(V/V_b)]. The higher porosity means that there is a less solid material per unit volume, and hence lesser conduction through the structure, while the force convection will be greater. The lower porosity means that there is less empty volume per unit volume, and hence lesser contribution of forced convection while conduction will be greater. Variation of pressure drop with porosity is shown in Fig. 10. It can be seen from the figure that the Squarebore1 structure depicts higher porosity due to its textile wire mesh like structure, and hence its convection heat transfer is found maximum, but offering higher pressure drop. The structures (squarebore microhole1, squarebore microhole2, squarebore microhole3) provide less pressure drop as compared to all other structures discussed here. The highest pressure drop is noticed for circularbore1 structure as the bore diameter is smaller as compare to other circularbore structures.

### 4.3 Thermal management index

The thermal management index is the ratio of amount of heat dissipated from the structure to the pressure drop across the structure. More pressure drop across the structure requires more fan power to be supplied. Ultimate goal for designing of the best cellular structure is to dissipate the maximum amount of heat with minimum pressure drop across the structure. Thermal performance is evaluated for each of the cellular structure discussed in this research. The thermal management index η of all the cellular structures is shown in Fig. 11. It can be seen from previous sections, that squarebore1 (Cu) is providing the maximum surface area density and larger pressure loss...
drop across the structure, which is reason of its low thermal management index $\eta$ of 0.31. This value is lesser than all other squarebore structures. The maximum thermal management index $\eta$ is achieved for squarebore microhole2 (Cu) which is 0.66 with structure height of 49 mm, while the optimized structure squarebore microhole3 (Cu) has thermal management index $\eta$ of 0.60 with structure height of 37 mm. The optimized structure squarebore microhole3 shows thermal management index $\eta$ of 0.60 and 0.54 by using Copper and Aluminum materials respectively.

Fig. 4 Temperature distribution of; a) squarebore1, b) squarebore2, c) squarebore3 d) squarebore microhole1e) squarebore microhole2
4.4 Effect of heat transfer rate with surface area density

High surface area densities are desirable for compact heat exchangers. A heat exchanger with $\alpha_{sf} > 700 \text{ m}^2/\text{m}^3$ is called a compact heat exchanger. All structures studied in this research were compact heat exchangers as their surface area density ($\alpha_{sf}$) were higher than 700 m$^2$/m$^3$. The variation in the amount of heat transfer rate with surface area density is shown in Fig. 12. Squarebore1 offered the maximum surface area density, which depicted the maximum heat transfer through convection is possible with this structure. The amount of heat transfer by circularbore1 and squarebore1 is higher but pressure drop is also higher as shown in Fig. 8 and Fig. 9. The minimum surface area density is noticed for circularbore3 with minimum heat transfer. Surface area density of squarebore structures is higher as compare to circularbore structures as shown in Fig. 12.

4.5 Effect of base temperature with surface area density and inlet velocity

The base temperature of electronic devices should not exceed 60-80 °C. The variation in base temperature with surface area density of all the structures studied in this research is shown in Fig. 13. It can be seen that circularbore structures offered minimum surface area density in comparison with the squarebore structures. This shows that there will be lesser convection and higher conduction in circular bore structures as surface area density is lesser as compare to squarebore structure. The maximum surface area density is offered by squarebore1 (Cu) which ensures the maximum convection heat transfer. The base temperature was recorded to be 62.1 °C for squarebore1 (Cu) structure, which is very high. It means that it requires higher inlet air velocity to remove the produced heat in the structure to reduce the base temperature. The data depicts that values of the base temperature of the optimize structure squarebore microhole3 are 49.5 °C and 53.4 °C with Copper and Aluminium respectively at the inlet velocity of 6 m/s. It is due to the fact that the thermal conductivity of Copper (387.6 W/mK) is higher than Aluminium (202.4 W/mK). Experimentally, we found the base temperature of the optimized squarebore microhole3 (Cu) to be 53.5 °C at air inlet velocity of 6 m/s with 150 W of heat generation. We found a close agreement between experimental results and numerical prediction for the base temperature with an error of 8.08% at 6 m/s. We have also compared the best achieved base temperature with reported values of the base temperature using air, water and nano-fluids. The values of the base temperature are reported to be 55 °C, 40.5 °C and 43.9 °C using air [1], water [2] and nano-fluids [27].

The variation in the base temperature with inlet velocity of the optimized structure is shown in Fig. 14. It can be seen from the figure that the curve showing numerical
Fig. 7 Schematic of experimental setup

Fig. 8 Variation of pressure drop with open area ratio

Fig. 9 Variation of pressure loss coefficient with open area ratio

Fig. 10 Variation of pressure drop with porosity

Fig. 11 Thermal management index
results is slightly below the experimental curve. We estimate an error of 8.71% between the numerical and experimental data at 10 m/s. The base temperature tends to reduce with an increase in the inlet velocity of air. The maximum values of the base temperature achieved are 49.5 °C and 53.5 °C numerically and experimentally respectively at inlet velocity of 6 m/s, while the minimum value achieved are 43.6 °C and 47.4 °C numerically and experimentally respectively at 10 m/s.

5 Conclusion

Rapid development in the microelectronic industry demands new modes to continuously dissipate the heat for reliable functioning of electronics chips. To remove the high heat flux new techniques are under study either by fluid or by increasing surface area. As a substitute of air cooled heat sinks, the nano-fluids and water cooled heat sinks are now in practice but require extra cooling mechanism for cooling of the fluid. Metal foams also participate in this race providing larger surface area but thermo-mechanical properties of cellular structures are better than the metal foams. Therefore, cellular structures can provide better performance with acceptable base temperature and with a very low pressure drop. In present research, we investigated the thermal performance of cellular structures using computational fluid dynamics (CFD). An optimal structure is obtain on the basis of parametric study in which effect of bore diameter, effect of design, effect of height, effect of materials is discussed in detail. The optimal structure is than validated experimentally. The core verdicts of this research work are as follows:-

- Base temperature and pressure drop noticed for circularbore structures is higher as compare to squarebore structures.
- The pressure drop is found 42.5% larger in circularbore1 structure than the squarebore2 structure, while the only difference between the two structures is of circular and square void cross-section.
- The squarebore microhole3 structure is the optimum structure among different studied cellular structures.
- Thermal management index $\eta$ for squarebore microhole3 is 0.60 and 0.54 using Copper and Aluminium respectively.
- The base temperature tends to decrease with increase in air flow velocity.
- The minimum base temperature for the optimized cellular structure is estimated to be 43.6 °C and 47.4 °C numerically and experimentally respectively with inlet velocity of 10 m/s and at heat flux of 35,503 W/m² (150 W of heat supplied).
- The maximum base temperature of the optimized structure is estimated to be 49.5 °C and 53.5 °C numerically and experimentally respectively at inlet velocity of 6 m/s.
- The numerical results for the base temperature are in close agreement with experimental results with an error of 8.71%.
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