A novel design of heat sink with PCM for electronics cooling

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Outline

• Introduction – PCM in electronics cooling devices
• Novel design of heat sink for electronics cooling - description
• Basic characteristics of proposed design in normal mode of operation
  – Methods of analysis
  – Results - basic properties of heat sink
• Thermal behavior of heat sink in transient thermal conditions
  – Mathematical and numerical model of heat transfer in transient conditions,
  – Thermal behavior of heat sink in case of fan accident
• Conclusions
Introduction

Cooling techniques for electronics devices

- **Single phase flow:**
  - natural convection air cooling,
  - forced convection air cooling,
  - liquid cooling (cold plates, jet impingement)

- **Two-phase flow cooling systems:**
  - pool boiling,
  - boiling in channels, (microchannels),
  - jet impingement with boiling,
  - spray cooling,

- **Auxiliary heat transfer devices:**
  - heat pipes,
  - thermoelectric modules
Introduction
Phase change materials (PCM) in electronics cooling application

- **Heat sinks in portable (mobile) electronic devices:**
  - palm pilots,
  - cellular phones,
  - personal digital assistants,
  (devices seldom used for more than a few hours continuously at peak load and their idle time is typically long enough to solidify the molten PCM)

- **PCM slurry as a heat transfer fluid in liquid cooling**

- **PCM as interface materials**
  provide the best filling of micro-gaps in the case-heat sink interface while material is melted. Basically a mixture of organic binders (e.g. waxes) and fine particle ceramic fillers for thermal enhancement.
Introduction

Examples of PCM application in electronics cooling

1. Composite heat sinks (CHS), constructed using vertical array of fins, made of large latent heat capacity phase change materials (PCM) and highly conductive base material (BM)

2. Heat storage unit containing PCM for cooling of portable electronics

3. PCM slurry as a heat transfer fluid in liquid cooling: paraffin ($C_{22}H_{46}$) + water + emulsifier,
   mass fraction of the paraffin slurry from 2.5 to 7.5%,
Heat sink with PCM for electronics cooling

Problems

- **High efficiency of dissipation of heat**
  - low total thermal resistance:
    - High heat transfer coefficient,
    - Large heat transfer surface area,

- **Efficient transfer of heat to PCM**
  incorporation of PCM in heat sink structure
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New design of heat sink with PCM

Top view and details of PCM incorporation

Fan

Base of heat sink

Aluminum pipe

PCM
New design of heat sink with PCM

Details

External walls
Channel for air flow
Basic characteristics

Normal mode of operation

Thermal resistance basic property of heat sink.

In this structure total thermal resistance consists of:
• Resistance of base plate
• Resistance of pipes bank
  (heat conduction along pipes and convective heat transfer to the air flow)
Thermal resistance of base plate

- **Method:**
  Numerical solution of steady state heat conduction equation across plate:

- **Input data:**
  - Temperature of bottom surface,
  - Temperature of air flow
  - Heat transfer coefficient

- **Output data:**
  - Heat flux dissipated by heat sink
  - Temperature profile of top surface of the plate.
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Thermal resistance of base plate
Example results

\[
\delta = 5 \text{ mm} \\
R = 0.244 \text{ K/W}
\]

\[
\delta = 10 \text{ mm} \\
R = 0.198 \text{ K/W}
\]

\[
\delta = 15 \text{ mm} \\
R = 0.196 \text{ K/W}
\]

Uniform surface temperature is crucial from the point of view of PCM operation
Thermal resistance of base plate

Example results – input data

- Dimensions of plate: 50×60 mm,
- Dimensions of contact area – surface of processor: 10×10 mm, in the middle of the plate’s lower surface,
- Dimensions of pipes/fins: diameter 3 mm, thickness of the wall 0.2 mm, length 50 mm, s = 6 mm; efficiency of such pipe/fin equals to 0.27,
- Boundary condition on the contact area – known, uniform temperature,
- Boundary condition on the upper surface – third kind b.c.: know convective heat transfer coefficient and air temperature. Effective heat transfer coefficient was determined from the formula
  \[ h_{\text{eff}} = h \pi d l \eta_{\text{fin}} / s^2 \]
  which takes into account real heat transfer coefficient on the surface of pipes/fins, efficiency of fins, their dimensions and density of fins on the surface (s – distance between pipes), real heat transfer coefficient \( h = 180 \text{ W/(m}^2\text{K)} \), its effective value: 630 W/(m²K)
- Other surfaces were considered as adiabatic (heat transfer is so small that can be neglected).

Material data (aluminum):
- \( k = 200 \text{ W/(m} \cdot \text{K)} \), \( \rho = 2800 \text{ kg/m}^3 \), \( c_p = 880 \text{ J/(kg} \cdot \text{K)} \),
Thermal resistance of pipes bank
Calculation procedure
Thermal resistance of pipes bank

Calculation procedure

Thermal resistance for heat transfer along pipes and heat convection to the air flow:

\[ R = \frac{T_b - T_0}{Q} \]

Total heat flux \( Q \):

\[ Q = \sum_{i=1}^{n} Q_i \]

Heat flux for \( i \)-th row of pipes

\[ Q_i = h_i A_i (T_b - T_{i-1}) \eta_{\text{fin}} \]

Heat transfer coefficient:

\[ \text{Nu} = \frac{h_i d_{\text{out}}}{k} = C \text{Re}_{\text{max}}^n \text{Pr}^n \left( \frac{\text{Pr}}{\text{Pr}_w} \right)^{0.25} C_i \]

Temperature increase of air flowing across \( i \)-th row is estimated from the energy balance

\[ m c_p (T_i - T_{i-1}) = Q_i \]

Pipe/fin efficiency

\[ \eta_{\text{fin}} = \frac{\tanh(ml)}{ml}; \quad m = \sqrt{\frac{4hd_{\text{out}}}{k(d_{\text{out}}^2 - d_{\text{in}}^2)}} \]
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Thermal resistance of pipes bank
Example results

Thermal resistance of heat sinks with pipes of different wall thickness; diameter of pipes 3 mm

<table>
<thead>
<tr>
<th>Internal diameter of pipes, $d_{in}$, mm</th>
<th>Thickness of pipe wall, $\delta_w$, mm</th>
<th>Thermal resistance, $R$, K/W</th>
</tr>
</thead>
<tbody>
<tr>
<td>2,6</td>
<td>0,2</td>
<td>0,538</td>
</tr>
<tr>
<td>2,4</td>
<td>0,3</td>
<td>0,463</td>
</tr>
<tr>
<td>2,2</td>
<td>0,4</td>
<td>0,415</td>
</tr>
<tr>
<td>2,0</td>
<td>0,5</td>
<td>0,385</td>
</tr>
</tbody>
</table>

Thermal resistance of heat sinks with different spacing between pipes $s$; diameter of pipes 3 mm, thickness of the wall 0,2 mm

<table>
<thead>
<tr>
<th>Distance between pipes, $s$, mm</th>
<th>Total number of pipes</th>
<th>Heat transfer area, $cm^2$</th>
<th>Thermal resistance $R$, K/W</th>
</tr>
</thead>
<tbody>
<tr>
<td>6</td>
<td>80 (8×10)</td>
<td>377</td>
<td>0,538</td>
</tr>
<tr>
<td>5,4</td>
<td>99 (9×11)</td>
<td>467</td>
<td>0,426</td>
</tr>
<tr>
<td>4,8</td>
<td>120 (10×12)</td>
<td>565</td>
<td>0,345</td>
</tr>
<tr>
<td>4,2</td>
<td>154 (11×14)</td>
<td>726</td>
<td>0,270</td>
</tr>
</tbody>
</table>
Thermal resistance of pipes bank

Example results

Thermal resistance of heat sinks with pipes of different outer diameter, thickness of the wall 0,2 mm, grid spacing $s = 2 \cdot d_{out}$.

<table>
<thead>
<tr>
<th>Outer diameter $d_{out}$, mm</th>
<th>Total number of pipes</th>
<th>Heat transfer area, cm$^2$</th>
<th>Efficiency of pipes-fins</th>
<th>Thermal resistance $R$, K/W</th>
</tr>
</thead>
<tbody>
<tr>
<td>3,0</td>
<td>80 (8×10)</td>
<td>377</td>
<td>0,268</td>
<td>0,538</td>
</tr>
<tr>
<td>2,8</td>
<td>80 (8×10)</td>
<td>352</td>
<td>0,269</td>
<td>0,578</td>
</tr>
<tr>
<td>2,6</td>
<td>99 (9×11)</td>
<td>404</td>
<td>0,262</td>
<td>0,495</td>
</tr>
<tr>
<td>2,4</td>
<td>120 (10×12)</td>
<td>452</td>
<td>0,255</td>
<td>0,439</td>
</tr>
</tbody>
</table>
Total thermal resistance of heat sink

Example results

Total thermal resistance of heat sinks with different base plates
(pipe bank – 3/2,6 mm, s = 6 mm)

<table>
<thead>
<tr>
<th>Thickness of base plate, mm</th>
<th>Max temperature difference on the surface, K</th>
<th>Thermal resistance of base plate, K/W</th>
<th>Total thermal resistance, K/W</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>5,7</td>
<td>0,244</td>
<td>0,782</td>
</tr>
<tr>
<td>10</td>
<td>2,5</td>
<td>0,198</td>
<td>0,736</td>
</tr>
<tr>
<td>15</td>
<td>1,2</td>
<td>0,196</td>
<td>0,734</td>
</tr>
</tbody>
</table>
Thermal properties of pipe heat sink

Auxiliary data

Temperature profiles along pipes for different internal dimensions

temperature at the bottom of pipes - 40°C, air temperature 30°C
Thermal behavior of heat spreader in transient thermal conditions

The purpose of the analysis:

to estimate temperature variations in crucial points of the spreader (base plate) caused by e.g.:

- rapid rise of heat load of the processor,
- sudden drop of heat transfer coefficient on the surface (e.g. damage of the fan),
Physical model

In normal operation mode – electronics work in stable conditions – excess heat is released to the ambient air through base plate and walls of pipes (high conductivity part of the spreader); PCM doesn't take part in heat transfer; however, it is heated to the same temperature as the wall at the same level (similar temperature profile along the axis)

In transient conditions – due to the change of temperature of the wall a part of dissipated heat is transferred to the PCM region causing heating and melting of this material.
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Mathematical model

Heat conduction equation (in polar coordinate system \((r, \theta)\))

\[
\rho_i c_{pi} \frac{\partial T_i}{\partial \tau} = k_i \left[ \frac{1}{r} \frac{\partial}{\partial r} \left( r \frac{\partial T_i}{\partial r} \right) + \frac{\partial^2 T_i}{\partial \theta^2} \right]
\]

index \(i\) denotes different materials (aluminum or PCM) or phases (solid or liquid of PCM).

Convective **boundary condition** on the surface of pipes

\[
-k_{Al} \frac{\partial T_i}{\partial r} \bigg|_{r=r_{out}} = h \left( T(r_{out}) - T_\infty \right)
\]

On the interfaces between metal and PCM **boundary conditions** of the fourth kind

\[
-k_{Al} \frac{\partial T_{Al}}{\partial r} \bigg|_{Al} = -k_{PCM} \frac{\partial T_{PCM}}{\partial r} \bigg|_{PCM}
\]

\[
-\lambda_{Al} \frac{\partial T_{Al}}{\partial \theta} \bigg|_{\theta=0} = -\lambda_{PCM} \frac{\partial T_{PCM}}{\partial \theta} \bigg|_{\theta=0}
\]
Mathematical model

Initial conditions

Heat transfer coefficient on the surface of pipes: 180 W/(m²K),

Given temperature of air flow (30°C)

Temperature profile along pipes

\[
T(x) = T_\infty + (T_b - T_\infty) \frac{\cosh m(L - x)}{\cosh mL}
\]

Given temperature of the upper part of base plate (40°C)

Heat flux dissipated to the air calculated for the above thermal conditions; this heat flux is set constant in subsequent calculations – thermal behavior of the heat sink in changed conditions (changed heat transfer coefficient)
Numerical solution

- CVFDM with explicit integration over time
- $100 \times 6$ control volumes of ring shape
- Relation between enthalpy and temperature for the PCM

\[
H(T) = \begin{cases} 
  c_{ps} T & \text{for } T < T_m - 1/2 \Delta T_m \\
  c_{ps} (T_m - 1/2 \Delta T_m) + C_m \frac{T - T_m + 1/2 \Delta T_m}{\Delta T_m} & \text{for } T \in (T_m - 1/2 \Delta T_m ; T + 1/2 \Delta T_m) \\
  c_{ps} (T_m - 1/2 \Delta T_m) + C_m + c_{pl} (T - T_m + \Delta T_m) & \text{for } T > T_m + 1/2 \Delta T_m 
\end{cases}
\]

where: $C_m$ – latent heat, $\Delta T_m$ – range of temperature in which melting occurs, $c_{ps}$ and $c_{pl}$ – specific heats of solid and liquid PCM.
Phase change material

**Lauric acid** CH₃(CH₂)₁₀COOH (fatty acid) – material which is considered as PCM in different energy storage applications, melting temperature fits the requirements of electronics cooling; material was tested experimentally in real radiator for electronics cooling.

Material properties – heat capacity measured by DSC technique (lauric acid of 99,5% purity made by Roth GmbH):

- **Melting point, °C** 41,5 ± 1
- **Latent heat, kJ/kg** 178
- **Specific heat of solid, kJ/(kg·K)** 2,34
- **Specific heat of liquid, kJ/(kg·K)** 2,17
- **Thermal conductivity, W/(m·K)** 0,2
- **Density, kg/m³** 800
Temperature variations vs. time in different points of the pipe with PCM in transient conditions after sudden drop of heat transfer coefficient from 180 to 10 W/(m²·K)
Results of numerical simulations

Temperature variations vs. time in different points of the pipe with PCM in transient conditions after sudden drop of heat transfer coefficient from 180 to 5 W/(m²·K)
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Results of numerical simulations

Comparison of foregoing results:

- Distinct difference of maximum temperature,
- Similar time of stabilization of temperature – the same amount of PCM, similar thermal conditions on the boundary of PCM region
Results of numerical simulations

Temperature variations vs. time in selected points in the pipes of different diameters
Solid lines – diameter of the pipe equals to 1,5 mm, dashed lines – 3 mm; thickness of the wall 0,2 mm; Differences in thermal performance are basically due to greater amount of PCM in the spreader with 3 mm pipes
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Temperature vs. time in different positions of the pipe, and for pipes of different dimensions

Temperature vs. time in different positions of the pipe, and for pipes of different dimensions.
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Temperature profiles along pipe in the PCM vs. time for different thickness of the wall; outer diameter 3 mm

Red line – initial temperature, blue lines – temperature after 1, 2, …, 6 min;
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Temperature profiles along aluminum pipe vs. time as in the previous slide

Red line – initial temperature, blue lines – temperature after 1, 2, …, 6 min;
Conclusions

• **Advantages of the cooling device:**
  – high cooling efficiency in nominal conditions, due to high heat transfer coefficient and relatively large area,
  – potential to stabilize the temperature of the electronics device,
  – potential to protect the electronics from burning caused by rapid changes (increase) in heat transfer rates,
  
  **have been proved theoretically**

• **Drawbacks** of the device – rather complex structure and manufacturing