

# Thermal Analysis and Experimental Studies of a Plate-Fin Heat Sink Integrated with U-Shaped Liquid-Cooling Channel

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**Abstract:** The continuous miniaturization and rising power densities of modern electronic modules demand advanced thermal management solutions capable of dissipating high heat fluxes safely. Conventional air-cooled natural convection systems often fail under elevated loads due to limited fluid heat capacity and restricted thermal conductivity. In this study, both experimental and numerical investigations were carried out to evaluate the transition from standard natural convection to a hybrid cooling configuration that integrates a plate-fin aluminium heat sink with a U-shaped liquid-cooling channel. Controlled trials were performed at heat inputs of 10 W, 20 W, and 30 W, with transient and steady-state temperatures recorded at the base, top fins, and coolant outlet. Under natural convection, base temperatures rose sharply to 56.11 °C (10 W), 78.06 °C (20 W), and a critical 93.02 °C (30 W), highlighting the severe limitations of pure air cooling. In contrast, the hybrid water-cooled channel significantly suppressed base temperatures to 43 °C (10 W), 59 °C (20 W), and 74 °C (30 W). Energy balance analysis showed that the liquid channel absorbed 3.40 W, 7.30 W, and 11.10 W across the respective loads, while fin conduction contributed 2.89 W, 5.46 W, and 8.57 W. Complementary CFD simulations performed in ANSYS Fluent exhibited strong agreement with experimental data, with deviations confined to 5–8%. These findings validate the hybrid cooling system's capability to overcome thermal bottlenecks, making it a reliable solution for high-density engineering and advanced electronic packaging applications.

**Keywords:** Plate-Fin Heat Sink, Natural Convection; Liquid Cooling Channel; Thermal Management; CFD Validation; Heat Dissipation; Compact Thermal Systems.

## I. INTRODUCTION

Managing heat effectively is a critical requirement in modern engineering systems, as excess thermal energy can reduce efficiency, shorten component life, and cause premature failure. Conventional natural convection cooling using air is often sufficient for low heat loads, but its limited thermal conductivity and heat capacity restrict performance at higher power levels. Heat sinks, typically made of aluminium or copper, are widely used to spread and dissipate heat through conduction and convection. Their effectiveness depends on geometry, fin arrangement, and cooling conditions. However, as devices become more compact and power-dense, natural convection alone cannot maintain safe operating temperatures.

Liquid cooling has emerged as a reliable alternative because water possesses a high specific heat capacity and favorable thermal properties. Integrating a water channel with a plate-fin heat sink creates a hybrid cooling system that combines conduction through the solid material with convective heat removal by both air and flowing water. In this arrangement, the heat source is positioned below the channel, while the heat sink is mounted above it. Heat is simultaneously absorbed by the coolant and dissipated through the fins, improving overall thermal performance. Monitoring the base, top, and outlet temperatures provides a clear indication of how effectively the system responds to different heat inputs. And Experimental setup is shown in fig 1

This study focuses on comparing natural convection cooling with a hybrid water-channel-assisted plate-fin heat sink under heat inputs of 10 W, 20 W, and 30 W. Experimental measurements are supported by CFD simulations to validate the numerical model. The results highlight the limitations of natural convection and demonstrate the enhanced cooling

capability of the hybrid system, establishing liquid-channel integration as a practical solution for managing higher heat fluxes in compact thermal devices.

## II. LITERATURE REVIEW

Choi and Eastman [1] introduced nanofluids by dispersing nanoparticles into conventional heat-transfer fluids, showing that thermal conductivity could be significantly enhanced compared with base liquids. Their work laid the foundation for advanced cooling systems. Pak and Cho [2] experimentally studied aluminium oxide and titanium dioxide suspensions in water under turbulent flow. They demonstrated that nanoparticle concentration improves heat transfer but also increases viscosity and flow resistance, highlighting the balance between thermal and hydraulic performance. Lee et al. [3] measured the thermal conductivity of oxide nanofluids using the transient hot-wire method. Their results confirmed that even small nanoparticle fractions enhance conductivity, while particle dispersion and agglomeration strongly influence performance. Xuan and Li [4] investigated heat-transfer enhancement mechanisms in nanofluids, emphasizing the role of particle size, shape, and dispersion stability. Their findings provided a theoretical basis for later studies on nanofluids in channels and heat exchangers. Koblinski et al. [5] discussed nanoscale mechanisms such as Brownian motion, clustering, and liquid layering, showing that nanofluid heat transfer is more complex than conventional effective-medium models. This work highlighted the importance of particle–fluid interactions. Das et al. [6] examined the temperature dependence of nanofluid conductivity, showing that enhancement varies with operating temperature. This finding is important for systems where thermal loads fluctuate, as properties cannot be assumed constant. Buongiorno [7] developed a theoretical model for convective transport in nanofluids, identifying slip mechanisms such as Brownian diffusion and thermophoresis. His work became a key reference for CFD modelling of nanofluid flow and heat transfer.

## III. MODELING AND SIMULATION

### HEAT SINK ASSEMBLY PARTS

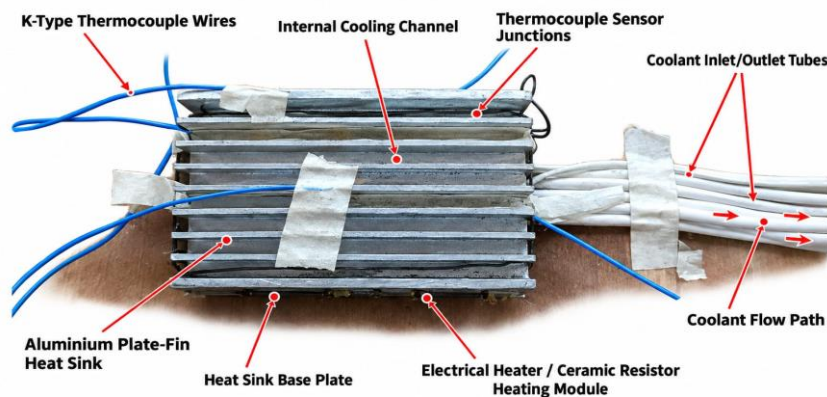


Fig.1 Experimental setup

Above Fig .1, shows the manual setup of thermocouples with heat sink which is having flat plate fins on its surface and the liquid is passed through micro channels inside the material at bottom of the surface by using pipes to carry out the fluid and thermos couples are indicate the temperature at different points of the heat sink surface like base, top, side of fins and entire body to measure the difference of temperature in entire body and different parts of the body (heat sink).

A 3D computational model was developed in CATIA and imported into ANSYS Fluent. High-resolution tetrahedral elements were used for discretization. The shear stress transport (SST)  $k-\omega$  turbulence model was applied, coupled with continuity, momentum, and energy conservation equations to simulate fluid–thermal interactions.

The experimental setup employed a precision-built plate-fin aluminium heat sink with fin length  $L=0.009$  m and single fin surface area  $A=6.00 \times 10^{-5}$  m<sup>2</sup>. Ceramic resistive heaters powered by a stabilized DC source delivered controlled thermal inputs of 10 W, 20 W, and 30 W. K-type thermocouples were mounted at the heat sink base, fin tips, and coolant inlet/outlet junctions to capture complete thermal profiles. Fig.2 & Fig.5.

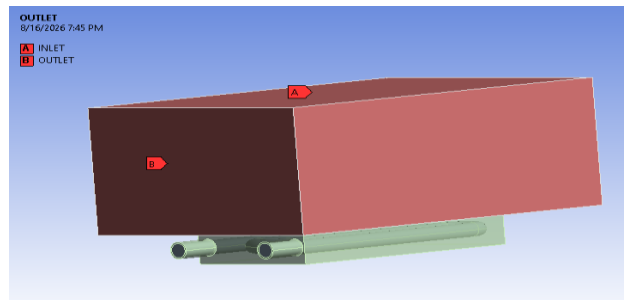


Fig .2 Boundary conditions for air domain

## Experimental Model and Data Processing

Two cooling methodologies were evaluated:

- Natural Convection Mode: Ambient air cooling relying solely on buoyancy-driven circulation across the fin arrays.
- Hybrid Liquid-Cooling Mode: Forced water circulation via a micro-centrifugal pump at a constant mass flow rate of  $\dot{m}=0.00006$  kg/s, with water specific heat capacity  $C_p=4184$  J/(kgK).

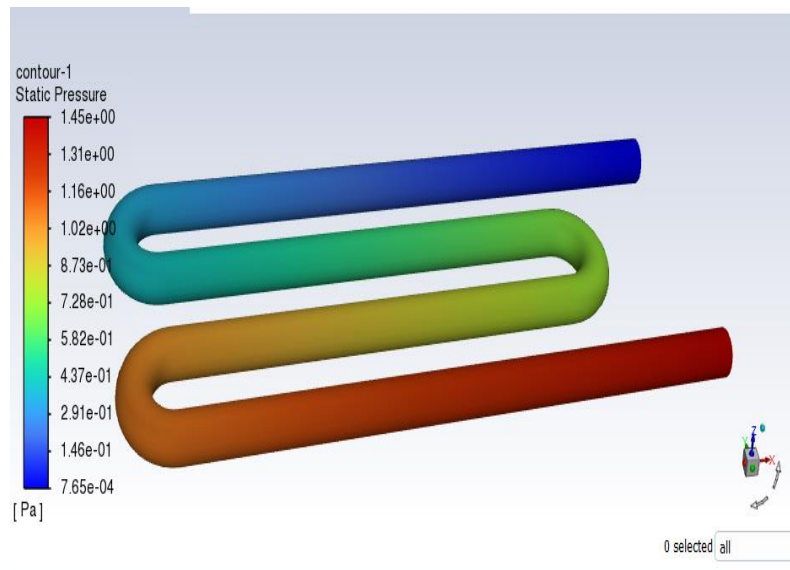


Fig .3 Pressure distribution in water passage

## For Governing Mathematical Equations

Coolant Heat Absorption ( $Q_{\text{water}}$ ):

$$Q_{\text{water}} = \dot{m} \cdot C_p (T_{\text{out}} - T_{\text{in}})$$

For the 30 W input test ( $T_{\text{in}}=30$  °C,  $T_{\text{out}}=74.23$  °C):

$$Q_{\text{water}} = 0.00006 \times 4184 \times (74.23 - 30) \approx 11.10 \text{ W}$$

Similarly, fluid heat gains were 3.40 W at 10 W input ( $T_{\text{out}}=43.54$  °C) and 7.30 W at 20 W input ( $T_{\text{out}}=59.07$  °C).

Fin Conduction Heat Transfer ( $Q_{\text{fin}}$ ):

$$Q_{\text{fin}} = k A (T_b - T_t) L$$

where  $k=167 \text{ W/(m K)}$ .

- Natural Convection (30 W):

$$Q_{\text{fin, single}} = 167 \times 6.00 \times 10^{-5} \times (93.02 - 91.82) 0.009 \approx 1.336 \text{ W per fin}$$

→ Total = 13.36 W (also 4.34 W at 10 W; 11.69 W at 20 W).

- Hybrid Cooling (30 W):

$$Q_{\text{fin, single}} = 167 \times 6.00 \times 10^{-5} \times (74 - 73.23) 0.009 \approx 0.857 \text{ W per fin}$$

→ Total = 8.57 W (also 2.89 W at 10 W; 5.46 W at 20 W).

## IV. RESULTS AND DISSUSSION

The comparative evaluation between conventional natural convection and the hybrid liquid-cooling layout highlights the significant performance gap in thermal regulation. Natural convection consistently exhibited high thermal resistance, while the integrated water-channel system demonstrated superior cooling efficiency across all tested loads.

### Natural Convection Evaluation

Operating under pure air cooling revealed severe limitations:

- Base Temperatures: 56.11 °C (10 W), 78.06 °C (20 W), 93.02 °C (30 W)
- Top Temperatures: 55.72 °C, 77.01 °C, 91.82 °C
- Fin Conduction:

$$Q_{\text{fin}} = 4.34 \text{ W (10 W), 11.69 W (20 W), 13.36 W (30 W)}$$

These elevated base temperatures confirm that passive air cooling cannot safely stabilize high thermal loads, especially at 30 W where the system approaches critical thermal limits is shown in fig.4.

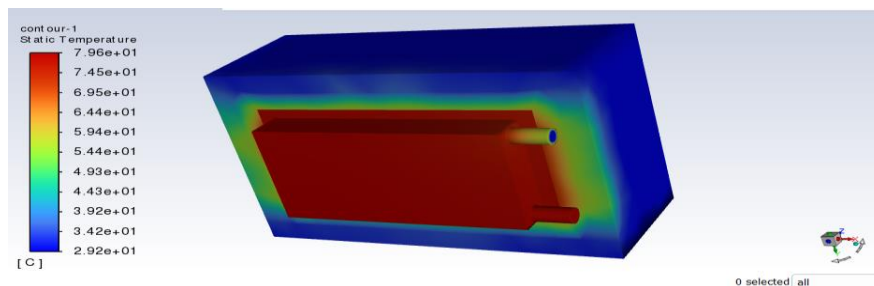


Fig .4 Temperature distribution for 30 W and water is a fluid

### Hybrid Water-Channel Evaluation

The U-shaped liquid channel drastically improved thermal regulation:

- At 10 W Input:
  - Base temperature: 43 °C (top: 42.74 °C)
  - Outlet temperature: 43.54 °C
  - Coolant absorption:  $Q_{\text{water}} \approx 3.40 \text{ W}$
  - Fin conduction: 2.89 W

- At 20 W Input:
  - Base temperature: 59 °C (top: 58.51 °C)
  - Outlet temperature: 59.07 °C
  - Coolant absorption:  $Q_{\text{water}} \approx 7.30$  W
  - Fin conduction: 5.46 W
- At 30 W Input:
  - Base temperature: 74 °C (top: 73.23 °C)
  - Outlet temperature: 74.23 °C
  - Coolant absorption:  $Q_{\text{water}} \approx 11.10$  W
  - Fin conduction: 8.57 W

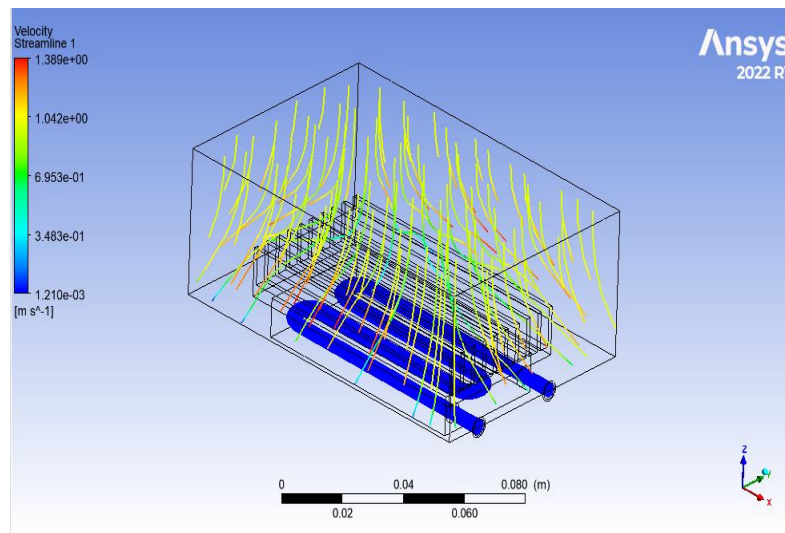


Fig .5 Air domain stream lines

## Discussion on System energy

The hybrid cooling system demonstrates a synergistic effect between forced liquid convection and fin conduction. At the maximum 30 W input, nearly two-thirds of the total energy is removed directly through fluid absorption and fin transfer. This dual mechanism ensures rapid heat removal from the source block, suppresses thermal resistance, and maintains safer operating temperatures compared to natural convection, is validated by above figure.5. The result graphs (fig.6,7&8), are validate that liquid-channel integration effectively bypasses the inherent limitations of passive air cooling, making it a robust solution for high-density thermal management applications and table 1,2,3

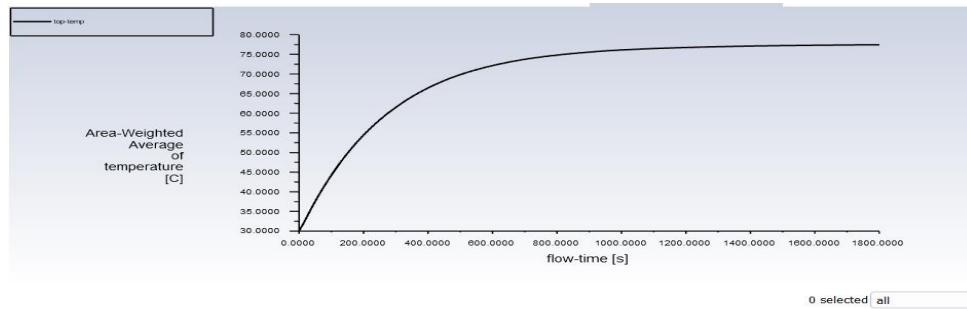


Fig .6 Temperature variation at the base centre of the heat sink

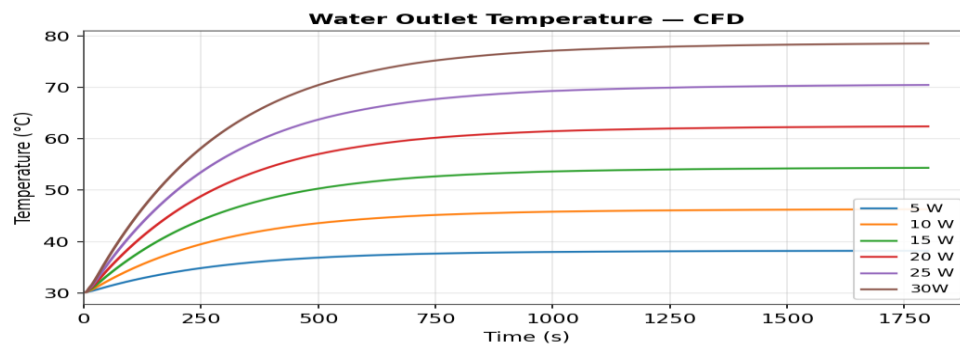


Fig .7 Water outlet temperature versus time for heat inputs of 5-30 W — CFD

Table 1 Results for natural convection through fins

| Heat input | Base temp in °C | Top temp in °C | Thermal conductivity (k) | Area of fin (A) | Length of fin (L) | Heat transfer rate ( $Q_{\text{per one fin}}$ ) | Heat transfer rate ( $Q_{\text{total}} (10 \times Q_1)$ ) |
|------------|-----------------|----------------|--------------------------|-----------------|-------------------|-------------------------------------------------|-----------------------------------------------------------|
| 10W        | 56.11           | 55.72          | 167                      | 6.00E-05        | 0.009             | 0.4342                                          | 4.34                                                      |
| 20W        | 78.06           | 77.01          | 167                      | 6.00E-05        | 0.009             | 1.169                                           | 11.69                                                     |
| 30W        | 93.02           | 91.82          | 167                      | 6.00E-05        | 0.009             | 1.336                                           | 13.36                                                     |

The study shows that as heat input increases from 10W to 30W, both base and top temperatures rise steadily. The temperature difference between base and top remains small, indicating effective conduction along the fin. Heat transfer rate per fin grows with input power, though the increase slows at higher levels. With ten fins, the total heat transfer scales proportionally, reaching up to 13.36W at 30W all this value shown in table 1.

Table 2: Results for Water cooling channel U shape.

| Heat input | Outlet temp (°C) | Mass flow rate (kg/s) | Specific heat (J/Kg K) | Heat gain by water (W) |
|------------|------------------|-----------------------|------------------------|------------------------|
| 10W        | 43.54            | 0.00006               | 4184                   | 3.40                   |
| 20W        | 59.07            | 0.00006               | 4184                   | 7.30                   |
| 30W        | 74.23            | 0.00006               | 4184                   | 11.10                  |

As the heat input rises from 10W to 30W, the outlet water temperature increases steadily from 43.54°C to 74.23°C. With a constant mass flow rate, the higher heat input directly enhances the energy absorbed by water. The heat gain grows

from 3.40W at 10W input to 11.10W at 30W, showing effective cooling performance. Overall, the water channel demonstrates strong capacity to absorb and transfer heat under increasing loads all this value shown in table 2.

Table3: Results for Heat sink.

| Heat input | Base temp (°C) | Top temp (°C) | Thermal conductivity (W /m k) | Single fin surface area (m <sup>2</sup> ) | Length of fin (m) | Heat gain per single fin (W) | Total Heat gain (W) |
|------------|----------------|---------------|-------------------------------|-------------------------------------------|-------------------|------------------------------|---------------------|
| 10W        | 43             | 42.74         | 167                           | 6.00E-05                                  | 0.009             | 0.289467                     | 2.89                |
| 20W        | 59             | 58.51         | 167                           | 6.00E-05                                  | 0.009             | 0.545533                     | 5.46                |
| 30W        | 74             | 73.23         | 167                           | 6.00E-05                                  | 0.009             | 0.857267                     | 8.57                |

As the heat input increases from 10W to 30W, the base and top temperatures rise gradually, showing effective thermal conduction through the fins. The temperature difference between base and top remains minimal, indicating efficient heat spreading. Heat gain per single fin grows steadily from 0.29W to 0.86W, while the total heat gain across all fins increases from 2.89W to 8.57W. Overall, the heat sink demonstrates strong performance in dissipating heat under higher loads all this value shown in table 3.

Table 4 Total heat transfer rate of hybrid model

| Heat transfer rate at U shape (Q1) | Heat transfer rate at fins (Q2) | Total Heat transfer rate $Q_{total} = (Q1+Q2)$ |
|------------------------------------|---------------------------------|------------------------------------------------|
| 3.40                               | 4.34                            | 7.74                                           |
| 7.30                               | 11.69                           | 18.99                                          |
| 11.10                              | 13.36                           | 24.46                                          |

The hybrid model combines heat transfer from the U-shaped channel and fins to enhance overall performance. As heat input increases, both Q1 and Q2 rise, leading to a higher total heat transfer rate. The combined effect shows significant improvement, with Qtotal reaching 24.46W at 30W input. This demonstrates that integrating fins with the cooling channel provides superior heat dissipation compared to individual methods all this value shown in table 4.

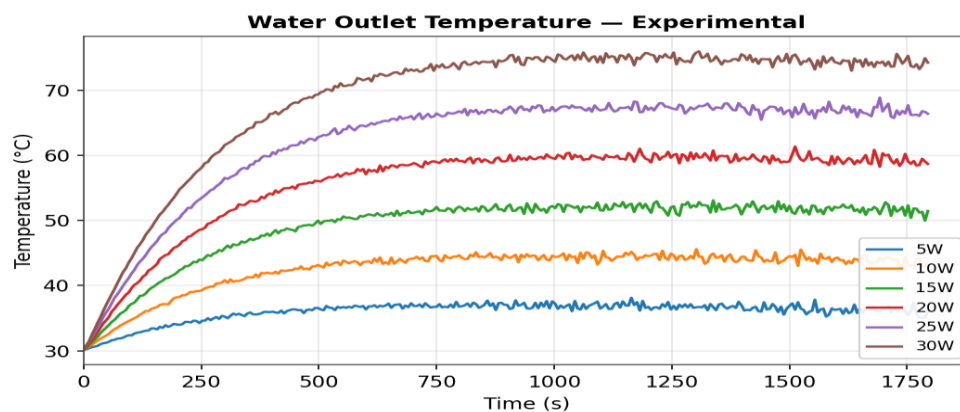


Fig .8 Water outlet temperature versus time for heat inputs of 5-30 W — Experimental

## V. CONCLUSION

This study investigated the performance advantages of transitioning from natural convection to a hybrid plate-fin and water-channel cooling system. Natural convection cooling proved inadequate for managing dense thermal loads, with base temperatures rising to 93.02°C at a 30 W input. In contrast, integrating a forced-flow water channel successfully reduced the 30 W base temperature to a safe operational limit of 74°C. Quantitative energy balance mapping at this load

demonstrated that the circulating water extracted 11.10 W while the aluminium fins contributed 8.57 W of conduction dissipation, thereby minimizing system thermal resistance. CFD simulation results verified close agreement with physical trials, with deviations limited to within 5–8%, confirming the accuracy of the numerical models. Ultimately, this work establishes hybrid liquid cooling as a superior thermal architecture compared to natural convection, laying a strong foundation for further enhancement using advanced nanofluids in subsequent research.

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