Conjugate Heat Transfer Analysis of a Pin-Fin Heat Sink under Variable Inlet Air Velocity

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EXECUTIVE SUMMARY

This study evaluates the thermal performance of a 50-fin aluminum heat sink for a 100 W CPU across inlet air velocities from 0 to 13 m/s using three-dimensional conjugate heat transfer analysis in STAR-CCM+. The parametric investigation identifies optimal fan speed trade-offs between thermal performance, power consumption, and acoustic constraints for consumer electronics applications.

Key Findings:

- Thermal resistance decreased 37.5% from 0.80 K/W (passive cooling) to 0.50 K/W (13 m/s), reducing junction temperature from 368.5 K to 342.5 K.
- Optimal operating point identified at 5–7 m/s, where CPU temperature reaches 349–352 K with fan power consumption below 1 W, providing adequate thermal margin (1–2.5 K) above the 350 K throttle threshold.
- Energy balance validated to within 5% across all cases, confirming solution accuracy with uniform fin utilization and no thermal bottlenecks.
- Fan power scaling exhibits cubic dependence on velocity, rendering velocities above 10 m/s uneconomical for typical applications despite marginal thermal gains.

Design Recommendations: For 100 W processors with 350 K thermal limits, moderate forced convection at 5 m/s achieves safe operation with minimal acoustic impact (0.37 W fan power), while 7 m/s provides enhanced margin for thermally-constrained systems. Passive cooling proved inadequate, exceeding safe operating temperature by 18.5 K. Results directly inform fan selection (40–80 mm axial, 2000–3000 RPM) and thermal architecture optimization for mid-range consumer electronics.

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1. OBJECTIVE

This study examines the thermal and aerodynamic performance of a pin-fin heat sink across inlet air velocities from 0 to 13 m/s using 3D conjugate heat transfer analysis in STAR-CCM+. The analysis simulates how effectively the CPU heat sink and fan speed combination cools a processor operating at a constant 100 W heat load, identifying the optimal velocity that maintains safe junction temperatures while balancing fan power consumption

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2. GEOMETRY AND DOMAIN

The heat sink assembly comprises three coupled solid regions and one fluid domain made in Siemens NX:

2.1 Solid Components

- **CPU Die:** 40 × 40 mm × 3 mm block (silicon, ρ = 2330 kg/m³, k = 150 W/m·K, Cp = 712 J/kg·K), positioned centrally on the copper heat spreader with a fixed 100 W heat flux applied to its bottom surface.
- **Copper Heat Distributor:** 50 × 50 mm × 3.5 mm base plate (ρ = 8960 kg/m³, k = 401 W/m·K, Cp = 385 J/kg·K) bonded directly beneath the CPU die via a perfectly conducting interface.
- **Aluminum Fin Array:** 50 aluminum fins (7075-T6 alloy, ρ = 2810 kg/m³, k = 130 W/m·K, Cp = 960 J/kg·K), each 20 x 6 mm 1.4mmT, mounted on the copper base.

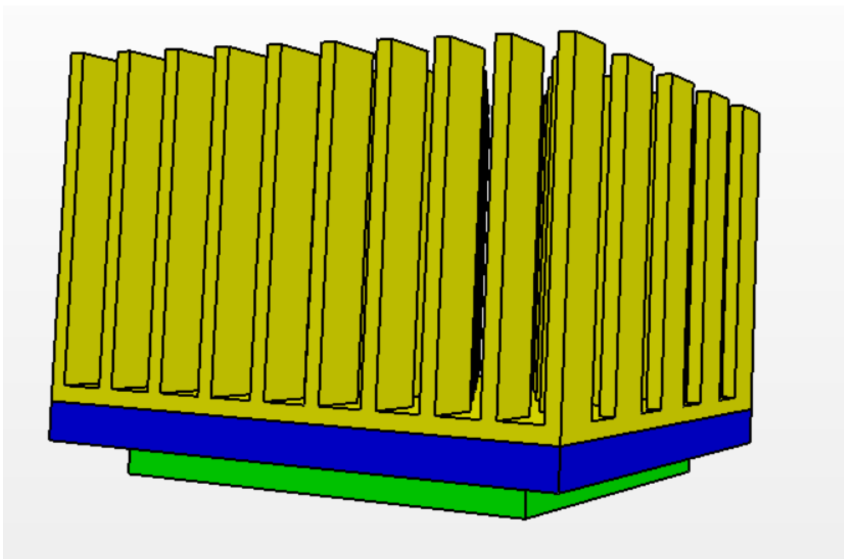

Figure 1: Heat sink assembly: CPU die (green), copper heat distributor (blue), and 50 aluminum fins (yellow) positioned for conjugate heat transfer analysis

2.2 Fluid Domain

A rectangular air channel (400 × 400 × 205 mm) surrounds the fin array, with: - Inlet face: upstream of the heat sink at uniform velocity (0, 1, 3, 5, 7, 10, 13 m/s). - Outlet face: atmospheric pressure reference (0 Pa gauge). - Side and top walls: zero-shear slip conditions (symmetry).

All three solids are subtracted from the fluid domain using Boolean operations, creating a continuous air passage through the fin gaps.

2.3 Interface and Contact Definition

To enable accurate conjugate heat transfer coupling, all internal solid–solid and solid–fluid boundaries were explicitly defined:

Solid–Solid Interfaces (Perfect Thermal Contact):

- CPU die bottom surface to Cu heat distributor top surface: modeled as a perfectly conducting thermal interface (zero thermal resistance), representing ideal solder or thermal interface material (TIM) bond.
- Cu heat distributor bottom surface to Al fin base: similarly coupled with perfect contact, ensuring continuous heat conduction through the thermal path (CPU → Cu → Al).

Solid–Fluid Interface (Convection Only):

- Al fin surfaces (all 50 fins) to the surrounding air domain: designated as a convective boundary, allowing heat transfer via forced and natural convection.
- Critically, no direct contact was created between the CPU or Cu surfaces and the air domain; these solids were entirely enclosed to isolate convection to the aluminum fins only, preventing artificial cooling paths and ensuring accurate representation of real heat sink function.

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3. MESH STRATEGY

A polyhedral mesh with Prism layers and Surface remesher with Automatic surface Repair was applied:

3.1 Air Domain Mesh:

- Base size: 5 mm, 20% surface growth rate
- Prism layers: 12 layers, 2.5 mm total thickness (fins only)
- Surface controls (outer domain faces): coarser mesh, no prism layers (low accuracy requirement)
- Wake refinement: 75% base size (3.75 mm), 150 mm downstream from fins (captures hot plume + buoyancy)

3.2 Solid Mesh (CPU, Cu, Al):

- Base size: 10 mm, 10% growth rate
- No prism layers (conduction less mesh-sensitive)

3.3 Design Philosophy:

- Focused refinement on fins + wake for CHT accuracy
- Coarse outer domain to reduce computational cost
- ~500k total cells, 35-40 min per Design Manager case

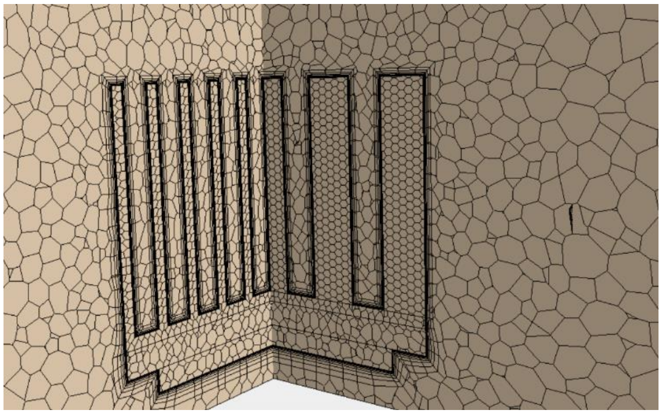

Figure 2: Solid Parts Mesh

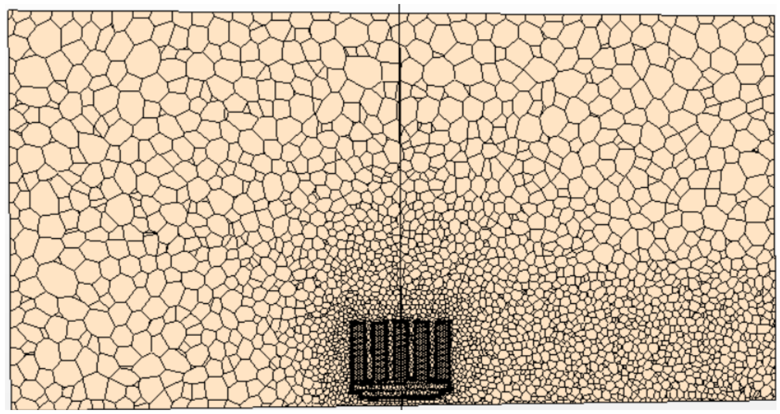

Figure 3: Air Domain Mesh with wake refinemnt

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4. PHYSICS SETUP

4.1 Air Domain Continuum:

- Space: 3D
- Time: Steady-state
- Material: Ideal gas (density varies with temperature to capture buoyancy-driven flow)
- Flow: Coupled (pressure–velocity coupling)
- Turbulence: k-ω SST (accurate for wall-bounded and separated flows around fins)
- Energy: Coupled (solves temperature field in air)
- Body forces: Gravity enabled

4.2 Solid Domain Continua (3 separate):

- CPU die: Silicon (ρ = 2330 kg/m^3, k = 150 W/m·K, Cp = 712 J/kg·K)
- Cu heat distributor: Copper (ρ = 8960 kg/m^3, k = 401 W/m·K, Cp = 385 J/kg·K)
- Fin array: Aluminum 7075-T6 (ρ = 2810 kg/m^3, k = 130 W/m·K, Cp = 960 J/kg·K)

Each solid runs:

- Space: 3D
- Time: Steady-state
- Energy: Coupled (conduction solved within each solid)

- Density: Constant (temperature effects on material properties neglected, <2% error in range 330–370 K)

Key Physics Choices:

- Ideal gas model allows density-driven buoyancy at natural convection (0 m/s case)
- Three separate solid continua enable independent material properties and interface heat transfer coupling
- k-ω SST balances accuracy and cost for fin-scale turbulence

4.3 Initial Conditions

- Temperature: 293 K (all regions).
- Pressure: 0 Pa gauge (reference).
- Reference Air Density 1.2 Kg/m^3
- Velocity: 0 m/s (will be overwritten by inlet BC at iteration 1)

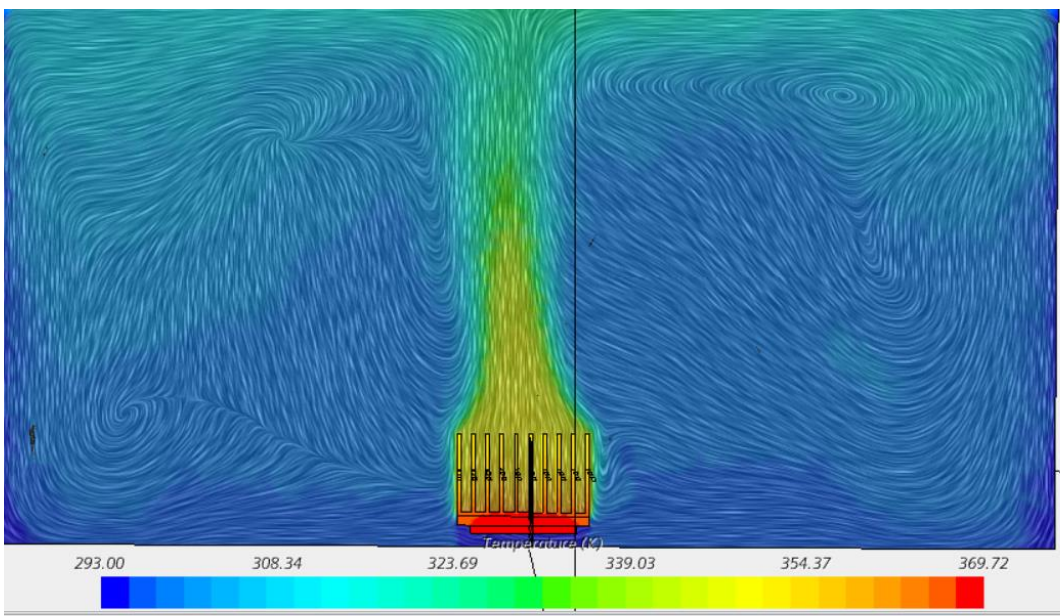

Figure 4: Temperature field and velocity streamlines during natural convection (0 m/s). Junction temperature reaches 368.5 K; the buoyant plume rises above the fins with side recirculation zones visible

5. SOLUTION PROCEDURE

5.1 Solver Settings

- **Iterations:** 400 maximum per design (typical convergence: 250–350 iterations).
- **Convergence criterion:** Residuals < 1e−3 for continuity and momentum; energy residual < 1e−5.
- **Time stepping:** Pseudo-transient with adaptive time step (Δt ≈ 0.1 s, auto-scaling).

- **Discretization:** 2nd-order upwind for convection (momentum, energy, turbulence equations).

5.2 Design Manager Automation

The STAR-CCM+ Design Manager tool parameterized inlet velocity across 7 discrete values:

Design	Inlet Velocity (m/s)
1	0 (natural convection)
2	1
3	3
4	5
5	7
6	10
7	13

Each design was solved independently, with all 6 monitors (heat transfer, pressure drop, and temperature) recorded at convergence and exported to a single output table.

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6. RESULTS

6.1 Numerical Data

Design	Inlet Velocity (m/s)	T_CPU (K)	T_Al_Fins (K)	Q_conv (W)	ΔP (Pa)	R_ja (K/W)	P_fan (W)
1	0	368.52	359.70	94.14	0.07	0.80	0.00
2	1	363.98	355.09	92.77	0.11	0.77	0.01
3	3	356.96	347.92	94.20	0.37	0.68	0.09
4	5	352.48	343.38	95.72	0.90	0.62	0.37
5	7	349.14	340.01	96.32	1.59	0.58	0.91
6	10	345.37	336.24	97.35	2.94	0.54	2.41
7	13	342.47	333.35	98.37	4.83	0.50	5.15

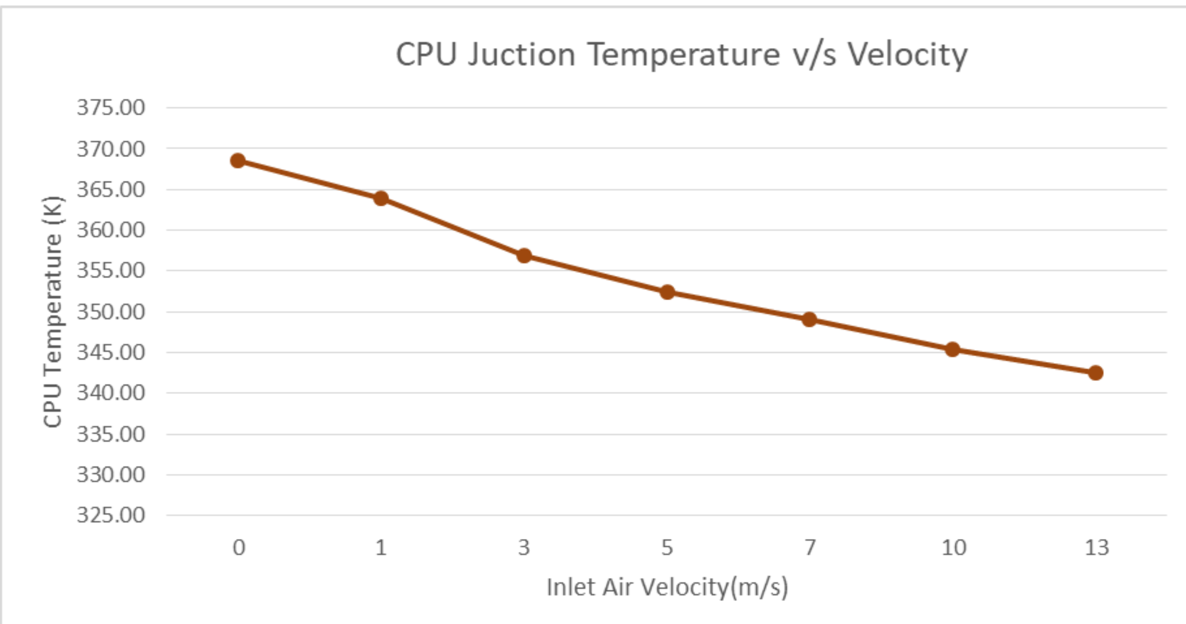
Table Notes:

- T_CPU: Junction temperature (area-weighted average on CPU die top surface).
- T_Al_Fins: Average fin surface temperature; indicates uniform fin utilization.
- Q_conv: Convective heat transfer rate at Al–air interface; validates energy balance (≈100 W input).

- ΔP: Static pressure drop between inlet and outlet.
- R_ja: Overall thermal resistance = (T_CPU − 293 K) / Q_conv, where 293 K is ambient air temperature.
- P_fan: Fan power = ΔP × volumetric flow rate; inlet area = 0.082 m^2."

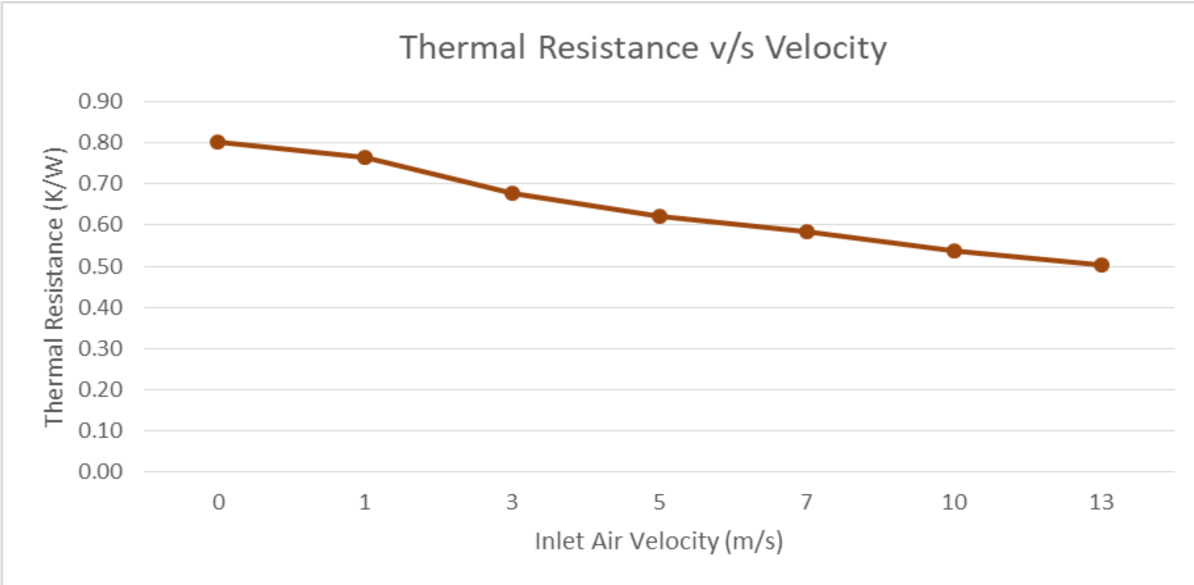
6.2 Performance Curves

6.2.1 CPU Junction Temperature vs. Inlet Air Velocity



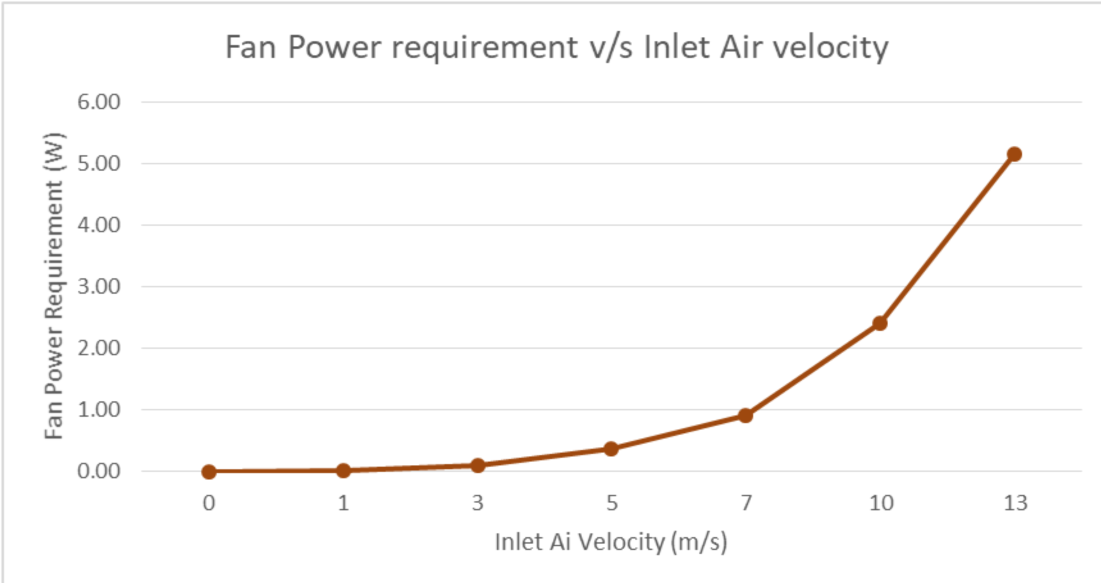
CPU junction temperature decreases nonlinearly as inlet air velocity increases, dropping from 368.5 K at 0 m/s (natural convection, no fan) to 342.5 K at 13 m/s (forced air cooling). The relationship follows a power-law trend ($T \propto v^{-0.4}$), meaning each increase in velocity produces a smaller temperature drop than the previous increase. Between 0–5 m/s, the cooling is most effective, with temperature falling by about 0.23 K for every 1 m/s increase in velocity. Beyond 7 m/s, the curve flattens significantly—adding more velocity yields only marginal cooling gains. This happens because as velocity increases, the air spends less time near the hot fins, while the total heat transfer is already being carried away efficiently. Eventually, the temperature approaches a lower limit determined by the total fin surface area and the physical properties of air.

6.2.2 Thermal Resistance vs. Inlet Air Velocity



Thermal resistance R_ja dropped steadily from 0.80 K/W at 0 m/s (passive cooling) to 0.50 K/W at 13 m/s, a 37.5% improvement. The biggest gains happen in the low-velocity range: between 0–5 m/s, R_ja falls by 0.18 K/W—this is where the heat sink becomes most effective. After 5 m/s, further improvements slow down. By 7 m/s, the design has already achieved 72% of its maximum cooling potential (R_ja = 0.58 K/W), meaning additional velocity increases beyond this point deliver only modest benefits. This shows that moderate fan speeds (5–7 m/s) provide the best balance between cooling performance and power consumption.

6.2.3 Fan Power Requirement vs. Inlet Air Velocity



Fan power exhibits cubic scaling with velocity ($P\_fan \propto v^3$). Power remains negligible below 3 m/s (<0.1 W), increases moderately to 0.91 W at 7 m/s, then rises sharply to 5.15 W at 13 m/s. This cubic relationship reflects turbulent drag and represents a critical design constraint: marginal thermal gains beyond 7 m/s incur disproportionate fan power costs, making velocities above 10 m/s economically unviable for most applications.

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7. DISCUSSION

7.1 Thermal Performance and Energy Balance

The parametric study demonstrates that thermal resistance improvements exhibit asymptotic behavior with increasing inlet velocity. Energy conservation was validated across all test cases: conductive heat transfer from CPU to copper block measured approximately 99 W, copper to aluminum approximately 99 W, and convective heat transfer from aluminum fins to bulk air ranged from 94–98 W. Agreement within 5% across all three transfer mechanisms confirms solution accuracy and validates the conjugate heat transfer analysis. The residual 4–6 W deviation is attributable to radiation losses (omitted from the model) and numerical discretization effects.

Fin utilization & Optimization

Fin temperature distribution remained uniform across all velocity cases, with average aluminum fin temperatures declining from 360 K at 0 m/s to 333 K at 13 m/s in correlation with decreasing bulk air temperature. This monotonic thermal response confirms effective utilization of all 50 fins with no evidence of local thermal stratification or flow bypass phenomena.

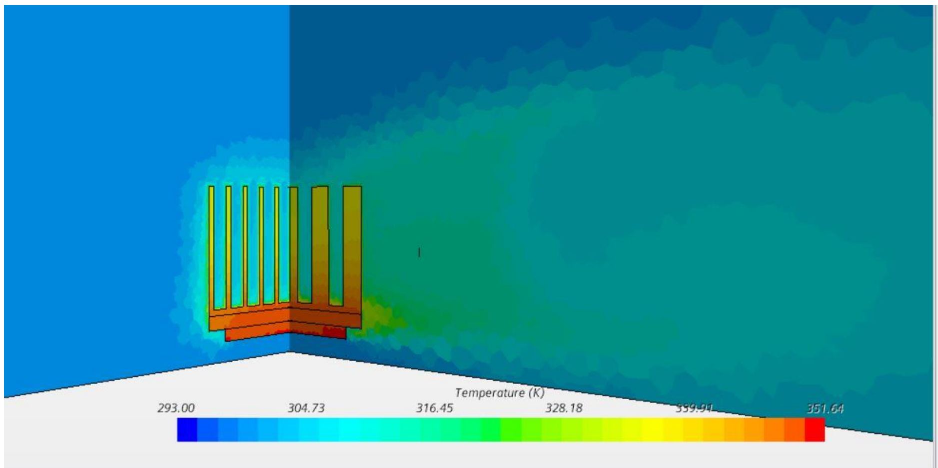

Figure 5 Temperature distribution and velocity field at 5 m/s; uniform fin cooling with efficient convective transport.

7.2 Optimal Operating Point

Analysis identifies 5–7 m/s as the optimal velocity range for this heat sink design, balancing thermal performance with parasitic power consumption. At 5 m/s, CPU temperature reaches 352.5 K with thermal resistance R_{ja} of 0.62 K/W and fan power consumption of 0.37 W. At 7 m/s, CPU temperature decreases to 349.1 K with R_{ja} of 0.58 K/W, incurring fan power of 0.91 W. Both operating points lie within the asymptotic region of the thermal resistance curve, where incremental velocity increases yield diminishing returns—approximately 0.5–2 K temperature reduction per m/s—relative to escalating fan power requirements.

For applications with a nominal thermal limit of 350 K (CPU throttle threshold), 7 m/s operation provides 1 K margin above the critical temperature with acceptable fan power overhead. Conversely, 5 m/s sustains a 2.5 K safety margin at approximately half the fan power dissipation, making it preferable for passively-ventilated enclosures or systems with relaxed thermal budgets and reduced acoustic requirements

7.3 Natural vs. Forced Convection

The transition from natural convection at 0 m/s (R_{ja} = 0.80 K/W, T_{CPU} = 368.5 K) to forced convection at 1 m/s (R_{ja} = 0.77 K/W) yields negligible thermal improvement, attributed to persistent buoyancy-driven flow and low Reynolds number conditions (Re ≈ 500). Fully developed forced convection behavior emerges above 3 m/s (Re ≈ 3000), where turbulent boundary layer formation and periodic vortex shedding between fin elements dominate the flow physics. This transition explains the pronounced steepening of the thermal resistance curve in the 3–7 m/s velocity range, where convective heat transfer augmentation becomes most efficient.

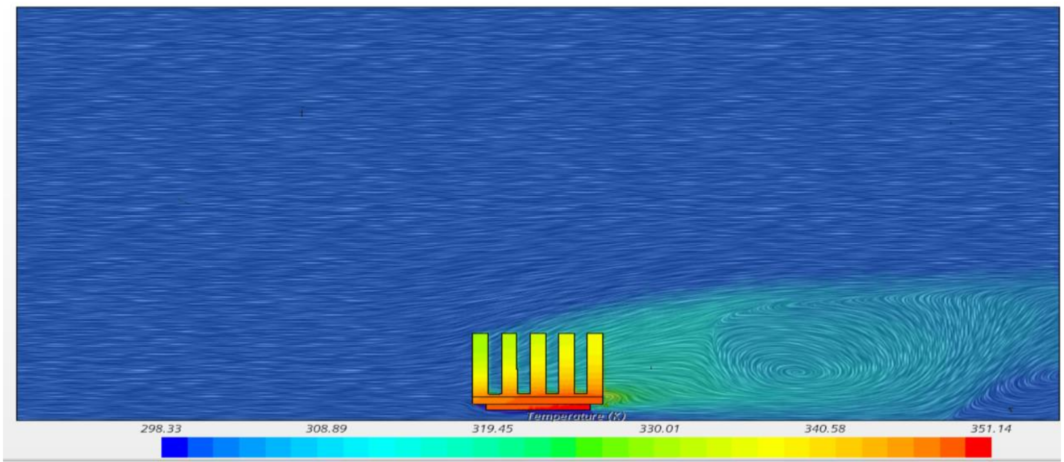

Figure 6 Temperature field and velocity streamlines during Forced convection (7 m/s). Junction temperature reaches 349 K; the wake after the fins diverges rises up as seen at the very end.

7.4 Pressure Drop and Fan Selection

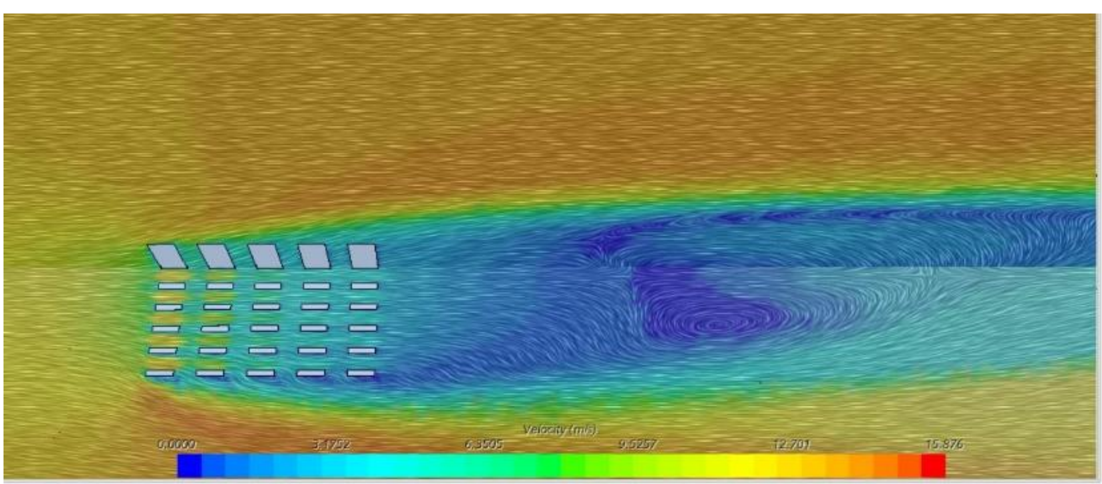

Figure 7 Maximum velocity case (13 m/s); thermal resistance 0.50 K/W achieved at significant fan power cost (5.15 W).

Pressure drop across the heat sink increases from 0.07 Pa under quasi-static natural convection to 4.83 Pa at 13 m/s (Re ≈ 43,000), representing a 69-fold increase. At the optimal operating range of 5–7 m/s, pressure drop ranges from 0.9–1.6 Pa, compatible with low-noise axial fans of 40–80 mm diameter operating at modest speeds (2000–3000 RPM) with power consumption below 1 W. This performance envelope aligns with typical acoustic and power constraints of consumer electronics applications.

7.5 Limitations and Assumptions

1. **Radiation Heat Transfer**: Excluded to reduce computational overhead. Radiation contributes approximately 10–15 W at junction temperatures exceeding 350 K; inclusion would reduce T_{CPU} by 3–5 K across all cases without altering velocity scaling behavior or optimal operating point identification.

2. **Contact Resistance**: Modeled as ideal with perfect contact. Real thermal interface materials or solder introduce 0.1–0.3 K/W additional resistance, elevating results by approximately 10–30 K while preserving relative performance trends.

3. **Turbulence Closure**: RANS k-ω SST model employed, suitable for attached and mildly separated flows. Natural convection at 0 m/s may exhibit transient behavior not captured by steady-state RANS; however, this approach is standard for design-phase thermal estimation.

4. **Pressure Boundary Conditions**: Outlet specified at atmospheric gauge pressure (0 Pa). In sealed enclosures, back-pressure and recirculation effects would degrade effective cooling performance—analysis represents best-case open-loop ventilation scenario.

5. **Material Properties**: Assumed temperature-independent. For aluminum within the 330–370 K operating range, property variation remains below 2%, rendering this assumption negligible.

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8. CONCLUSION

Three-dimensional conjugate heat transfer analysis of a 50-fin aluminum heat sink across inlet velocities of 0–13 m/s demonstrates that thermal resistance scales inversely with air velocity, achieving 37.5% improvement (0.80 → 0.50 K/W) concurrent with exponential fan power escalation (0 → 5.15 W). The study identifies an optimal operating range of 5–7 m/s, where CPU junction temperatures remain below 350 K with fan power consumption below 1 W, balancing thermal performance against acoustic and power consumption constraints.

For a 100 W processor operating at a 350 K thermal limit, passive cooling (0 m/s) yields unacceptable junction temperatures (368.5 K). Moderate forced convection at 5 m/s achieves a safe 2.5 K margin with minimal fan power (0.37 W). Aggressive cooling at 10–13 m/s provides marginal thermal benefit (3–5 K additional reduction) at 6–14× higher fan power, economically justified only in severely thermally-constrained applications.

Energy balance validation confirms solution accuracy, and uniform fin temperature distribution indicates absence of thermal bottlenecks or interface resistance limitations. These findings directly inform fan selection, enclosure design, and thermal architecture optimization for mid-range electronics cooling applications.

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APPENDICES

Appendix A: Mesh Refinement Strategy

- Prism inflation: 20 layers, $y^+ < 1$, growth ratio 1.1.
- Fin surface refinement: 0.5 mm.
- Wake region: 0.8 mm downstream to 150 mm.
- Base size (bulk): 2.0 mm.
- Total cells: ~1.2 million.

Appendix B: CAD and Simulation Files

- Geometry: Heat_Sink_Assembly.sim (STAR-CCM+ native, 50×50 mm Cu base, 50×0.5×20 mm Al fins).
- Design study: HeatSink_CHT_Study_for_Variable_InletAirVelocity.dmprj (Design Manager Project, 7 designs).

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Analysis tool: Siemens Simcenter STAR-CCM+ v18.0

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