TECHNICAL CASE STUDY
Natural Convection · Radiator Feasibility

Fanless Radiator Designed for Extreme Ambient Conditions

At a Glance

Advanced Thermal Solutions, Inc. (ATS) conducted a first-principles analytical study on the feasibility of a custom fanless radiator designed to dissipate 400 W to ambient without forced convection. The work evaluated natural convection performance at ambient conditions up to 50°C and compared radiator geometries to identify a more effective passive architecture.

CUSTOMER OVERVIEW

The customer needed a custom liquid cooling radiator capable of rejecting 400 W to ambient with no rotating fans. The system was intended for extreme outdoor conditions, including desert operation up to 50°C ambient, making passive heat rejection the governing design constraint.

- Fanless radiator with no forced convection
- Operating ambient range from -30°C to 50°C
- Liquid inlet/outlet temperatures of 58°C and 55°C

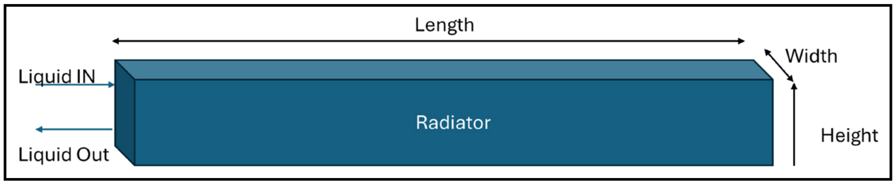

Figure 1.
A Liquid Cooling Radiator Could Not Use Forced Convection Via Rotating Fans. Calculations Helped Determine a More Effective Air-Cooled Heat Sink.

CHALLENGE

The design needed to reject 400 W by natural convection alone while remaining practical in size. The original geometry target favored a tall, narrow radiator (Figure 1), but the key question was whether that approach could achieve the required heat extraction under 50°C ambient conditions.

Design requirements:

- Heat load: 400 W
- Ambient range: -30°C to 50°C
- Liquid inlet/outlet: 58°C / 55°C
- No rotating fans allowed
- Initial geometry target: height ≤ 300 mm, width ≤ 100 mm

The central challenge was determining whether passive buoyancy-driven airflow could support the required thermal duty, and which geometry would make natural convection most effective.

METHODOLOGY

ATS used first-principles heat transfer analysis to calculate coolant flow, air volumetric flow, air-side and liquid-side heat transfer coefficients, overall heat transfer coefficient, and heat exchanger effectiveness for a fanless radiator architecture.

Analytical Investigation Areas

- Calculated fluid mass flow rate from known heat load and liquid temperatures
- Calculated natural-convection air volumetric flow through the fin channels
- Evaluated air-side and liquid-side heat transfer coefficients
- Calculated overall heat transfer coefficient and NTU-based effectiveness
- Determined required heat sink length for 400 W power extraction

Calculating Fluid Mass Flow Rate

Because the heat load (400 W) and inlet and outlet fluid temperatures are known, the fluid mass flow rate can be calculated:

$$Q = m.C_p.\Delta T$$

$$Q = 400W$$

$$\Delta T = 58\text{-}55 = 3°C$$

$$m = 0.0333 \text{ kg/sec}$$

Calculating Air Volumetric Flow Rate

Velocity was calculated from:

$$U \approx \sqrt{g\,\beta\,\Delta T\,H}$$

$\beta = 1/T_{surface}$

$T_{surface} = (58+55)/2 = 56.5\ °C = 329.5\ K$

$H =$ fin gap

$\Delta T = T_{surface} - T_{ambient}$

Volume flow rate =

U x Fin pitch x width x (number of fins -1)

Calculating Air Side Heat Transfer Coefficient

$$Gr = \frac{g\,\beta\,\Delta T\,L_c^3}{\nu^2}, \quad Ra = Gr \cdot Pr.$$

$$\mathrm{Nu}_L = \left[0.825 + \frac{0.387 Ra^{1/6}}{(1 + (0.492/Pr)^{9/16})^{8/27}}\right]^2$$

$$h = \frac{\mathrm{Nu}_L k}{L_c}.$$

Gr = Grashof number
Ra = Railey number
Nu = Nusselt number
Pr= Prandtl number
h = heat transfer coefficient
K =air thermal conductivity
L_c = plate height

Calculating Liquid Side Heat Transfer Coefficient

$$Nu = 0.023\,Re^{0.8}\,Pr^n$$

hl = K.NU /D

D = tube inside diameter
K = Liquid thermal conductivity
hl= liquid heat transfer coefficient
Re = Reynolds number
Pr = Prandtl number

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Calculating the Overall Heat Transfer Coefficient

$$C_{air} = \dot{m}_{Air} C_{PAir}$$

$$C_{Wat} = \dot{m}_{Wat} C_{PWat}$$

$$C_{\min} = \min(C_{Air}, C_{\max})$$

$$C_r = \frac{C_{\min}}{C_{\max}}$$

$$NTU = \frac{h_{Air} A_{Air}}{C_{\min}}$$

Overall heat transfer coefficient

$$U = \frac{1}{\frac{Aa}{Aw} * \frac{1}{hw} + 1/ha}$$

Aa = air side total surface area
Aw = Liquid side total surface area
Liquid = EG50%

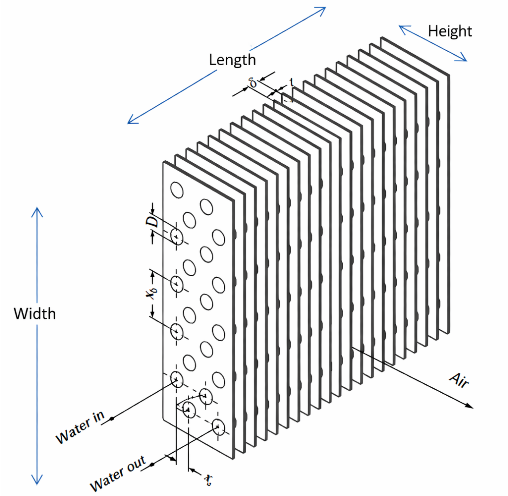
Calculating the Heat Exchanger Heat Transfer Effectiveness

$$\varepsilon = 1 - \exp\left[(1/C_r)(NTU)^{0.22} \left\{ \exp[-C_r (NTU)^{0.78}] - 1 \right\} \right]$$

$$q_{\max} = \frac{q}{\varepsilon}$$

$$Q_{max} = C_{min}(T_{inlet} - T_{ambient})$$

$$Q = \epsilon \times Q_{max}$$



Original Radiator Design

METHODOLOGY

Geometry and Spacing Study

- Natural convection fin spacing study identified 6 mm as the best overall spacing
- Compared tall, narrow geometry against short, wide geometry
- Tested required length as a function of width and height at 50°C ambient

This approach quantified the physical feasibility limits of passive radiator sizing before moving into CFD or detailed mechanical design.

SOLUTION

ATS identified that a wider, shorter radiator geometry was more effective than the customer’s original tall, narrow concept for natural convection heat rejection. The analysis showed that geometry had a strong effect on buoyancy-driven airflow and required surface area (Figure 2 & 3).

- Recommended 6 mm fin spacing for best overall natural convection performance
- Showed that a 300 mm width and 100 mm height outperformed a 100 mm width and 300 mm height concept
- Quantified required radiator length for each geometry under 50°C ambient conditions
- Provided a first-principles basis for passive radiator feasibility and architecture selection

This redirected the design from a less effective tall, narrow concept toward a geometry better aligned with buoyancy-driven flow.

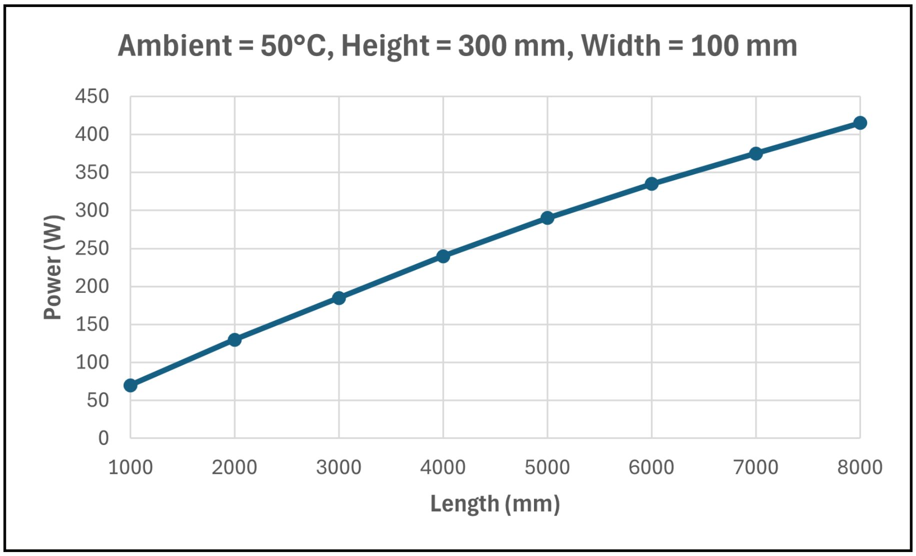


Figure 2.
Graph Shows a 7.5 m Heat Sink with a Height of 300 mm is Required to Extract 400 W of Heat

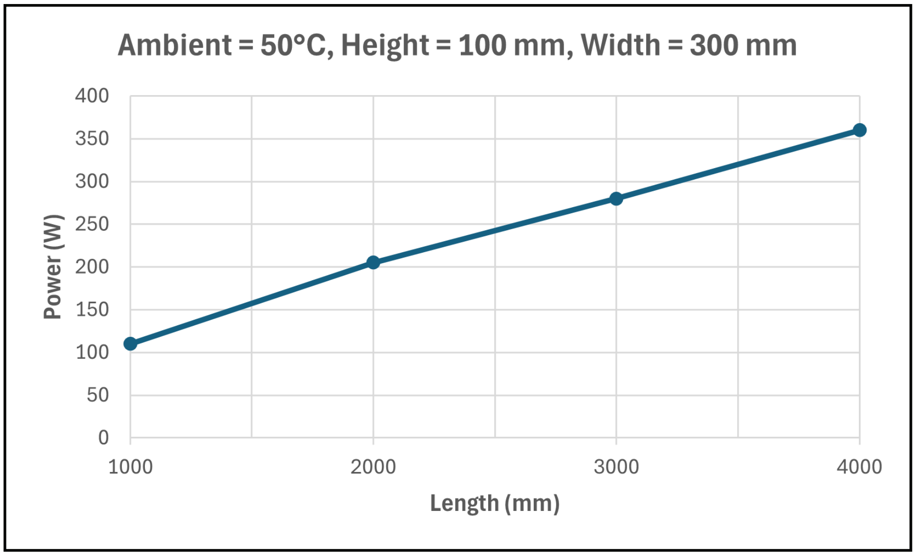


Figure 3.
Graph Shows a 4.0 m Heat Sink with a Height of 100 mm is Required to Extract 400 W of Heat

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RESULTS & DATA

The study showed that natural convection strongly limited heat rejection at the worst-case ambient condition, and that geometry choice had a major impact on required radiator length.

Key findings:

- Natural convection severely limits heat rejection at 50°C ambient
- Original 100 mm × 300 mm geometry was impractical, requiring about 8 m of length
- Wider 300 mm × 100 mm geometry improved buoyancy-driven airflow
- Shorter, wider geometry reduced required length to about 4 m
- Ambient temperature strongly drove required surface area

The analysis established that passive radiator feasibility was governed not just by total surface area, but by how geometry influenced airflow development under natural convection.

Thermal engineering outcomes:

- Calculated mass flow and air-side flow behavior for a fanless radiator
- Applied NTU-based heat exchanger sizing under extreme ambient conditions
- Completed a parametric geometry study comparing height and width tradeoffs
- Defined physical feasibility limits before CFD phase
- Provided data-driven architectural recommendations

ANALYSIS & CONCLUSION

This study shows that designing a fanless radiator for extreme ambient conditions requires more than maximizing area within an arbitrary footprint. Under natural convection, geometry has a first-order effect on airflow development, heat transfer coefficient, and practical feasibility.

- Natural convection was the dominant design constraint at 50°C ambient
- The original tall, narrow concept was not physically practical for the required heat load
- A wider, shorter radiator improved buoyancy-driven airflow and reduced required length
- Analytical modeling established feasibility limits before further design investment

ATS delivered a first-principles feasibility study that clarified the limits of passive radiator performance and identified a more effective geometry for fanless heat rejection in extreme conditions.

Take control of your thermal performance with expert analysis and design services, contact ATS to speak with our engineers and start optimizing your system today.