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Better Pin-Fin Cooling for Power Electronics Engineers

Mapping the trade between thermal resistance and pressure drop for pin-fin cold plates shows that a better pin field beats more flow.

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Method
Correlation-based cold-plate model with design sweep

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Abstract

Pin-fin cold plates for power modules trade thermal resistance against pressure drop. We map that trade for a representative 100 × 50 mm copper pin plate carrying 1.5 kW with 50/50 glycol at 8 L/min, using the Žukauskas heat transfer and Gaddis–Gnielinski pressure-drop correlations, fin efficiency and effectiveness analysis across 136 inline and staggered designs. Model components are checked against a finite-difference fin solution and a marching energy balance.

The baseline inline plate (3 mm pins at 6 mm pitch, 6 mm tall) has 11.3 K/kW at 1.43 kPa. Doubling flow lowers this to 7.6 K/kW, while spending the same pressure drop on a denser inline field at the original flow reaches 5.9 K/kW. Staggered pins transfer slightly more heat at equal geometry but need about twice the pressure drop, and at equal pressure drop inline layouts match or beat them.

Keywords: pin fin, cold plate, power electronics, pressure drop, thermal resistance, liquid cooling

1. Introduction

Liquid-cooled power modules in traction inverters, chargers and drives commonly use a baseplate with an integrated pin-fin array. Pin diameter, pitch, height and arrangement set both the thermal resistance and the pressure drop the coolant loop must supply, and the two pull in opposite directions. Teams that need a cooler module often reach for more flow. This paper maps the trade for a representative cold plate and compares the gain from a better pin field with the gain from more flow.

2. Model and Method

The cold plate (Figure 1) is 100 mm long in the flow direction and 50 mm wide, with copper pins (k = 390 W/(m·K)) on a 3 mm copper base carrying 1.5 kW from a power module. Coolant is 50/50 ethylene glycol and water entering at 65 °C at 8 L/min, with representative properties (ρ = 1040 kg/m³, c_p = 3500 J/(kg·K), k = 0.40 W/(m·K), μ = 1.4 mPa·s, Pr = 12.2) [5]. Pins span the channel height H and the cover is treated as adiabatic.

The pin-side heat transfer coefficient follows the Žukauskas correlation for tube banks in crossflow, $Nu = C\,Re_{max}^{\,m}\,Pr^{0.36}$ with row correction [1], [2], where Re_{max} uses the velocity in the narrowest cross-section. Pins are fins with adiabatic tips, η = tanh(mH)/(mH), m = √(4h/(kd)) [1]. With an isothermal base the plate is a heat exchanger of effectiveness

$$\varepsilon = 1 - \exp(-UA/(\dot{m}c_p)), \quad UA = h(A_{base} + \eta A_{pins}), \quad T_{base} = T_{in} + Q/(\varepsilon \dot{m} c_p) \tag{1}$$

[1], and the thermal resistance reported is $(T_{base} - T_{in})/Q$. Pressure drop follows the Gaddis–Gnielinski correlation for tube bundles, which blends laminar and turbulent contributions and covers inline and staggered layouts [3], [4]. Designs are swept over pin diameter 1.0 to 3.0 mm, pitch 1.5 to 3.0 diameters (square pitch; staggered rows offset by half a pitch), height 3 to 8 mm, and both layouts, limited to height-to-diameter ratios of 4 or less for manufacturability (an assumption), giving 136 designs. The baseline is a common inline layout: 3.0 mm pins at 6.0 mm pitch, 6 mm tall.

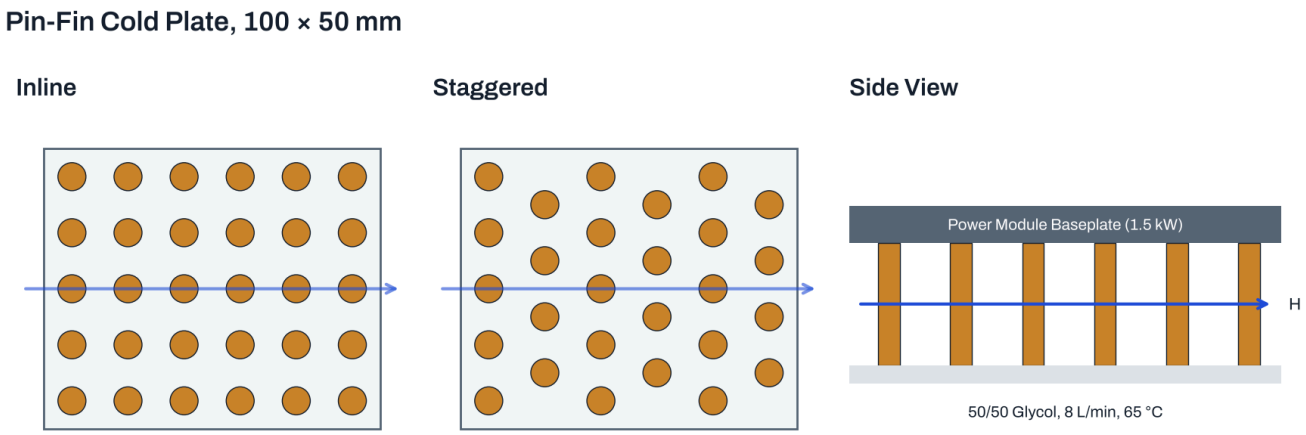


Figure 1. Pin-fin cold plate: inline and staggered pin layouts (top views, flow left to right) and side view with the module baseplate above and the channel cover below.

3. Verification

Check	Reference	Result
Pin fin efficiency	Finite-difference solution of the 1D fin equation	0.6067 against 0.6067
Effectiveness relation	Marching integration of the coolant energy balance along the plate	1500.2 W recovered of 1500 W
Pressure-drop equations	Published form of the Gaddis–Gnielinski correlation [6]	Implemented term by term; per-row coefficient at pitch 2d and Re 1,000: inline 0.21, staggered 0.47, the same order as the Žukauskas charts in [1]

Table 1. Verification of the cold-plate model.

4. Results

4.1 Baseline Plate

The baseline plate has a thermal resistance of 11.3 K/kW (16.9 K at 1.5 kW) and a pressure drop of 1.43 kPa. Its pins run at Re_{max} = 1981 with fin efficiency 0.71. Of the 16.9 K, only 3.1 K is the coolant's own temperature rise; the rest is the limited conductance of the pin field (ε = 0.18).

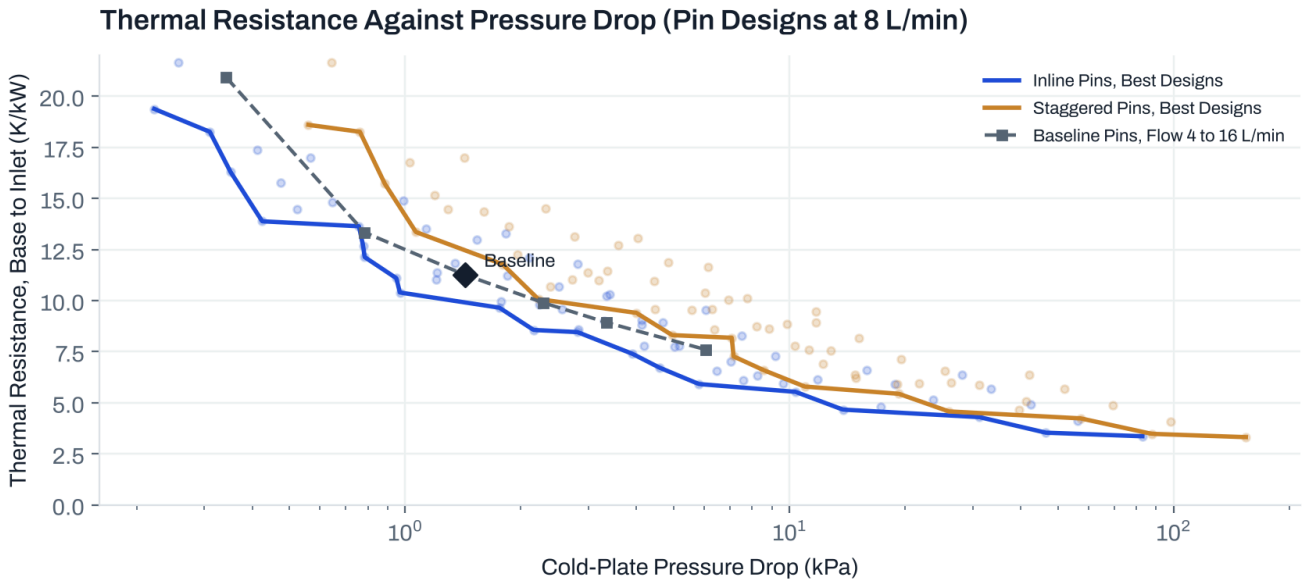


Figure 2. Thermal resistance against pressure drop for all 136 pin designs at 8 L/min (dots), with the best designs at each pressure drop for inline and staggered pins (solid). Dashed: the baseline pin field with flow raised from 4 to 16 L/min.

4.2 Thermal Resistance Against Pressure Drop

Figure 2 shows the full trade. At the baseline's pressure drop, the best design (inline, 2.5 mm pins at 2.0d pitch, 8 mm tall) lowers resistance to 10.4 K/kW; alternatively, a 2.0 mm, 2.5d design matches the baseline resistance at 0.95 kPa, 34 % less pressure drop. The larger gains come from spending pressure on the pin field. Doubling flow to 16 L/min with the baseline pins gives 7.6 K/kW at 6.1 kPa; at the same or lower pressure drop and the original 8 L/min, a 2.0 mm, 1.5d, 8 mm inline field gives 5.9 K/kW at 5.8 kPa, without doubling the pump's flow.

4.3 Inline and Staggered Layouts

At equal geometry, staggered pins transfer slightly more heat than inline pins but at more than twice the pressure drop (Figure 3: 8.3 against 8.6 K/kW and 5.0 against 2.2 kPa for 2.0 mm pins); at equal pressure drop the inline front lies at or below the staggered one over most of the range (Figure 2). Smaller pins help down to about 2 mm at this pitch and height, below which the lower fin efficiency of slender pins offsets the higher heat transfer coefficient.

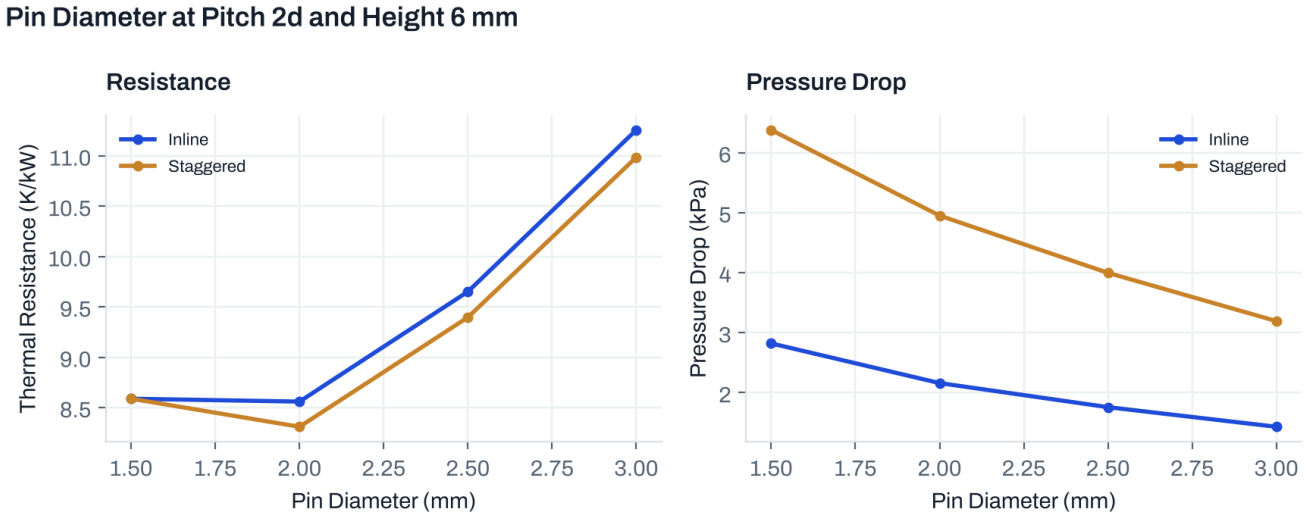


Figure 3. Thermal resistance (left) and pressure drop (right) against pin diameter for inline and staggered layouts at a pitch of two diameters and 6 mm height, 8 L/min.

5. Discussion

For this class of plate, the pin-field conductance, not the coolant's heat capacity, limits performance, so a denser, taller inline field is the most efficient use of a pressure budget. Raising flow is the expensive path: it needs a larger pump and higher flow everywhere in the loop, while a denser pin field only raises pressure drop locally. Staggered layouts, often assumed better, pay for their extra mixing in pressure drop in this laminar-to-transitional range. Teams should set the plate's pressure budget first and then choose pins on the inline front.

Limitations. Correlations derived for long tube banks are applied to short pins between end walls; end-wall heat transfer and flow bypass are not modeled; the base is assumed isothermal, which ignores spreading under discrete die; property values are representative and constant, and the wall-to-bulk viscosity correction $(Pr/Pr_s)^{0.25}$ is omitted, which is conservative for a heated coolant; and the Žukauskas correlation changes regime at Re = 1,000, which makes some trends step-like. Manufacturing limits on pin aspect ratio and spacing vary by process. A conjugate CFD model or a flow-bench test should confirm the selected design.

6. Conclusions

1. A common inline pin plate (3 mm pins, 6 mm pitch and height) carrying 1.5 kW at 8 L/min has 11.3 K/kW and 1.43 kPa.
2. Spending a pressure budget of about 6 kPa on a denser inline pin field gives 5.9 K/kW, against 7.6 K/kW from doubling the flow through the original pins.
3. At equal pressure drop, inline pins match or beat staggered pins in this range.

References

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