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THERMAL PERFORMANCE OF DIFFERENT HEAT SINK FIN GEOMETRIES: A REVIEW

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Abstract
The fin shape is a significant design parameter of air-cooled heat sinks since it influences concurrently the heat transfer area, the boundary layer behavior and the pressure drop . The current research investigates the performance of plate, pin, wavy, folding, perforated and twisted fins in relation to typical indicators like temperature, heat transfer coefficient, thermal resistance, pressure drop and performance evaluation criteria (PEC). The mechanisms of each geometry are outlined and the sample numerical and experimental investigations are tabulated with the operating circumstances and the observed improvements mentioned against their own baselines. The comparison demonstrates that no shape is the best in all scenarios. Gains in heat transfer are always accompanied by increased flow resistance or decreased fin conduction and findings rely greatly on the mode of convection and the baseline. Research Gaps Common benchmark conditions Direct comparison of many geometries Combined impact of perforations and shapes Standardized PEC reporting.

Keywords: Heat sink; fin geometry; thermal resistance; pressure drop; electronics cooling; heat transfer enhancement.

Introduction

The downsizing and power densification of electronic devices are ongoing and result in heat fluxes at which thermal management is the direct limiter of reliability, performance and service life. The excess heat not properly evacuated decreases the gadget's life and may hinder its regular functioning. The air cooled heat sinks are still the most widely used in electronics industry due to their low cost and high reliability [1]. Conventional plate-fin heat sinks have been widely used due to their ease of production and low cost, but, have been claimed to be inadequate to meet the dissipation needs of modern high power systems [2]. This has prompted a lot of research on modified fin shapes. There are two more things that make the choice of fin not trivial. First, the dominant heat transfer mechanism changes the ranking of the geometries. Natural convection heat sinks run on buoyancy-induced flow and are dependent on how the fins are spaced. Forced convection designs are limited by the pressure the fan can supply. Second, computational fluid dynamics has become a common method to visualize the flow and temperature fields around solid and perforated fins [2], so a substantial part of current work is numerical and its credibility relies on validation against experiments [1,2].

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The fin shape, spacing, height, and surface modification strongly influence the thermal performance of a finned heat sink. These factors define the effective heat transfer area, the formation or disruption of the thermal boundary layer and the flow resistance imposed on the coolant. In general geometries which increase the convective heat transfer coefficient also tend to increase the pressure drop, therefore the choice of shape is a trade-off between thermal gain and fan power. The performance of plate-fin heat sinks has been optimized by varying the fin number and thickness [3], the flow direction and the filet shape at the fin-base junction [4], and with innovative configurations such as cross fins under natural convection and radiation [5].

The pin fins have been extensively studied because of their ability to improve flow mixing and their ability to tailor the cross-section. Comparative studies, both experimental and numerical, have been reported in the literature for circular, square, elliptical, diamond and hexagonal pins [6–10]. The effect of inline versus staggered arrangement and pitch have been studied for natural, forced and mixed convection [11–13]. Micro-PIV and numerical simulation [14] and friction-factor and Nusselt-number correlations [15] have been applied to study smaller-scale pin-fin arrays. Wavy fins exhibit a greater heat transfer rate than plate fins, but this is achieved at the expense of a higher pressure drop [16], and triangular folded fins have been characterized by experimental correlations as functions of fin pitch, fin height and Reynolds number [17,18]. Perforations reduce fin mass and break the boundary layer and their effect has been studied for rectangular, plate and pin fins in natural and forced convection [2,19–23]. More recently, geometric combinations like twisted fins with perforations have been proposed [24,25].

Despite this activity, comparison across studies remains difficult. The studies vary in the mode of convection, working conditions, base area, heat input and metrics reported. For instance, pin-fin work covers natural, forced and mixed convection [11-13] and some new fin configurations are evaluated only under mixed convection [26]. The optimization of heat sink thermal design has been discussed in previous reviews in general [27] and the behavior of nanofluids in micro channel heat sink [28]. A dedicated comparison of single-phase fin geometries normalized by common performance indicators seems to be missing. This review tries to fill that gap.

The review is based on single-phase finned heat sinks with specific fin shapes, and the majority of the reviewed studies are done with air and water in micro-scale or liquid cooled cases [29]. Nanofluids, phase change materials and two phase cooling are excluded so that the comparison isolates the effect of geometry. The contribution is threefold: A unified description of the mechanisms by which each geometry acts on heat transfer and flow resistance; Tables that record the operating conditions next to each reported result, so that results are not read out of context; And a list of research gaps derived from the reviewed evidence. The heat transfer enhancement can be achieved via the coolant, e.g. nanofluids [28], phase change or the solid surface. Passive surface modification does not change the working fluid, and is compatible with existing fans and ducts, but the benefit of such modification has to be considered against the manufacturing complexity and flow resistance. The other variable most directly controlled by the designer, once the footprint and the fan are fixed, is the geometry of the fin. These features give reasons for the ongoing research on modified fins even in the presence of alternative cooling methods, and for the usefulness of a structured comparison of the geometries for both designers and researchers. A baseline in the paper is a reference geometry from the source study, typically an unmodified plate-fin or cylindrical-fin heat sink. Improvements are reported, not re-calculated. If only the heat transfer

coefficient or some temperature is given in a source, this is pointed out, since these quantities are not sufficient to determine which geometry is better at equal fan power. Studies are grouped by fin family and within each family by convection mode so that the reader may follow the evidence for one design question at a time. The tables refer to basic studies establishing a geometry or providing a correlation . Before 2015 Marked . The present work aims at classifying the main fin geometries and describing their heat transfer mechanisms, at defining the thermal and hydraulic metrics used to compare them (thermal resistance, pressure drop and PEC), at comparing representative numerical and experimental studies including their operating conditions and reported enhancements, and at identifying research gaps and future directions.

The paper is organized as follows. Section 2 describes the review methodology. Section 3 presents the fin geometries and heat transfer mechanisms. Section 4 defines the performance metrics. Section 5 compares the geometries. Section 6 discusses research gaps, and Section 7 concludes.

2. Review Methodology

2.1 Search strategy

The literature search is designed for Scopus, Web of Science, and ScienceDirect for studies published between 2015 and 2026. Earlier foundational studies are cited where they define a geometry or a correlation. The following search string is applied to titles, abstracts, and keywords:

"heat sink" AND ("fin geometry" OR "wavy fin" OR "folded fin" OR "plate fin" OR "pin fin") AND ("thermal resistance" OR "pressure drop")

Backward and forward citation tracking of key papers is used to identify additional studies.

2.2 Eligibility criteria

Inclusion criteria:

- peer-reviewed journal articles in English;
- numerical, experimental, or combined studies of single-phase heat sinks with a defined fin geometry;
- reporting of at least one thermal metric and, preferably, pressure drop.

Exclusion criteria:

- studies on nanofluids, phase change materials, or two-phase cooling;
- studies lacking quantitative thermal or hydraulic results;
- conference abstracts, theses, and duplicate records.

2.3 Screening, extraction, and comparison approach

Screening of records by title and abstract, then full-text assessment (Fig. 1). For each of the included studies, the following data are extracted: fin type, working fluid, type of study (CFD or experimental), mode of convection, Reynolds number or air velocity range, geometric parameters, key results and reported advantages and limitations. Results are not pooled because the studies have different base areas, heat inputs, and baselines. Each reported improvement is quoted in relation to the baseline defined in the source study and the convection mode is stated with it. Results are shown in Fig. 1 (marked as 47), and are done after the search is run.

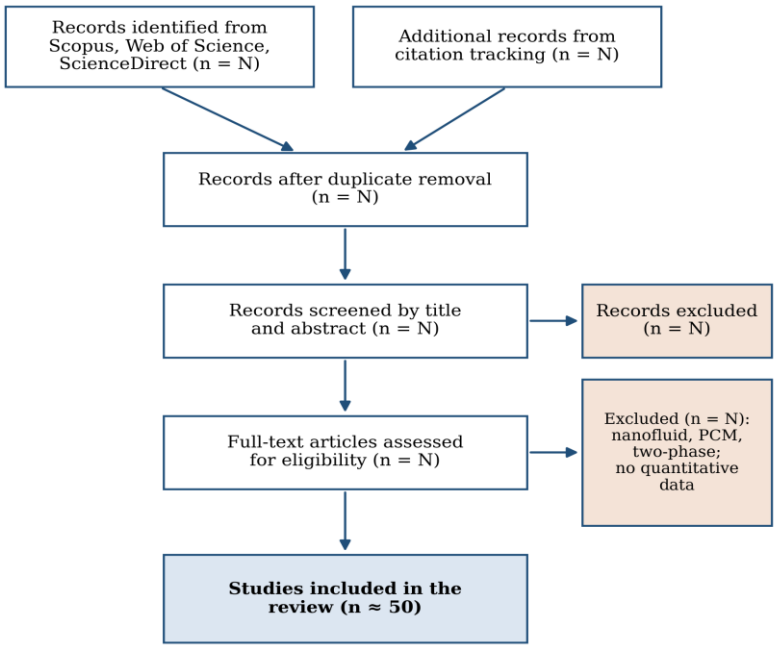


Fig. 1. Simplified PRISMA flow diagram of the study selection process (counts to be completed).

3. Fin Geometries and Heat Transfer Mechanisms

3.1 Heat transfer mechanisms

The heat from the source is conducted through the base into the fins and then removed by convection and, when significant, by radiation to the coolant (Fig. 2b). Therefore, the total resistance is the sum of the conduction resistance of the base and fins and the convective resistance at the fin surface. The fin shape acts on the second term, mainly. On a continuous surface the thermal boundary layer gets thicker in the flow direction and the local heat transfer coefficient decreases. Interrupted, curved or perforated surfaces disturb or re-start the boundary layer (Fig. 2a) and promote mixing in the wake, but at the expense of form drag. The same geometric modifications change the buoyancy driven flow under natural convection. In the study of perforated flat fins, both the turbulent flow developed around the perforations and the buoyancy effect in the enclosure were found to affect the heat dissipation [2]. Perforation also removes metal, thus reducing conduction along the fin [2]. Hence, a perforated fin has an increase in convective performance and a decrease in conductive performance simultaneously.

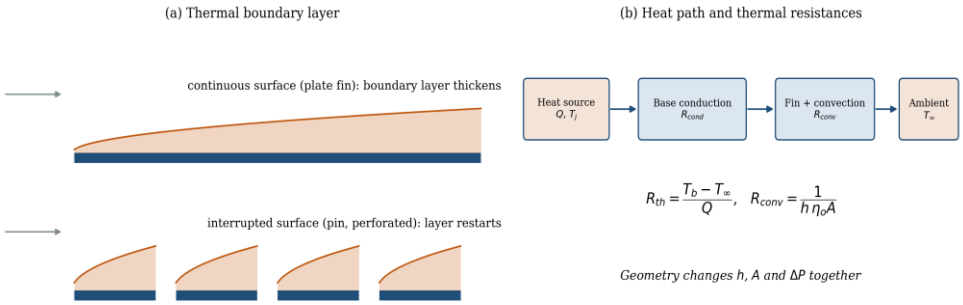


Fig. 2. Heat transfer mechanisms: (a) thermal boundary layer on a continuous and on an interrupted surface (schematic, not to scale); (b) heat path and thermal resistances in a finned heat sink.

3.2 Fin families

The five families of fins are discussed (Figs. 3 and 4). Plate fins are extended surfaces parallel to each other and form a baseline for most comparisons [3,4]. Pin fins are separate columns with a circular, square, elliptical, diamond or hexagonal cross section, and are arranged inline or staggered [6,7,9-11]. Wavy fins are characterized by wave number and amplitude [16] and follow a sinusoidal path along the flow. Folded fins are made of a folded sheet, with the most studied one being a triangular fold [18] experimentally. The perforated fins have holes, the shape, number, size and position of which are the design variables [2,20,30,31]. These families are extended by hybrid forms such as twisted fins with diamond shape perforations [24] and perforated twisted tapes inserted in plate-fin heat sinks [25].

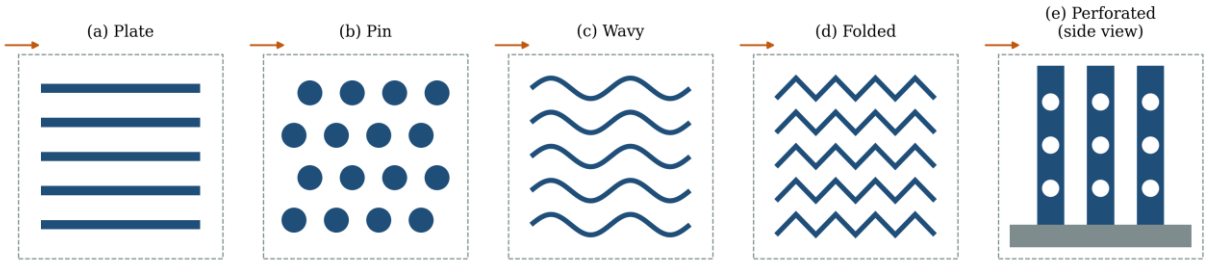


Fig. 3. Schematic of the main fin geometries (top view unless noted; arrow indicates flow direction): (a) plate, (b) staggered pin, (c) wavy, (d) folded, (e) perforated.

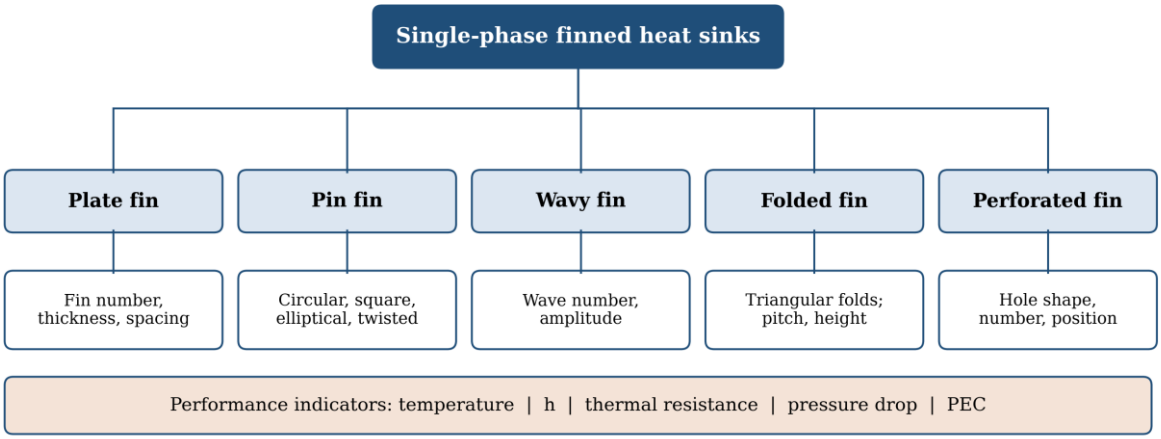


Fig. 4. Classification of single-phase fin geometries covered in this review and the performance indicators used for comparison.

3.3 Design variables and convection modes

Each fin family has its own design variable set. For plate fins they are fin number, thickness, spacing, height and the filet at the base [3,4]. For pin fins they are the cross-section, the aspect ratio H/d, the longitudinal and transverse pitch and the inline or staggered arrangement [1,13,29]. The wave number and amplitude are contributed by wavy fins [16], the fin pitch and height by folded fins [18], the hole shape, number, size, position and orientation by perforated fins [2,19,30], and the twisting angle and the perforation perimeter by twisted fins [24]. The convection mode adds a 2nd axis. Natural convection heat transfer from pin-fin arrays [11], perforated rectangular fins [20], cross fins [5] and perforated flat-

plate fins in an enclosure [2] have been investigated. Pin-fin arrays [12], perforated vertical fins [22] and folded fins in duct flow [18] are studied under forced convection. The mixed convection has been studied for elliptical pin fins [1,13] and for a new fin configuration [26]. Thus a given geometry is rarely tested in all three modes and this is one reason why the literature does not yet permit a single ranking of fin families.

4. Thermal and Flow Performance Metrics

The most reported thermal metrics are the base or maximum temperature, the average heat transfer coefficient, the Nusselt number, and the thermal resistance [1,32]. The mean heat transfer coefficient and the Nusselt number are

$$h = Q / [A (T_w - T_\infty)] \quad (1)$$

$$Nu = h D_h / k_f \quad (2)$$

where Q is the convective heat transfer rate, A the heat transfer area, T_w the mean wall temperature, T_∞ the inlet or ambient fluid temperature, D_h the characteristic (hydraulic) diameter, and k_f the fluid conductivity. The flow regime is set by

$$Re = \rho u D_h / \mu \quad (3)$$

The thermal resistance of the heat sink is defined with the mean base temperature T_b [1]:

$$R_{th} = (T_b - T_\infty) / Q \quad (4)$$

Rth includes conduction and convection, and thus is the single figure of merit to compare different geometries at the same heat load. For example, in a study of elliptical pin fins, thermal resistance was discovered to be an almost insensitive function of input heat flux over the range tested [13], indicating its utility for comparisons across heat loads for a single configuration. The hydraulic cost is given by the pressure drop ΔP between inlet and outlet, or by a friction factor, or by a drag coefficient [1]. A friction factor based on the channel length L is

$$f = 2 \Delta P D_h / (\rho u^2 L) \quad (5)$$

and the pumping power is $W_p = Q_v \Delta P$, with Q_v the volumetric flow rate. The efficiency of a fin of height H is

$$\eta_f = \tanh(mH) / (mH), \quad m = (hP / k_s A_c)^{1/2} \quad (6)$$

where P is the fin perimeter, A_c the cross-sectional area, and k_s the solid conductivity. Fin efficiency is the quantity that perforation and tall slender pins degrade most directly, because it depends on conduction along the fin [1,2].

To combine heat transfer and flow resistance in one number, the performance evaluation criterion compares a modified surface with a reference surface at equal pumping power.

$$PEC = (Nu / Nu_0) / (f / f_0)^{1/3} \quad (7)$$

where the subscript 0 denotes the reference geometry, usually the plate fin. A related hydrothermal performance factor (HTPF) is used in some studies [24]. Because the exponent and the baseline differ between papers, PEC values from different sources should be compared with caution; this is revisited in Section 6.

In practice, the choice of the metric should be the design constraint. When the heat load and allowable junction temperature are fixed, thermal resistance is the natural target. When the fan or pump is fixed, the limiting variables are the pressure drop and the pumping power. The heat dissipated per unit fin mass is also informative when doing a choice based on mass or material cost, for example for perforated fins,

as perforation reduces weight while it reduces conduction [2,23]. Studies reporting only the heat transfer coefficient, without pressure drop or thermal resistance, cannot be placed on the same footing as those reporting all three, and such studies are therefore indicated in the tables of Section 5.

5. Comparison of Fin Geometries

5.1 Plate fins

In a numerical study of an air-cooled plate-fin heat sink, the maximum temperature, the pressure drops and the thermal resistance decreased with increasing the fin number and the fin thickness, and an optimal fin number and thickness were identified [3]. A study based on Fluent [4] showed that the flow direction changes and the addition of a filet at the fin and base junction improved the thermal performance than a traditional plate-fin heat sink. A cross-fin arrangement was tested under natural convection and radiation conditions and in a numerical model validated by experiments [5] it gave better performance than a conventional parallel-fin heat sink. These results indicate that for plate fins, the improvements are due primarily to arrangement and junction details and not to changes to the fin surface.

5.2 Pin fins

Pin cross-section has an obvious effect. It was found that the elliptical pin fin had the highest heat transfer coefficient with low drag among plate fins and circular, elliptical and square pin fins of the same perimeter. The rectangular plate fin had the best entropy-generation value and the square pin performed worst [6]. Numerical studies on six pin cross sections showed the elliptical profile to be the best performer [7,8]. An experimental comparison of elliptic, square and circular pins reported that the staggered elliptic arrangement gave the lowest pressure drop and thermal resistance [9]. Diamond-shaped micro pins had better heat transfer than circular and square micro pins in a micro-PIV and numerical study and vortices were more easily entrained into the main flow [14]. Also the arrangement is important: in natural convection inline arrays were reported to perform better than staggered arrays [11], while the lowest pressure drop and thermal resistance was achieved with the staggered elliptic arrangement in an air-side comparison [9]. This difference may be due to the mode of convection and the test conditions as discussed in Section 1.

There is one geometry that has the optimum aspect ratio. The three-dimensional simulation for elliptical fins under mixed convection (Re = 50–250) gave an optimal aspect ratio of around 8.16, after which the thermal performance degraded owing to the temperature gradient near the fin tips [1], which is consistent with the optimum found in the experimental study used for validation [13]. The insertion of metal foam around the elliptic pins improved the thermal efficiency at the expense of a modest pressure drop increase [33]. The optimum pin diameter and number were also obtained for cylindrical pin-fin heat sinks [34]. Pin length and longitudinal pitch were shown to influence heat transfer and pressure drop in a water-cooled pin-fin heat sink [29]. Variable-density arrangements have been used to improve temperature uniformity in micro pin-fin heat sinks [35] and hemispherical pins have been numerically optimized [36].

The earliest and the latest pin-fin work agree on the importance of the position in the array. An experimental investigation of pin fins in an oncoming flow which turns to cross flow has shown that the fins at the periphery of a heat sink exhibited higher heat transfer coefficients than the fins in the core [37]. A theoretical model for a vertical array of pin fins in combined forced and natural convection with

impinging flow gave an optimum fin spacing and agreed with experiments [38]. The dependence of the Nusselt number on the Reynolds number and the fin spacing was explored for forced-convection pin-fin arrays [12]. A comparison of cylindrical, elliptical, sprocket, kite, channeled and triangular pins based on COMSOL showed that the kite and elliptical pins had the most effect on the Nusselt number and heat transfer coefficient [39]. The conclusions are extended to the comparison of staggered elliptical, circular and diamond pins in a rectangular channel at the micro scale [40] and a study of square micro pins under natural convection [41]. Natural-convection heat transfer for pins of different shape at different vertical fin spacing is also investigated [42].

5.3 Wavy and folded fins

For wavy fins, a finite-volume study with the k–ε turbulence model compared wavy-fin and plate-fin heat sinks and varied the wave number and amplitude; wavy fins raised the heat transfer rate, accompanied by a pressure-drop penalty [16]. Folded fins are less studied. Experiments on triangular folded fins in duct flow (Re = 212–1974, fin height 19–36 mm) showed that the cooling performance increases markedly as the fin pitch decreases and as the Reynolds number and fin height increase, and correlations for the Nusselt number and the friction factor with ±6.5% and ±20% accuracy were provided [18]. An earlier conference study of the same configuration reported higher cooling performance than a conventional extruded plate-fin heat sink of the same height [17]. The available evidence therefore supports wavy and folded fins as enhancement options, but direct, same-condition comparisons with pin or perforated fins are missing.

5.4 Perforated fins

Perforation studies are mostly under natural convection. Circular perforations in rectangular fins increased the heat transfer rate and coefficient with the number of perforations [20], and increasing the perforated area increased the heat transfer coefficient in an experimental study of perforated rectangular fin arrays at several inclinations [21]. Under forced convection, perforations of circular, rectangular, and triangular shape on vertical heated fins all improved the temperature difference compared with solid fins, with the circular shape giving the greatest improvement [22]. A numerical study with the RNG k–ε model found superior heat transfer with perforated fin arrays compared with solid fins, together with a weight reduction [23]; the orientation of square perforations also affected the heat dissipation [19]. An analytical model has been proposed for laterally perforated-finned heat sinks [43].

Shape and position interact. In a CFD study of flat-plate fins in an enclosure, circular perforations at the top of the fin gave the best overall performance, with the heat transfer coefficient rising by up to 29.6% relative to solid fins, whereas triangular perforations gave the largest temperature difference along the fin [2]. The authors attributed the loss of conduction to the removed metal, which is a limitation of perforation [2]. For pin fins, multiple circular perforations raised the Nusselt number by about 11% compared with the solid pin, as summarized in [2] from [44], and perforation shape and diameter have been optimized numerically [31]; elliptical perforations were also optimized with a hybrid neuro-fuzzy and genetic algorithm [45]. Perforation of micro pin-fin heat sinks has also been analysed computationally [46].

5.5 Twisted and hybrid fins

A Fluent-based study of twisted fins with diamond-shaped perforations found that a 540° twist raised the Nusselt number by up to 46% and the hydrothermal performance factor by 25% relative to conventional cylindrical fins; varying the perforation perimeter at 540° gave a further 28% and 36%, respectively, relative to twisted fins without perforations [24]. A plate-fin heat sink equipped with twisted tape and perforated twisted tape has also been tested [25], and multi-branching fins have been proposed for natural convection [47]. These hybrid forms are promising, but the reported gains are relative to different baselines and cannot be ranked against the other families on the present evidence.

5.6 Cross-geometry comparison

Representative studies and operating conditions are shown in Table 1. Quantitative changes reported by the sources are presented in Table 2, each in relation to its own baseline. Table 3 summarizes the advantages, disadvantages and the usual working conditions for each geometry.

Table 1. Representative previous studies on heat sink fin geometries.

Ref.	Fin type	Study type	Fluid / conditions	Main findings
[3]	Plate	Numerical (Icepak)	Air; fin number and thickness varied	Maximum temperature, pressure drop, and thermal resistance decreased with fin number and thickness; an optimum was identified.
[4]	Plate with fillet	Numerical (Fluent)	Flow direction and fillet profile varied	Better thermal performance than the conventional plate-fin heat sink.
[5]	Cross fin	Numerical, validated experimentally	Natural convection and radiation	Superior to a conventional parallel-fin heat sink.
[6]†	Plate vs. circular, elliptical, square pins	Not stated in source	Equal fin perimeter	Elliptical pin: highest h, low drag; rectangular fin: best entropy generation; square pin: lowest h.
[9]	Elliptic, square, circular pins	Experimental	Air side; inline and staggered	Staggered elliptic arrangement gave the lowest pressure drop and thermal resistance.
[7]	Six pin cross-sections	Numerical	In-duct flow	Elliptical profile performed best among the six shapes.
[13]	Elliptical pin	Experimental	Mixed convection; aspect ratio 5.1–9.18; void fraction 0.534–0.884	Thermal resistance almost independent of input heat flux; optimum aspect ratio identified.
[1]	Elliptical pin	Numerical (3-D, Fluent)	Mixed convection; Re = 50–250	Optimal aspect ratio ≈ 8.16; new Nu correlation.
[14]	Circular, square, diamond micro pins	Micro-PIV and numerical	Laminar flow	Diamond micro pins gave the best heat transfer.

Ref.	Fin type	Study type	Fluid / conditions	Main findings
[16]	Wavy	Numerical (FVM, k–ε)	Turbulent flow; wave number and amplitude varied	Higher heat transfer rate than plate fins, with a pressure-drop penalty.
[18]	Triangular folded	Experimental	Duct flow; Re = 212–1974; height 19–36 mm; pitch 5–9 mm	Performance rose with smaller pitch and higher Re and height; Nu and f correlations proposed.
[20]†	Rectangular with circular holes	Experimental	Natural convection; 6–220 W; 24–56 holes	Heat transfer rate and coefficient increased with the number of perforations.
[22]	Perforated vertical fins	Experimental	Forced convection; circular, rectangular, triangular holes	All shapes improved on solid fins; circular gave the greatest improvement.
[23]	Perforated fin array	Numerical (RNG k–ε)	Turbulent convection	Better heat transfer than solid fins, with lower weight.
[2]	Perforated flat-plate fins	Numerical (Fluent), validated	Natural convection in enclosure; 1.0–2.5 kW/m²	Circular, top-positioned holes best; h up to +29.6% vs. solid fins.
[24]	Twisted, perforated	Numerical (Fluent, k–ε)	Twist angle and hole perimeter varied	540° twist: Nu +46%, HTPF +25% vs. cylindrical fins.

† Published before 2015; retained as a foundational study. HTPF: hydrothermal performance factor. Entries report only quantities stated in the cited sources.

Table 2. Quantitative changes reported in the literature (each relative to the baseline of its own study; not comparable across rows).

Ref.	Geometry change	Baseline	Metric	Reported change
[24]	540° twist	Cylindrical fin	Nu; HTPF	Up to +46%; +25%
[24]	Diamond perforation at 540°	Twisted fin, unperforated	Nu; HTPF	Up to +28%; +36%
[2]	Circular perforation, top position	Solid flat fin	Heat transfer coefficient	+10% to +29.6% (1.0–2.5 kW/m²)
[2]	Circular perforation, top position	Solid flat fin	Fin temperature difference	+15.6% (minimum increment)
[44]	Multiple circular perforations in pin	Solid pin fin	Nu	≈ +11% (as summarized in [2])
[22]	Circular / rectangular / triangular holes	Solid vertical fin	Temperature difference	51.29% / 45.57% / 42.28% (as summarized in [2])

Percentages are quoted as given in the sources and, where marked, as reported in a secondary summary. They should be checked against the original papers before being used quantitatively.

Table 3. Summary of advantages, limitations, and operating conditions by fin geometry.

Geometry	Advantages	Limitations	Operating conditions in the studies
Plate	Simple and low-cost [2]; tunable by fin number, thickness, fillet [3,4]	Reported inadequate for very high dissipation [2]; boundary layer grows along the fin	Forced and natural convection [3,5]
Pin	Mixing; tailorable section; elliptical pins combine high h with low drag [6,9]	Optimum aspect ratio exists [1]; arrangement conclusions differ between studies [9,11]	Natural, forced, and mixed convection [11–13]
Wavy	Higher heat transfer than plate fins [16]	Pressure-drop penalty [16]	Turbulent air flow [16]
Folded	Correlations available; performance rises with Re and height [18]	Few recent studies; fabrication of fold geometry	Duct flow, Re = 212–1974 [18]
Perforated	Higher h; lower weight [2,23]	Lower fin conduction from removed metal [2]; shape and position sensitive	Natural convection [2,20]; forced convection [22,23]
Twisted / hybrid	Large Nu and HTPF gains vs. cylindrical fins [24]	Baselines differ; limited validation across studies	CFD, k–ε [24]

Statements on fabrication and boundary-layer growth are general engineering remarks and are not attributed to a specific source.

5.7 Effect of convection mode and operating range

Table 1 shows that the geometry families have been tested in largely different regimes. Elliptical pins were simulated in laminar mixed convection at Re = 50–250 [1], folded fins were tested in duct flow at Re = 212–1974 [18], wavy fins and perforated fin arrays were modelled in turbulent flow [16,23], and perforated flat fins were studied under natural convection with heat fluxes of 1.0–2.5 kW/m² [2] and heater powers of 6–220 W [20]. The characteristic length used in the Reynolds number also differs between studies, for example the mean pin diameter in the elliptical-pin work [1] and the hydraulic diameter in the folded-fin work [18]. As a result, the numerical values of Re in different rows of Table 1 cannot be compared directly, and a ranking of fin families from the published results alone is not justified.

Within each regime, the trends are consistent. Larger heat transfer area, through more or thicker plate fins, reduces the thermal resistance [3]. A smaller pitch and a taller fin improve folded-fin performance [18]. A larger number or area of perforations raises the heat transfer coefficient under natural convection [20,21]. Gains in heat transfer are paired with higher pressure drop for wavy fins [16], and they are paired with reduced conduction for perforated fins [2]. These recurring pairings are the quantitative content of the trade-off discussed in Section 1.

5.8 Numerical modelling and validation practice

The recent experimental studies are numerical and the modeling choices are summarized in Table 4, stated in the sources. Two practices are notable. Validation is reported in a non-uniform way. The simulation of the elliptical pin was compared with published experiments, it was run on four meshes before the fine mesh was selected [1]. The perforated fin model was validated against experiments on solid fins with a maximum deviation of about 2% and measurement uncertainty of up to ±0.69 °C [2]. Several other studies [3,16,24] do not report the validation procedure in the abstract . Secondly, the turbulence model used is different for k–ε [16,24] and RNG k–ε [2,23] which may influence the heat transfer prediction around perforations and in wakes. Therefore, when comparing the absolute values of these studies, the choice of model must be considered as a source of uncertainty.

Table 4. Modelling choices stated in representative numerical studies.

Ref.	Solver	Flow and turbulence model	Validation / uncertainty
[1]	ANSYS Fluent (finite volume, SIMPLE, QUICK)	Steady incompressible, Boussinesq buoyancy; no turbulence model stated	Compared with experiments [13]; four meshes tested, fine mesh ≈ 5.9×10^6 nodes
[2]	ANSYS Fluent 2019 R1 (SIMPLE, second-order upwind)	RNG k–ε with Boussinesq approximation	Validated on solid fins, max. deviation 2.14%; uncertainty up to ±0.69 °C; 4×10^5 elements
[16]	Finite volume, SIMPLE	k–ε	Compared with a plate-fin heat sink; experimental validation not stated in abstract
[23]	Numerical, 3-D	RNG k–ε	Not stated in the available summary
[24]	ANSYS Fluent	RANS, k–ε	Not stated in abstract
[3]	ANSYS Icepak, 3-D	Not stated in abstract	Not stated in abstract

"Not stated" indicates that the item is not given in the abstract or summary consulted, not that it was not done.

5.9 Dependence of the verdict on the chosen metric

A recurring result is that the preferred geometry is metric-dependent. In comparison of plate fins with circular, elliptical and square pins of equal perimeter, elliptical pin gave highest heat transfer coefficient with low drag but rectangular plate fin gave best entropy-generation value [6]. For the perforated flat-plate, the best overall performance, i.e., the smallest temperature difference along the fin, was achieved with circular holes. Triangular holes still increased the heat transfer coefficient but had the largest temperature difference and a less even temperature distribution [2]. In the work with folded-fin [18], increasing the Reynolds number and fin height improved the cooling performance, but these also increase the flow resistance. Thus, the net benefit is dependent on whether there is a fan power constraint. Therefore, a statement like geometry A is better than geometry B" makes sense only when the metric, the baseline and the operating condition are specified together. These three are therefore listed side by side in the tables in this review.

5.10 Practical design considerations

Three practical considerations in addition to thermal and hydraulic metrics are involved regarding geometry selection. The first is weight. The perforation reduces the weight of the fin, which was mentioned as possibly beneficial for mobile and high-technology applications [23]. The second is the manufacturability, since plate fins are simple to manufacture and inexpensive [2] whereas wavy, folded, twisted and perforated fins require additional forming or machining operations. Costs of these operations are not reported in the abstracts and summaries reviewed for this review and therefore a ranking of costs is not undertaken. Third is installation: both the inclination of perforated rectangular fin arrays [21] and the orientation of square perforations [19] have been studied in natural-convection studies, showing that mounting orientation is a design variable, not a fixed condition, for perforated fins. Thus, a designer applying a geometry from Table 3 should check that the source study matches the desired orientation, convection mode and heat load.

6. Research Gaps and Future Directions

In section 5, results of the present study are compared with previous studies, pointing out where they agree or differ, or where they are silent. The main gaps are:

1. No common baseline. Different base areas, heat loads, convection modes are used in studies and reports are often limited to a single mode (natural, forced or mixed) [11–13,26]. The risk of transferring conclusions between conditions is shown by the differing arrangement conclusions from a natural convection study and an air-side comparison [9,11]. A typical reference case with constant base area, heat load and flow range would allow direct comparison.
2. There are few studies of multiple geometries under the same conditions. For the most part, plate, pin, wavy, folded and perforated fins are discussed in separate papers. Specifically, there are no same-condition comparisons of wavy and folded fins with pin and perforated fins [16, 18].
3. Combined effects of perforation. It was stated that the combined effect of perforation shape and position was not well enough investigated in the CFD study that did address it [2]. Extensions to size, number and orientation, and to forced convection are still open [19,22,30].
4. Trade-off between conduction and convection. Tall slender pins have an optimal aspect ratio [1], and removed metal reduces conduction and fin efficiency [2]. The convective gain and the fin efficiency are not coupled yet as a design rule.
5. Non uniform indices of performance. Various baselines are used such as PEC, HTPF, entropy generation, and thermal resistance [6,24]. Studies should report Nu, f, Rth and PEC in relation to a stated plate-fin baseline.
6. Radiation and enclosure effect. Natural convection studies often assume the enclosure to be adiabatic or still air [2], but radiation can be significant in natural convection [5]. Both warrant explicit treatment.

6.1 Future directions

Benchmark cases. The most useful next step is a common reference case with fixed base area, defined heat-flux range and a plate-fin baseline, to which experimental and numerical groups can report Nu, f, Rth and PEC. Raw curves of thermal resistance versus pressure drop instead of single point

improvements would allow a direct comparison between families tested in different Reynolds-number ranges [1,18].

Multiobjective optimization Optimization algorithms have been applied to elliptical perforations with hybrid neuro-fuzzy and genetic approach [45] and variable-density pin arrangements for temperature uniformity [35]. Such methods could be extended to objectives involving thermal resistance, pressure drop and mass, and several fin families in one design space, to address the trade-off at the heart of this review.

Hybrid geometries. Encouraging gains are reported for perforated twisted fins [24], perforated twisted tapes in plate-fin heat sinks [25] and multi-branching fins [47], but each is compared to a different baseline. A direct comparison with the best pin-fin and perforated-fin designs identified in Section 5 is necessary to recommend these forms.

Experimental discipline Mesh studies and measurement uncertainty should be routinely reported and validated with experiments [1,2]. It is also important that experiments are done with more than one mode of convection for the same geometry, so that the dependence discussed in Section 5.7 can be quantified.

6.2 Limitations of the present review

The reference set was selected by targeted searching of 47 sources and not the full protocol of Section 2. We also need to fill in the PRISMA counts. We should also expand the table of studies to the final set of ~50 included studies. In the case of a number of entries we have relied on abstracts or summaries given in other papers, and the corresponding values are noted in the tables. Studies prior to 2015 are included as foundations. These limitations do not affect the qualitative conclusions but they do limit the quantitative use of the tabulated values. While no universal ranking is possible, the individual results can be compiled in a qualitative guide for early-stage design (Fig. 5). The guide indicates for each priority and regime which geometry is favored by the reviewed evidence. It does not replace a case-specific simulation or test.

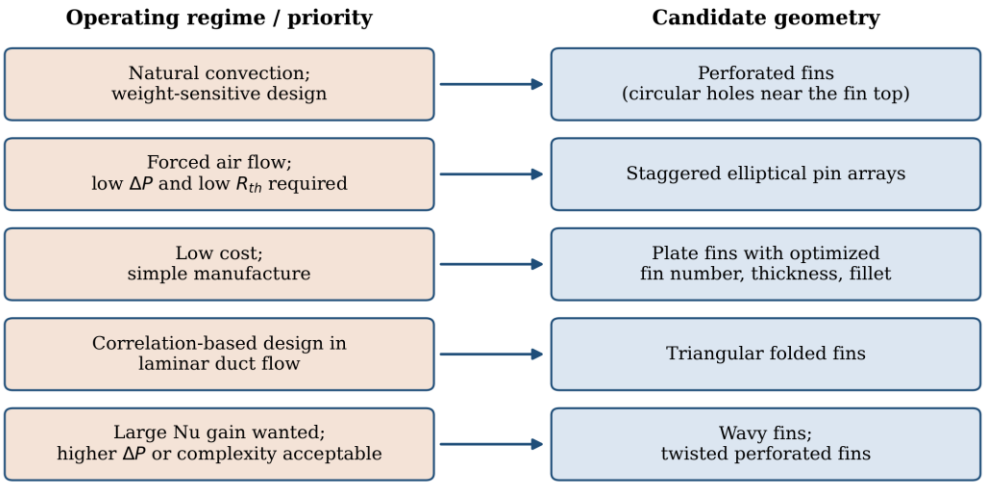


Fig. 5. Qualitative guide linking operating regime and design priority to candidate fin geometries, arranged from individual studies reviewed in Section 5 (not validated across studies).

Table 5. Summary of research gaps, supporting evidence, and proposed actions.

No.	Gap	Evidence in the reviewed literature	Proposed action
1	No common benchmark	Different convection modes and Re definitions across families [1,2,18]	Define a reference base area, heat load, and flow range
2	Few same-condition multi-geometry studies	Wavy and folded fins studied separately from pin and perforated fins [16,18]	Run plate, pin, wavy, folded, and perforated cases in one validated model
3	Combined perforation effects	Shape and position studied together in only one CFD study [2]	Extend to size, number, orientation, and forced flow
4	Conduction–convection coupling	Removed metal lowers conduction [2]; optimum aspect ratio for pins [1]	Report fin efficiency alongside Nu and Rth
5	Inconsistent indices	PEC, HTPF, entropy generation, and Rth all in use [6,24]	Report Nu, f, Rth, and PEC against a stated plate-fin baseline
6	Validation and uncertainty	Validation not visible for several numerical studies; reported where present [1,2]	Report mesh study, validation error, and measurement uncertainty

The table restates the gaps of Section 6 in compact form.

7. Conclusions

The temperature, heat transfer coefficient, thermal resistance, pressure drop and PEC were used to compare the plate, pin, wavy, folded, perforated and twisted fins. The main results are:

1. The geometry of fins simultaneously influences heat transfer and flow resistance. The wavy fins enhance heat transfer at the cost of pressure drop while the perforation increases the heat transfer coefficient and removes conductive material.
2. Elliptical sections are repeatedly the best compromise between heat transfer and drag among pin fins and there is an optimum aspect ratio.
3. Shape and position of the perforation are important: the best results in the CFD study varying both were for circular holes at the top of the fin; circular holes also gave the greatest improvement in forced convection.
4. Hybrid twisted and perforated fins exhibit the largest reported gains relative to the baselines, but the baselines are different and no cross-family ranking is warranted.
5. Conclusions are sensitive to convection mode and baseline as well as the lack of benchmark conditions and standardized PEC reporting hamper direct comparison.

The recommendation for designers is to select the geometry by operating regime: staggered elliptical pin arrays in air-side flow; perforated fins where weight and natural convection dominate; and plate fins with optimized number, thickness and junction details where simplicity and cost dominate.

Nomenclature

Symbol	Quantity	Symbol	Quantity
A	heat transfer area	ρ	fluid density
A_c	fin cross-sectional area	μ	dynamic viscosity
D_h	hydraulic (characteristic) diameter	η_f	fin efficiency
f	friction factor	H	fin height
h	average heat transfer coefficient	k_f, k_s	fluid, solid conductivity
L	channel length	Nu	Nusselt number
P	fin perimeter	ΔP	pressure drop
PEC	performance evaluation criterion	Q	heat transfer rate
Re	Reynolds number	R_{th}	thermal resistance
T_b, T_w, T_∞	base, wall, ambient temperature	u	air velocity
W_p	pumping power	HTPF	hydrothermal performance factor

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