

Cooling of server electronics: A design review of existing technology



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HIGHLIGHTS

- Review heat transfer/flow parameters from studies on server cooling technologies.
- Perform direct quantitative comparison between cooling technologies.
- Identify heat transfer limitations, power requirements, etc.
- Summarize industry implementations of each cooling solution from a design perspective.
- Identify the capacity of cooling solution relative to heat loads expected by 2020.

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ABSTRACT

This review quantitatively examines and compares the heat transfer characteristics of several cooling technologies with potential application in the server electronics industry. Strategies that have been examined include traditional air cooling, single and two-phase indirect liquid cooling, heat pipes, pool boiling, spray cooling, and jet impingement. The characteristics that have been examined include heat flux values, coolant temperatures, and coolant flowrates; which serve as indicators of the heat transfer limitations and power requirements of each cooling solution. A direct comparison against anticipated server heat loads has shown that some form of liquid cooling is necessary in high performance computing applications; where individual processor heat loads are expected to reach 300 W by the year 2020. While in the case of general purpose computing, where individual processor heat loads are expected to reach 190 W, air cooling remains a viable option; although other factors such as operating costs, chip reliability, and waste heat recovery may still encourage the use of liquid cooling.

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1. Introduction

Cooling of server electronics is a growing concern for businesses and government organizations involved with data processing, storage, and telecommunication. The main basis for this concern is the increasing cost and complexity of thermal management in server housing facilities. As the industry continues to adopt newer generations of processors that offer improved computational performance and capabilities, the rise in generated heat becomes unavoidable.

In a typical data center, roughly 33% of the total electricity consumed is allocated to thermal management of server electronics [1]. Concurrently, total global electricity usage in data centers has increased from approximately 152 billion kW h/year in 2005 to 238 billion kW h/year in 2010; roughly 1.3% of global electricity use [2]. In the United States alone, total data center energy usage

amounted to 91 billion kW h/year in 2013 and is projected to reach 140 billion kW h/year in 2020; resulting in operating electricity costs of \$13 billion/year and annual carbon emissions of 150 million metric tons [3]. These trends have exacerbated issues pertaining to energy consumption, operational costs, and the overall carbon footprint of server housing facilities; thus generating major discussions on the limitations and practicality of traditional air cooling strategies. Consequently, substantial efforts have been made to research and develop alternative liquid cooling solutions in an attempt to mitigate such issues.

Several reviews have been published in recent years dealing with thermal management challenges within data centers as well as available cooling technologies. Kant [4] chiefly reviews storage, networking, and management challenges within large virtualized data centers; however detailed discussions regarding standard server rack layouts and traditional air cooling systems are also provided. Garimella et al. [2] identify major drivers necessary for progress at both business and technological levels within data centers. Implementations of liquid cooling within industry are

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Nomenclature

A_{HS}	heat spreader area (cm ²)
A_w	heated surface area (cm ²)
h_{eff}	effective heat transfer coefficient (W/cm ² K)
P_{fan}	fan power consumption (W)
P_{pump}	pumping power consumption (W)
ΔP	pressure drop (kPa)
Q	coolant flowrate (cm ³ /s)
q	total heat transfer rate (W)
q''_{HS}	heat spreader heat flux (W/cm ²)
R_{HS}	heat spreader thermal resistance (K/W)
R''_{TIM}	TIM thermal resistance (K cm ² /W)
T_{bath}	coolant bath temperature (°C)
T_i	coolant inlet temperature (°C)
T_o	coolant outlet temperature (°C)
T_{sat}	coolant saturation temperature (°C)

T_w	heated surface temperature (°C)
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Acronyms

CDU	coolant distribution unit
CRAC	computer room air conditioning
CRAH	computer room air handling
HP	high power
HPC	high performance computing
LHP	loop heat pipe
LP	low power
LTS	loop thermosyphon
MCHS	microchannel heat sink
PM	porous medium
RDHx	rear door heat exchanger
SCHx	side car heat exchanger

discussed with focus upon energy savings resulting from waste heat recovery. Garimella et al. [1] provide a comprehensive qualitative review of the various thermal management technologies available within industry. Solutions examined include air cooling, liquid cooling (direct and indirect, single and two-phase), and miscellaneous cooling technologies such as thermoelectrics and synthetic jets. Ebrahimi et al. [5] focus upon methods and technologies available for data center waste heat recovery. Solutions such as air cooling and indirect single and two-phase liquid cooling are discussed quantitatively in terms of heat loads and system operating temperatures. Zhang et al. [6] review various technologies and configurations available for data center free cooling; estimated costs and energy savings are presented for airside, waterside, and heat pipe free cooling systems. Fulpagare and Bhargav [7] provide a general review on recent advances in data center thermal management. Numerical and experimental studies regarding air cooled data centers are summarized with focus upon facility layouts, efficiencies, performance metrics, and lifecycle analyses. Liquid cooling solutions such as on-chip water cooling and rear door heat exchangers are also discussed briefly. Oró et al. [8] focus upon the integration of renewable energy technologies into existing data center infrastructures and their respective impacts upon data center energy efficiencies and carbon footprints. Air cooling and various liquid cooling strategies are described along with their potential for free cooling. Finally, the *Data Center Handbook* [9] provides a comprehensive overview of the various cooling technologies and configurations available to data center engineers. Emphasis is placed upon air cooling and its numerous arrangements while alternative liquid cooling strategies such as on-chip single and two-phase cooling, pool boiling, and sealed dielectric systems are also discussed in great detail. Estimated energy savings associated with these technologies are also presented.

Collectively, these reviews paint a comprehensive picture of the various technologies available for data center thermal management. There is however a lack of direct quantitative comparison between the numerous cooling solutions proposed by researchers or industry. It is the aim of this design review to provide such an assessment through a fixed set of quantitative evaluation criteria. Additionally, the heat transfer mechanisms of each cooling solution are briefly reviewed while emphasis is placed upon their benefits and drawbacks as seen in industry applications. The overall intent is to provide a useful design tool for thermal engineers involved in server cooling applications; particularly at a time when new and innovative cooling technologies are frequently emerging.

2. Evaluation criteria

In this review, cooling solutions are evaluated quantitatively through their heat transfer limitations and power requirements. The following sections summarize the motivation and significance behind these criteria.

2.1. Heat transfer limitations

The foremost characteristic of any cooling solution is its heat transfer limitation. This is defined as the maximum rate at which heat may be generated by a processor while die operating temperatures remain below acceptable limits; typically recommended to be 85 °C [5]. This definition must also assume that a practical set of operating conditions apply. For example, an air cooling system's heat transfer limitation may be increased considerably by reducing inlet air temperatures; however this approach is rendered impractical beyond a certain point due to amplified vapor-compression cooling costs.

Heat transfer rates of proposed cooling solutions are well documented in literature and are usually quantified through either the wall heat flux metric or the heat transfer coefficient. The wall heat flux metric is simply the total heat transfer rate divided by the heated surface cross sectional area. The heat transfer coefficient instead employs the total convective heat transfer area (*i.e.* base and fin surface areas) while also taking into account the heated surface and coolant temperatures.

Although very commonly used to rate the efficacy of a given cooling solution, these metrics fail to provide sufficient information necessary for direct comparison between alternative technologies. The wall heat flux metric provides no indication of the heat spreading area required for effective heat dissipation. An air cooled heat sink for instance, will require larger base and fin areas in relation to a waterblock when cooling processors of identical power and size. The waterblock is therefore a more effective cooling solution as it requires less server area for similar heat dissipation; the chip heat flux metric fails to identify this. The heat transfer coefficient faces a similar problem. The total convective heat transfer area, which accounts for extended surface areas, does not directly characterize the server area required for a given cooling solution.

In sight of this, this review quantifies the heat transfer rate of a given cooling solution through the *heat spreader heat flux metric*; the total heat transfer rate divided by the overall base area of the heat spreader. This is not a perfect metric as it has relinquished

Table 1
ASHRAE predictions for processor/server heat loads expected by the year 2020 [10].

Component/server	Number of CPUs	Power (W)			2020 Rack heat load (kW)
		Year 2010	Year 2015	Year 2020	
High power HPC processor	1	250	300	300	–
Low power HPC processor	1	165	210	210	–
General purpose 4S processor	1	140	175	190	–
General purpose 2S processor	1	140	140	140	–
1 U rackmount server	4/server	1000	1100	1200	50.4
2 U rackmount server	4/server	1400	1800	2000	42.0
4 U rackmount server	4/server	2300	3100	3300	33.0
7 U blade server chassis	2/blade	5500	6500	7500	45.0
9 U blade server chassis	2/blade	6500	8000	9500	38.0
10 U blade server chassis	2/blade	8000	9000	10,500	42.0

information related to extended surfaces or fin properties. As a result, it may not be used as a linear scaling factor when hypothesizing larger heat spreader areas for the same cooling solution; the conventional heat transfer coefficient is appropriate for such a calculation. The heat spreader heat flux metric does however allow for a direct comparison of heat transfer rates and required server areas among different cooling solutions. The experimental studies examined in this review are ultimately compared through an *effective heat transfer coefficient* calculated as follows:

$$h_{\text{eff}} = \frac{q}{A_{\text{HS}} \cdot (T_w - T_i)} \quad (1)$$

where q is the total heat transfer rate, A_{HS} is the heat spreader area, and T_w and T_i are the heated surface and inlet temperatures, respectively. In the case of pool boiling, T_i is replaced with the coolant bath temperature, T_{bath} .

2.2. Power requirements

Capital and operating expenditures of liquid cooling solutions are seldom documented in literature; hence prohibiting any form of direct quantitative comparison. An indirect quantitative assessment is possible however through examination of *coolant temperatures and flowrates*; properties that are appropriately documented in experimental studies.

Simply put, higher coolant temperatures promote the use of free cooling and waste heat recovery, thus reducing the overall operating cost of vapor-compression cooling [6]. Reduced coolant flowrates decrease the mechanical fan or pump work required for adequate thermal management; thus contributing to a further reduction in operating costs [5].

One consideration readers should keep in mind is that many of the parameters necessary to perform such evaluations are not presented in a clear or effective manner in many experimental publications. This is particularly true for parameters related to coolant flow and temperatures. As a result, the authors have attempted to the best of their abilities to estimate these values based upon available data.

3. Server component heat loads

Major heat generating electronics within servers include processors, memory modules, voltage regulators, chipsets, and power supplies [10]. With respect to thermal management, processors are deemed the most problematic of these components due to their high heat flux; they account for as much as 50% of total server power consumption while taking up minimal server area [10]. Consequently, the majority of experimental studies found in literature consider only processor geometry and heat loads in order to test and evaluate cooling technologies; remaining server components are seldom considered.

As a benchmark for the heat transfer limitations presented in this study, ASHRAE's predicted heat loads for several processor and server classes are presented in Table 1. By the year 2020, individual processor heat loads are expected to reach 140–190 W in general purpose servers and 210–300 W in high performance computing (HPC) applications [10]. Server heat loads are expected to range between 1.2 and 3.3 kW in rackmount servers and between 7.5 and 10.5 kW in blade server chassis [10]. The resulting total heat generated by a single 42 U server rack may therefore range between 33 and 50 kW [10].

In brief, the 42 U server rack is the most common configuration for mounting servers and modular assets within data centers [5]. It measures up to 78 in. in height and up to 25 by 30 in. in width and depth [5]; these dimensions may fluctuate depending upon the presence of additional rack entities such as vented rear and front doors. The minimum thickness of a single modular asset to be mounted onto a standard server rack is 1 rack unit (U), which is equivalent to 1.75 in. A 42 U rack may thus support up to 42 individual 1 U rackmount servers.

The geometric differences between the aforementioned server classes are demonstrated in Fig. 1. Rackmount servers are horizontally orientated modules containing dedicated fans, power supplies, and management infrastructure [4]. The thickness of a

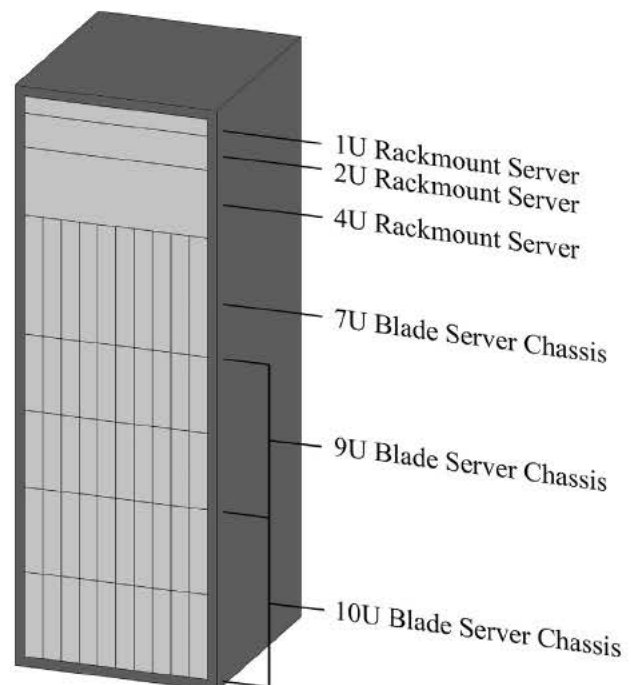


Fig. 1. Size comparison of common server types.

rackmount server may vary depending upon the number of CPUs contained within (and thus the size of the cooling equipment required) or upon the size and quantity of other components such as serial hard drives [5]. Blade server chassis, on the other hand, minimize spatial requirements by distributing power and airflow to several blade servers contained within a single chassis [5]. Blade servers may be horizontally or vertically oriented; regardless of this, the removal of dedicated fans and power supplies permits the storage of a larger number of server modules per rack unit [5]. As shown below however this does not necessarily lead to higher heat dissipation per rack unit.

The server and rack heat loads presented in Table 1 are shown graphically in Fig. 2. The server heat load represents the total heat dissipated by a single server module of a given type; the rack heat load represents the total heat dissipated by a 42 U rack fully

occupied by identical server modules of a given type. 1 U rackmount servers possess the lowest server heat loads (1–1.2 kW); yet their corresponding rack heat load (50 kW) exceeds those of blade server chassis (38–42 kW). The aim of this comparison is solely to discourage the misconception that blade servers are necessarily more problematic from the standpoint of heat dissipation.

4. Cooling solutions

A hierarchy of the cooling solutions discussed in this review is presented in Fig. 3. All cooling solutions may be categorized into three distinct groups: air cooling, indirect liquid cooling, and direct liquid cooling. The following sections summarize the heat transfer characteristics of each cooling solution based on existing experimental work.

4.1. Air cooling

Air cooling is currently the most well established cooling solution adopted by the data center industry and it is commonly implemented through the hot/cold aisle arrangement represented in Fig. 4 [11]. Heated facility air is cooled by either computer room air conditioning or air handling units (CRAC/CRAH) and delivered to allocated cold aisles through an underfloor plenum [9]. The difference here is that CRAH units utilize water–air heat exchangers to cool facility air while CRAC units rely upon refrigerant-filled evaporators [9]. At the server level, all electronics are cooled through direct forced air convection with heat sink attachments required for high heat flux components [9]. Dedicated server fans drive cold aisle air through individual servers and release the heated exhaust air back into the facility through allocated hot aisles [11]. The efficiency of this process may be augmented through the method of aisle containment; where either the hot or cold air streams are confined to prevent air mixing and recirculation [11]. Although not quantified in this review, server fan operation within data centers yields extreme acoustic noise levels that are approaching limitations set by health and safety standards [1].

A detailed schematic of a processor and heat sink assembly typically employed in traditional air cooling systems is shown in Fig. 5. In its most basic form, the chip package consists of a functional die attached to a base substrate via appropriate solder joints

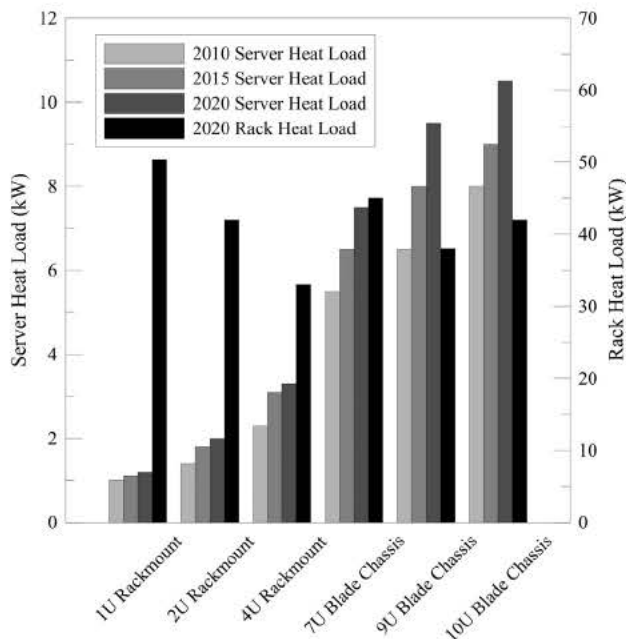


Fig. 2. Comparison of server and rack heat loads for various server types.

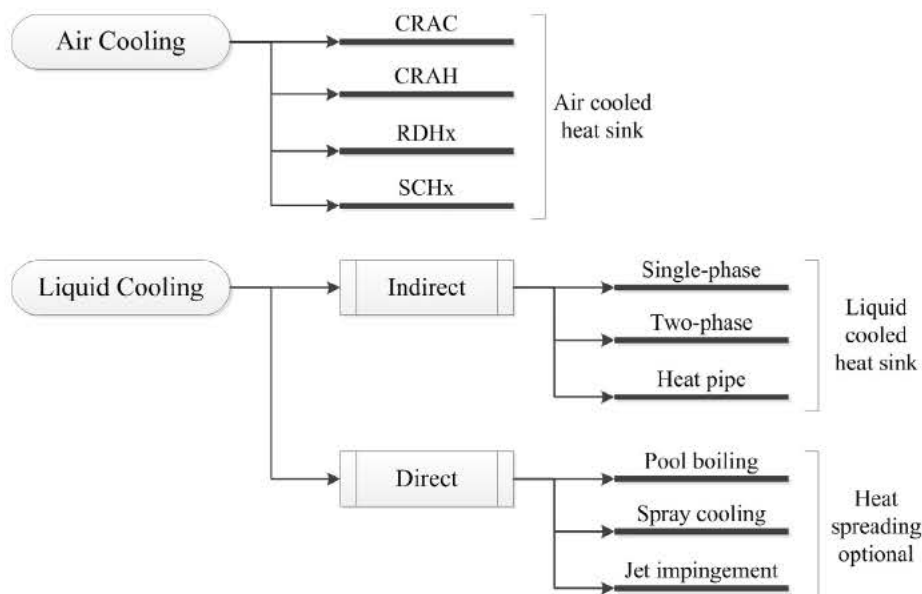


Fig. 3. Air and liquid cooling solutions examined in this review.

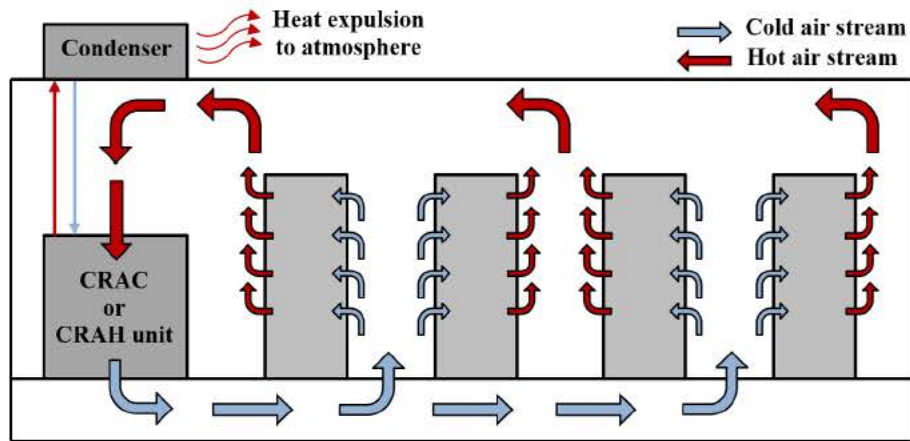


Fig. 4. Hot/cold aisle air cooling arrangement.

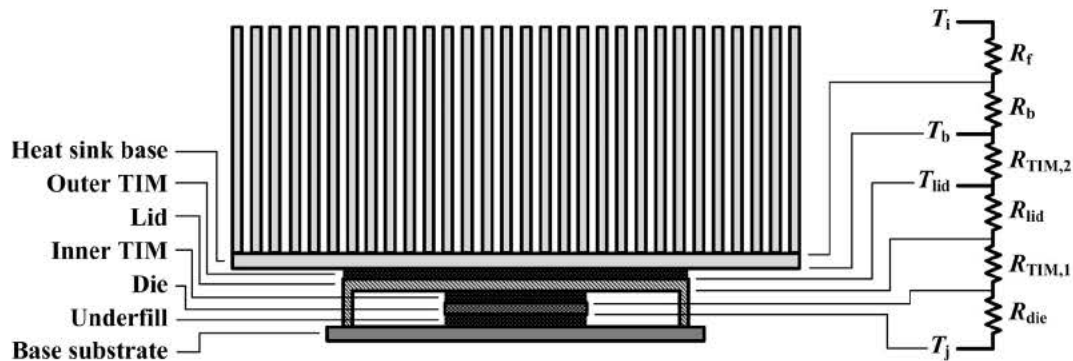


Fig. 5. CPU internal geometry and air cooled heat sink assembly.

Table 2
Air cooling operational characteristics based on experimental work and case studies.

Ref.	Analysis	q (W)	q''_{HS} (W/cm ²)	h_{eff} (W/cm ² K)	Q (cm ³ /s)	T_i (°C)	T_o (°C)	A_{HS} (cm ²)	A_w (cm ²)	T_w (°C)
[13]	Heat sink	250	2.50	0.0417	24,000	30.0	38.6	100	4.00	90
[15]	Heat sink	300	4.00	0.0667	20,000	20.0	32.4	75	4.00	80
[16]	Heat sink	110	1.33	–	7740	–	–	83	1.44	50
[17]	Heat sink	103	1.62	0.0463	12,000	35.0	42.1	64	2.56	70
[18]	Heat sink	227	–	–	–	16.0	40.0	–	–	72
[19]	Rack	6000	–	–	–	16.7	43.1	–	–	–
[11]	Rack	45,000	–	–	–	16.6	32.0	–	–	–
[20]	Data center	–	–	–	–	24.4	–	–	–	–
[20]	Data center	–	–	–	–	20.9	–	–	–	–

[12]. Underfill epoxy encapsulates the solder bumps and absorbs thermal stresses caused by thermal expansion disparities between the die and base substrate [12]. The term *junction temperature* (T_j) is defined as the temperature at the interface between the die and underfill (i.e. the bottom surface of the die) [13]. On the opposing side, the die is enclosed by a lid (also serving as a heat spreader) with thermal interface materials applied on both its inner and outer surfaces [13]. A thermal resistance network between the junction temperature and air inlet temperature (T_i) has also been included in Fig. 5. The sole intention here is to demonstrate the resistive interfaces that exist between the die and cooling equipment (i.e. R_{die} , $R_{TIM,1}$, R_{lid} , $R_{TIM,2}$) regardless of the cooling solution; only R_b and R_f are affected by implementing an enhanced air or liquid cooling solution. The resistive interfaces within the chip package may be subsided using direct liquid cooling strategies however.

Due to the absence of sealed enclosures and facility/server level piping, air cooling is more versatile in relation to liquid cooling strategies [14]. In this context, versatility defines the level of ease with which a cooling solution is implemented across servers of varying forms and sizes. A highly versatile solution facilitates retrofitting and installation processes [1], while also minimizing the need for customization. Additionally, the absence of liquid coolants and fluidic connectors presents the opportunity for low-maintenance, highly reliable hot swapping of entire servers as well as their internal components [14]. These factors, along with simplicity in design and installation, have rendered air cooling a prominent thermal management solution in large data centers.

4.1.1. Heat transfer limitations of air cooling

Heat transfer and flow characteristics of air cooled systems based on recent experimental work are presented in Table 2.

This table examines various domains of analysis including individual heat sinks, racks, and complete data centers. The heat transfer parameters listed include the overall heat transfer rate (q) and the heat spreader heat flux (q''_{HS}) which is calculated based upon the base area of a given heat sink (A_{HS}) rather than the heated surface area (A_w). The temperature parameters listed include the heated surface temperature (T_w), and the inlet and outlet air temperatures (T_i and T_o) which in this case correspond to cold and hot aisle temperatures. The flow parameters listed include the volumetric flowrate of air (Q) and the fan power (P_{fan}) required to induce this flow.

In spite of the popularity of air cooling, only a limited number of studies have actually focused upon its heat transfer limitation. Xu et al. [13] numerically predicted the heat transfer limitation of air cooling to be 340 W at a die temperature of 90 °C and with a heat sink base area of 100 cm². The numerical approach involved three dimensional finite element modelling of the processor package and heat sink base combined with analytical modelling of convective heat transfer across heat sink fins. The heat transfer limitation was identified through a parametric study of material properties, assembly configurations, and fin geometries. The highest heat spreader heat flux of 4.0 W/cm² was obtained from Katoh et al. [15] where a traditional heat sink was compared experimentally to an enhanced pulsating heat pipe heat sink assembly. The pulsating heat pipe repeatedly contacts the heat source and heat sink fins in a serpentine fashion; thus enhancing heat spreading and yielding a 40% reduction in thermal resistance relative to traditional heat sinks.

Given that the heat generation rate of general purpose processors is expected to range from 140 to 190 W by the year 2020 [10], the current heat transfer limitation of air cooling is clearly not an inhibiting factor to server cooling. In the case of HPC however, where heat generation rates currently range from 210 to 300 W [10], the heat transfer limitation of air cooling does not permit further increase in chip power densities.

The heat transfer limitation of air cooling may be extended by means of increasing heat sink base and fin areas; although this is counterproductive given the nature of the industry to minimize spatial requirements [1]. Limitations may also be extended by reducing cold aisle air temperatures; however this challenges ASHRAE facility temperature specifications for operator comfort and safety [21]. Reduced cold aisle temperatures also increase operating costs associated with vapor-compression cooling.

4.1.2. Power requirements

Due to poor thermophysical properties, air cooling generally requires lower coolant temperatures and higher flowrates relative to liquid cooling strategies; both of which increase operating expenses. Cold aisle temperatures have typically ranged from 16 to 25 °C while hot aisle temperatures have ranged from 30 to 45 °C [8,9,22]. In order to attain feasible free cooling, outdoor air temperatures must remain sufficiently lower than cold aisle temperatures [6]. In particular, case studies have indicated that outdoor air temperatures between 9 and 16 °C are ideal for free cooling of air cooled data centers [23,24]. These climatic requirements limit the free cooling capability of air cooled data centers in warmer regions.

Unfortunately, studies involving air cooling seldom provide the information necessary to calculate fan power requirements. As a result, technical documentation from fan manufacturers SUNON [25] and DELTA [26] has been examined for 80 × 80 mm server fans (which are standard for 2 U servers) that deliver the flowrates observed in experimental studies. Rather than providing a specific range of fan power requirements, it suffices to say that their magnitudes are in the order of watts and not hundredths or thousandths of watts.

A prominent method for reducing power consumption in air cooled data centers is to utilize rear door (RDHx) or side car (SCHx) heat exchangers [27–29]. These technologies embody water–air heat exchangers located along the rear or sides of individual server racks, and serve to cool heated exhaust air prior to its re-entry into the facility [29]. This approach of delivering cooling water to individual racks alleviates the power consumed by CRAC/CRAH fans and is estimated to reduce operating and capital costs by as much as 30–40% [29].

4.2. Indirect liquid cooling

Indirect liquid cooling involves heat dissipation in a manner such that no direct contact exists between the liquid coolant and electronic components. This strategy is implemented by replacing traditional air cooled heat sinks with some form of liquid cooled heat sink or evaporator [9]. Recent research in this field has been focused on the performance of microchannel heat sinks (MCHS) due to their enhanced heat transfer characteristics relative to traditional cold plates or waterblocks [30,31].

In a typical implementation shown in Fig. 6, coolant distribution units (CDU) exchange heat between an external cooling source (i.e. supply water, water tower, chiller, etc.) and the closed internal coolant loop that is thermally attached to the electronics [28]. Within the facility, coolant is delivered to individual racks and servers where heat is extracted through forced convection; additional heat transfer mechanisms may exist at the chip level, however these are specific to the type of liquid cooling solution being discussed. Typically, only processors are liquid cooled due to their high heat flux and simplified geometries while remaining server components are air cooled [14]. Nonetheless, certain cooling solutions, such as Asetek's RackCDU D2C™ [32] and IBM's Power 775 Supercomputer [28], have extended liquid cooling to additional server components.

Indirect liquid cooling enables the use of coolants with substantially superior thermophysical properties relative to air [1]; thus extending heat transfer limitations and decreasing power consumption by permitting higher coolant temperatures and reduced flowrates. The trade-off however is a reduction in versatility due to the addition of sealed enclosures and facility/server level piping required to maintain and direct liquid flow [14]; thus cooling systems must now be tailored to specific server and facility layouts. Furthermore, fluidic connections prohibit the easy hot swap access to server electronics that is customarily permitted with air cooling [14]. Quick-disconnect fluidic connectors still permit low-maintenance hot swap access; however they raise reliability concerns by presenting an additional opportunity for leakage [1]. The following subsections provide further details on the three major types of indirect liquid cooling: single-phase, two-phase, and heat pipe cooling.

4.2.1. Single-phase cooling

Single-phase cooling specifically involves sensible heating of the circulating coolant; hence no phase change is expected or permitted to occur. Water is accordingly the most practical choice of coolant here due to its high boiling point and superior thermophysical properties relative to common dielectric fluids and refrigerants [33]. There are however added risks pertaining to coolant leakage that have discouraged enthusiasm for water cooling systems within data centers [1]. Selected industry implementations of water cooling in servers have been developed by IBM [34], Asetek [35], and CoolIT Systems [36]. A recently common feature in water cooled systems is the integration of cold plates and centrifugal pumps into a single package. This reduces spatial requirements while increasing reliability in servers with multiple processors that are liquid cooled in series [35].

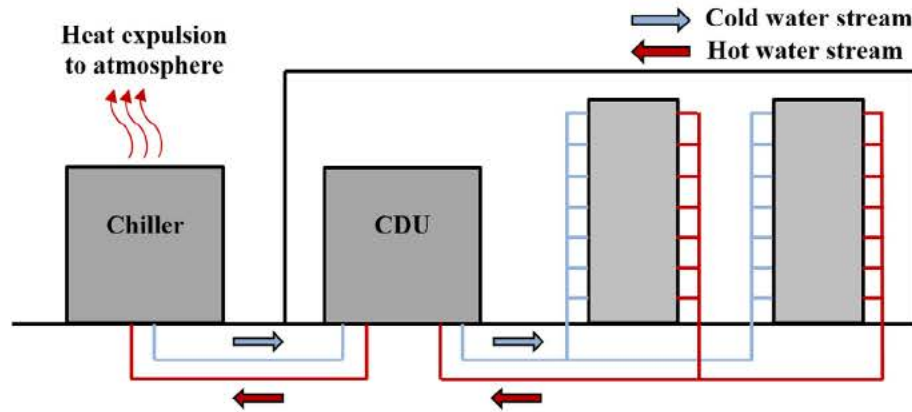


Fig. 6. Indirect liquid cooling arrangement.

Table 3
Single-phase water cooling operational characteristics based on experimental work.

Ref.	Type	q (W)	q''_{HS} (W/cm ²)	h_{eff} (W/cm ² K)	Q (cm ³ /s)	T_i (°C)	T_o (°C)	A_{HS} (cm ²)	A_w (cm ²)	T_w (°C)	ΔP (kPa)	P_{pump} (W)
[39]	PM	140	50	0.625	4.20	20	30	2.8	0.5	100	35	0.14
[37]	MCHS	1150	43	1.87	27	20	30	27	27	43	0.90	0.025
[40]	MCHS	36	100	4	6.40	20	33.6	0.36	0.36	45	115	0.75
[41]	MCHS	448	100	2.44	10.1	15	28	4.48	4.48	56	30	0.3
[42]	MCHS	272	121	3.46	16.7	35	39	2.25	2.25	70	55.3	0.921
[43]	MCHS	220	137	2.12	31.7	20	21.7	1.61	0.15	85	21	0.67
[38]	MCHS	273	170	3.62	2.70	25	50	1.61	1.61	72	10	0.027
[44]	MCHS	25	2.68	0.103	0.33	20	38	9.45	9.45	46	38.6	0.013
[45]	MCHS	30	9.87	0.395	0.40	25	42.3	3.04	3.04	50	4.0	0.0016
[46]	MCHS	625	100	1.82	6.67	20	42	6.25	6.25	75	1.0	0.0067
[47]	MCHS	375	93.8	1.25	16.7	50	55	4	3.75	125	19.5	0.33
[48]	MCHS	170	47.8	2.39	16.7	60	62	3.55	2.31	80	14	0.23

Heat transfer and flow characteristics of single-phase water cooling systems based on recent experimental work are presented in Table 3. These studies have focused upon chip level thermal management through some form of cold plate design such as microchannel or porous medium (PM) heat sinks. Additional parameters included in this table are the heat sink pressure drop (ΔP) which, when multiplied by the coolant volumetric flowrate (Q), yields the required pumping power (P_{pump}) necessary to maintain the given flow. The highest heat transfer rate of 1150 W has been obtained from Rimbault et al. [37] where the performances of CuO–water nanofluid and water were compared experimentally using a 27 cm² microchannel heat sink. Interestingly, pure water yielded the largest heat transfer coefficient as well as the lowest heat sink pressure drop relative to all tested volume fractions of CuO–water nanofluid. The highest heat spreader heat flux of 170 W/cm² has been obtained from Lee et al. [38] where sectional oblique fins were used instead of continuous channels to modulate flow. This discontinuity in the extended surfaces forced re-initialization of thermal boundary layers as well as secondary flow; both of which contributed to enhanced heat transfer. Overall, heat spreader heat flux values in single-phase water cooled systems have ranged from 2.68 to 170 W/cm² which represents a substantial improvement relative to air cooling.

Inlet water temperatures have ranged from 15 to 35 °C in experimental studies concerning microchannel heat sinks; however an impressive inlet temperature range of 45–60 °C has been observed in industry implementations [34,48–50]. Clearly, these temperature ranges promote free cooling and waste heat recovery in a wider spectrum of global climates relative to traditional air cooling. Liquid cooling of processors also alleviates the load on

air conditioning units as higher air temperatures may now sufficiently cool the remaining low heat flux server components; air temperatures as high as 50 °C have been observed in industry implementations [49,50]. This extends the potential of free cooling and waste heat recovery to the high temperature circulating air.

Considering the trade-off between heat recovery and chip thermal reliability, Sharma et al. [51] have recommended (via optimization strategies) an optimal water inlet temperature range of 40–50 °C with a flowrate of 1 L/min (16.6 cm³/s). A recent experimental study by Carbó et al. [52] complements and drives this point further. Using a 1.2 kW micro data center test bench involving four liquid cooled servers (each employing two Intel Xeon 8 core processors), the waste heat recovery potential of a liquid cooled data center was characterized. Inlet water temperatures of 20 and 40 °C yielded outlet temperatures of 32.5 and 50 °C, respectively, at a flowrate of 1 L/min. The higher end of this range clearly maximized heat reuse potential; however it also corresponded to a 3% increase in IT power consumption (resulting from leakage currents inside silicon chips and increased fan power consumption) and a 9% reduction in liquid cooling capacity (i.e. the portion of generated heat that was dissipated by water reduced from 81% to 72%). These results are indicative of the fact that the design of an ideal liquid cooling solution is largely dependent upon the priorities of data center management.

Water flowrates for cooling a single processor have ranged from 0.3 to 32 cm³/s. Based on the measured pressure drop across various heat sinks, pumping power requirements have been calculated to range from 0.002 to 0.9 W per processor. This reduction in power consumption relative to air cooling is clearly attributed to the increased density and specific heat capacity of water [33];

however large pressure drops generated across microchannel heat sinks still yield significant pumping power requirements in certain cases [42].

4.2.2. Two-phase cooling

Two-phase cooling involves latent heating of the circulating coolant; hence liquid–vapor phase change is exploited. Choices of coolants commonly encompass various refrigerants and dielectric fluids with boiling temperatures lower than acceptable die operating temperatures [53]. The added benefits of this solution include enhanced heat transfer rates and reduced temperature gradients across heated surfaces, both of which are attributed to the isothermal nucleation processes present in two-phase flow as well as the associated latent heat of vaporization [30,31].

The major concerns with this strategy are related to flow instabilities which induce pressure and temperature fluctuations, flow reversal, and several other behaviors that may ultimately result in superheating and burnout of heated surfaces [30]. Two-phase cooling solutions in industry are scarce; a selected implementation has been developed by Ebullient [54]. Their technology consists of jetted cold plates that are capable of dissipating 500 W of heat at an ultimate heat rejection temperature of 55 °C.

Heat transfer and flow characteristics of two-phase cooling systems based on recent experimental work are presented in Table 4. The outlet temperature in this table has been replaced by the coolant saturation temperature (T_{sat}) as all studies maintained the fluid quality below 1. The highest heat transfer rate of 2160 W has been obtained from Pranoto and Leong [55] where flow boiling heat transfer characteristics within graphite foam structures have been examined. Dielectric liquid FC-72 with a saturation temperature of 56 °C and no subcooling was driven through a 48 cm² porous heat sink maintaining a heated surface temperature of 85 °C. This study also examined the effects of adjusting the evaporator gap above the porous structure. Smaller evaporator gaps caused bubble confinement and vapor layer formation over foam surfaces which significantly increased wall temperatures. The highest heat spreader heat flux of 222 W/cm² has been obtained from Agostini et al. [56] where the saturated critical heat flux of refrigerant R236fa within a 4 cm² straight finned microchannel heat sink has been examined. High operating pressures yielded a saturation temperature of 25 °C for R236fa which maintained a heated surface temperature of 75 °C with no subcooling. Overall, heat spreader heat flux values in two-phase cooling systems have ranged from 24.2 to 222 W/cm² which represents an improvement relative to single-phase water cooling.

Coolant inlet temperatures in experimental studies have ranged from 0 to 63 °C with saturation occurring at temperatures ranging from 25 to 63 °C; these temperature ranges are comparable to those of single-phase water cooling. Coolant flowrates have ranged from 0.66 to 12.2 cm³/s and pumping power requirements have been calculated to range from 0.002 to 0.62 W per processor. This indicates a slight reduction in pumping power requirements relative to water cooling which is attributed to the latent heat transfer associated with two-phase working fluids [72].

4.2.3. Heat pipe cooling

A heat pipe is a passive two-phase heat transfer device driven strictly by a temperature difference across its terminals. Vapor formed at the heat pipe evaporator effectively transports energy to the condenser by means of pressure driven advection; while condensed fluid formed at the condenser is driven back to the evaporator through either capillary action (wick heat pipe) or gravity (thermosyphon) [73]. The near-vacuum pressure within the sealed assembly maintains coolants in a consistently saturated state; thus permitting the use of water as a two-phase coolant in electronics cooling applications [74].

Loop heat pipes (LHP) with flat evaporators in particular have gained precedence in cooling applications due to their compatibility with processor geometries as well as their effectiveness in transferring heat over large distances relative to traditional heat pipes [74]. The operational characteristics of loop heat pipes are demonstrated in Fig. 7. Heat transferred to the evaporator base (q_{in}) vaporizes the liquid coolant preserved within the porous wick structure in the vicinity of the vapor removal channels; this in turn produces the necessary pressure to drive the newly formed vapor out of the evaporator [75]. However due to a combination of surface tension and adhesive forces between the liquid and wick pores, the menisci formed at the liquid–vapor interface generate sufficient capillary pressure to prevent vapor incursion into the wick structure [75]. The result is vapor accumulation within the vapor collection chamber until there is sufficient pressure to drive the vapor through the vapor line and to the condenser [73]. Heat extraction at the condenser (q_{out}) condenses the vapor flow, allowing liquid coolant to return to the compensation chamber located near or within the evaporator assembly [75]. The purpose of the compensation chamber is to store excess liquid and uncondensed vapor much like a reservoir in two-phase cooling applications [75]. This entire process can only be maintained while the pressure generated through capillary action across the wick equates the cumulative pressure drop associated with each

Table 4

Two-phase cooling operational characteristics based on experimental work.

Ref.	Type	Coolant	q (W)	q''_{HS} (W/cm ²)	h_{eff} (W/cm ² K)	Q (cm ³ /s)	T_i (°C)	T_{sat} (°C)	A_{HS} (cm ²)	A_w (cm ²)	T_w (°C)	ΔP (kPa)	P_{pump} (W)
[57,58]	MCHS	FC-72	250	40	0.721	2.1	29.5	56	6.25	6.25	85	5.0	0.0103
[57,58]	MCHS	FC-72	375	60	1.08	2.9	29.5	56	6.25	6.25	85	7.0	0.020
[59,60]	MCHS	R134a	425	188	8.55	6.57	63	63	2.64	2.26	85	94	0.62
[56,61]	MCHS	R236fa	888	222	4.44	7.47	25	25	4	4	75	100	0.50
[62]	MCHS	R134a	1080	180	2.05	12.2	22	25	6	4	110	–	–
[63]	MCHS	R134a	216	88.2	5.88	3.26	29.1	29.1	2.45	2.45	44.1	–	–
[63]	MCHS	R245a	198	80.8	3.23	2.96	26	26	2.45	2.45	51	–	–
[64]	MCHS	R134a	1100	183	–	5.34	20	25	6	4	–	–	–
[64]	MCHS	R236fa	812	135	5.4	3.21	30	40	6	4	55	–	–
[64]	MCHS	R245fa	860	143	–	2.30	25	40	6	4	–	–	–
[65]	MCHS	FC-72	84.9	24.2	1.72	0.663	54	56.1	3.5	3.5	68.1	–	–
[66]	MCHS	R134a	61.9	27.5	1.96	2.07	20	25	2.25	2.25	34	–	–
[67]	MCHS	R407C	64.7	28.75	1.85	2.20	20	25	2.25	2.25	35.5	–	–
[68]	MCHS	HFE-7100	227	105.8	4.41	2.28	61	61	2.15	2.15	85	1.0	0.0023
[55]	PM	FC-72	2160	45	1.55	–	56	56	48	48	85	–	–
[69]	MCHS	HFE-7100	84.9	173	2.16	1.32	20	59.6	0.491	0.491	100	10	0.0132
[70,71]	MCHS	HFE-7100	98	200	2.5	1.32	0	60	0.49	0.49	80	10	0.0132

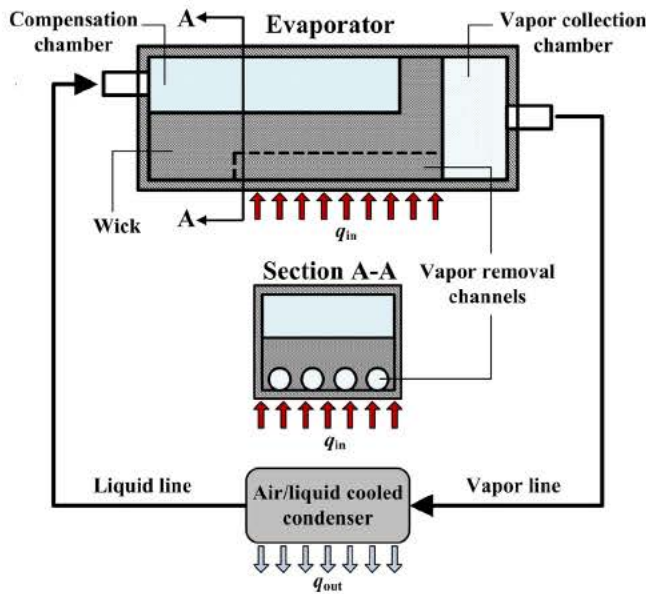


Fig. 7. Loop heat pipe process structure and process diagram.

subsequent step [73]. There exist more sophisticated loop heat pipe designs; however the current discussion suffices given the scope of this review. Maydanik et al. [74] review various forms of loop heat pipes with flat evaporators. Ku [75] provides a very elaborate account of the operating characteristics of loop heat pipes with secondary wicks (which has been omitted from the present discussion).

One added benefit of heat pipe cooling is the mitigation of leak risks within servers; heat pipes are highly reliable hermetically sealed devices that require no fluidic connectors at the chip level [76]. Additionally, the passivity of heat pipes nullifies pumping power requirements at the chip level while increasing reliability due to the absence of moving parts [76]. A selected industry implementation of loop heat pipe cooling in servers has been developed by Calyos [77] where various condenser cooling approaches are

offered; these include water and air cooled condensers, as well as a conductive thermal contact heat exchanger which excludes fluidic connectors for easy hot swapping. A selected implementation of traditional heat pipe cooling in servers has been developed by Xyber Technologies [78] where the side walls of an actual server enclosure are replaced by finned heat sinks in order to act as large air cooled condensers.

Heat transfer and flow characteristics of heat pipe cooling systems based on recent experimental work are presented in Table 5. This table lists the primary coolant within the heat pipe as well as the heat transfer medium used to cool the heat pipe condenser (i.e. air or water). The inlet and outlet temperatures (T_i and T_o) in this case represent the condenser coolant temperatures. The highest heat transfer rate of 700 W has been obtained from Maydanik et al. [79] where a 31 cm long compact copper–water loop heat pipe with a 10.24 cm² flat-oval evaporator was tested. A heated surface temperature of 87 °C was maintained by cooling the loop heat pipe condenser with 20 °C water. The highest heat spreader heat flux of 68.4 W/cm² has been obtained from the same study. Overall, heat spreader heat flux values in heat pipe cooling systems have ranged from 2.4 to 68.4 W/cm². This indicates a significant reduction in heat transfer limitations relative to water cooling. Nevertheless, this range exceeds the heat transfer limitations of traditional air cooling and permits effective thermal management of both general and HPC servers with a higher dependency upon heat spreading than in the case of single or two-phase liquid cooling systems. With regards to coolant temperatures, heat pipe condensers have been cooled using both air and water with temperatures ranging from 5 to 55 °C in both cases. As with active two-phase cooling, these temperature ranges are also comparable to those of water cooling.

4.3. Direct liquid cooling

In contrast to indirect liquid cooling, direct liquid cooling permits contact between liquid coolants and electronic components by exploiting the electrical insulation provided by dielectric fluids [53,97]. One immediate benefit of such a strategy is an increase in the versatility of the cooling solution as piping and sealed

Table 5
Heat pipe cooling operational characteristics based on experimental work.

Ref.	Heat pipe coolant	q (W)	q''_{HS} (W/cm ²)	h_{eff} (W/cm ² K)	Condenser coolant	Q (cm ³ /s)	T_i (°C)	T_o (°C)	A_{HS} (cm ²)	A_w (cm ²)	T_w (°C)
[80]	Water	200	46.95	1.1	Air	–	22.5	–	4.26	4	65
[80]	Water	150	35.21	1.41	Air	–	55	–	4.26	4	80
[81]	Water	200	14.3	0.231	Air	15,000	23	34.1	14	14	85
[81]	Methanol	175	12.5	0.202	Air	15,000	23	32.72	14	14	85
[81]	R141b	150	10.7	0.173	Air	15,000	23	31.3	14	14	85
[82]	Water	100	8.3	0.163	Air	–	25	–	12	12	75.8
[82]	NF	150	12.5	0.242	Air	–	25	–	12	12	76.7
[83]	Water	60	2.4	0.12	Air	–	50	–	25	25	70
[83]	SiO ₂ –H ₂ O	60	2.4	0.16	Air	–	45	–	25	25	60
[84]	Methanol	120	10	0.222	Air	–	20	–	12	12	65
[85]	Water	138	8.27	0.136	Air	–	24	–	16.62	9	85
[86]	Water	197	27.8	1.11	Air	–	25	–	7.1	7.1	50
[87]	Water	350	38.9	0.635	Air	22,300	23.7	36.78	9	6.25	85
[88]	Ammonia	130	8.13	0.163	Air	6040	20	37.94	16	16	70
[88]	Ammonia	230	14.38	0.288	Water	8.67	20	26.35	16	16	70
[88]	Ammonia	320	20	0.476	Water	8.67	20	29.83	16	16	62
[89,90]	Water	60	8.49	0.135	Air	–	22	–	7.07	6.25	85
[91]	Water	300	33.3	0.564	Air	40,116	26	32.23	9	6.25	85
[92]	Water	80	10	0.167	Water	–	25	–	8	8	85
[79]	Water	700	68.4	1.02	Water	8.33	20	40.10	10.24	9	87
[93]	Water	100	4.35	0.087	Water	38.9	20	20.62	23	16	70
[93]	Water	100	4.35	0.145	Water	38.9	55	55.62	23	16	85
[94]	Methanol	100	3.7	0.0463	Water	33.3	5	6.48	27	25	85
[95]	Acetone	60	5	0.141	Air	–	16.5	–	12	12	52
[96]	Water	140	12.35	0.274	Air	–	15	–	11.34	9	60

enclosures are no longer required at the server level to maintain and direct liquid flow [14]. Furthermore, a hybrid air–liquid cooling solution is no longer necessary for cooling remaining low heat flux server components; these components may be directly and more effectively cooled by dielectric fluids [14]. Direct liquid cooling solutions involving phase change also yield uniform surface temperatures due to the associated latent heat transfer [97].

Although the direct contact and phase change phenomena involved in direct liquid cooling applications may result in reduced thermal resistances [97], the thermophysical properties of dielectric fluids are notably inferior to those of water [33]. It is therefore invalid to immediately associate direct liquid cooling with having enhanced heat transfer characteristics relative to traditional indirect liquid cooling strategies.

Direct liquid cooling solutions that are open to atmospheric pressure generally permit low-maintenance hot swapping due to the absence of hermetically sealed enclosures [14]; however additional maintenance procedures are required to prevent fluid losses and to degas systems of infiltrating air and humidity [98,99]. This issue is less prevalent in hermetically sealed direct liquid cooling solutions. However these systems require additional maintenance and servicing procedures for hot swapping electrical components; enclosures must first be evacuated and then refilled with coolant prior to being re-commissioned [14]. The following subsections provide further details on the three major types of direct liquid cooling: pool boiling, spray cooling, and jet impingement.

4.3.1. Pool boiling

Pool boiling, also referred to as two-phase passive immersion cooling, is a passive all-liquid (no air) cooling solution where electronics are fully immersed in a dielectric bath [97]. The temperature rise of high heat flux surfaces above the saturation temperature of the coolant induces nucleate boiling within the dielectric bath; thus cooling components through latent heat transfer, bubble induced flow mixing, and gravity driven two-phase convection [100]. In a typical implementation shown in Fig. 8, vapor bubbles that are formed at heated surfaces rise to an allocated vapor space above the dielectric bath where the coolant is condensed through a water-cooled heat exchanger [14]. Recent research in this field has been focused on the effects surface enhancements upon the heat transfer characteristics of pool boiling [101].

A primary benefit of open bath immersion cooling in specific is the absence of hermetically sealed enclosures, server level piping, and fluidic connectors; thus yielding a highly versatile solution that also permits low-maintenance hot swapping of server components [14]. Additionally, the passive heat transfer characteristics

eliminate chip level pumping power requirements while improving reliability [14]; essentially emulating heat pipes while operating at atmospheric pressure.

Heat transfer and flow characteristics of pool boiling systems based on recent experimental work are presented in Table 6. This table lists the type of surface enhancement (if any) tested in each study. A heat spreader area (A_{HS}) labelled as *not applicable* (n/a) indicates that no heat spreading was used in the given study; the heated surface was cooled directly by the dielectric coolant. Inlet and outlet temperatures (T_i and T_o) have been replaced by the dielectric fluid saturation temperature (T_{sat}) and the bath temperature (T_{bath}) which may differ from the former when the coolant is subcooled. The highest heat transfer rate of 156 W has been obtained from Misale et al. [102] where the impact of confining a heated surface with a face-to-face parallel unheated surface was studied. Dielectric liquid HFE-7100 with a saturation temperature of 61 °C and no subcooling was used to maintain a roughened 7.07 cm² heated surface at 72.5 °C. The major findings indicated that a smaller confinement region drastically reduced heat transfer at high surface temperatures while such an effect was less apparent at surface temperatures closer to the fluid saturation temperature. The highest heat spreader heat flux of 98.3 W/cm² has been obtained from Yu and Lu [103] where the effects of rectangular fin arrays on pool boiling heat transfer were studied. Dielectric liquid FC-72 with a saturation temperature of 56 °C and no subcooling was used to maintain a 1 cm² finned heated surface at 85 °C. It was identified that closely packed and taller fins engendered greater flow resistance against vapor liftoff between adjacent fins causing the heat transfer coefficient to rapidly decay. Overall, heat spreader heat flux values for pool boiling systems have ranged from 11 to 98.3 W/cm². This indicates an improvement in heat transfer limitations relative to loop heat pipe cooling systems; yet this range of heat flux values is still inferior relative to active indirect liquid cooling strategies.

Information regarding water temperatures within condensers is seldom documented in experimental studies; however atmospheric boiling temperatures for commonly used dielectric coolants have ranged from 49 to 61 °C while certain studies have subcooled dielectric baths to as low as 11 °C. Industry implementations of open bath pool boiling have recorded water temperatures as high as 60 °C within condenser coils while maintaining chip temperatures below 85 °C [14]. As with all liquid cooling strategies reviewed thus far, these temperature ranges are comparable to those of water cooling.

These characteristics have rendered pool boiling a major contender to air cooling; which justifies the recent growth of immersion cooling within the data center industry. A selected implementation of open bath two-phase immersion cooling in servers has been developed by 3M [14]. Alternative approaches to immersion cooling within industry have been implemented by LiquidCool Solutions [115] and Green Revolution Cooling [116] where heat transfer is dominated by dielectric coolant flow rather than nucleate boiling. Studies examining the combined effects of forced convection and nucleate boiling in immersion cooling systems have been performed by Gess et al. [106,117]. A comparison between pure pool boiling and a flow condition of 660 mL/min was performed within a dielectric bath. The addition of forced convection raised the heat transfer limitation of the system by 45% at a heated surface temperature of 80 °C.

4.3.2. Spray cooling

Spray cooling involves the dispersion or atomization of liquid coolants into fine droplets prior to achieving contact with a heated surface [118]; this is accomplished through the large pressure differential generated across nozzle orifices. The heat transfer mechanisms involved with spray cooling are complex and are still

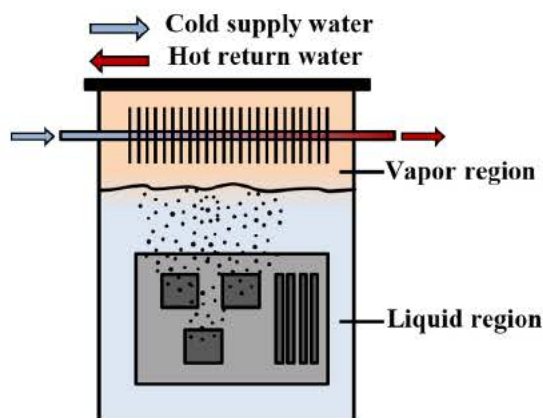


Fig. 8. Pool boiling arrangement.

Table 6

Pool boiling operational characteristics based on experimental work.

Ref.	Surface	Coolant	q (W)	q''_{HS} (W/cm ²)	h_{eff} (W/cm ² K)	T_{sat} (°C)	T_{bath} (°C)	A_{HS} (cm ²)	A_w (cm ²)	T_w (°C)
[104]	Smooth	FC-72	70.95	11	0.393	56	56	n/a	6.45	84
[104]	Particles	FC-72	103.2	11.12	0.383	56	56	9.29	6.45	85
[105]	Smooth	Novec 649	71.88	11.5	0.479	49	49	n/a	6.25	73
[105]	Smooth	HFE-7100	75	12	0.545	61	61	n/a	6.25	83
[105]	Microfins	Novec 649	109.38	17.5	1.46	49	49	n/a	6.25	61
[105]	Microfins	HFE-7100	109.8	17.5	1.94	61	61	n/a	6.25	70
[106]	Smooth	Novec 649	70	12.15	0.419	49	49	n/a	5.76	78
[106]	Microporous	Novec 649	78.75	13.67	1.24	49	49	n/a	5.76	60
[106]	Microfins	Novec 649	80	13.89	1.26	49	49	n/a	5.76	60
[99]	Smooth	FC-72	17	17	0.81	51.5	51.5	n/a	1	72.5
[99]	Porous	FC-72	28	28	2.33	51.5	51.5	n/a	1	63.5
[99]	Smooth	FC-72	30	30	0.545	51.5	21.5	n/a	1	76.5
[99]	Porous	FC-72	58	58	1.16	51.5	21.5	n/a	1	71.5
[107]	Microporous	PF-5060	25	25	3.85	51.4	51.4	n/a	1	57.9
[108]	Porous	FC-72	30	30	1.07	56	56	n/a	1	84
[108]	Porous	HFE-7100	30	30	1.25	61	61	n/a	1	85
[108]	Porous	FC-72	57.5	57.5	0.991	56	26	n/a	1	84
[108]	Porous	HFE-7100	65	65	1.2	61	31	n/a	1	85
[109]	Smooth	FC-72	17	17	0.633	56	56	n/a	1	83
[109]	Nanoparticle	FC-72	22	22	0.88	56	56	n/a	1	81
[102,110]	Roughened	HFE-7100	155.54	22	1.91	61	61	n/a	7.07	72.5
[111]	Microporous	FC-72	8.2	20	1.25	56	56	n/a	0.41	72
[111]	Microporous	FC-72	15.6	38	0.585	56	21	n/a	0.41	86
[112]	Micro-cavity	FC-72	30	30	1	56	56	n/a	1	86
[103]	Fin array	FC-72	98.3	98.3	2.89	56	56	n/a	1	90
[113]	Rough pin fin	FC-72	30	30	1.03	56	56	n/a	1	85
[113]	Rough pin fin	FC-72	50	50	1.06	56	31	n/a	1	78
[113]	Rough pin fin	FC-72	63	63	0.913	56	11	n/a	1	80
[114]	Hoodoo	FC-72	25	25	0.714	56	56	n/a	1	91

subject to investigation. Mechanisms such as single-phase convection, thin film evaporation, and secondary nucleation have been suggested by researchers attempting to justify the enhanced heat transfer rates observed in spray cooling systems [118,119].

Direct spray cooling may be implemented in a versatile manner that is capable of universally cooling all server components. One particular design suggested by Yan et al. [120] employs inclined gas-assisted nozzles to induce multiple sprays covering the majority of a server board's area. The issue here however is that server electronics are now sealed within the vapor chamber that maintains two-phase coolant flow. This will require high-maintenance hot swapping procedures as with any hermetically sealed direct liquid cooling solution [14].

In contrast, spray cooling may be implemented in an indirect manner where the heat transfer phenomena take place within specialized cold plates. A selected example of such an approach has been developed for military environments by SprayCool™ [121]

where air cooled heat sinks within servers are replaced with spray cooled cold plate modules. This strategy permits easy hot swap access to electronics by means of quick-disconnect fluidic connectors; however this solution bears more resemblance to indirect liquid cooling and is subject to similar drawbacks in versatility.

Heat transfer and flow characteristics of spray cooling systems based on recent experimental work are presented in Table 7. The highest heat transfer rate of 503 W has been obtained from Chen et al. [122] where the heat transfer characteristics of refrigerants R134a and R22 are compared in spray cooling applications. R22 with a saturation temperature of 11 °C and an inlet temperature of 0 °C was used to maintain a smooth 2.01 cm² heated surface (503 W) at 46 °C. On the other hand, R134a at a saturation temperature of 26.7 °C and an inlet temperature of 0 °C maintained the same heated surface at 72.7 °C with a heat transfer rate of only 235.17 W. Although R22 is preferable in terms of heat transfer characteristics, R134a is considered a prospective alternative due

Table 7

Spray cooling operational characteristics based on experimental work.

Ref.	Surface	Coolant	q (W)	q''_{HS} (W/cm ²)	h_{eff} (W/cm ² K)	Q (cm ³ /s)	T_i (°C)	T_{sat} (°C)	A_w (cm ²)	T_w (°C)	ΔP (kPa)	P_{pump} (W)
[123]	Smooth	R134a	231.15	115	2	5.93	9.37	0.63	2	48.13	–	–
[124]	Smooth	PF-5060	17.5	3.97	0.064	0.36	26	56.7	4.4	88	207	0.0751
[125]	Smooth	R134a	240	120	4.8	6.50	10	10	2	35	480	3.121
[125]	Macro-pins	R134a	340	170	6.8	6.50	10	10	2	35	480	3.121
[125]	Micro-pins	R134a	420	210	8.4	6.50	10	10	2	35	480	3.121
[126]	Smooth	PF-5060	160	80	1.62	3.33	20.5	31	2	70	379	1.262
[126]	Fins	PF-5060	250	125	2.53	3.33	20.5	31	2	70	379	1.262
[127]	Smooth	R134a	137.45	28	1.04	0.67	10.7	15.7	4.9	37.7	80	0.0538
[127]	Cu Foam	R134a	196.36	40	2.58	1.12	10.7	15.7	4.9	26.2	240	0.269
[128]	Smooth	Methanol	100	16	0.604	–	28.5	64.7	6.25	55	100	–
[129]	Smooth	R134a	222.46	138	8.12	4.83	20.3	20.3	1.61	37.3	105	0.507
[122]	Smooth	R134a	235.17	117	1.61	6.04	0	26.7	2.01	72.7	400	2.416
[122]	Smooth	R22	502.5	250	5.43	5.87	0	11	2.01	46	400	2.35
[130]	Smooth	R134a	290	145	2.99	8.04	0.6	0.6	2	48	474	3.809
[131]	Cu Foam	R134a	147.27	30	2.31	0.68	10.7	15.7	4.9	23.7	80	0.0545
[131]	Grooved	R134a	132.54	27	1.17	0.68	10.7	15.7	4.9	33.7	80	0.0545
[132,133]	Roughened	R134a	309.7	192	8.53	8.33	20.3	20.3	1.6	42.8	300	2.5

to its zero ozone depletion potential, low global warming potential, non-toxicity, and non-inflammability. The highest heat spreader heat flux of 250 W/cm² has been obtained from the same study with R22 as coolant. Overall, heat spreader heat flux values for spray cooling systems have ranged from 3.97 to 250 W/cm².

This range of heat flux values exceeds that of indirect two-phase cooling; however this comes at the cost of increased pumping power requirements attributed to the large pressure drop generated across spray nozzle orifices. Flowrates for cooling a single processor have ranged from 0.4 to 8.3 cm³/s and pumping power requirements have been calculated to range from 0.05 to 3.8 W per processor. Coolant inlet temperatures within these studies have ranged from 10 to 29 °C with saturation occurring at temperatures ranging from 0 to 65 °C. These temperature ranges indicate a reduction in free cooling potential relative to single-phase cooling; although industry driven experimentation may disprove this conclusion. The reduction in coolant inlet temperatures may be attributed to the popularity of refrigerants with low saturation temperatures in spray cooling experimentation.

4.3.3. Jet impingement

Jet impingement exploits high-velocity forced convection by ejecting coolant directly onto a heated surface through a convergent nozzle [97]. The high speed impingement results in the formation of thin temperature and velocity boundary layers which, when combined with bubble induced flow mixing and latent heat transfer, yield desirable heat transfer rates [100,134].

Similar to spray cooling, jet impingement is currently scarcely implemented in server cooling applications. Core similarities also exist between the two cooling solutions; namely, the necessity of a sealed vapor chamber, two-phase coolant flow, and direct contact with electronic components. As a result, jet impingement is subject to the same conclusions as spray cooling regarding its versatility and hot swap capabilities.

Heat transfer and flow characteristics of jet impingement systems based on recent experimental work are presented in Table 8. The highest heat transfer rate of 347 W has been obtained from Joshi and Dede [135] where the impact of subcooling upon jet impingement onto a 6.45 cm² porous pin fin array is investigated. Dielectric HFE-7100 at a saturation temperature of 61 °C and an inlet temperature of 55.5 °C was used to maintain a heated surface temperature of 66 °C. Major findings indicated that porous coated fins yield a 79% enhancement in the heat transfer coefficient relative to plain fins. Additionally, the heat transfer performance of the

system was enhanced by reducing inlet subcooling. The highest heat spreader heat flux of 275 W/cm² has been obtained from Ndao et al. [136] where jet impingement heat transfer on smooth surfaces, circular pin fins, square pin fins, and hydrofoil pin fins was investigated. The largest heat transfer rate of 11 W was achieved through hydrofoil pin fins with an array coverage area of 0.04 cm² and a heated surface temperature of 69 °C. The coolant used was R134a at a saturation temperature of 36 °C and an inlet temperature of 26 °C. Overall, heat spreader heat flux values for jet impingement systems have ranged from 16 to 275 W/cm².

This range of heat flux values further exceeds that of spray cooling while showing a reduction in pumping power requirements. Flowrates for cooling a single processor have ranged from 1 to 30 cm³/s and pumping power requirements have been calculated to range from 0.006 to 1.5 W per processor. This is attributed to reduced pressure differentials generated across jet nozzles in relation to atomizing nozzles used in spray cooling applications. As with most liquid cooling solutions reviewed, coolant temperatures for jet impingement are comparable to those of single-phase water cooling. Inlet temperatures have ranged from 20 to 61 °C with saturation occurring at temperatures ranging from 31 to 68 °C.

5. Comparative design review

The examined cooling technologies are compared directly by plotting the effective heat transfer coefficients attained from each study as a function of their corresponding inlet temperatures and pumping power requirements. The solutions are first compared within their respective categories followed by a general comparison of all systems.

5.1. Air and active indirect cooling

First, air and active indirect cooling solutions (*i.e.* water, and two-phase cooling) are compared as a function of inlet temperatures in Fig. 9a. Effective heat transfer coefficients of air cooling systems have ranged from 0.04 to 0.07 W/cm² K with the higher end of the range corresponding to an inlet temperature of 20 °C and the lower end corresponding to an inlet temperature of 35 °C. This serves as an appropriate benchmark given that HPC processor heat loads expected by the year 2020 have approached the heat transfer limitation of practical air cooling.

Effective heat transfer coefficients of water cooling systems have ranged evenly from 0.1 to 4 W/cm² K with most data points

Table 8

Jet impingement operational characteristics based on experimental work.

Ref.	Surface	Coolant	q (W)	q''_{HS} (W/cm ²)	h_{eff} (W/cm ² K)	Q (cm ³ /s)	T_i (°C)	T_{sat} (°C)	A_{HS} (cm ²)	A_w (cm ²)	T_w (°C)	ΔP (kPa)	P_{pump} (W)
[137]	Smooth	FC-72	57.6	40	1.82	2.5	47	50	n/a	1.44	69	240	0.6
[137]	Fins	FC-72	119.52	83	3.95	2.5	47	50	n/a	1.44	68	240	0.6
[137]	Pin fins	FC-72	122.4	85	5	2.5	47	50	n/a	1.44	64	240	0.6
[138]	Microchannel	HFE-7100	200	100	1.67	7.68	20	68	2	2	80	150	1.152
[139]	Smooth	PF5060	24	30	0.698	6.2	31	56	n/a	0.8	74	–	–
[140]	Smooth	HFE-7100	64	16	0.5	7.5	51	61	n/a	4	83	2.5	0.0188
[140]	Smooth	HFE-7100	100	25	0.893	30	51	64	n/a	4	79	9.5	0.285
[136]	Smooth	R134a	4.6	115	6.97	–	26	31	n/a	0.04	42.5	–	–
[136]	Square pins	R134a	7	175	8.14	–	26	31	n/a	0.04	47.5	–	–
[136]	Hydrofoil pins	R134a	11	275	6.4	–	26	36	n/a	0.04	69	–	–
[141]	Smooth	HFE-7100	30.18	60	2.5	1	61	61	n/a	0.50	85	–	–
[134]	Micro pin fins	FC-72	120	120	2.4	–	31	56	n/a	1	81	–	–
[142]	Smooth	FC-72	110	18.3	0.654	1.807	56	56	n/a	6	84	–	–
[143]	Smooth	R134a	2.55	255	9.62	1.675	37	47	n/a	0.01	63.5	187	0.313
[135]	Smooth	HFE-7100	216.59	33.58	1.34	7	51	61	6.45	3.61	76	0.9	0.0063
[135]	Pin fins	HFE-7100	339.34	52.61	1.75	7	51	61	6.45	3.61	81	2.7	0.0189
[135]	Porous fins	HFE-7100	346.56	43.73	4.16	7	55.5	61	6.45	3.61	66	3.6	0.0252
[144]	Smooth	HFE-7100	37	37	3.08	7.15	56	61	n/a	1	68	210	1.502
[144]	Smooth	Nanofluid	46	46	3.83	7.15	56	61	n/a	1	68	210	1.502

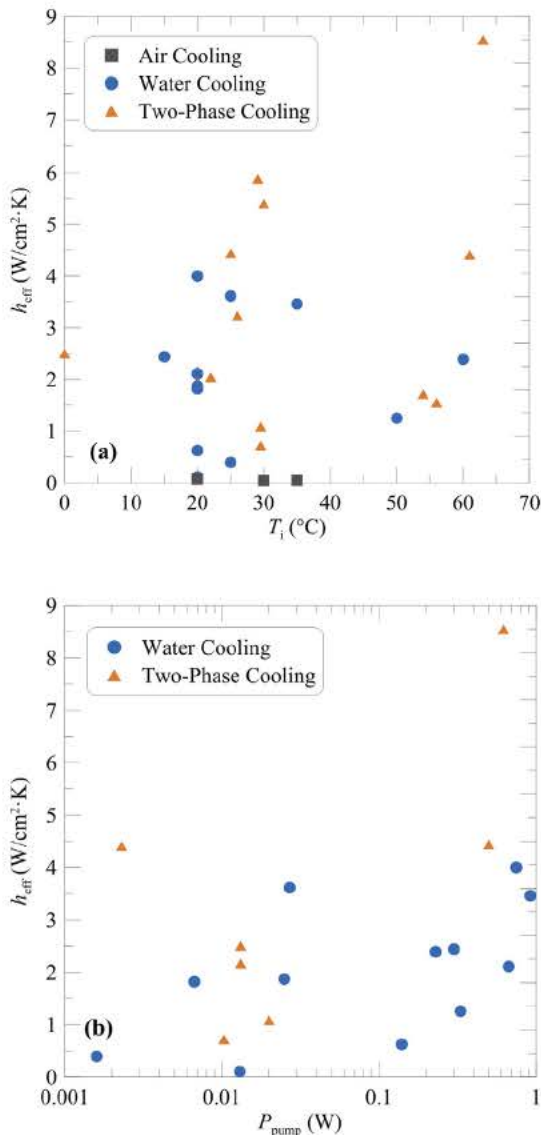


Fig. 9. Effective heat transfer coefficients derived from active indirect cooling studies.

falling within the 20–30 $^{\circ}C$ inlet temperature range; however even with inlet water temperatures as high as 50 and 60 $^{\circ}C$, effective heat transfer coefficients as high as 1.3 and 2.4 $W/cm^2 \cdot K$ have been attained. These values are more than an order of magnitude greater than those of air cooling with an inlet air temperature of 20 $^{\circ}C$. This amplification of cooling capacity and waste heat recovery potential comes as no surprise given the poor thermophysical properties of air relative to water as a heat transfer medium.

Two-phase cooling results show further improvement relative to water cooling with effective heat transfer coefficients ranging evenly from 0.7 to 6 $W/cm^2 \cdot K$ (excluding an anomalous data point at 8.6 $W/cm^2 \cdot K$) with most data points again falling within the 20–30 $^{\circ}C$ inlet temperature range. Effective heat transfer coefficients as high as 1.7 and 4.4 $W/cm^2 \cdot K$ have also been attained at inlet temperatures between 50 and 60 $^{\circ}C$. Therefore even though water possesses thermophysical properties preferable to those of typical dielectric coolants and refrigerants, two-phase systems have demonstrated higher cooling capacities due to the presence of nucleation and latent heat transfer phenomena.

When comparing water and two-phase cooling systems as a function of pumping power requirements in Fig. 9b, it is difficult

to draw any conclusions due to the lack of pressure drop data available in two-phase cooling experimental studies. It is evident however, that pumping power requirements are generally less than 1 W for both solutions; even in cases with effective heat transfer coefficients as high as 4 $W/cm^2 \cdot K$. Although fan power requirements in air cooling systems were not calculated, it is expected that they are greater than 1 W while yielding effective heat transfer coefficients of less than 0.1 $W/cm^2 \cdot K$. Again, this reduction of pumping power requirements in liquid cooling systems is mainly attributed to the superior thermophysical properties of liquid coolants relative to air.

5.2. Passive cooling

Passive cooling solutions (i.e. heat pipes and pool boiling) are compared as a function of inlet temperatures in Fig. 10; no graph is dedicated to pumping power requirements here as these passive systems consume no energy at the server level. Effective heat transfer coefficients of loop heat pipe cooling systems have ranged evenly from 0.1 to 1.4 $W/cm^2 \cdot K$ while those of pool boiling systems have ranged evenly from 0.4 to 2 $W/cm^2 \cdot K$ (excluding anomalous data points between 2 and 4 $W/cm^2 \cdot K$). The larger cooling capacities of pool boiling systems may be attributed to the absence of thermal interfaces between heated surfaces and dielectric coolants. Regardless, both solutions have yielded cooling capacities lower than those of water and two-phase cooling systems while still substantially larger than those of air cooling.

The majority of loop heat pipe systems have required condenser coolant temperatures ranging from 15 to 25 $^{\circ}C$ while pool boiling systems have required bath temperatures ranging from 50 to 60 $^{\circ}C$. It is tempting to conclude that pool boiling offers higher potential for waste heat recovery; however it should be noted that a direct comparison between heat pipe condenser temperatures and pool boiling bath temperatures is not entirely valid. For this comparison to be warranted, inlet water temperatures within pooling boiling condensers must be presented instead; however this data is seldom available in pool boiling literature. Regardless, industry implementations of pool boiling have successfully made use of inlet condenser water temperatures as high as 60 $^{\circ}C$ [14] and a few heat pipe systems reviewed have done so with air and water condenser inlet temperatures as high as 55 $^{\circ}C$. It is therefore difficult to draw any meaningful conclusions until condenser temperatures are better documented within pool boiling literature.

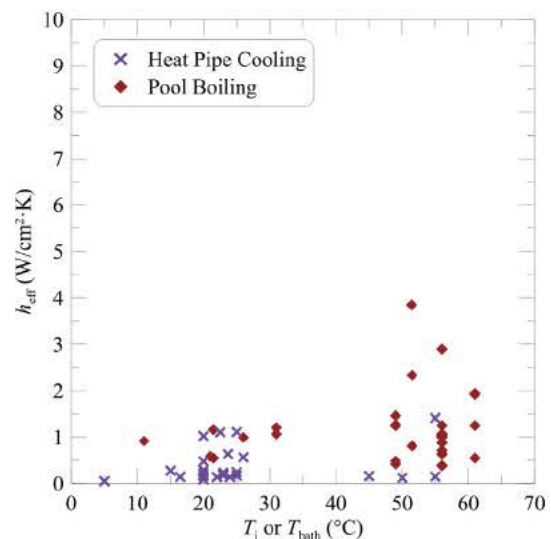


Fig. 10. Effective heat transfer coefficients derived from passive cooling studies.

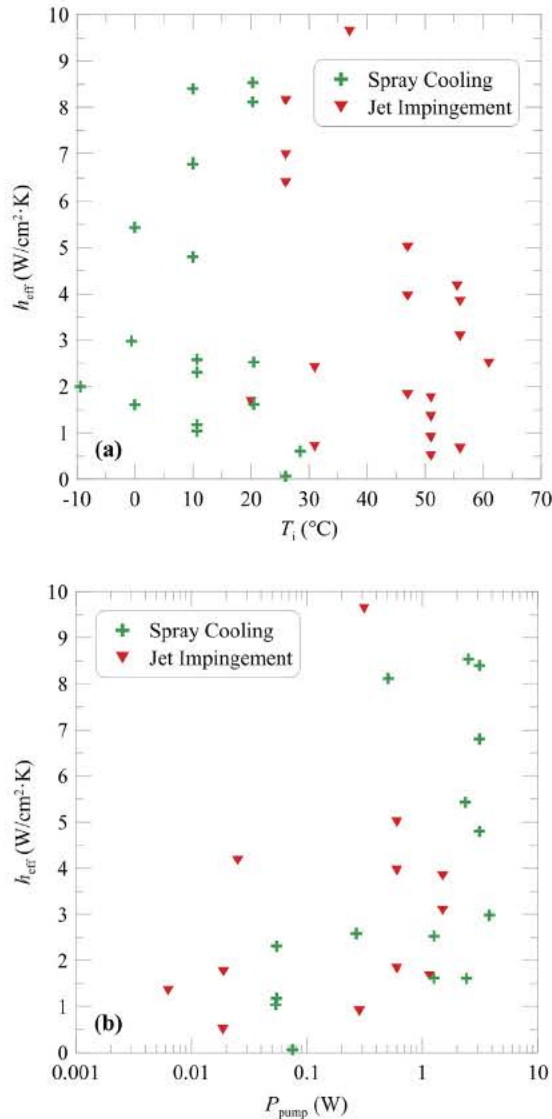


Fig. 11. Effective heat transfer coefficients derived from active direct cooling studies.

5.3. Active direct cooling

Active direct cooling solutions (*i.e.* spray cooling and jet impingement) are compared as a function of inlet temperatures in Fig. 11a. Effective heat transfer coefficients for both systems have ranged evenly from less than 0.4 to 9 $W/cm^2 \cdot K$; interestingly however, a clear distinction is apparent in their respective inlet temperatures. The majority of spray cooling studies required inlet temperatures ranging from 0 to 20 $^{\circ}C$ while most jet impingement studies utilized inlet temperatures ranging from 45 to 60 $^{\circ}C$ (a number jet impingement data points also appear in the vicinity of 30 $^{\circ}C$). The reasons for this discrepancy are not immediately clear and require deeper investigation into the heat transfer mechanisms of these cooling solutions. It is likely however that the popularity of refrigerants in spray cooling studies versus the popularity of dielectric fluids (with higher saturation temperatures) in jet impingement studies plays a certain role. Regardless, this data indicates that jet impingement yields higher potential for waste heat recovery than spray cooling. A comparison of pumping power requirements shown in Fig. 11b reveals a similar arrangement. The majority of spray cooling systems have required

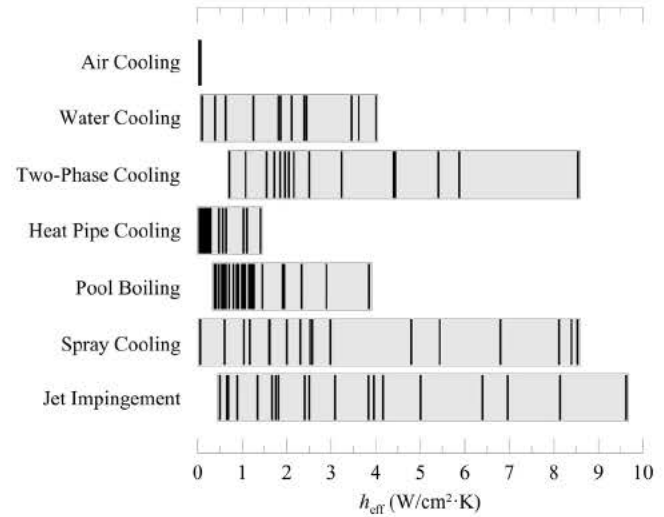


Fig. 12. Ranges of effective heat transfer coefficients from all reviewed studies.

pumping powers ranging from 1 to 4 W while most jet impingement systems have operated with less than 1 W (less than 0.1 W in some cases).

5.4. General review

The effective heat transfer coefficients presented above are collected in a single one dimensional bar chart in Fig. 12 in order to facilitate a general comparison; this figure also indicates the effective heat transfer coefficients from individual studies within each bar. With respect to heat transfer limitations, active direct liquid cooling solutions – namely, spray cooling and jet impingement – exhibit the most promising results. This superiority over indirect liquid cooling strategies is attributed to the absence of conductive thermal interfaces between the coolant and heated surface, as well as the evaporative and latent heat transfer processes involved [97]. Nonetheless, these solutions are also associated with the highest pumping power requirements of any liquid cooling solution reviewed; a characteristic that is attributed to the large pressure drop generated across jet and spray nozzles [145]. As an additional drawback, the necessary containment of server electronics within sealed vapor chambers prohibits easy hot swap access to server components as evacuation and refill processes are mandatory [14].

Active indirect liquid cooling solutions – namely, single and two-phase cooling – stand as the next best strategies in terms of heat transfer limitations. In spite of this inferiority, these solutions exhibit a substantial reduction in pumping power requirements while permitting easier hot swap access to server electronics through quick-disconnect fluidic connectors [28]. Furthermore, the heat flux values attained through these solutions are more than sufficient for electronics cooling applications when compared to traditional air cooling. Conclusively, indirect single and two-phase cooling solutions are more practical in server cooling applications than active direct liquid cooling solutions (spray cooling and jet impingement). A drawback with these solutions however is a lack of versatility as custom piping layouts are required at both facility and server levels [14].

Due to their passivity, heat pipe cooling solutions require no server level pumping power and at the same time yield greater reliability [76]; however they also possess the lowest heat transfer limitations of all liquid cooling solutions reviewed. Nonetheless, heat pipes present several advantages when compared to single and two-phase cooling solutions. First, fluidic connectors within servers are eliminated as heat pipes must be hermetically sealed

prior to being commissioned [76]; this mitigates issues pertaining to leakage during assembly. Second, heat pipe condensers may be effectively cooled by means of air or thermal contact heat exchangers [76,77]; both of which permit easy hot swap access to electronics without terminating any fluidic connections. As an indirect liquid cooling solution however, heat pipe cooling systems remain subjected to a lack of versatility.

The most impressive liquid cooling solution reviewed is undoubtedly open bath two-phase passive immersion cooling or pool boiling. This approach overcomes the disadvantages of its competitors by presenting an all-liquid cooling solution that is passive and versatile, while providing easy hot swap access to electronics in a manner that is comparable to that of air cooling [14]. In spite of maintenance issues pertaining to coolant degassing and vapor loss [98], immersion cooling is a highly practical choice for general purpose server cooling within data centers. For practical application in HPC servers however, further investigation is required into the effects of heat spreading on pool boiling heat transfer [99].

On a final note, air cooling may be considered the least effective cooling strategy solely from the perspective of heat transfer and operating costs. The poor thermophysical properties of air yield the lowest heat transfer limitation of any cooling solution reviewed along with undesirable coolant temperatures and flow-rates; all of which exacerbate the issue of thermal management and increase operating expenses. From the standpoint of reliability and implementation however air cooling is a prominent strategy for thermal management. The absence of liquid coolants, sealed enclosures, piping, and fluidic connectors bolsters reliability while also rendering a highly versatile system; this facilitates component hot swapping, installation, and maintenance. It is evident from the popularity of air cooling within industry that these qualities take precedence over heat transfer and flow characteristics as the scale and complexity of data centers intensifies.

6. Conclusions

A review of the heat transfer characteristics of cooling strategies for server electronics has been presented. Liquid cooling solutions have exhibited significant improvements in terms of heat transfer limitations, coolant temperatures, and coolant flowrates relative to traditional air cooling; this has been attributed to the superior thermophysical properties and heat transfer mechanisms associated with liquid cooling.

In the case of general purpose computing, it has been concluded that the heat transfer limitation of air cooling is not an inhibiting factor to thermal management. Therefore the implementation of liquid cooling strategies in such scenarios should be motivated by other factors such as the reduction of operating costs, improved chip performance, and the need for waste heat recovery. It is evident from the cold aisle temperatures observed that waste heat recovery is less practical, and in some cases impossible, with traditional air cooling. Solely a heat transfer medium with enhanced thermophysical properties can engender the desired leap in heat reuse potential.

The above conclusion does not hold in the case of high performance computing, where the rate of chip heat generation has approached the limitation of air cooling. In such scenarios, the potential heat transfer limitation of a given cooling solution should play a larger role in design processes. It has been concluded that with appropriate heat spreading, all liquid cooling solutions examined in this study permit the effective thermal management of high performance processors available until the year 2020. As a result, alternative factors must be considered when selecting an ideal liquid cooling solution; these may include reliability, versatility,

hot-swap capability, maintenance requirements, and capital expenditures.

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