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Air Flow Measurement and Management for Improving Cooling and Energy Efficiency in Raised-Floor Data Centers: A Survey

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ABSTRACT Recently, the rapid growth in both data center power density and scale poses great challenges to the cooling system. On one hand, the data center operators try to over provision cooling resources for fear of server failures induced by accumulated heat. On the other hand, they also want to reduce the energy cost as the cooling system takes up a significant portion of overall energy consumption. Among all available cooling solutions, air cooling dominates the data center industry due to its simpleness. However, its cooling efficiency has been questioned due to the low air density and specific heat. In this paper, we provide an overview for current endeavors to improve the air cooling efficiency. We group existing researches according to the locations where they can be applied from the perspective of air flow cycle. We also discuss the thermal measurement issues. We hope this paper can help researchers and engineers to design and control their data center air cooling systems.

INDEX TERMS Air-cooled raised-floor data center, cooling and energy efficiency, air flow management.

I. INTRODUCTION

Techniques to reduce the energy consumption of data centers have been one of the most active research areas for decades. Despite of the emergence of numerous energy-saving technologies in recent years, however, it seems that increases both in number and in size of data centers do not stop due to a surging computing resource demand. As a result, the trend of overall energy consumption for the data center industry is still increasing. According to a report of National Resources Defense Council (NRDC) [1], U.S. data centers are expected to consume around 140 billion kWH electricity in 2020. The expenditure for electricity has been comparable to that for purchasing new servers [2]. To reduce the negative environmental impact stemmed from the huge energy consumption of data centers, several solutions have been proposed, among which the Energy Internet [3], [4] is a promising candidate since it has the ability to integrate various renewable and clean energy sources with plug-and-play functionality.

The major power consumers inside data centers are Information Technology (IT) system and cooling system. While the former offers the required computing productivity, the latter is often considered as a secondary or auxiliary system since its aim is only to remove heat from the machine room. Unfortunately, the cooling system needs as much as 50% of the overall data center energy consumption to operate [5]. In addition, its configuration also has noticeable impact on IT system energy consumption, as the relationship between chip leakage power and temperature has been proven to be super-linear [6], [7]. Therefore, thermal management mechanisms are indispensable to improve the cooling efficiency and cut the cooling cost. Furthermore, they are also key measures to prevent the system from reliability degradation and thermal attack [8], [9]. There have been a number of research projects fully or partially dedicated to investigate energy efficient cooling management techniques in data centers, e.g., GENiC [10], GreenDataNet [11], RenewIT [12], CoolEmAll [13], All4Green [14], etc.

There are various cooling solutions available for data center operators, among which the air cooling approach undoubtedly dominates the current industry [15]. A standard configuration of air-cooled data center is the raised-floor supply with cold aisle/hot aisle layout, where the air flow starts from the Computer Room Air Handler (CRAH), passes through several system components like the underfloor plenum, perforated tile, cold aisle, server rack, hot aisle, and finally returns to CRAH. How to measure and manage the air flow in this context are the focus of this work. More specifically, we first investigate the air flow measurement methodologies and instrumentations, and then survey the existing works for two key control knobs, i.e., supplying air temperature and fan speed. Finally, techniques for air flow management in each system component on the air flow cycle are reviewed.

The rest of this paper proceeds as follows: related surveys are discussed and compared in section II; a brief overview of air-cooled data center and air flow cycle is presented in section III; sources of air cooling inefficiency are identified in section IV; section V provides a survey on air flow measurement strategies and instrumentations; sections VI and VII review works on temperature and fan control, respectively; sections VIII—XII elaborate techniques for improving the cooling efficiency in every component along the air flow cycle; section XIII concludes the paper.

II. RELATED SURVEYS

There have been a number of existing surveys on energy- and thermal-related issues for data centers. Here we group these works into three categories as shown in Table 1: system modeling, performance metrics, and energy-saving techniques.

The goal of system modeling is to develop mathematical approaches such that the thermal and energy characteristics for a given management decision can be precisely predicted. Some works in this area include [5], which summarized and organized a large body of energy models for both cooling and IT systems. The air flow modeling and validation techniques were discussed in [22], [30], [35], [44], and [52]. These techniques often employ a Computational Fluid Dynamics (CFD) approach, which typically requires massive computing resources and time to solve the model. To reduce the complexity of these models, some low-dimensional and measurement-based models were proposed, which were surveyed by Samadiani and Joshi [43] and Hamann and Lopez [53].

The thermal and energy performance can be evaluated by a set of performance metrics. Daim *et al.* [45] introduced some organizations initiated by government, industry, and academia who play vital roles in ranking data center energy efficiency, drafting standards, and developing performance metrics. Wang and Khan [47] and Schodwell *et al.* [46] classified metrics into two categories: computer level and data center level. Apart from basic metrics, Whitehead *et al.* [50] and Patterson [54] surveyed environmental metrics such as carbon and water efficiency, etc. Capozzoli *et al.* [49] and Ni and Bai [21] reviewed thermal metrics and presented evaluation standards for them. Chinnici *et al.* [51] provided a novel classification of thermal metrics, i.e., global, local, and "glocal". The relationships among metrics were investigated in [48].

The thermal management of data centers can be implemented from two perspectives. The first perspective is the thermal-aware IT system level solution which has direct impacts on the heat load production and distribution. Kong *et al.* [20] covered chip level thermal topics such as temperature monitoring, microarchitecture, and floorplanning, etc. Orgerie *et al.* [16] surveyed energy-efficient techniques for computing and networking facilities. Zhuravlev *et al.* [18] focused on energy-aware scheduling issues and classified scheduling algorithms as reactive, proactive, and mixed. This family of algorithms, however, only tries to minimize the energy consumption of IT equipments and probably results in poor cooling efficiency. To address this problem, various thermal-aware scheduling algorithms were proposed, which were reviewed by Chaudhry *et al.* [19] and Kong and Liu [17]. The second perspective is the cooling system solution seeking to improve the cooling efficiency for a given heat load. Literatures [21], [22], [25], [29] provided reviews for thermal management techniques in traditional air-cooled data centers. Other advanced topics in this area include liquid and hybrid cooling [22], [26], [30], [31], free cooling [28], and waste heat recovery [23], [24], [27], etc. There are also some surveys which covered both perspectives, such as [36]–[38] and [40]–[42]. While most of works discussed the two perspectives independently, an apparent fact is that IT and cooling systems are coupled via heat. Treating them separately may only lead to limited impacts [55] and suboptimal solutions [56]. As a consequence, joint optimization techniques which were reviewed in [39] become a cutting-edge research direction.

Existing surveys which have the closest relationship with this paper are dedicated to the air flow measurement and management. Bhattacharya [57] presented a list of measurable items for the cooling system. Hamann and Lopez [53] provided operating principles for measurement equipments. Arghode and Joshi [34] presented some thermal measurement tools which are commercially available or developed by themselves. They also analyzed the performance of perforated tiles and containment systems. Schmidt and Iyengar [33] reported research results for ventilation systems, underfloor plenum, perforated tiles, and rack layouts, etc. A set of design guidelines were also recommended. Kumar and Joshi [58] discussed the factors affecting the air flow distribution in underfloor plenum, server rack, and machine room. Schmidt *et al.* [35] summarized the impact of several factors, such as the data center layout, underfloor air distribution, and rack position, etc, on cooling efficiency. Patankar [32] paid a special attention to the underfloor plenum which is the crux of "cooling battle." Patterson [15] reviewed techniques and codes/regulations for the containment system. Although these surveys provided detailed

TABLE 1. Summery of related surveys.

Literature	Area of focus	Source	Year
	Energy-saving techniques: thermal-aware IT system solutions		
[16]	Techniques for improving the energy efficiency of IT systems	ACM Computing Surveys	2014
[17]	Green-energy-aware power management	ACM Computing Surveys	2015
[18]	Energy and thermal management through OS-level scheduling	IEEE Transactions on Parallel & Distributed Systems	2013
[19]	Thermal-aware scheduling, thermal modeling, monitoring, and profiling	ACM Computing Surveys	2015
[20]	Chip level thermal management	ACM Computing Surveys	2012
	Energy-saving techniques: cooling system solutions		
[21]	Energy performance of air conditioning systems	Renewable & Sustainable Energy Reviews	2017
[22]	Thermal models, measurement studies, and new cooling technologies	Journal of Electronic Packaging	2015
[23]	Thermal load characteristics, cooling system management, and waste heat recovery	Renewable & Sustainable Energy Reviews	2014
[24]	Waste heat utilization	Renewable & Sustainable Energy Reviews	2018
[25]	Cooling solutions for computer and telecommunications equipment room	IEEE Transactions on Components & Packaging Technologies	2003
[26]	Cooling strategies for high power-density server electronics	Applied Thermal Engineering	2016
[27]	Various data center cooling solutions	Heat Transfer Engineering	2015
[28]	Free cooling	Renewable & Sustainable Energy Reviews	2014
[29]	Thermal management challenges and solutions	IEEE Transactions on Components, Packaging, and Manufacturing Technology	2012
[30]	Thermal management for air-cooled raised floor data centers, energy and thermal metrics, and newly emerged technologies	Renewable & Sustainable Energy Reviews	2015
[31]	Thermal management of air- and liquid-cooled data centers	Applied Energy	2017
[32]	Air flow characteristics and management in raised floor data centers	Journal of Heat Transfer	2010
[33]	Air flow management and guidelines	ASHRAE Transactions	2007
[34]	Air flow measurement and management	Springer Briefs in Applied Sciences & Technology	2016
[35]	Air flow measurement, modeling, and management	IBM Journal of Research & Development	2005
[15]	Air flow management, containment techniques and regulations	IEEE Transactions on Components, Packaging, and Manufacturing Technology	To appear
	Energy-saving techniques: mixed solutions		
[36]	Energy-efficient datacenter architecture, resource provisioning, power and thermal management	IEEE Transactions on Computer-Aided Design of Integrated Circuits & Systems	2012
[37]	Optimization techniques for data center energy consumption	Renewable & Sustainable Energy Reviews	2016
[38]	Energy sustainability in data centers	Renewable & Sustainable Energy Reviews	2016
[39]	Joint cooling and IT systems management	IEEE Communications Surveys & Tutorials	2016
[40]	Energy-efficient mechanisms for IT and cooling systems	IEEE Systems Journal	2016
[41]	Cooling, power, and renewable energy management	Renewable & Sustainable Energy Reviews	2015
[42]	Technological drivers and future challenges for energy efficient data centers	Applied Energy	2013
	Modeling		
[5]	Energy modeling for IT and cooling systems in data center	IEEE Communications Surveys & Tutorials	2016
[43]	Reduced order air flow and thermal modeling	International Journal of Numerical Methods for Heat & Fluid Flow	2010
[44]	Air flow and heat transfer modeling	Distributed Parallel Databases	2007
	Performance metrics		
[45]	Metrics for data center energy efficiency	Management of Environmental Quality: An International Journal	2009
[46]	Green performance indicators	Americas Conference on Information Systems	2013
[47]	Energy, thermal, and other performance metrics	Journal of Supercomputing	2011
[48]	Energy metrics and relationships among metrics	International Workshop on Energy Efficient Data Centers	2014
[49]	Thermal metrics and evaluation standards	International Conference on Sustainability in Energy and Buildings	2014
[50]	Data center energy usage, environment impact, and related metrics	Building & Environment	2014
[51]	Energy, thermal and productivity metrics, and measurement methodologies	Book chapter: Pervasive Computing	2018

TABLE 2. Comparison of air flow management surveys.

Literature	Measurement	Temperature control	Fan control	Underfloor plenum	Perforated tile	Air containment	Rack layout	Air flow distribution
[32]				✓	✓	✓		
[33]				✓	✓	✓	✓	✓
[34]	✓				✓	✓		✓
[35]	✓			✓			✓	✓
[15]		✓	✓			✓		
Ours	✓	✓	✓	✓	✓	✓	✓	✓

reviews on various kinds of air flow management techniques, none of them covered this topic from the perspective of whole air flow cycle. Our work is intended to bridge this gap and to incorporate the most recent works in this field. A comparison of these surveys with our work can be found in Table 2.

III. DATA CENTER COOLING SYSTEM AND AIR FLOW CYCLE OVERVIEW

A large scale data center is typically cooled by chilled water cooling system, which consists of three loops [59], as shown in Fig. 1.

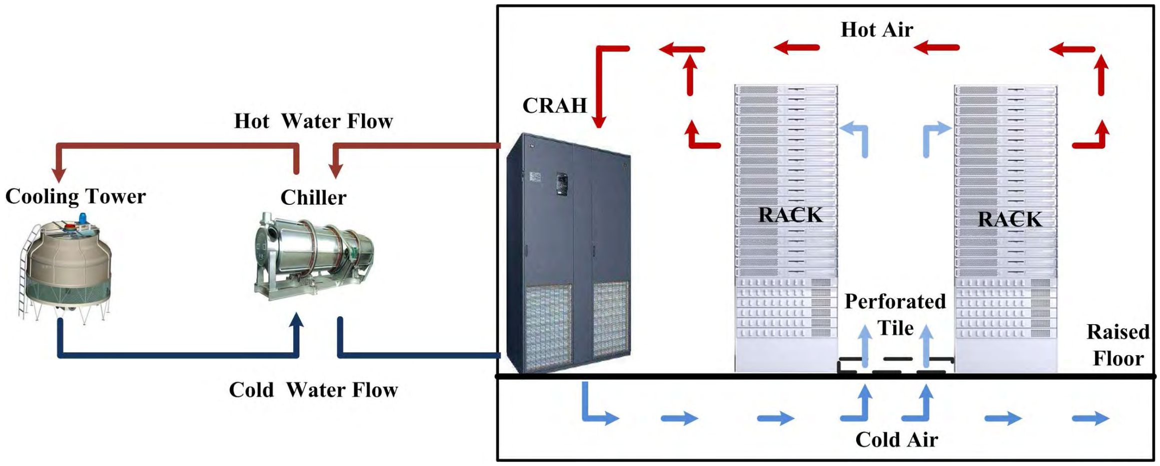


FIGURE 1. A typical air-cooled raised-floor data center.

1) Air side loop. The cold air supplied by CRAH traverses through the underfloor plenum and is ejected from perforated tiles due to the pressure differential between the plenum and machine room. Racks hosting IT equipments are arranged in a cold aisle/hot aisle style, where the cold air is sucked into servers from rack inlets facing the cold aisle and is pushed toward the hot aisle after absorbing the waste heat generated by servers. The hot air is then collected by CRAH via either open paths (flooding return) or dedicated ducted channels, and turns to cold air by a heat transfer process with the chilled water. This air flow loop is driven by a set of fans, e.g., CRAH blowers, tile fans, rack fans, server fans, etc.
2) Chiller-CRAH chilled water loop. The chilled water produced by the chiller plant flows into cooling coils of CRAH with the help of pumps, and returns to the chiller plant after warming up. In the chiller plant, the heat in the warm water is dumped into the second condenser water loop. A storage tank is usually presented for caching extra amount of chilled water serving as a temporary cooling source for emergency circumstances like power outage or chiller failure, etc.
3) Chiller-cooling tower condenser water loop. This loop is responsible for moving the heat to the outside atmosphere by the process of evaporation in the cooling tower. If outside environment is favourable, the chiller plant can be turned off and bypassed to save energy.

Note that in some advanced data center designs, the water side loops can be replaced by the cold water from natural sources to improve the energy efficiency. For example, the Facebook data center in Lulea, Sweden [60] and the Google data center in Hamina, Finland [61], use nearby river and ocean water to produce the cold air, respectively. The Microsoft Project Natick [62] even submerges the data center pod deep into the sea to leverage the huge cooling capacity of sea water.

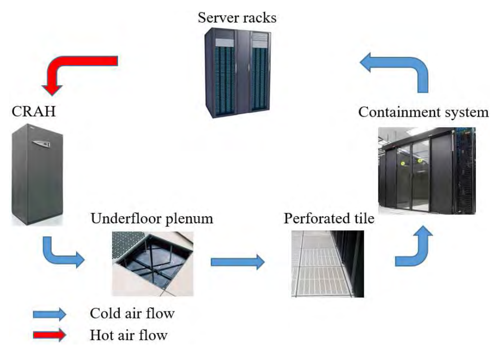


FIGURE 2. Data center air flow cycle.

What we consider in this paper is the air side loop. The air flow inside data center machine room passes though system components in the order of CRAH, underfloor plenum, perforated tile, containment system (cold aisle), server rack, and finally returns back to CRAH, as plotted in Fig. 2. Management of air flow can be achieved by applying advanced techniques on these components along the air flow path [63]. Note that some techniques, such as changing underfloor plenum height and installing containment, etc., are static, since they are less likely to be changed once applied. However, the remaining ones are dynamic. The reason for this is that the data center thermal load is changing over time. Moreover, IT equipments are not installed into racks permanently. Typically, 10% of the equipments are replaced each month [32]. Therefore, dynamic cooling management (a.k.a. cooling capacity right-sizing [59]), which is the counterpart of IT right-sizing technique [64], is critical for data center energy efficiency. Some manageable items are listed in Table 3.

IV. INEFFICIENCIES IN AIR COOLING

The main contributor for inefficient air cooling is the mixing of supplying cold air and return hot air [22]. The reason for this harmful air mixing can be attributed to three aspects [65]:

TABLE 3. Manageable items in each component.

Component	Manageable item
CRAH	Blower speed and supplying air temperature
Underfloor plenum	Plenum height, partitions, blockages, sealing techniques
Perforated tile	Porosity, layout, position, damper state (for adaptive vent tile), tile fan speed (for active tile)
Containment	Partial/full containment, sealing techniques
Server rack	Rack layout, equipments mounted onto the rack, rack fan speed (if installed)

cold air bypass, hot air recirculation, and negative pressure. Although there are numerous efforts from academia and industry to fight against these issues, e.g., dynamic fan control to balance the pressure differential [66]–[71], aisle containment or ducted solution to physically separate air flows [70], [72]–[80], advanced layout design to optimize the air flow pattern [32], [81]–[83], etc., the current air cooling practices are far from perfect. Below we will discuss air cooling inefficiencies in more details.

A. COLD AIR BYPASS

Cold air bypass is the result of high air flow velocity, over provisioning, and leakage, etc. The existence of bypass increases the overall air volume required for cooling servers, and hence the energy consumption of air moving devices rises. Alissa *et al.* [69] observed up to 37.62% bypass if the data center layout and airflow distribution system are not well engineered. A survey conducted by Hamann *et al.* [83] also showed that more than 50% cold air bypassed perforated tiles due to the structural leakage in majority of data centers. Bypass channels can be cable cutouts [84], sleeping or unpowered servers [85], channels near equipment rails in each rack [70], [86], rack top and bottom [87], sides of rack [88], and disabled CRAHs, etc. Some simple measures like sealing the most significant leakage paths can considerably reduce the bypass [89].

Bypass in open aisle systems is rather severe. A measurement study by Salim and Tozer [90] on 40 open aisle data centers revealed that the average amount of cold air bypassed the servers is 50%. Even if the aisle containment technique is employed, bypass can still be observed in the containment structure, especially when the cold air is over provisioned [88]. Apart from the small space below, above, and between doors, the most critical leakage channel in the containment structure is the under rack gap [71], [91]. Rack manufacturers have invented a wide range of measures to reduce the leakage through these channels, e.g., filling the space with sheet metal, foam, rubber strips, and brushes [88], etc.

The pressure differential plays an important role in bypass reduction. Sundaralingam *et al.* [92] found that around 10% of the air supplied to the cold aisle bypassed the racks at aisle pressure differential of 6.2 Pa. Tatchell-Evans *et al.* [88] varied the pressure differential and observed up to 20% bypass, with the rack bypass dominating the containment bypass. They also showed that with a proper pressure differential, the data center can save as much as 16% cooling energy. An effective way to control the pressure differential is the adjusting of fan speed. It was found that decreasing the CRAH fan speed leaded to a reduction of bypass from 13.4% to 4.6% [75].

B. HOT AIR RECIRCULATION

Hot air recirculation occurs when there is a deficit of cold air supply. In this case, the hot air from rack rear will return back to the server inlet and mix with the supplying cold air. As a consequence, the server inlet temperature rises. Lower temperature setpoints are needed for the CRAH outlet air flow and chilled water to meet the server inlet temperature requirement. This will degrade the cooling efficiency and increase the energy usage.

In open aisle systems, the recirculation is often found at the top of racks where only partial of the supplying cold air can reach. The recirculation destroys the uniformity of rack inlet temperature distribution, i.e., a higher temperature is observed at the rack top. Other places where the recirculation exists include ends of open cold aisles and unpowered servers, etc. Two straightforward solutions to prevent the recirculation are 1) increasing the supplying cold air volume, and 2) installing containment systems such that air flows are physically isolated. However, implementing these solutions needs careful engineering since performance degradations emerge if the system is not well controlled. For example, if containment systems are installed, a poor pressure control may lead to a low rate of air flow passed through servers. In extreme cases, unexpected reverse air flow can be found even if server fans are well functioning [93]–[95]. Effective recirculation-suppression measures also include placing additional perforated tiles near aisle ends to create an air curtain [32], increasing the ceiling height [96], and adding blanking panels in racks where servers are not installed [97], etc.

C. NEGATIVE PRESSURE

The current industrial practices generally over provision the cold air in order to prevent the hot air recirculation. While reducing the temperature at the upper part of rack, this approach actually damages the cooling efficiency of servers at lower locations near perforated tiles due to the negative pressure created by a phenomenon known as the Venturi effect. The Venturi effect is the pressure reduction when the air stream is ejected from small openings (e.g., perforated tiles) at high speed. In case of air over provisioning, the supplying air volume grows but the cross section area of perforated tiles remains unchanged. Therefore, the air flow velocity through perforated tiles will be higher, and the negative pressure is produced [85]. For standard raised floor data centers, the under rack recirculation induced by the negative pressure can be observed in both open and contained aisle systems if the gap between rack and floor is not well sealed [87], [98], [99]. Actually the negative pressure shows its impact not only at the rack bottom, but at each opening where the high speed cold air traverses through. For example, the reverse air flow is also found at the upper part of rack for the ceiling supply system [97] and in perforated tiles close to CRAH [100], [101].

V. THERMAL MEASUREMENT IN DATA CENTERS

To evaluate the thermal performance of the cooling system, two common approaches are the Computational Fluid Dynamics (CFD) simulation and real-time measurement. CFD is very useful in the data center construction stage for design assessment and trouble-shooting [102]. However, it only characterizes the stationary thermal behavior and cannot reflect the changes [103], e.g., server replacement, equipment upgrade, workload fluctuation, and time-varying outdoor weather condition, etc., that frequently take place. Limitations of the CFD approach can be summarized as follows [104], [105]:

1) Hard to verify the CFD model at the data center scale due to computational complexity.
2) Many structure details like cable cutouts and underfloor obstructions are prone to be overlooked but have significant effect on air flow patterns.
3) Traditional server block models cannot capture air flow characteristics precisely due to the device diversity.
4) The CFD approach is too time-consuming to provide a timely prediction for the transient behavior or cooling components failure.
5) Optimal cooling control strategies cannot be easily obtained via CFD simulations due to a large number of variables.

Because of these drawbacks, it is common to observe over 10% prediction errors in CFD simulations [106]. The Proper Orthogonal Decomposition (POD) [43] is an alternative approach. Compared to CFD, it has a desired advantage of faster running time [107]. However, it needs data from CFD or field measurement to calibrate the model. Reduced order methods based on thermodynamics first principles [105], [108] are also effective in evaluating the data center thermal and energy performance. A model validation procedure against the real system measurement is still necessary.

Real-time measurement [109], on the other hand, leverages either dedicated measuring apparatuses or built-in sensors inside servers and Building Information Systems (BIS) to monitor the instant data center thermal status. The operational data are fetched directly from the production system rather than simulation, and hence it is more precise and reliable. In addition, these monitored data can be used to adaptively control the cooling system in response to system dynamics [22], [110]. It should be noted that if the real-time measurement is implemented at the data center scale, huge volume of data will be accumulated. In order to effectively abstract useful information from these raw measurement data, big data analytics techniques [3] and advanced data processing architectures [111] should be employed.

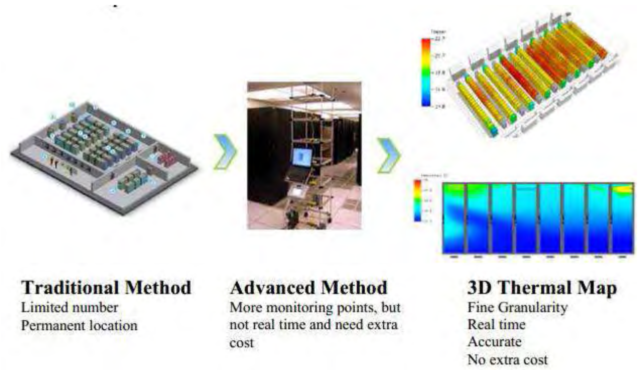


FIGURE 3. Three measurement strategies in data centers [67]. Traditional methods use sensors integrated into BIS. Advanced measuring tools are often in-house developed. Therefore, developers can customize the tool by adding mobility and sensor density. The 3D thermal map is more intuitive to data center operators to fast locate hot spots. However, it is not suitable for cooling control because of the data availability and accuracy.

A. MEASUREMENT STRATEGIES

Measurement strategies inside the data center can be generally categorized into three types [67], as shown in Fig. 3:

1) Traditional method with limited number of sensors installed on fixed locations like CRAH supply/return openings and hot/cold aisles, etc.
2) Advanced method using increased number of sensors (perhaps with some mobility) measuring a wide range of locations.
3) 3D thermal map created by built-in sensor readings from various components inside the server, e.g., CPU, HDD, memory, etc.

Compared to the traditional method, the advanced method achieves a finer space resolution at the cost of higher monetary expenditure. The built-in sensor based approach estimates the big thermal picture at the data center scale without extra cost [112]–[114]. However, it requires server cooperations which are often unavailable for multi-tenant data centers where servers belong to users rather than the data center [115]. Therefore, for current multi-tenant data centers the advanced method maybe the best option. In addition, the readings from built-in sensors may suffer from serious accuracy issues [116] which often lead to overcooling [117]. Hence, they should be preprocessed by some advanced techniques [118] to eliminate the noise before usage.

During the deployment stage, the space granularity of sensors must be high enough to capture a precise system thermal state [103], [119]. A group of sensors installed at various locations in CRAH, hot/cold aisles, and underfloor plenum (see Fig. 4) was extensively used in the literatures to analyze the thermal performance [74], [94], [120]. The industry also leverages high-density sensor systems to monitor and control the cooling system. For example, the sensor density of HP data center in Bangalore, India, is 1.15 sensors/m^3 [104]; the inter-sensor distances for lateral and z-direction are 8 and 12 inches for the Mobile Measurement Technology (MMT) in

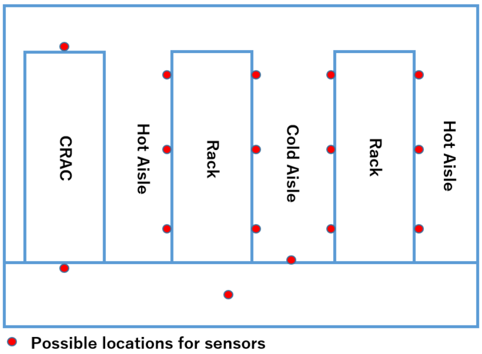


FIGURE 4. Locations for installing sensors. These locations include: 1) CRAC inlet; 2) CRAC outlet; 3) underfloor plenum; 4) cold aisle right above perforated tiles; 5) rack inlets (various heights); 6) rack outlets (various heights).

IBM data centers [121]. For microscale (rack, perforated tile, and CRAH opening, etc.) investigations, grid-based sensor networks are usually used to measure the air flow speed and temperature distribution [69], [122], [123]. Some novel methods like the super-multipoint temperature sensing technology [124], which measures the temperature using Raman-scattering light in an optical fiber, can even achieve a space granularity as low as 10cm.

TABLE 4. Monitoring intervals.

Literature	Monitoring interval
[129]	1 min.
[115]	20 sec for power consumption, 1 min for mechanical information, 5 mins for weather, and 1 month for water usage.
[130]	20 min.
[131]	30 min.
[132]	15 min.
[76]	5 sec and 1 min.
[127]	2 min for tile flow rate, 30 sec for cold aisle and CPU temperatures, 5 sec for cold aisle pressure.
[133]	5 sec.
[134]	1 sec for temperature and 5 sec for pressure.
[135]	0.1 sec to 1 sec.
[84]	10 sec.
[136]	30 sec.
[34]	30 sec.
[137]	1 min.

The time granularity also matters when configuring the sensor system. On one hand, it helps researchers to understand why and how thermal state evolves along with system variations such as changes of CRAH operating temperature point and blower speed; On the other hand, in production data centers it provides timely discovery and response to unexpected events such as critical equipment failures or power outages. Although some industrial data center operators (e.g., Facebook [128]) advocate a finer time granularity (in the order of seconds), we observe a wide palette of monitoring intervals in the literatures ranging from 0.1 sec to 30 min (refer to Table 4 for details). The problem of determining the optimal time granularity needs further investigations.

The location of sensors needs extra care. This is not as straightforward as it appears to be and does not receive enough attention. Recent reports [113], [138] revealed that most of the data center operators installed the sensors in wrong places. For example, the CRAH temperature sensor should be placed near the rack inlet to ensure the inlet temperature constraint [139]. However, over 75% operators only measure the CRAH return air temperature, which often results in inadequate controls of cooling equipments [140]. Advanced techniques like [141] and [142] can help to optimize the location of sensors.

B. INSTRUMENTATIONS

Monitoring systems can be wired [104], [137] or wireless [120], [129], [145]–[147]. Hybrid systems are also available in the industry [148], [149]. Advanced Industrial Internet of things (IIoT) technology [150] can be applied to wireless solutions to enhance the system flexibility and scalability. However, some issues still need to be addressed to ensure the system robustness for wireless systems: 1) Limited communication range. There are many obstacles like metal rack, server box, wall, etc., which block the wireless signal. The maximal communication distance for sensors can be determined by increasing the sensor-to-sink distance while continuously sending `ping` massage [120]. 2) Power outage, node failure, and security issues. Since wireless sensing nodes are often powered by battery, sleep scheduling techniques are necessary to save energy and extend system lifetime. Wang *et al.* [150] proposed a novel scheduling strategy which significantly reduced the energy consumption of sensing systems. Node failure and security can be addressed by using periodical probing message and encryption techniques [131].

A more flexible implementation is robot mobile sensor systems [126], [151], [152] shown in Fig. 5c which probe around the data center, identify the tile type (standard or perforated), and generate the data center layout automatically. As the probing process completes, the robot will return the dynamic thermal profile evolution as well as photographic information to the operator.

1) TEMPERATURE AND VELOCITY MEASUREMENT

Perhaps the most extensively studied metrics reflecting the data center thermal performance are the air flow temperature and velocity. The temperature and velocity measurement tools can be categorized into three levels: rack and aisle level, CRAH level, and perforated tile level.

- Rack and aisle level tools. Rack and aisle level measurements are usually used to determine the temperature and flow distributions. Therefore, measurement tools are generally designed as grid style systems. Figs. 5a and 5b are examples of such systems used by Nada *et al.* [125] and Fakhim *et al.* [101] to read the rack inlet and outlet temperatures at different heights. The Cold Aisle Temperature Acquisition System (CATAS) in Fig. 5d [127] is a 3D sensor grid, which extends to both ends of cold

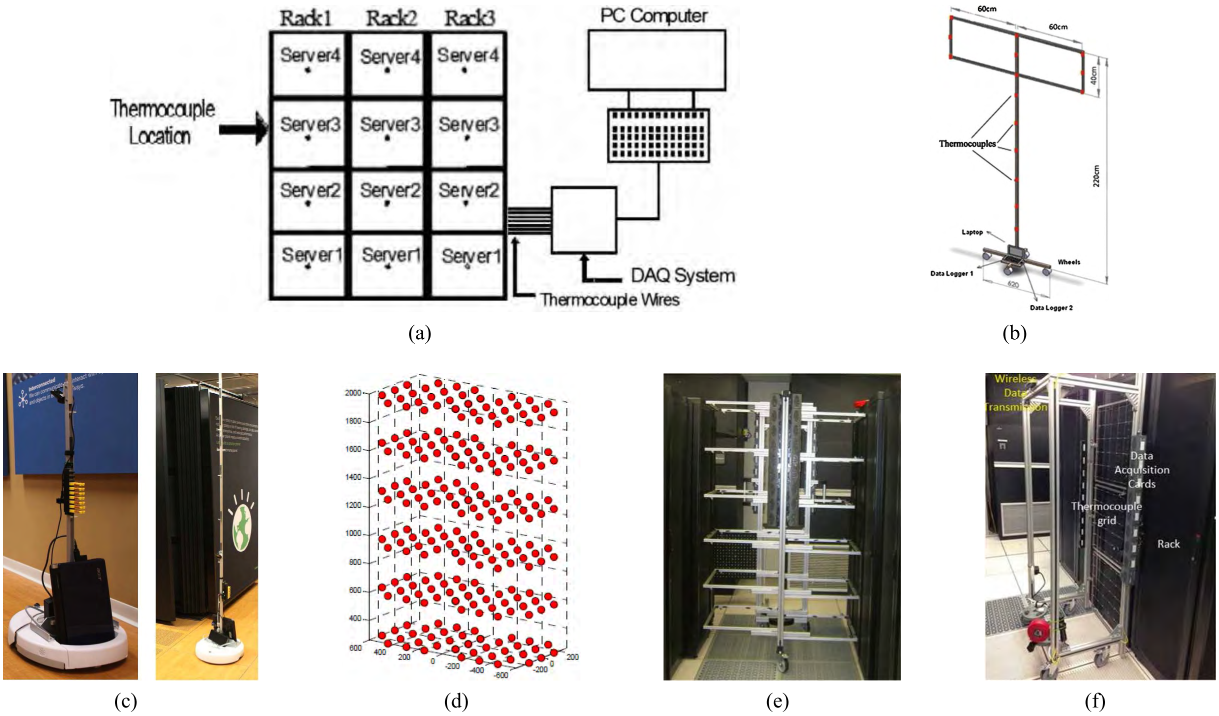


FIGURE 5. Temperature measurement tools. (a) Rack temperature measurement tool with fixed location [125]. Four type-T thermocouples attached on a plastic frame are placed at rack front and rear with a 2cm sensor-rack distance. Monitored data are routed to PC via a Data Acquisition (DAQ) interface. (b) Mobile rack temperature measurement tool [101]. Eight type-K thermocouples are placed at the lower part of the equipment. A large horizontal frame with another eight K-type thermocouples are installed at the top (180 – 220cm) to measure the detailed thermal map. (c) Robot rack temperature measurement tool [126]. K-type thermocouples can move from the bottom to top along the rail mounted on an iRobot. As the robot traverses across the data center, this tool will generate thermal maps at different heights. (d) CATAS aisle temperature measurement tool [127]. The width of CATAS equals to the width of cold aisle. Sensors are divided into multiple horizontal layers, with a 0.61m interlayer distance. The first layer lies 0.26m above the floor. (e) Mobile 3D temperature field measurement tool [34]. A 3D array of 256 type-T thermocouples is deployed into a mobile grid with 122cm width, 191cm height, and 61cm depth. A 3D thermal map of the whole cold aisle can be produced when the tool slides through the aisle. (f) Mobile planar temperature field measurement tool [34]. This tool contains 108 type-T thermocouples distributed in a plane. It is used for measuring the thermal field with finer spatial resolution (10.2cm).

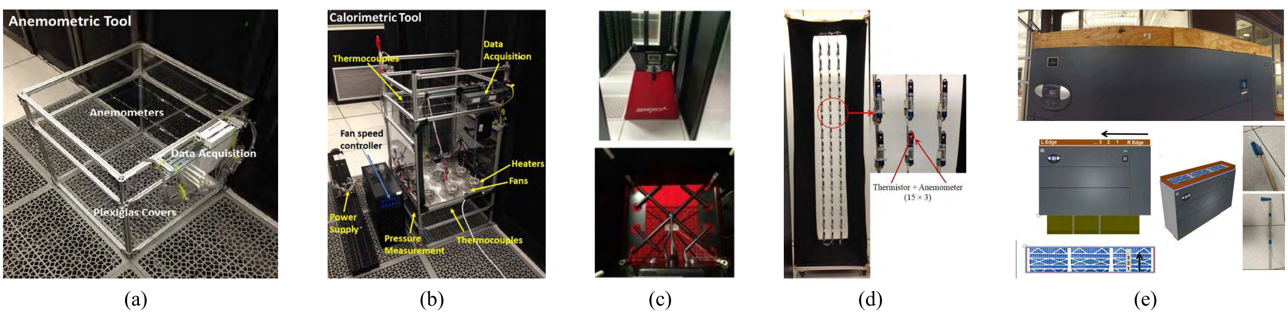


FIGURE 6. Velocity measurement tools. (a) Anemometric tool [143]. The tool has Plexiglas walls on all four sides except top and bottom ends. A grid of 16 velocity sensors is placed 46cm above the perforated tile. (b) Calorimetric tool [144]. The frame structure of this tool is similar with the anemometric tool. 3 × 3 electric heaters are placed on the air flow path. Variable Frequency Drive (VFD) fans are installed below heaters to obtain a near zero pressure differential across heaters. Two 4 × 4 thermocouple grids are installed before and after heaters to measure the temperature rise of air flow. (c) Flow hood [69]. A cloth skirt is used to guide the air flow toward the velocity grid, which derives the average flow rate by measuring the air temperature and the difference between total and static pressures. (d) Rack flow measurement tool [122]. The tool includes a grid of 45 anemometers and a cloth shield to direct the air flow. (e) CRAC flow measurement tool [69]. A 6-inch traverse is installed on the return face of CRAH to regulate the air flow direction. The sensing plane, which is 2 inches above the filter's face, consists of 13 zones with totally 195 measuring points.

aisle with depth of one tile, to generate a detailed thermal map. The mobile 3D temperature field measurement tool (Fig. 5e) developed by Arghode and Joshi [34] can capture the 3D temperature field for an entire cold aisle. If a finer spatial resolution is required, the mobile planar temperature field measurement tool (Fig. 5f) with inter-sensor distance of 10.2cm can be used. The rack level flow volume rate can be calculated by the product of the size of measured area and average air flow velocity obtained by an array of 16 thermal anemometers installed on a hard frame (Fig. 6d). The frame is covered by cloth skirts to prevent the air flow from bypassing

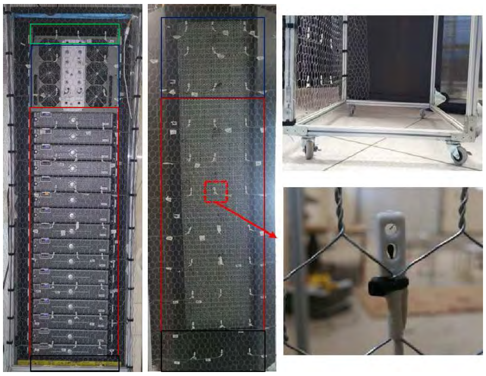


FIGURE 7. **The integrated Mobile Temperature/Velocity Mesh (MTVM) measurement tool [153]. 3 (column) × 12 (row) temperature/velocity sensors are deployed on a wire mesh on one vertical side of the frame, which covers the area of rack front/rear. The size of frame base equals to a perforated tile to avoid air flow interference.**

sensors [122], [127]. The MTVM tool (Fig. 7) in [153] measures both rack level temperature and velocity distribution.

- CRAH level tools. Since openings of CRAHs are quite large, measurement of these equipments also employs a sensor grid resembling the rack flow measurement tool in Fig. 6d. The temperature and velocity can be computed as the average value from all sensor readings [123]. Alissa *et al.* [122] used a more delicate method to measure the CRAH flow rate where the CRAH return face was horizontally divided into 13 zones with 15 measuring points for each zone (Fig. 6e). Further, a 6-inch traverse was placed on the return face to force the airflow direction to be vertical.
- Perforated tile level tools. The temperature measurement of perforated tiles can be also achieved by a thermocouple grid as long as the cross section area equals to the tile size [154]. For the air flow velocity measurement, using a single sensor was proven to be error inclined since nearly 50% air flow rate difference was observed in different measuring points for a single perforated tile [84]. The vane anemometer measures the average velocity over the measuring area, but the position and direction must be carefully considered to achieve accurate results [155]. The velocity sensor grid logs the velocity at multiple points simultaneously over a given period so that fluctuations are remarkably reduced. However, since the hot wire anemometer can only read the point velocity, it is recommended that the sensor grid should be placed at the height where tile flow jets merge [155]. The flow hood [156] (Fig. 6c) is a commercially available tool used to estimate more accurate results. However, since the flow hood always underestimates the actual flow rate due to the presence of back pressure in the instrument, an air flow balance system is needed to compensate the back pressure [69]. VanGilder *et al.* [157] found that previous correction methods are inadequate for tiles with a large opening area. To address this issue, they proposed a new family of semiempirical two-measurement correction methods. Arghode *et al.* [143], [144] also proposed two alternative tile flow measurement tools, i.e., the anemometric tool (Fig. 6a), which consisted of an array of thermal anemometers, and the calorimetric tool (Fig. 6b), which used the principle of temperature rise across a known heat load to measure the tile flow rate. It was found that these tools have more precise readings for high porosity tiles compared to the commercial flow hood.

2) PRESSURE MEASUREMENT

Pressure monitoring is also an important issue in data center to estimate the state of air supply-and-demand as well as leakage, especially for contained systems. Nemati *et al.* [70] argued that it was impractical to investigate containment systems without keeping an eye on the pressure differential between cold and hot aisles. Unfortunately, after some interviews with a number of production data center operators, we found that the pressure monitoring is missing in most of them.

When deploying the pressure measurement system, sensors can be installed at various locations, such as the mid height of racks [70], the center of cold aisle and underfloor plenum [127], and the drop ceiling [80], etc. In order to measure the pressure precisely, the orientation of sensor matters, which can be determined by the help of air flow direction [120]. To measure the pressure of duct flows, the sensor should be aligned with the local flow direction. In underfloor or drop ceiling plenums, however, since the local flow direction is unknown, multiple sensors should be placed in different directions. Radmehr *et al.* [156] monitored the pressure of underfloor plenum for eight different directions with the angle of adjacent directions being 45 degree. The minimum reading form sensors was treated as the static pressure. For deep plenums, a pressure sensor tube facing upward can be used to circumvent the effect of dynamic pressure [133], [134].

3) OTHER MEASUREMENT TOOLS

Besides above mentioned tools, there are some alternative tools to characterize some other aspects of the air flow which are hard to be captured by traditional measurement systems. For example, smoke which visualizes the air flow path, direction, and velocity [69], Particle Image Velocimetry (PIV) system which produces fine grain flow maps [55], [89], [158], and infrared camera which generates the temperature map to fast locate the hot spots and yield insights on cooling performance [159], etc.

C. ADVANCED PERFORMANCE METRICS

The basic air flow temperature, velocity, and pressure data can be used to derive some advanced metrics. These advanced metrics have been summarized in some published literatures, e.g., energy metrics [5], [37], [45], [47], [50], [160], [161],

TABLE 5. Energy related metrics.

Name	Definition	Discription
Power Usage Efficiency (PUE) and Data Center infrastructure Energy DCiE [166]	$PUE = \frac{\text{Total data center power consumption}}{\text{IT equipment power consumption}}$ $DCiE = \frac{1}{PUE}$	PUE, which is the ratio between total energy consumption and IT energy consumption, is a commonly used metric to evaluate the energy efficiency of data centers. DCiE is the reciprocal of PUE.
PUE level 4 (PUE4) [167] (A. K. A. Revised PUE [168])	$PUE4 = \frac{\text{Total data center power consumption}}{\text{IT power}}$	IT power is different from IT equipment power appeared in the definition of PUE in the sense that it excludes the power consumption of fans and power supply units inside IT equipments.
Energy Usage Efficiency (EUE) [169]	$EUE = \frac{E_t}{E_s} = \frac{\int_{t_1}^{t2} P_t dt}{\int_{t_1}^{t2} P_s dt}$	EUE is the ratio of the total electrical energy consumed by the data center E_t to the energy consumed by the servers E_s. Note that EUE is obtained over a significant time period, compared to PUE which is an instantaneous metric.
Data Center Energy Performance Metric (DCEPM) [160]	$DCEPM = \frac{\text{Average server dynamic power}}{\text{Sum of average IT and cooling power}}$	The server dynamic power, which is an increasing function of CPU utilization, is the power dedicated to execute user applications. DCEPM measures the energy efficiency in terms of useful works performed.
Energy Usage Intensity (EUI) [140]	$EUI = \frac{\text{Annual total energy consumed}}{\text{Total area of the building}}$	EUI characterizes the energy usage per space unit.
Cooling Capacity Factor (CCF) [65]	$CCF = \frac{\text{Total running cooling capacity}}{\text{IT consumption} \times 1.1}$	CCF is computed by dividing the total running nameplate cooling capacity by 110% of the IT critical load. It can be used to diagnose the mismatch between thermal load and cooling capacity.
Datacenter Performance Per Energy (DPPE) [170]	$DPPE = \frac{\text{throughput at the data center}}{\text{energy consumption}}$ $= \text{ITEU} \times \text{ITEE} \times \frac{1}{\text{PUE}} \times \frac{1}{1 - \text{GEC}}$	DPPE expresses the amount of throughput produced by per unit energy. It is a comprehensive index linked to four independent sub-indicators: IT Equipment Utilization (ITEU), IT Equipment Energy Efficiency (ITEE), PUE, and Green Energy Coefficient (GEC).
Space, Watts and Performance (SWaP) [47]	$SWaP = \frac{\text{Performance}}{\text{Space} \times \text{Power Consumption}}$	SWaP describes the energy efficiency for a data center, which is defined by three parameters: performance (evaluated by industry-standard benchmarking), space (measured using server height in terms of rack units), and watts.
Data Center Energy Productivity (DCEP) [171]	$DCEP = \frac{\text{Useful Work Produced}}{\text{Total Data Center Energy Consumed over time}}$	DCEP evaluates the productivity of energy. The "useful work" is defined by users and is application specific.
Compute Power Efficiency (CPE) [172]	$CPE = \frac{\text{IT Equipment Utilization}}{PUE}$ $= \frac{\text{IT Equipment Utilization} \times \text{IT power Consumption}}{\text{Total Power Consumption}}$	CPE measures the proportion of total facility power used by IT workloads. Lower CPE often indicates an over provision of computing and cooling resources.
The Data Center Cooling System Efficiency (CSE) [173]	$CSE = \frac{\text{Average cooling system power usage}}{\text{Average cooling load}}$	CSE quantifies the amount of energy consumed by per cooling load on average. It characterizes the cooling system efficiency.
CRAC Cooling Load Deviation (CLD) [101]	$CLD = \frac{Q - \bar{Q}}{\bar{Q}}$	CLD is the deviation of sensible cooling load for a particular CRAC (Q) from the mean load ($\bar{Q}$). It can be used to discover the thermal load imbalance and plan the local cooling capacity.

environment and sustainability metrics [50], [54], thermal and cooling efficiency metrics [47]–[49], [162], [163], etc. Here, we categorize these metrics into three groups.

The first group is energy related metrics, which include metrics reflecting the energy distribution among various system components (PUE, DCiE, PUE4, EUE, DCEPM), energy productivity (DPPE, SWaP, DCEP, CPE), cooling performance (CSE and CLD) and capacity (CCF), and energy density (EUI). The second and third groups are thermal metrics computed based on measured temperature and velocity/pressure data to evaluate the performance of air flow delivery system. Details for these metrics are given in Tables 5, 6, and 7. There are also compound performance metrics reflecting multiple facets of the system. An example of such metric is the Performance Indicator (PI) [164], [165], proposed by The Green Grid. PI includes three submetrics: 1) PUE ratio (PUEr), which compares the actual PUE against design objective; 2) IT thermal conformance, which is the

TABLE 6. Thermal metrics I.

Name	Definition	Discription
Supply Heat Index (SHI) and Return Heat Index (RHI) [174]	$SHI = \frac{T_{RackInlet} - T_{Ref}}{T_{RackOutlet} - T_{Ref}}$ $RHI = \frac{T_{RackOutlet} - T_{RackInlet}}{T_{RackOutlet} - T_{Ref}}$	T_{Ref} is the reference temperature (CRAH supplying temperature). A lower SHI indicates a more uniform cold air distribution, and thus a better cooling efficiency. RHI is complement to SHI, with $SHI + RHI = 1$.
High End Rack Cooling Index ($\mathrm{RCI_{HI}}$) and Low End Rack Cooling Index ($\mathrm{RCI_{LO}}$) [175]	$RCI_{HI} = 1 - \frac{\text{Total over temperature}}{\text{Max allowable over temperature}} = 1 - \frac{\sum(T_i - T_{MaxRec})_{T_i > T_{MaxRec}}}{n \times (T_{MaxAll} - T_{MaxRec})}$ $RCI_{LO} = 1 - \frac{\text{Total under temperature}}{\text{Max allowable under temperature}} = 1 - \frac{\sum(T_i - T_{MinRec})_{T_i < T_{MinRec}}}{n \times (T_{MinRec} - T_{MinAll})}$	T_i is the temperature at intake i. T_{MaxRec} is the maximum recommended temperature. T_{MaxAll} is the maximum allowable temperature. n is the number of intakes. $RCI_{HI} < 100\%$ means at least one intake temperature is less than the maximum recommended temperature. $RCI_{HI} < 90\%$ indicates serious hot air recirculation. RCI_{LO} is complement to RCI_{HI} to measure the temperature lower bound.
Capacity (CAP) and Resilience (RES) [176]	$CAP = \frac{\text{IT equipment load with } T_{MaxInlet} \leq 27°\text{C}}{\text{Total facility IT equipment load}}$ $RES = \frac{\text{IT equipment load with } T_{MaxInlet} \leq 32°\text{C}}{\text{Total facility IT equipment load}}$	CAP and RES represent the proportion of load that is operating within the desired cooperating conditions defined by ASHRAE's guideline. of CAP or RES less than 1 indicate that there are risks when installing IT equipment up to the design limit.
Return Temperature Index (RTI) [177]	$RTI = \frac{\text{Total airflow through CRAH}}{\text{Total airflow through IT equipment}} = \frac{T_{return} - T_{supply}}{\Delta T_{equip}}$	$T_{return} - T_{supply}$ is the average temperature difference in the CRAH side and $\Delta T_{equipment}$ is the average temperature difference in the IT equipment side. A 100% RTI means a perfect cold air delivery. A larger or smaller value means the existence of hot air recirculation or cold air bypass, respectively.
Thermal Correlation Index (TCI) [178]	$TCI_{ij} = \frac{\text{Temperature change at rack } i}{\text{Temperature change at CRAH } j}$	TCI is a steady state metric which reflects the temperature change at rack i in response to tuning the supply air temperature at CRAH j.
β index [35]	$\beta = \frac{\Delta T_{inlet}}{\Delta T_{rack}} = \frac{T^r_{in}(z) - T^C_{sup}}{T^r_{out} - T^r_{in}}$	Compared to macro metrics like SHI and RHI, β index is a micro metric which detects the local recirculation. ΔT_{inlet} is the temperature difference between a particular position z at rack inlet and the chilled-air entry, and ΔT_{rack} is the average air temperature increment across the rack. The average value of β always lies between 0 (no local recirculation) and 1 (temperatures are the same for rack inlet and outlet).
Bypass (BP), Recirculation (R), and Negative Pressure (NP) [65]	$BP = \frac{T_{exit,rack} - T_{return}}{T_{exit,rack} - T_{raised,floor}}$ $R = \frac{T_{intake,rack} - T_{raised,floor}}{T_{return} - T_{raised,floor}}$ $NP = \frac{T_{raised,floor} - T_{supply,CRAH}}{T_{return} - T_{supply,CRAH}}$	BP, R, and NP measure the extent of cold air bypass and hot air recirculation in the air distribution system. NP describes whether there is return air leaked into the underfloor plenum due to the negative pressure. The ideal value for these three metrics is 0.
Balance ratio (BAL) [65]	$BAL = \frac{T_{exit,rack} - T_{intake,rack}}{T_{return} - T_{supply,CRAH}} = \frac{1 - R}{(1 - BP) \times (1 + NP)}$	BAL is the ratio between the CRAH airflow and IT equipment airflow. The ideal value of BAL is 100%. Any deviation from 100% is caused by the bypass or recirculation.

percentage of suitable IT inlet temperatures during normal operations; 3) IT thermal resilience, which is the percentage of suitable IT inlet temperatures under cooling failures and planned maintenances. PI visualizes the relationship among submetrics and reflects the difference between actual performance and design goal using a triangle.

VI. TEMPERATURE CONTROL

In the Thermal Guideline published by American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) [181], the IT environmental condition of data center was categorized into one recommended and four allowable classes as shown in Fig. 8. One of the most conspicuous modifications in this guideline compared to its previous versions is the rise of rack inlet temperature. An optimization process was also proposed to choose a proper temperature range. In this section, we briefly review issues related to the supplying air temperature control.

A. SUPPLYING AIR TEMPERATURE AND COOLING EFFICIENCY

Considerations for reliability determined the supplying air temperature setpoint at the early age. The relationship

TABLE 7. Thermal metrics II.

Name	Definition	Discription
Cold Aisle Capture Index (CI_C) and Hot Aisle Capture Index (CI_H) [179]	CI_C = The fraction of rack inlet air which originates from local cooling resources (for example, perforated floor tiles or coolers). CI_H = The fraction of air exhausted by a rack which is captured by local extracts (for example, local coolers or return vents).	CI is a rack level air delivery efficiency metric which is commonly computed by CFD simulation. A higher CI indicates better performance.
Aisle Pressure Differential (ΔP) [70], [71]	$\Delta P = P_{Cold} - P_{Hot}$	The definitions of cold and hot aisle are different for CAC and HAC. In CAC, $\Delta P = P_{Containment} - P_{Room}$; in HAC, $\Delta P = P_{Room} - P_{Containment}$.
Airflow Provisioning Ratio (APR) [70]	$APR = \frac{\text{Total air flow required by IT equipments}}{\text{Total air flow provided by CRAC}}$	APR specifies the state of air provisioning, e.g., under, perfect, or over provisioning.
Leakage Impact Factor (LIF) [71]	$LIF_j = \frac{\dot{Q}_{L_j} \times (T_{P_j} - T_s)}{\sum_{i=1}^{n} \lvert\dot{Q}_{L_i}\rvert \times (T_{P_i} - T_s)}$	LIF is a metric which characterizes the significance of leakage channel j, where n is the total number of leakage channels, T_s is the supply temperature, $\dot{Q}_{L_j}$ and T_{P_j} are the leakage volumetric flow rate and penetrating temperature for channel j, respectively. A positive LIF reflects the cold air bypass and a negative LIF refers to the hot air recirculation.
Flow Reduction Factor (FRF) [180]	$FRF = \left\lvert \frac{Q_t}{FD} \right\rvert$	FRF measures the proportional reduction of IT air flow rate based on its design airflow with no backpressure, where Q_t is the real instant air flow rate within IT device and FD is the design air flow rate with no backpressure.

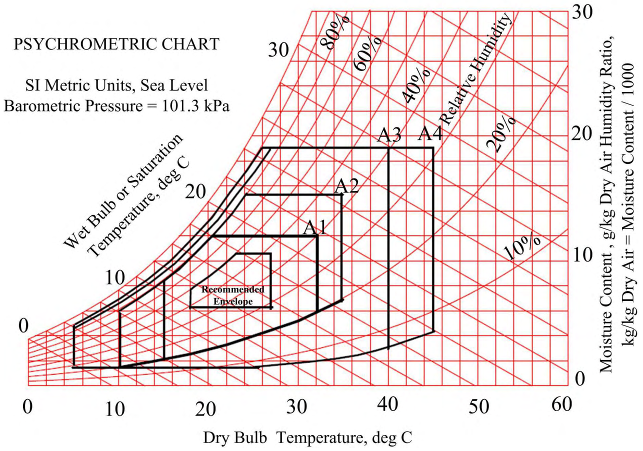


FIGURE 8. ASHRAE's supplying air condition range [181]. The recommended class specifies a strict air condition, which lies within 18 – 27°C dry-bulb temperature and 5.5°C dewpoint to 15°C dewpoint and 60% relative humidity. It is used to ensure the high reliability and performance of IT equipments in data centers. Allowable class A1 is designed for enterprise servers and storages. It still needs tight control of the environment. Allowable classes A2-A4 have less stringent requirements and only need limited control.

between temperature and reliability of IT equipments has been extensively investigated in the past decades. The accumulated heat can induce server shutdown [112] and remarkably decrease the lifetime [182]–[184]. It also leads to a shortened response time in case of cooling failures [59], [185]. Hardware manufacturers define the IT device reliability by the inlet temperature, which is usually up to 35°C [15]. In production data centers, however, operators often choose a much lower temperature. A survey by Emerson [138] indicated that nearly 93% of the operators set the supplying air temperature at or lower than 22°C. While keeping a good indoor environment for IT devices, however, this practice induces serious concerns from the energy consumption perspective, since the low temperature setpoint compromises the energy efficiency of cooling system [72], [186].

Nevertheless, many field studies suggested that commercially available computers are much more durable than expected. They can still function well in extreme conditions, e.g., hotter (40°C or more [187]) or colder (less than −20°C [130]) ambient temperatures. Even higher temperature is also acceptable if Hard Disk Drives (HDDs) are replaced by Solid-State Drives (SSDs). Therefore, there is plenty of room for operators to save the cooling energy by raising the supplying air temperature.

In recent years, High Ambient Temperature Data Centers (HADC), whose operating setpoint is higher than 25°C, are proposed to improve the cooling efficiency and evade the overcooling [188]. Raising the supplying air temperature saves the cooling energy in following manners: 1) Higher chiller efficiency. It was shown that every 1°F (≈ 0.56°C) increase in the chilled water temperature associated with the increased air temperature will bring 2% improvement for the chiller efficiency [189]. As a consequence, raising the supply air temperature alone can save 15 − 25% in chiller energy. 2) More free cooling hours. The primary cooling source will shift from the chiller to cooling tower or direct outside air cooling [187], and the chiller can be shut off most of the time [188]. Temporarily increasing 3°C beyond the ASHRAE's allowable upper bound makes the free cooling with evaporative cooler feasible in almost each part of the world [190]. Moreover, the increase in temperature reduces the air density, which improves the air flow delivery efficiency in open aisle systems with a low tile flow rate: the cold air can reach higher parts of the rack, and the hot air exited from racks can approach closer to the ceiling [191].

B. ENERGY AND PERFORMANCE CONSIDERATIONS

Although raising the supplying air temperature is proven to be a valid approach to improve the cooling efficiency, it is not a panacea which can be used everywhere without restrictions. The optimal temperature setpoint varies across different data centers since the indoor and outdoor thermal characteristics, IT workload, and equipments are all different [192]. Some issues should be considered carefully to avoid unexpected results.

1) TEMPERATURE VS. ENERGY

The first energy consideration which is often overlooked by cooling engineers is the leakage power induced by the leakage current in electronic chips. In the past, it did not draw much attention since the leakage current is an order of magnitude less than the chip normal operating current. However, as the scale of semiconductor technology enters the nano-meter era, the leakage is exacerbated by smaller geometries [193] and is now very significant [194]. Yan *et al.* [195] reported that the percentage of leakage power in holistic chip power consumption rised from 22% in 70nm technology to around 70% in 35nm technology. A number of analytical models for the leakage power, ranging from micro scale [193], [196] to macro scale [6], [7], [197], show that there is a superlinear relationship between the leakage power and temperature. Raising the ambient temperature increases both the chip operating temperature and leakage power.

The second energy consideration is from air moving devices, e.g., fans and blowers, which directly control the air flow rate injected to server cabinets. Some initial results showed that for the high ambient temperature environment, keeping the server temperature within the safety range required more fan power [56], [198], [199]. It was also revealed that the fan power was directly proportional to the supplying air temperature [200]. As a consequence, the overall energy consumption of cooling and ventilation systems is not minimized under the high ambient temperature. Therefore, raising the supplying air temperature is not always conducive to energy reduction.

2) TEMPERATURE VS. PERFORMANCE

Temperature and performance are actually contradictory issues. High supplying air temperature only provides limited cooling capacity for performance-critical equipments. As a result, the system performance degrades. Studies suggested that every 15°C temperature increase led to $10-15\%$ growth in circuit delay [201]. A high temperature above some thresholds also triggers the CPU or memory throttling as well as the Read-after-Write (RaW) mode for the hard disk [183] which reduce the system throughput.

The above considerations clearly indicate that cooling, ventilation, and IT systems must be jointly optimized to determine the data center ambient temperature. Simply raising the temperature setpoint is surely not enough. Beaty and Quirk [202], [203] provided comprehensive discussions on the workflow for raising the ambient temperature from the cooling system perspective. El-Sayed *et al.* [183] analyzed a large field data set and presented insights on the impact of ambient temperature on IT system energy, reliability, and performance. The energy and performance optimizations at the data center scale need joint efforts from both cooling and IT experts. Some pioneering works have been done in the past [39], but applying them into the practice still has a long way to go.

C. PRACTICAL ISSUES

Since the supplying air temperature is controlled at CRAHs, here we discuss some practical issues for CRAH management.

1) DYNAMIC TEMPERATURE CONTROL

Dynamically adjusting the CRAH temperature setpoint according to the instant thermal load is an effective way to save the energy consumption [56], [204]–[211]. Generally speaking, tuning the temperature is based on the readings of CRAH temperature sensors. Unfortunately, many data centers install sensors in wrong places, i.e., return faces of CRAHs [103], [113], [139], [212]. Since what we care is the rack inlet temperature, CRAH sensors should be placed near the rack inlets of cold aisles

The temperature changing rate cannot be too fast. The temperature variability in data centers is an equally important or even more important factor for disk reliability than the actual operating temperatures [183]. The maximum temperature changing rate should not exceed 5°C and 20°C per hour for tape devices and disk devices, respectively [213]. Therefore, minimizing the temperature variation should also be an optimization objective to ensure the data center reliability [214].

Contrary to many model-based studies which assumed that CRAHs provided cold air flows at a constant preset temperature, the supplying air temperature is actually fluctuating around the setpoint and is affected by many factors like the hot return air temperature, cold fluid temperature, and flow rates, etc., since CRAHs are basically heat exchangers [215]. The temperature differential between the simple constant model and Thermal Characteristic (TC) model can be as high as 9°C, indicating that the constant assumption may not be applicable in some situations.

2) MULTI-CRAH MANAGEMENT

Large-scale data centers typically over provision the cooling capacity by hosting a number of CRAHs in a machine room to facilitate possible future expansions. However, production data centers are underutilized most of the time [216], [217]. The mismatch between thermal load and design capacity results in a low cooling efficiency [115], since the cooling system is designed to meet the peak thermal load [218]. One feasible way to address this issue is to identify underutilized CRAHs by measuring the temperature differential of air flow traversing across CRAH[1], and completely shut them down [137], [219]. However, some studies like [119] found that turning off CRAHs will significantly change the air flow pattern within the data center and may only yield limited power savings. Another approach is to under provision the cooling resource such that the cooling efficiency rises remarkably with a little sacrifice in performance and reliability [190].

When multiple CRAHs cooperate to perform the cooling duty in a large machine room, their interactions under the raised floor should not be neglected [15], [103]. The Thermal Correlation Index (TCI) [119], physics-based methodology [220], or CFD simulation [221] can be used to quantify the effect of a CRAH on a particular rack and partition the data center into regions. Local temperature management within a region is then possible by tuning the temperature setpoint of CRAH which most affects the inlet temperature of this region [112], [221]. Note that the effective region should be dynamically adjusted to reflect system changes such as CRAH failures [221].

VII. FAN CONTROL

Temperature control alone is not sufficient to optimize the efficiency of cooling systems. Fans, which constantly drive the cold air to heat sources like CPU, memory, and hard drive, etc., consume energy comparable to the chiller (39% of the total cooling system energy consumption) in a data center [222]. An unnecessarily high fan speed leads to excessive energy waste due to the cubic power-speed fan law. If fans run too slow, on the other hand, negative pressure differential may be created which damages the rack inlet temperature [103]. Modern data centers prefer to use Variable Frequency Drive (VFD) fans to dynamically adjust the volume of cold air supply according to real time demands [94]. It was shown that the energy saving by properly reducing the air flow rate was sometimes even larger than raising the supplying air temperature [199].

A. FAN TYPES AND CONTROL STRATEGIES

There are a lot of fans in the data center, ranging from small built-in server fans to large facility fans. Here we survey fan control strategies for various kinds of fans.

1) TRADITIONAL SERVER FANS

The server fan speed is the most critical factor affecting the air flow rate inside servers [122]. However, field experiments showed that for some specific locations, other factors also matter. For example, the tile flow rate significantly influences the server flow rate in the lower part of racks [98], [99] due to a large negative pressure differential between server inlet and outlet. The small fan-to-obstruction distance inside the server and interactions among multiple fans also have pronounced impact [69]. Therefore, the traditional flow curve model, which is obtained by subtracting the system impedance curve from the fan curve and is valid for large CRAH blowers, may overestimate the flow rate by as much as 50% for small server fans. Alissa *et al.* [69] presented an experimentally calibrated effective flow curve model to capture the server fan behavior more accurately. They also proposed an active flow curve model which predicted the server flow rate based on internal and external impedance effects of IT fans [180]:

$$Q = \begin{cases} \alpha_{r_1} f^3 \Delta P^2 + \beta_{r_1} f \Delta P + \gamma_{r_1}/f & , P_C > \Delta P \\ \alpha_{r_2} f^3 \Delta P^2 + \beta_{r_2} f \Delta P + \gamma_{r_2}/f & , P_C \leq \Delta P \end{cases} \quad (1)$$

where f is the normalized fan speed. P_C and ΔP are the critical backpressure for a specific IT device and the pressure differential between inlet and outlet, respectively. A similar model was used in [223] to predict the air flow in Open Compute storage devices with errors less than 5%.

Piatek *et al.* [167] designed a fan controller which operated according to an analytical thermal model. An energy saving of 12% was reported in a data center scale simulation. Zapater *et al.* [6] proposed a fan control strategy to minimize the fan plus leakage power for a single server. Zheng *et al.* [224] combined the Thermoelectric Cooler (TEC) with the server fan. In their system, the fan speed was reduced since local hot spots were removed by TECs. Han and Joshi [225] developed a reduced Proper Orthogonal Decomposition (POD) model for CPU and heatsink based on which a fan controller was designed. To reduce the probability of CPU overheating due to the approximation in POD model, the temperature was constantly monitored and the fan speed was set to maximum unless the temperature droped below the limit. Li *et al.* [226] proposed a proactive server fan control strategy using a user specified parameter P_p to express the user's preference in the tradeoff between temperature and power/performance. VFD fans can also work with partitions [227] to direct the air flow to the most heated components inside server cabinet, thus reduce the air flow rate by 13.8% and save 32.6% fan power.

To stabilize the fan speed, a safe CPU operating temperature range can be defined, and the fan speed changes only if the boundaries of the temperature range are violated [228]. Lee *et al.* [229] designed a fuzzy fan controller to maintain the component temperature while suppressing the fan speed fluctuation. Kim *et al.* [230] presented an adaptive Proportional-Integral-Derivative (PID) server fan controller which eliminated the fan speed oscillation induced by time lags and quantization in temperature sensors.

[1] A small temperature differential indicates low efficiency.

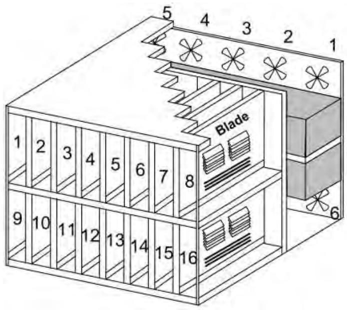


FIGURE 9. An example of fan wall in a blade enclosure [63]. The thermal distribution inside enclosure may not be uniform due to the imbalance of computing workload. Fans should coordinate to effectively remove the heat from enclosure, since a blade node receives the air flow from multiple fans on the fan wall.

2) BLADE SERVER FANS

There are many data centers equipped with blade servers rather than traditional servers. Multiple blade servers are usually installed into a single enclosure (see Fig. 9) which is finally mounted onto the rack. A fan wall in the front side of enclosure is responsible for sucking the cold air in. In this case, the air flow blowing over a blade server is affected by multiple fans on the fan wall. To estimate the air flow received by a single blade server, a linear combination model [63] can be used, i.e.,

$$V_j = \sum_i \eta_{ij} FS_i, \tag{2}$$

where V_j is the air flow rate for blade server j, FS_i is the air flow rate of fan i, and η_{ij} is a coefficient related to the fan-to-blade distance and enclosure layout.

Wang *et al.* [231] developed transient and stationary thermal models for the server temperature and proposed a proactive fan control algorithm to minimize the server fan energy. Compared to traditional approaches, the proposed controller saved 20% fan energy. They extended their work to a hierarchical framework which integrated workload, server, and fan control under power cap and temperature constraints for blade enclosures [63]. A distinct characteristic of this approach is that its implementation relies on the cooperation of many distributed controllers with different time granularities. The idea of cooperation was also used in other literatures. For example, Kodama *et al.* [232], [233] found that the overheat of some nodes in the blade system would generate a high fan power. They proposed a method which jointly considered the fan control, workload dispatching, and node relocation to balance the node temperature distribution.

3) RACK FANS

In traditional servers, the temperature and power (mainly the server fan power) rise drastically under high ambient temperature, since these servers are not designed for operating in such environment [187], [235]. The solution is to use large and more energy efficient rack fans instead of small server fans. The air flow control inside servers, which is often unapplicable in multitenant data centers [236], is also possible if rack fans are installed. Industrial examples of fan-equipped racks include Scorpio rack [187], CloudRack [234], ECS rack [235], and Rack Scale Architecture [237], etc., as shown in Fig. 10.

FIGURE 10. Examples of server rack equipped with shared cooling fans. (a) Scorpio rack [187]; (b) CloudRack [234]; (c) ECS rack [235]. The fan size in the Scorpio rack is 140 × 140 × 38mm (corresponds to 3U). Fans in the CloudRack are even larger. For ECS racks, the cold air enters the rack at the rack bottom with the help of 8 rack fans (92 × 92 × 38mm each) blowing upward.

In [238], a row-based rack fan control scheme was obtained by a linear function of weighted sum of internal Pulse Width Modulation (PWM) signals from all servers, where weights were determined by measuring the influence zone for each fan row. With a shared and centralized rack level fan control, the fan power and total cooling power were reduced in the orders of 50% and 10%, respectively, as compared to the traditional server design [187], [238]. Sahini *et al.* [239] showed that rack fans alone could drive the air flow in data centers even without CRAH blowers, and the CRAH-less data center saved 73% fan energy. In addition, the rack fan wall is also more robust against a single fan failure [240]. A problem for shared rack fans is that some servers may be overcooled, since the thermal load is not usually uniformly distributed inside the rack. However, even in this case the fan power consumption was found to reduce by around 35% compared to traditional server fans [241]. In practice, the distance between rack fans and servers should be large enough (20mm for example) to ensure a uniform air flow rate across all servers [240].

4) FACILITY FANS

Facility fans are usually large fans to move massive volume of air. Typical examples of facility fans include CRAH blowers [242] and economizer fans to overcome the filter resistance [243], etc. Endo *et al.* [244] proposed a facility fan control algorithm for fanless servers in modular data centers, which achieved 22.8% energy saving compared with servers with built-in fans. Patterson *et al.* [113] used the temperature differential between rack top and bottom as the input parameter to determine the CRAH blower speed. The control framework for CRAH blowers can also be integrated with various forms of objectives. For example, Das *et al.* [242], [245] proposed two families of utility functions, i.e., multiplicative and additive utility functions, to satisfy different preferences of data center operators. Multiplicative utility functions expressed the objective of energy minimization subjected to soft temperature constraints, and additive utility functions considered the economic costs of energy consumption and temperature-induced equipment lifetime reduction. If there are multiple CRAHs in the computer room, all fans are typically tuned simultaneously with the same speed [139], [246] at the cost of a little loss of control flexibility.

B. COORDINATED FAN CONTROL

The data center operates in suboptimal states if various kinds of fans are not well coordinated. This is because the mismatch between air supply and demand severely degrades the cooling performance [122], [247]. Coordinated fan control is an effective approach to balance the air supply and demand.

1) DRAWBACKS OF UNCOORDINATED FAN CONTROL

The uncoordinate fan control generally results in under or over provision of the cold air, for which the drawbacks are briefly discussed in the followings.

- Under provision case. The lack of cold air creates insufficient cooling. For open aisle systems where the cold and hot air are free to mix, the hot air recirculation will be exacerbated due to the low pressure of cold aisle [99]. The server temperature will quickly surpass the secure threshold which directly leads to CPU and memory throttling [248]. As a consequence, the server reliability and system performance are compromised. For contained aisle systems, the assumption of unlimited air supply, based on which the built-in server level cooling system is designed [176], is no longer valid. A negative pressure differential between cold and hot aisles is generated due to the shortage of air supply, and may finally result in the reverse air flow in IT equipments even if server fans are working at full speed [180]. To detect the existence of under provision without the help from pressure sensors, a convenient way is to measure the rack inlet temperature to see whether there are recirculations or reverse flows.
- Over provision case. Many data center operators would rather over provision the cold air [15], [249], [250] at the cost of higher energy expenditure to ensure the reliability. It was reported that the current industrial practice used 2.5 times air flow than required [32]. However, this still does not necessarily improve the cooling efficiency. Song *et al.* [87] showed that the over provision of cold air could lead to an increased air flow speed through perforated tiles, and a higher inlet temperature in rack bottoms was found due to the Venturi effect. Over provision also increases the cold air leakage from the cold aisle and underfloor plenum [87]. It is not an easy task to discover the over provision. A practical approach is to regularly reduce the air flow and monitor whether the reading of temperature sensors installed on the overflow ports increases [135].

2) COORDINATE CONTROL STRATEGIES

A prerequisite for coordinate fan control is that the CRAH blower speed can be adjusted independently from the supplying air temperature [15]. The speed of CRAH blower can be determined based on many factors such as the airflow rate, temperature, and pressure differential between cold and hot aisles, etc [66]. If the thermal state inside servers (such as server fan speed) is available, the airflow demand for each server can be accurately estimated [67]–[69], [237]. More specifically, if there are totally n fans inside the server, then the server airflow rate can be written as [247]

$$Q = \sum_{i=1}^{n} c_i \times RPM_i + c_{n+1}, \tag{3}$$

where c_i is a coefficient and RPM_i is the fan speed. The speed of CRAH blower can then be configured as the sum of air flow rate required by all servers.

In other cases where this information is unavailable, the pressure differential is used to determine the extent of cold air provision [66], [69], [79]. To do this, a number of pressure sensors are distributed into the underfloor plenum and cold/hot aisles [70], [251]. Some threshold values such as -10, 0, and 10 Pa representing the under, perfect, and over provision can be used to set the CRAH blower speed [70], [71]. In practice a slightly over provision is needed [71]. The extent of over provision is site-specific and depends on many factors such as the control system response and accuracy, variability of IT workloads, and leak-tightness of the containment system [15], etc. For example, the fan control algorithm developed by Hackenberg and Patterson [135] maintained a positive differential pressure below 5 Pa in their downflow plenum system, and Nemati *et al.* [153] kept the pressure differential between $1-3$ Pa.

Alternatively, Khalili *et al.* [252] provided an approach to estimate the Critical Under Provisioning Flow Rate (CUPFR) for a cold aisle. CUPFR is the lower bound of total required flow rate, i.e., some devices are not able to draw air from the cold aisle if the provisioned air flow is less than CUPFR. Letting Q_c be the CUPFR for a cold aisle, it can be computed as

$$Q_c = \sum_{i} N_i Q_{i@p} - Q_{L@p}, \tag{4}$$

where i is used to index the device type, N_i is the number of type i devices, $Q_{i@p}$ is the flow rate of type i device (related to fan speed) given the differential pressure p, and $Q_{L@p}$ is the leakage air flow rate into the cold aisle under p. Here, p is the critical pressure (a pressure threshold under which the equipment cannot suck in any cold air [180]) of devices with the weakest air moving systems, e.g., network switches.

C. PRACTICAL ISSUES

The time granularity of dynamic fan control policy should be in accordance with the mechanical and physical constraints.

The turning on/off cycle of large fan motors should not exceed the order of an hour [253], and the small fan speed can be adjusted in orders of seconds or tens of seconds [63], [228].

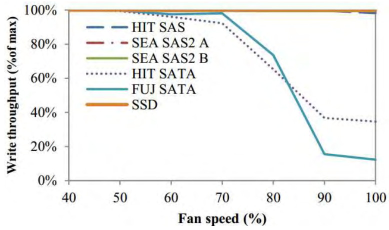


FIGURE 11. Relationship between server fan speed and performance of various storage systems [255]. The write throughput of HDD systems is relatively stable when the fan speed is slow. However, it degrades drastically when the fan speed exceeds a threshold (70%) due to the increased vibration amplitude. In contrast, SSD systems are almost insensitive to the fan speed.

The vibration-induced hard disk performance degradation should be paid more attention while designing the server and rack fan control strategy. It was shown in [254] that although the vibrational amplitude was a linear function of fan speed, the relationship between HDD performance degradation and fan speed was superlinear (see Fig. 11). An unoptimized fan control algorithm decreases the hard disk performance up to 60%. A hybrid SSD-HDD system accompanied with a novel fan control policy was proposed in [255] to mitigate the impact of harmful vibration, where SSD played a role of data cache. As shown in Fig. 11, the HDD performance is less sensitive to the fan speed while the fan speed is low. The fan control policy took advantage of this feature by a double-state operating mode: 1) At moderate temperatures and low fan speeds, the fan control policy is generally temperature-aware; 2) When the temperature increases and requires a high fan speed, the control policy's focus will shift to balance the temperature and performance.

The noise is also a serious problem since there are a large number of fans inside the data center. The main sources of noise are the flow turbulence and mechanical vibrations [256]–[258]. It was reported that 20% faster air flow speed increased the noise by 4dB [259]. The Heating, Ventilation, and Air Conditioning (HVAC) system and a single server typically produce 70 dBA and $40-70$ dBA noise, respectively. The data center scale noise lies within $70-80$dBA, with higher level observed in the hot aisle [260]. Prolonged exposure in such noisy environment impedes effective vocal communication and potentially harms the hearing of workers. Novel fan control algorithms like the phase-aware controller [261] were proven to be effective in noise suppression.

VIII. UNDERFLOOR PLENUM

Today most of the large scale data centers adopt the raised floor architecture as a standard part of air flow delivery system. The cold air flow driven by CRAHs first enters the underfloor plenum before reaching overfloor cold aisles. Therefore, although invisible to most of data center operators and users, the underfloor plenum has significant impact on the performance of air flow distribution.

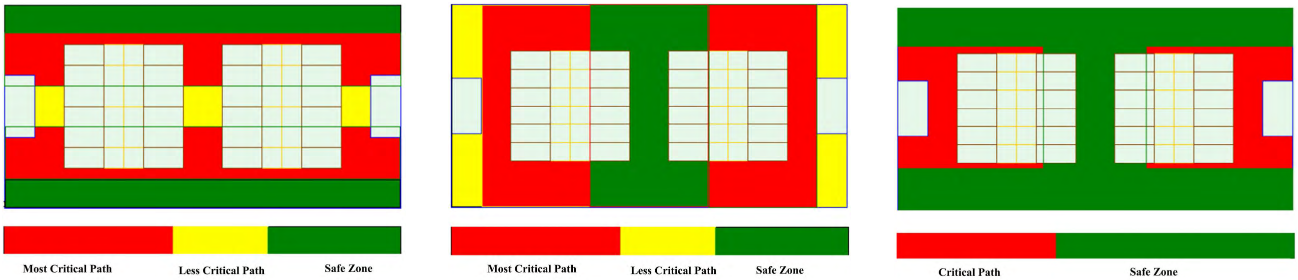
A. LEAKAGE

The air leakage exists in the underfloor plenum [87]. The air flow temperature typically rises 3°C when it passes through the underfloor plenum [78], [154]. The leakage area is around 0.2% and 8% of the floor area and open area, respectively [32]. The leakage flow in a typical data center is between $5-15\%$ [156]. For data centers with a limited number of perforated tiles, the amount of leakage is comparable to the air flow through vent tiles. Radmehr *et al.* [156] developed a method to measure the amount of leakage and found that the leakage flow when there were only 2 vent tiles was over two times larger than the one with 20 vent tiles. Karki *et al.* [262] proposed a CFD-based model to estimate the floor leakage under the assumption that the leakage area was uniformly distributed over the entire raised floor except for the areas covered by solid objects. They further simplified the model by assuming a uniform underfloor pressure distribution, which showed that the ratio of floor leakage can be estimated by the ratio of leakage area. Containment technique is available for the underfloor plenum to reduce the leakage. Experiments showed that the underfloor containment system increased the tile flow rate by 9% in Cold Aisle Containment (CAC) systems [84].

B. PRESSURE DISTRIBUTION AND OBSTRUCTIONS

The underfloor plenum pressure is the major determinant of the tile flow rate. It is the pressure variation in the plenum which accounts for the nonuniformity in airflow distribution [32]. Randomizing the direction of underfloor supplying air flow uniforms the pressure. However, its implementation requires additional sensors and equipments [263]. A more popular and simpler method to mitigate the pressure variation is to alter the geometry of plenum. A change in the plenum length only affects the frictional resistance, and a change in the plenum height modifies both frictional resistance and pressure [264]. Therefore, increasing the plenum height generally homogenizes the tile flow rate and rack inlet temperature distribution [82], [96], [265]. However, the uniformity of air flow may be undermined if the tile porosity is too high (over 50%) [266]. Once the plenum height surpasses a threshold, any further increase does not affect the flow distribution. The typical depth of underfloor plenum in current data centers lies between $12-36$ inches ($0.305-0.914$m), although the optimal range is $30-36$ inches [30]. Zhang et al. [267] investigated the effect of plenum height in various air flow delivery configurations. It was shown that the optimal heights were $1.0-1.2$m, $0.6-0.8$m, and $0.4-0.6$m for open aisle systems, cold aisle containment systems, and hot aisle containment systems, respectively.

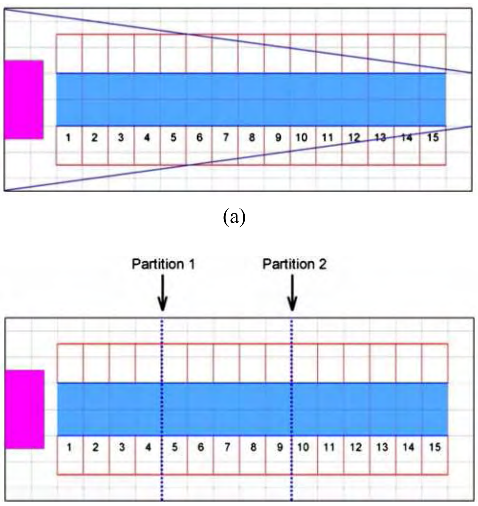
In addition to the role of a tunnel for cold air transportation, the underfloor plenum also serves as a container to host cables, pipes, and structural beams, etc. Investigations showed that these obstructions notably affected the pressure



(a) (b) (c)

FIGURE 12. Safe zones for (a) parallel obstructions; (b) perpendicular obstructions; (3) random obstructions [81].

distribution in the plenum. In particular, these obstructions decrease the air flow rate by as much as 80% and increase the rack temperature by up to 2.5°C [268]. Bhopte *et al.* [81] performed a series of CFD analysis to identify safe places for piling obstructions such that their harmful effects were minimized. It was found that parallel blockages should be placed near the walls (Fig. 12a) and perpendicular blockages should be placed away from the space between CRAH and cold aisle (Fig. 12b). Sammakia *et al.* [52] futher investigated an unsymmetrical data center layout. They concluded that perpendicular blockages were more detrimental than parallel blockages. In addition, they also supported that blockages should not be placed in the spaces under perforated tiles and between CRAH and cold aisle.



(a)

(b)

FIGURE 13. Beneficial underfloor blockage [32]. (a) Inclined partition. (b) Perforated partition.

Some properly-placed obstructions can improve the pressure distribution and air flow delivery. When stacked into the safe zone as shown in Fig. 12c, random blockages packaged into bins with full plenum height reduce the average rack inlet temperature by up to 0.3°C [81]. Patankar [32] proposed to deliberately set inclined partitions in the plenum (Fig. 13a) to reduce the available area for the horizontal air flow at plenum end far away from CRAH so that a uniform tile flow distribution is obtained. However, this approach requires solid partitions which is unfavourable in case of CRAH failure. To address this issue, he advocated to use perforated plate to balance the pressure distribution (Fig. 13b).

C. EXTENSIONS

The underfloor plenum can be further divided into smaller compartments by installing partitions, such that each CRAH only feeds air to a specific compartment [265]. This approach is more flexible in terms of the local air provision and under floor pressure control. In case of CRAH failures, the partitions can be removed manually or automatically as the underfloor pressure drops.

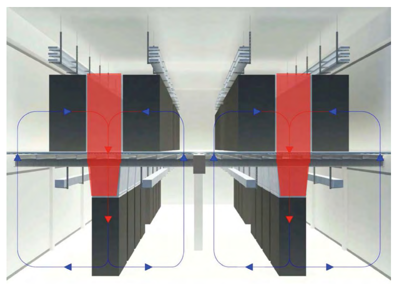


FIGURE 14. The downflow plenum design [139].

Hackenberg [139] extended the concept of plenum into a full-height building story which separated the locations of IT devices and supporting facilities (such as cooling and electrical distribution equipments) into distinct floors, as shown in Fig. 14. CRAHs are placed directly under the hot aisle, and the cooling capacity grows with the size of computer room. This creates an obvious security advantage where IT and non-IT stuffs have different working spaces. In addition, the cross-section area increases, and the cold air path and pressure drop decreases, leading to a reduced CRAH power. By careful design, the volume size of data center with downflow plenum even decreases by 10%. It was expected that the partial PUE (pPUE) based on CRAH fans alone was less than 1.02 [135].

The underfloor plenum also can be used as a space where beneficial air mixing takes place. Contrary to a common viewpoint that the mixing of cold and hot air degrades the cooling efficiency, a recent study by Demetriou and Khalifa [269] found that bypassing the hot air from CRAH and directly into the underfloor plenum actually saved the CRAH energy in data centers with contained cold aisles, because it decreased the amount of air traversing across CRAH where a high flow resistance should be overcame. As a consequence, the CRAH blower power, which is a cubic function of fan speed, is reduced. However, this will introduce a hot air flow into the underfloor plenum and raise the plenum temperature. CRAH should be set to a lower temperature to guarantee the rack inlet temperature constraint, which results in a lower refrigeration efficiency. Therefore, a tradeoff should be considered between the CRAH blower power and refrigeration power. In their study, Demetriou and Khalifa demonstrated that the cold and hot air could be well mixed in the underfloor plenum, and an optimal operating condition (the ratio of bypass) existed [270], [271]. An overall cooling energy saving of over 35% is expected if efficient chillers are used [269]. When integrated with the indirect air side economizer, the saving grows to 58% due to the slower air flow rate in the economizer [272].

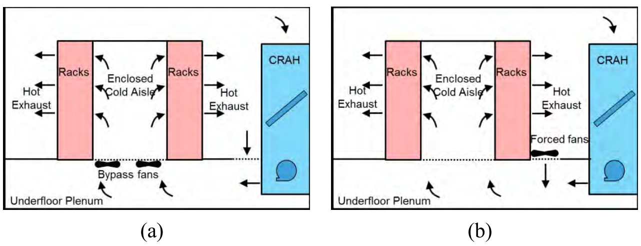


FIGURE 15. Hot air bypass configurations [270]: a) Induced, and 2) Forced.

There are several approaches [270], [272], [273] to implement the underfloor air mixing as shown in Fig. 15: 1) Induced bypass, where the bypass flow enters the underfloor plenum via a bypass branch in CRAH or perforated tiles with the help of bypass fans. 2) Forced bypass, where the hot air is pushed into the underfloor plenum through perforated tiles or bypass ducts by low-lift forced fans. Numerical studies showed that both approaches achieved considerable overall energy saving although the induced bypass had an advantage over the forced approach in terms of the reduced underfloor plenum pressure and less leakage [270], [272].

IX. PERFORATED TILES

Perforated tiles are the gateway where the cold air is delivered into the cold aisle. Based on whether there are auxiliary fans attached to the tile, they can be generally categorized as passive tiles and active tiles.

A. PASSIVE TILES

The air flow through passive tiles is driven only by the pressure differential between the underfloor plenum and machine room. Therefore, the tile flow can be controlled through adjusting the CRAH blower speed and tile opening ratio.

1) AIR FLOW CHARACTERISTICS

The tile flow distribution is a complex function of many factors [106], [252], [274], of which the underfloor pressure and tile pressure drop are the most important. Section VIII has elaborated the impact of underfloor pressure. The pressure drop for perforated tile is proportional to the square of approaching air flow velocity and pressure loss factor [264]. The pressure loss factor is affected by the tile porosity, i.e., the lower porosity, the higher pressure loss factor. A more detailed pressure drop model was given in [275]. Compared to the pressure drop at CRAH and subfloor turning, the pressure drop at perforated tiles is trivial [32]. Hence, the opening ratio has little effect on the overall air flow volume rate. However, it does affect the air flow velocity near the tile and rack top [133], and subsequently influences the pressure at rack bottom and the cold air bypass in open aisle systems [89], [276].

For vent tiles with low opening area, the pressure drop over tiles increases and becomes much larger than the horizontal pressure variation in the underfloor plenum [32]. Therefore, the air flow distribution is nearly uniform [94], [127], [277], [278]. Low porosity tiles are favorable in exact or under provision cases since they help the cold air to reach the rack top and reduce the hot air recirculation [279]. However, they increase the air flow momentum above the tile surface and cold air bypass at the rack top if the cold air is over provisioned [34], [252]. In addition, the high air velocity generates a low pressure region near the tile surface and degrades the cooling efficiency for servers located at rack bottom.

If the opening area is high, the uneven pressure distribution and complex swirling flow pattern in the underfloor plenum appear to be dominant factors for the air flow velocity. This is the reason why remarkable air flow rate variations were observed in [174], [277], and [280]. In particular, tiles near CRAH usually have low or even reverse air flows because of the high air flow velocity and low static pressure near the underfloor CRAH outlet [32], [96], [100], [101], [106], [266], [281]. The tile flow rate in the center of cold aisle is larger if CRAHs installed near aisle ends are blowing against each other [282]. The heterogeneous tile flow rate leads to substantial cooling capacity variation, i.e., nonuniform inlet temperature distribution at rack inlet. In practice, an opening ratio of 20% − 30% is recommended [266].

In order to guide more cold air into servers rather than bypassing racks, directional tiles, where angled vanes are attached to the bottom side of vent tiles, are proposed [252]. Following the angle of vanes, the tile flow hits the lower part of the rack and increases the local pressure. Therefore, the under rack recirculation is reduced when the cold air is well provisioned. However, if the cold aisle containment system is installed and the cold air is under provisioned, the angled air flow exacerbates the under rack recirculation and raises the inlet temperature of the lowest server.

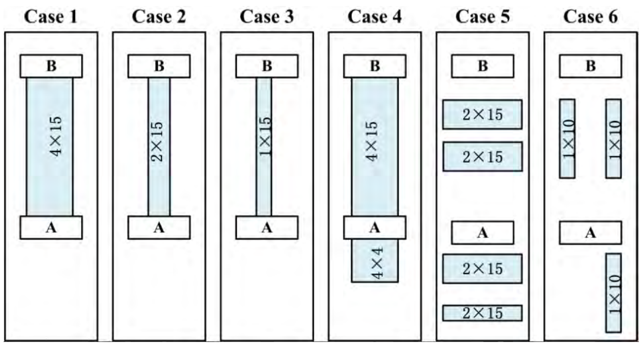
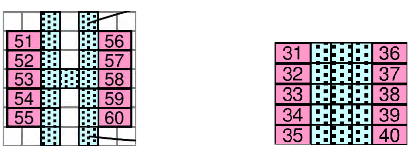


FIGURE 16. **Tile layouts investigated in [106], where A and B are CRACs and blue parts are perforated tiles.**

Other configurations, such as the aisle containment, tile layout, tile structure, and Angle of Approaching underfloor flow (AoA), etc., also affect the tile flow performance. The existence of containment changes the pressure differential between under- and overfloor spaces. As a result, the tile flow rate varies as well. For under provision case, the tile flow rate is faster in the open aisle system compared to the contained aisle system. However, this relationship is reversed for the over provision case [127]. Schmidt *et al.* [106] studied 6 tile layouts (Fig. 16) and draw some general conclusions: 1) As the number of vent tile decreases, the underfloor plenum pressure grows, leading to a faster tile flow rate. 2) Perforated tiles should be installed between two CRAHs blowing towards each other in order to enhance the tile flow rate, and locations close to CRAHs should be avoided due to the negative pressure. The guard layout [83], which adds perforated tiles to the corner or end racks as shown in Fig. 17, reduces the hot air recirculation for open aisle systems. Finally, we should note that even the air flow rate for a single tile is not uniformly distributed [84], [283]. Honeycomb structures can be attached to the bottom of tile to straighten the flow, improve the uniformity, and remove the large scale eddies from the flow approaching the tile [133]. Understructure scoops [155], [252] are used to capture more air from the underfloor plenum, make the flow direction perpendicular to the tile surface, and reduce the effect of AoA.



(a) (b)

FIGURE 17. **Guard (a) and traditional (b) tile layouts [83].**

A recipe of models with different computational complexity can predict the tile flow performance [89], [284]. Arghode and Joshi [34] elaborated these models and evaluated the impact of pore size, blocked edges, and tile width on the flow rate. Detailed tile models should consider geometry, orientation, and other factors to capture the flow characteristic precisely [102].

Both the underfloor plenum level and tile level solutions are available to make the air flow rate more uniform. The former has already been discussed in the previous section. Tile level solutions include the followings: 1) Populating the floor with low porosity tiles. This will increase the pressure of underfloor plenum, the amount of leakage, and the tile flow speed. 2) Deploying tiles of different porosities at different locations according to the underfloor pressure distribution [82], [285]. 3) Substituting traditional perforated tiles with controllable tiles [286] or active tiles [122], as discussed later.

2) CONTROLS

Controllable passive tiles are often referred to as Adaptive Vent Tiles (AVTs) [286], which are extensions of passive tiles where the opening ratio or open/closed state can be finely adjusted dynamically according to the cooling demand by a damper installed beneath the tile, as shown in Fig. 18. Besides the opening area, the damper can also control the tile flow direction [149]. It was shown that the integration of AVTs with other advanced technologies reduced the cooling cost by 30% [287].

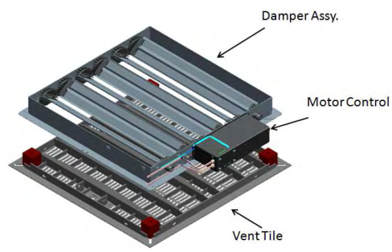


FIGURE 18. **An example of Adaptive Vent Tile (AVT) [289].**

The tile flow rate is proportional to the opening area [212]. Although adjustment of a single tile is a local control action, it impacts the global air flow distribution. This is because AVTs only change the distribution of air flow, rather than the amount of overall supply air volume [286]. For example, closing the tile vent notably increases the adjacent rack inlet temperature but reduces the inlet temperatures of nearby locations [103], since more cold air is ejected from nearby locations to achieve the air flow conservation. The air flow rate m_i of a particular AVT i can be computed as [288]

$$m_i = m_{CRAH} \times \bar{U}_i, \tag{5}$$

where m_{CRAH} is the mass flow rate of CRAH. $\bar{U}_i = \frac{U_i}{\sum_{j=1}^{N} U_j}$ is the normalized opening for AVT i where U_i is the mechanical opening ratio ranging from 0% to 100% and N is the total number of AVTs.

Beitelmal *et al.* [286] found that a constant linear model was not accurate enough to characterize the relationship between rack inlet temperature and tile opening.

They designed a PI-MIMO controller with an online estimator to update model parameters in real time. KRATOS [290] used an MPC controller to guarantee the inlet temperature while cutting down the overall supplying air volume. Wang *et al.* [212] designed another integrated MPC controller for CRAH blower and tile damper to minimize the aggregated opening area and stabilize the system. However, the plenum pressure, which was used as a media to reflect the extent of air flow provisioning and distribution, is hard to determine in practice. Their successive work [288] incorporated the supplying air temperature rather than pressure to formulate a dynamic model for capturing the evolution of rack inlet temperature. To characterize the flow cross correlation effect, they used a linear model to approximate the air flow rate for rack r

$$m_r = \sum_{i \in V(r)} b_i m_i, \tag{6}$$

where b_i is a coefficient and $V(r)$ is the set of AVTs near rack r. One problem of this approach is that all AVTs are individually controlled, leading to scalability issues in large data centers. They further developed a two-step optimization method [289] where all AVTs within the same cold aisle were controlled simultaneously, so that the problem complexity was significantly reduced. In this method, the CRAH blower was adjusted in a finer time scale than AVTs to satisfy the real time cooling demand.

FIGURE 19. An example of active tile [122].

B. ACTIVE TILE

The active tile is an ordinary passive tile attached with fans blowing toward the cold aisle, as shown in Fig. 19. Advantages of deploying active tiles include

- More flexible tile flow control and higher thermal efficiency. The flow rate of passive tiles exhibits great diversity due to the complex underfloor pressure distribution. The data center operator has to use the global control approach, i.e., setting a higher CRAH blower speed, to guarantee the thermal performance for all locations. However, this strategy increases the air flow rate for all vent tiles and often leads to the overcooling for some racks. With the help of tile fans, on the other hand, a finer control of local air supply can be achieved by accelerating the tile fan speed according to the tile location and thermal load of nearby racks. It was observed that active tiles improved the cooling performance in both open aisle systems [99], [122] and contained aisle systems [134].
- Sufficient cold air supply and higher energy efficiency. A phenomenon observed in practice is the deficiency of cold air even if CRAH runs at full speed because of the large flow resistance of the passive tile [122]. Fan-assisted tiles can increase the air flow volume entered into the cold aisle at the cost of limited additional energy consumption of tile fans, which is trivial compared to the CRAH blowers. It was found that in general the ''CRAH + active tile'' configuration was more energy efficient to deliver the air flow than using CRAH alone. More specifically, the total fan energy (including CRAH blower and active tile fan) was reduced by more than two-thirds to attain a same flow rate.
- Less floor leakage. While aisle containment techniques in modern data centers alleviate the air mixing, they add pressure to the air distribution system. More floor leakage was observed in contained systems compared to open systems [69]. Active tiles decrease the underfloor pressure, thus reduce the floor level leakage [122], [134]. On the other hand, extra care should be paid to avoid a negative relative plenum pressure due to the overspeed of tile fans [291], since it will draw the warm room air back into the plenum.
- Increased ride-through time in case of cooling failures. Athavale *et al.* [291] observed an extended ride-through time for data centers deployed with active tiles when the chilled water pump and the CRAC blower failed. Possible reasons for this are a growth in thermal mass available in the room and/or air circulation. In particular, tile fans can partially drive the air flow, thus improve the thermal condition even if the CRAC blower malfunctions.

The air flow speed of active tile is different from the passive tile in that it is an increasing function of both CRAH blower and tile fan speed, but the contribution of tile fan diminishes as the CRAH blower speed grows [122]. The tile flow rate is also strongly correlated to the tile location, and the flow cross correlation effect between adjacent active tiles is limited. The fan-to-tile distance, flow angle, and existence of containment structure also affect the tile flow rate. Song investigated the impact of tile flow angle and/or fan-to-tile distance in open aisle systems [292] and contained aisle systems [98]. Fan positions either too close to or far away from the tile result in performance degradation [99]. For instance, the under-rack recirculation is more conspicuous if fans are placed too close to the tile. A larger fan-to-tile distance increases the uniformity of the tile flow, but may lead to under- and over-cabinet maldistribution. A regression-based model was proposed as follows to estimate the zonal maximum temperature T_{mzm}

$$T_{mzm} = \alpha_0 + \alpha_1 x_1 + \alpha_2 x_2 + \alpha_{12} x_1 x_2, \tag{7}$$

where x_1 and x_2 are the straightening effect and fan-to-tile distance, respectively. The cooling efficiency is reduced as the fan tile angles away from the server inlets. The containment structure has positive effect on the uniformity of tile flow distribution [293].

An extension of active tile is the reversible tile [294], where the tile fan can blow either upward or downward. The tile flow direction is changed from upward to downward when the cold air is over provisioned to reduce the CRAH flow and improve the energy efficiency.

X. AIR CONTAINMENT

The idea of air containment is to physically separate the return hot air from the supplying cold air such that the air mixing is significantly reduced and a more uniform inlet temperature distribution is achieved [70]. With the containment system, data centers are rendered more economizer running hours and opportunities to increase the rack power density. Further, the CRAH temperature setpoint rises, and the fan speed slows down, yielding a better cooling efficiency [15], [78]. It was reported that adding the containment alone reduced PUE from 1.8 to 1.18 [15]. Due to these advantages, many regulations/standards, such as California Energy Code [295], E.U. Code of Conduct [296], ASHRAE 90.4 [297], and Data Center Optimization Plan by U.S. Department of Energy [298], advocate or even mandate to use containment systems in high density data centers.

The containment solution is not perfect as well. The installation of containment systems adds more flow resistance and hence potentially requires higher CRAH fan speed. The leakage still exists even if the containment structure is well sealed [299]. The performance of containment system is susceptible to network devices, which may generate reverse air flows [15], [119] due to the side-to-side or side-to-back (instead of the standard front-to-back) airflow used by routers and switches [139]. The air flow rate at rack bottom inside the containment may suffer from a reduction since it is closer to the low pressure region created by the jet flow from perforated tiles [71]. In addition, there are some extra requirements for the containment system [78]. For example, updated refrigeration systems to tackle a higher return air temperature, increased room geometry space for hosting the containment system, and additional fire suppression mechanisms, etc. The reliability issue for some equipments in the hot aisle, such as electrical gears and network devices, should also be considered [15].

There are some data centers where both contained and open structures exist. This hybrid configuration is not good for cooling efficiency since the containment increases the pressure in the enclosed cold aisle. As a result, the cold air in containment is reduced and the air flow in open aisle is strengthened [69].

A. COLD AISLE CONTAINMENT (CAC)

Implementing CAC (Fig. 20a) increases the cooling efficiency and reduces the energy consumption [72], [73]. In a simulation study by Choo *et al.* [304], CAC made possible a rise of 9.1°C in CRAH supplying air temperature and associated 132MWh annual energy saving. It was estimated that the saving in CRAH is roughly a fifth [305]. CAC also reduces the rack inlet temperature [101] and air flow rate [75]. It was reported that the typical cooling energy saving of CAC was up to 40%, and the power density for CAC systems could achieve 25.2 kW per cabinet, as compared to 14.6 kW in open aisle systems [74]. CAC also prolongs the uptime in case of cooling failure [15]. Shrivastava and Ibrahim [76] reported that the uptime increased from 4 mins to 19 mins when the data center shifted from open aisle systems to CAC systems.

CAC tends to balance the supply-and-demand of air flow [69], [134], [306]. Generally speaking, CAC has minimal effect when the tile flow matches the rack flow. However, if the air flow is under provisioned, CAC increases the tile flow and suppresses the rack flow. On the contrary, if the cold air is over provisioned, the tile flow rate decreases due to the increased cold aisle pressure.

The under rack gap is considered as the main source of leakage in CAC [70], [71], [252]. The leakage is also found in the door and ceiling seams, mounting rails, clearance between racks, and common boundaries between containment panels [75], etc. The amount of leakage significantly affects the performance of CAC systems. It was shown that when the leakage area ratio grew beyond a threshold (typically 15% [75], [299]), benefits of CAC were lost and the temperature distribution at rack inlet became nonuniform.

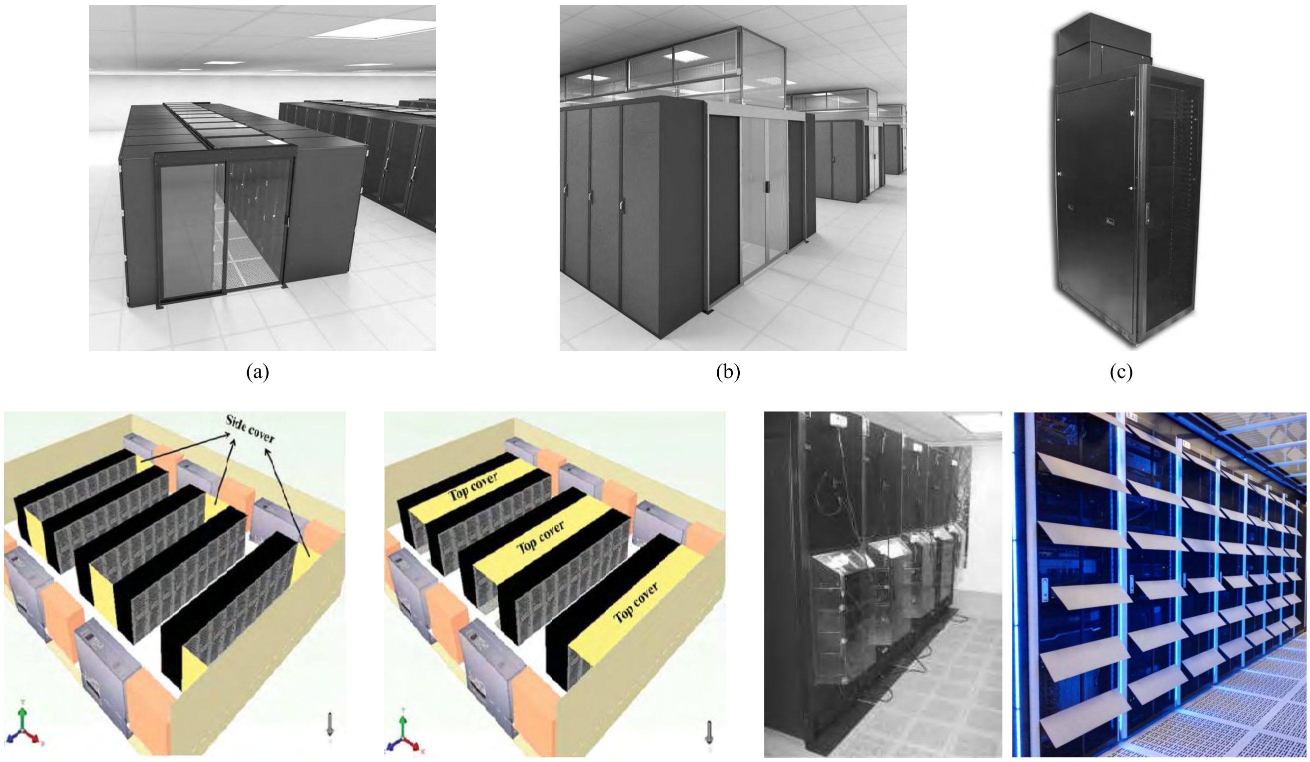
B. HOT AISLE CONTAINMENT (HAC)

HAC seals the space in rack rear and directs the hot air back to CRAH without mixing with the supplying cold air. There are no differences between CAC and HAC in terms of the thermal efficiency [15]. However, HAC is more favorable in practice since

1) For the same air provision level, a less aisle pressure differential is required for HAC compared to CAC, hence a less leakage is expected [71].
2) Nearly the entire room space in the HAC solution is casted into the cold aisle, which potentially yields more cooling capacity [58].
3) HAC takes smaller space for the hot air transportation and is more temperature friendly for data center maintenance staff [139].

The ambient temperatures in all working areas except HAC are all close to the supplying air temperature. Therefore, the CRAH temperature setpoint can be further raised in HAC to achieve the same working environment temperature compared to CAC [72]. It was estimated that 43% annual cooling energy could be saved (corresponding to 15% PUE reduction) by using HAC [77]. Other advantages of HAC include more uptime in CRAH failure and higher heat exchanger efficiency [70], etc.

HAC can be implemented using Ducted Hot Aisles (DHAs) to the drop ceiling (Fig. 20b) or Vertical Exhaust Ducts (VEDs, a.k.a. cabinet chimneys) for individual racks (Fig. 20c). For DHAs, it was suggested that the hot aisle width



(a) (b) (c)

(d) (e) (f) (g)

FIGURE 20. Air containment systems. (a) Cold aisle containment [300]; (b) Hot aisle containment: ducted hot aisle [301]; (c) Hot aisle containment: VED or cabinet chimney [302]; (d) Partial containment: top only containment [100]; (e)Partial containment: side only containment [100]; (f) Rack level containment: half length snorkels installed on the lower half of racks [242]. (g) Rack level containment: baffles [303].

should be larger than 1.8m [239], since a small dimension of hot aisle elevated the pressure and required more server fan energy to overcome it. it The VED solution provides a cooler working condition in hot aisles compared to DHA. It is also an option to capture the hot exhaust air for data centers where DHAs cannot be installed due to space or location issues [78]. However, solid rear doors and side panels are indispensable for VED racks as the internal pressure of racks is very high [70], and additional fans may be necessary to overcome the increased pressure drop [79]. The average cooling performance is generally better for DHAs than VEDs [80].

While the pressure distribution in CAC is rather uniform, the pressure of hot aisles in HAC exhibits variations, i.e., a lower pressure is observed in the upper part for racks close to CRAH [70]. In VEDs, the rack pressure variation is more drastic: lower pressure for racks closer to CRAH and higher pressure for racks far away from CRAH. The performance of HAC also shows a strong correlation with the drop ceiling pressure [80]. To obtain a more uniform ceiling plenum pressure, a depth of $18 - 24$ inches ($0.46 - 0.61$ m) and regular/symmetic layouts are favourable.

Well sealing the containment system is instrumental in the enhancement of HAC performance. Special attentions should be paid for the drop ceiling since many investigations suggested that the ceiling system, especially integrated with lighting fixtures, was substantially leakier than estimated by ASHRAE [80], [307]. The typical leakage of the ceiling should not exceed 10%.

C. PARTIAL CONTAINMENT

The partial containment is a compromised solution used in situations where it is not easy or impossible to implement the full containment. Commonly used partial containment systems include door-only containment [70], [308] (Fig. 20d) and top-only containment [127] (Fig. 20e), etc. The top-only option outperforms the door-only option in terms of both rack flow rate [127] and inlet temperature [100], [309], especially in the fully provisioned case [310]. Wang *et al.* [97] investigated 6 partial CAC options for a containerized data center with a non-standard air distribution system, i.e., ceiling supply and flooded return. They found that keeping the openings at positions farthest away from CRAH to increase the hot air circulation path was the most effective way to improve the air flow performance. In particular, this configuration is even more effective compared with an under provisioned and poorly sealed full containment system, since a shortage of cold air supply in contained aisle will induce severe hot air leakage [127]. Cho and Kim [311] and Nada *et al.* [312] proposed a partial partition system consisted of vertical walls installed at the top of cold aisle (but did not reach the ceiling) to mitigate the hot air recirculation at upper locations. Some advantages of this approach include its fast response to fire and easier relocation of IT and other auxiliary equipments, etc.

D. RACK LEVEL CONTAINMENT

Rack level containment systems take less space and are more flexible to implement. For example, snorkels are cost-effective rack level partial containment systems (see Fig. 20f) designed by IBM for data centers with layouts and structures not suitable for standard aisle containment systems. It was shown that half-length and full-length snorkels reduced the inlet temperature at rack top by 5°F and 15°F, respectively [242]. There are two kinds of snorkels, i.e., flat-top snorkel and angle-top snorkel, which are different only on the degree of the top side. The flat-top snorkel works better than the angle-top snorkel when deployed with CAC simultaneously [313]. The performance of both types of snorkels can be improved by further extending the length down and closer to perforated tiles. Tilted baffles installed at rack inlets (Fig. 20g) can also improve the efficiency of air flow delivery. Yuan *et al.* [303] revealed that larger baffles achieved better performance and the best angle was 75°. It was also shown that the performance was insensitive to the baffle location.

There are many leakage sources in current commercial racks which should be carefully considered in the rack level containment. Blanking the gaps under and inside racks increases the containment pressure and reduces the hot air recirculation [176]. It also remarkably improves the temperature distribution [97], [313]. Other sealing techniques include panels installed in the empty space between servers [74], sealing grommets (flaps or brush-like structures) over cable cutouts [84], side air dams between the cabinet side panels and mounting rails, etc. Absence of these measures significantly degrades the cooling efficiency. It was found that the removal of blanking panels and sealing grommets from cabinets increased the fan power by 29% and 47%, respectively.

Recent advancements in cloud computing technology, e.g., Virtual Machine (VM) migration [314]–[316] and workload management [317]–[319], pose new challenges to the rack containment system. Data centers nowadays are more dynamic under the real-time resizing techniques [64], [320] which adjust the number of active servers and redistribute the workload according to the instant resource demand. Apparently, servers in off/sleep mode do not need cold air flow to keep them cool. However, the current practice takes no specific measures for these servers, making them as sources of leakage [127]. It was observed that servers could leak cold air into the hot aisle at a rate of 23% − 135% of their designed flow rate [246]. Blanking panels do not suit for this scenario because they are installed manually and cannot be frequently removed. In this case, it is more feasible to use active shutters [85], [321], which finely tune the shutter angle or completely block the cold air if the server temperature drops below a threshold (e.g., 40°C), or dampers [246], which are installed inside the server and eliminate the leakage during server shutdown.

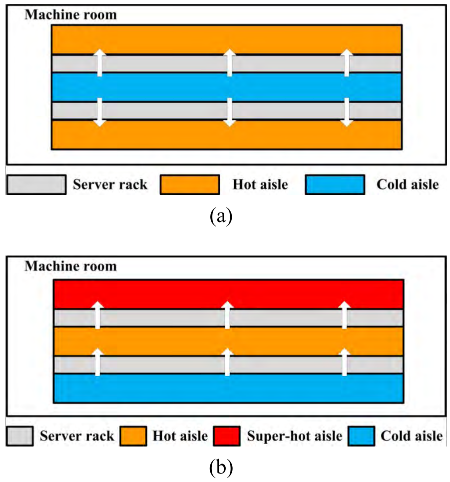


FIGURE 21. Rack placement in a machine room.: a) standard HA/CA, and 2) tandem. Arrows represent the direction of air flow.

XI. RACK CONFIGURATION

Arbitrarily placing the rack may significantly affect the air flow pattern and create unexpected hot spots and other thermal issues [101], [323]. Therefore, nearly all data centers adopt the Hot Aisle/Cold Aisle (HA/CA) layout to organize their server racks. As shown in Fig. 21a, all rack fronts face the cold aisle to let the server inlets draw in cold air, and all rack rears discharge the hot air into the hot aisle. There are also some alternative layout designs. For example, in the tandem layout [324] shown in Fig. 21b, the exhaust air from the first rack row feeds the inlet of second rack row such that a third super-hot aisle (around 50°C) is formed to facilitate the waste heat reuse. This layout obviously requires that IT equipments in the second rack row can endure higher inlet temperatures. Fortunately, state-of-the-art IT systems support inlet temperatures up to 40°C [325].

The rack power density has an indirect impact on the air flow. A low density ($1 - 4$ kW/rack) requires less cold air. As a result, the air flow from perforated tiles may hardly reach the rack top due to the slow air flow speed, and hence the hot air recirculation occurs. On the contrary, the air flow speed has to be enhanced for high power density racks (≥ 10 kW). Here, hot spots may be created at the lower cabinets and more cold air will bypass the rack top [280], [326]. Schmidt and Cruz [327] found that a much lower tile flow rate was required if high-powered racks were sparsely populated among low-powered racks. Azimi *et al.* [328] developed two heuristics and an Integer Linear Programming (ILP) based optimal solutions to the rack placement problem for the cooling power minimization.

The server layout inside racks pronouncedly affects the cooling performance. Racks populated by servers with various cold air demands were investigated by Kumar *et al.* [158] (Fig. 23). The best cooling efficiency was achieved when all servers were uniformly loaded (case A). Clustering active servers (case D) was better than separating them apart

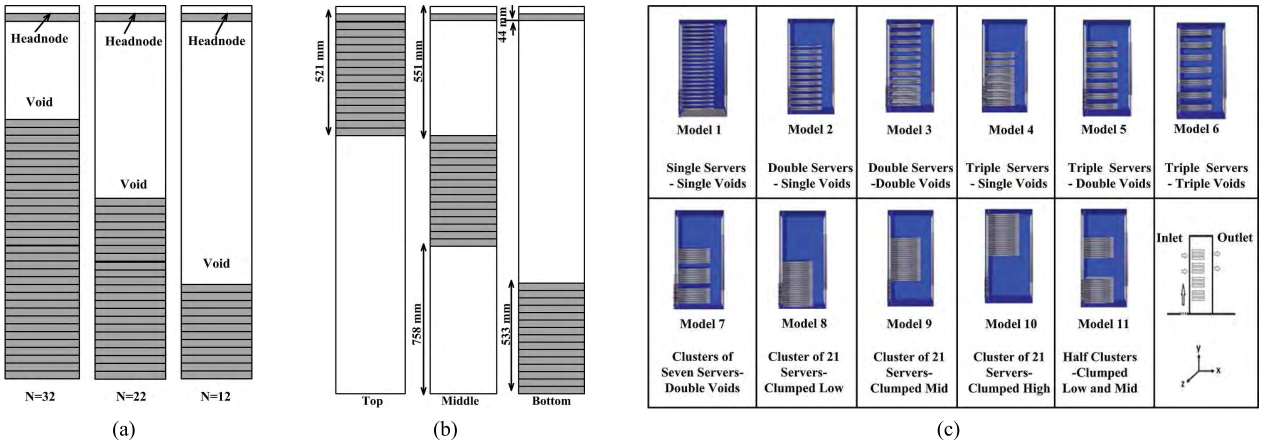


FIGURE 22. Server layouts inside the rack. (a) Rack mounted with different number of server on the bottom [154] (*N* is the number of servers). (b) 3 locations for server groups investigated in [154]. (c) 11 server layout models in [322].

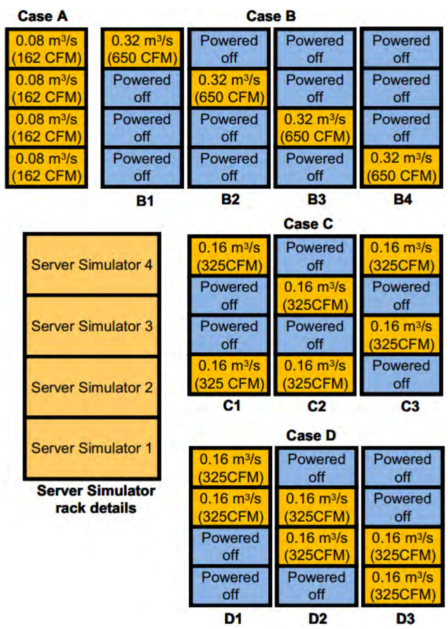


FIGURE 23. Server rack populated by server simulators with various cold air demand [158].

(case C). The same conclusion was also given in [326]. Rambo and Joshi [329] studied the optimal server layout for a sparsely populated rack with 12 1U servers and 3 blade servers. The best practice is to separate 3 high-powered blade servers as far as possible by inserting 1U servers between them. Furihata *et al.* [330] showed that racks replete with servers had better cooling performance since the recirculation in the upper part was reduced. Ghosh *et al.* [154] measured the cooling performance for various server population schemes (Fig. 22a) and revealed that both the exhaust temperature and temperature difference between cold and hot aisles increased as the number of servers grew. They further concluded that among the 3 server cluster layouts in Fig. 22b, placing all servers on the top of rack reduced the server fan speed, CPU temperature, and total energy consumption. However, later studies reached an opposite conclusion. Fakhim *et al.* [322] evaluated the thermal performance of 11 rack layouts (see Fig. 22c) by calculating SHI. It was found that model 3 and 10 had the best and worst rack level performance, respectively. The poor performance of model 10 was caused by placing all racks in the recirculation zone (top section of the rack). Nada *et al.* [125] also proposed to use high-powered IT equipments in the bottom or to increase the server flow rate from the bottom to the top, since the server surface temperature was increasing along with the height even under uniform server power and air flow distributions due to the accumulation of energy absorbed by the cooling air rising from the bottom to exhaust fans. To find the optimal location for servers with different power profiles, Sun *et al.* [331] presented an optimization problem where the objective was to reduce the maximum inlet temperature. They proved that this problem was NP-hard and proposed a greedy heuristic algorithm to solve the problem approximately. Since the server power consumption varies with time due to the workload fluctuation, Demetriou and Khalifa [199], [332] developed a dynamic power/workload allocation strategy within racks to achieve a better cooling efficiency. In their approach, the workload was assigned based on the chassis temperature obtained by built-in sensors.

The type of IT equipments mounted on the rack also matters. Alissa *et al.* [180] found that populating the rack with 2U servers required less air flow compared to 1U servers. Blade enclosures has a large air flow demand which may lead to air starvation in adjacent devices like 1U switches. These switches usually have the weakest air moving systems, making them susceptible to low or even reverse flows in case of negative pressure [252]. In addition, many network devices use nonstandard ventilation schemes [139], [176], which will damage the cooling efficiency if their locations are not well engineered.

Optimal rack level design can be formulated as optimization problems. Due to the complexity of these problems, classical gradient-based methods do not work well.

Genetic Algorithms (GAs) appear to be powerful candidates to solve such problems since they can handle general nonsmooth and nonconvex functions. Li *et al.* [333] proposed a POD-based multi-objective optimization framework, which employed a kriging-guided GA algorithm as the solution method, to obtain the optimal power distribution inside the rack cabinet. Compared to traditional GAs, their algorithm has a lower computational complexity since simulation calls are reduced by a half. GAs can also be extended to larger optimal design problems, e.g., machine room level optimization problems [334].

XII. AIR FLOW DISTRIBUTION

In addition to the standard raised floor supply system, there are a number of air flow distribution options like the side wall supply [176], overhead supply [101], and hard floor system [309], etc. Since airflow distribution systems are often built in the initial data center construction stage and are almost impossible to be altered once this stage is completed, it is essential to evaluate the cooling performance of various options in advance.

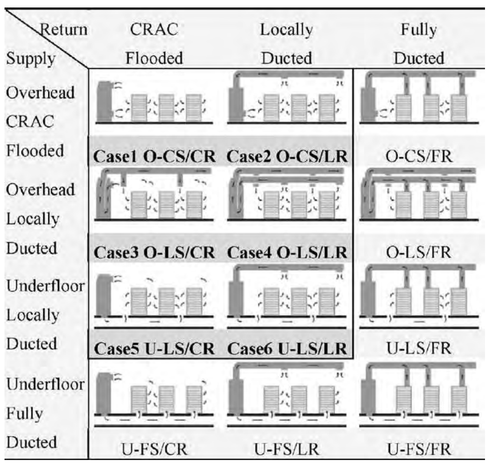


FIGURE 24. 12 air flow distribution layouts [323].

The supply and return paths of air flow can be classified into three categories [335]: flooded, locally ducted, and fully ducted. When incorporated with the location of supply vents (overhead or underfloor), totally 12 air flow distribution systems are generated [323], as shown in Fig. 24. For data centers with a hard floor, the flooded supply and return layout is unbefitting due to a severe air mixing [100]. Cho *et al.* [323] examined all 12 layouts and argued that fully ducted options for both supply and return sides were infeasible in large scale data centers (although later study [94] showed that fully ducted was possible). The remaining 6 layouts were investigated and results indicated that the underfloor locally ducted supply and locally ducted return was the best option if the recirculation in upper part of racks was restrained. In line of this study, they also simulated the performance of these 6 layouts with partially contained cold aisles, where it was found that the air distribution system was less important there [311].

They further added the sidewall ventilation option and evaluated 4 systems [72]: 1) underfloor supply + ceiling return; 2) underfloor supply + sidewall return; 3) overhead supply (ducted) + sidewall return; 4) underfloor supply + ceiling return (ducted). CFD simulation showed that the third scheme outperformed the others in the open aisle machine room, and all 4 schemes performed well under the cold aisle containment system. Generally speaking, the ducted supply and return solution improves the air distribution efficiency and can be used for high power density racks. However, some major concerns should be considered [32], e.g., inconvenience for CRAHs and racks relocation, more carefully engineered air flow rate for CRAHs and racks, and potential catastrophic consequences in case of CRAH failures, etc.

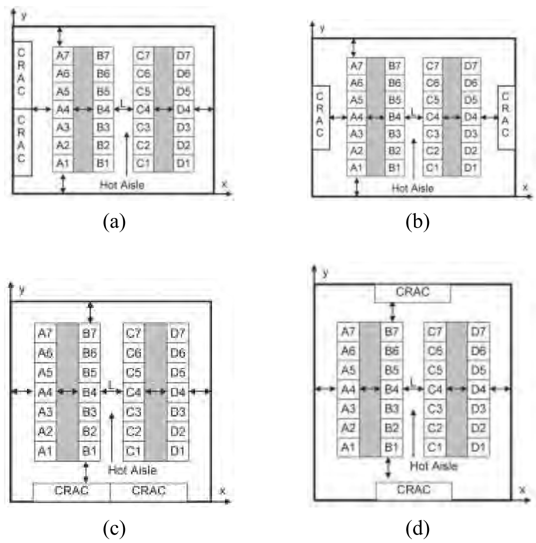


FIGURE 25. CRAC locations and orientations in [337]. (a) 1N; (b) 2N; (c) 1R; (d) 2R.

Field study also showed that the air flow distribution was a strong function of the machine room layout [106], such as CRAH location, rack row orientation, perforated tile position, and room size, etc. Improper CRAH placement and hot/cold aisle arrangement will lead to a heat load imbalance and poor cooling performance [336]. The current practices often use machine room layouts shown in Fig. 25. Rambo and Joshi [337] investigated 4 air flow distribution systems under these layouts: 1) raised floor supply with standard flooding overhead return (RFP-STD); 2) raised floor supply with ducted overhead return (RFP-OH); 3) overhead ducted supply with flooding room return (OH-RR); 4) overhead supply and return (OH-OH). They concluded that OH-RR-2N and OH-RR-2R appeared to be the best options. Nada and Said [282] confirmed that installing CRAHs to positions such that the air flow was blowing perpendicular to rack rows (2N layout as shown in Fig. 25b) not only improved the tile flow uniformity and cooling performance in terms of RTI, SHI, and RHI, but reduced the hot air recirculation at rack row ends and the cold air bypass in the middle, compared to the parallel location (2R layout as shown in Fig. 25d).

Bhopte *et al.* [96] studied the 2N layout with various CRAC-cold aisle distances. It was found that both tile flow and rack inlet temperature distributions were improved as the cold aisle moved away from CRAH. The optimal location of CRAH can be determined by CFD simulations [279], [338].

XIII. CONCLUSION AND OUTLOOK

The huge energy consumption of cooling systems is always a conundrum for data center operators. Witnessing the fact that air-cooling systems are adopted by majority of data centers, in this paper we try to conduct a comprehensive survey of current air flow management techniques for improving energy and cooling efficiency. Since thermal measurement data are inputs to various management strategies and benchmarks reflecting the cooling performance, we first reviewed existing measurement methodologies and instrumentations. Then, we discussed air flow management techniques for various cooling components along the air flow cycle.

Although the data center air flow management has been investigated for decades, it can be seen that a vast number of literatures are still emerging these years, indicating that our understanding of air cooling behaviours is far from adequate. New ideas such as novel infrastructure designs and cooling resource control methods are being proposed. While the infrastructures are unlikely to be changed once the data center has been constructed due to a high retrofit cost, intelligent cooling control techniques show a strong adaptability. Specifically, machine learning technologies such as Deep Reinforcement Learning (DRL), which have been successfully applied to IT and power resource management problems [339]–[341], begin to attract attentions from cooling engineers [342], [343]. By leveraging the recent breakthrough in reinforcement learning aided by deep neural networks [344], the controller can naturally deal with the notoriously complex air flow and heat transfer models. In addition, the controller is also adaptive to system changes, e.g., workload variation and outdoor temperature fluctuation, etc., because of its learning capability. With the help of such techniques, Google announced that the cooling bill was reduced by 40% [345]. We anticipate a more widely application of machine learning techniques in the data center air flow management in the future.

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