Optimization of row-and-rack level airflow management in data centers

Xiaolei Yuan
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Abstract

Building energy use is responsible for around 40% of total primary energy use, and about 36% of total global carbon emission, causing significant environmental impacts (e.g., climate change). Data centers (DCs) belong to the special building sector accounting for a very small proportion of the building volume but have a very high energy use. Electricity used for DCs accounts for about 2% of the global total electricity use, whose figure is predicted to reach 5% by 2024. Thus, energy conservation methods should be applied in DCs under the premise of safe operation. This thesis adopts some research methods (e.g., literature review, experimental tests, numerical studies, model validation, and formal analysis). A literature review can engender new directions and innovations and is the ground for future research and thesis. Thus, this thesis firstly conducts a literature review about the cooling system and optimization in DCs. Considering the current huge market share of air-side cooling and its high cooling reli-ability & applicability and normalized model matching design compared with liquid-side cooling, this thesis proposes row-and-rack level airflow management is worth further studying. Then, 4 different row-and-rack level methods are proposed and studied to optimize the thermal environment through experimental & numerical studies and model validation. The nu-merical models are validated by the experimental results, and the results of both experimental and numerical studies are presented and analyzed as well as energy saving potentials. Finally, the literature review is conducted again based on phase change cooling (PCC) in DCs, and PCC is recommended for future study. The results show that the numerical results are in good agreement with the experimental results, and the reliability and feasibility of numerical models are validated. In addition, all the row-and-rack level airflow management methods can improve the thermal environment in DCs to varying degrees. The rack hotspot temperatures can be decreased by 1.5-2.5 K with different methods, which reduces the risk of servers’ down-time and extends their lifespan. Furthermore, some of these methods can achieve considerable energy savings (98-146 kWh electricity use per day) under the premise of safe operation. The proposed methods (server terminal baffles and tilted server placement) can be used directly in the operation phase of DCs, and only need simple modifications (e.g., adding baffles and adjusting server angles) for safe operation and thermal environment improvement, while the rest (in-rack UFAD and step-like server placement) are applicable in the DC design phase, which also just needs simple modifications (e.g., changing the positions of perforated floor and server rack, and rack length).

Keywords Data centers, row-and-rack level airflow management, tiled servers, server terminal baffle, in-rack UFAD, step-like servers, phase change cooling, energy saving
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In addition to data center related research, I have also been involved in HUKATON & FINEST project on building energy management (e.g., waste heat recoveries and demand response) during Aalto doctoral studies. HUKATON project was funded by European Regional Development Fund (ERDF), City of Helsinki and Helen Ltd. and FINEST Twins project was co-funded by European Union (Horizon 2020 programme, Grant No. 856602) and the Estonian government. I also gratefully thank these financial supporters.

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I have been a qualified driver for many years, while a doctoral degree is somehow like a research driver’s license for me. For the future, I wish I could drive fast but smoothly on the research highway and let’s see how far I can go.

Espoo, Helsinki, 11 April 2022
Xiaolei Yuan (Damon)
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List of Abbreviations and Symbols

**Abbreviations**

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<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
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<tbody>
<tr>
<td>CA</td>
<td>cold aisle</td>
</tr>
<tr>
<td>CAC</td>
<td>cold aisle containment</td>
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<tr>
<td>CFD</td>
<td>computational fluid dynamics</td>
</tr>
<tr>
<td>COP</td>
<td>coefficient of performance</td>
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<tr>
<td>CRAC</td>
<td>computer room air-conditioning</td>
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<tr>
<td>DC</td>
<td>data center</td>
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<tr>
<td>FM</td>
<td>facility management</td>
</tr>
<tr>
<td>HP</td>
<td>heat pipe</td>
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<tr>
<td>HVAC</td>
<td>heating, ventilating and air-conditioning</td>
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<tr>
<td>IR-CA</td>
<td>in-rack cold aisle</td>
</tr>
<tr>
<td>ICT</td>
<td>information and communications technology</td>
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<tr>
<td>MTP</td>
<td>measured temperature point</td>
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<tr>
<td>MVP</td>
<td>measured velocity point</td>
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<tr>
<td>OHA</td>
<td>open hot aisle</td>
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<tr>
<td>PCC</td>
<td>phase change cooling</td>
</tr>
<tr>
<td>PCM</td>
<td>phase change material</td>
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<tr>
<td>RSM</td>
<td>Reynolds stress model</td>
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<tr>
<td>SAT</td>
<td>supply air temperature</td>
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<tr>
<td>STP</td>
<td>simulated temperature point</td>
</tr>
<tr>
<td>SVP</td>
<td>simulated velocity point</td>
</tr>
<tr>
<td>UFAD</td>
<td>under floor air distribution</td>
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**Symbols**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tr>
<td>$A$</td>
<td>area of air vent of each CRAC, m$^2$</td>
</tr>
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</table>
\( \text{COP}_{\text{CRAC}} \) COP of the CRAC units

\( c_p \) specific heat capacity of air, kJ/(kg·K)

\( \vec{g} \) gravitational acceleration vector

\( k_{\text{eff}} \) effective thermal conductivity, W/(m·K)

\( \rho \) air density, kg/m\(^3\)

\( p \) static pressure, Pa

\( Q_1 \) daily cooling energy saving, kWh

\( Q_2 \) daily total electricity saving for the CRAC units, kWh

\( S \) volumetric heat sources

\( \text{SAT}_{c1} \) SAT of original case, °C

\( \text{SAT}_{cx} \) SAT of the selected case, °C

\( T \) static temperature, °C

\( \Delta t \) SAT difference between original case and optimal case, K

\( \bar{u} \) average velocity vector

\( v_{\text{eff}} \) effective fluid viscosity

\( v \) air velocity through the air outlet of the CRAC units, m/s

\( V \) volume of air from the CRAC units, m\(^3\)

\( x \) number of the optimum case
List of Publications

This doctoral thesis consists of a summary and of the following publications which are referred to in the text by their Arabic numerals.


Author’s Contribution

Publication 1: Design and validation of an airflow management system in data center with tilted server placement

Xiaolei Yuan (Damon) is the principal author of the paper. The author put forward the concept of tilted server placement by himself, and carried out the measurements in accordance with the guidelines, advice, coaching and supervision of Prof. Jinxiang Liu from Nanjing Tech University. The author was responsible for the conceptualization, methodology, software, measurements, data curation, formal analysis, literature sortation, writing original draft, figure and table drawing, and discussion & summary. Prof. Risto Kosonen from Aalto University, Prof. Yu Wang, Prof. Hao Cai and Xiejie Xu from Nanjing Tech University contributed to the scientific discussion and comments.

Publication 2: Experimental and numerical investigation of an airflow management system in data center with lower-side terminal baffles for servers

Xiaolei Yuan (Damon) is the principal author of the paper. The author put forward the concept of lower-side terminal baffles for servers by himself, and carried out the measurements in accordance with guidelines, advice, coaching and the supervision of Prof. Risto Kosonen from Aalto University and Prof. Jinxiang Liu from Nanjing Tech University. The author was responsible for the conceptualization, methodology, software, measurements, data curation, formal analysis, literature sortation, writing original draft, figure and table drawing, and discussion & summary. Prof. Yu Wang, Xuetao Zhou and Xiejie Xu from Nanjing Tech University contributed to the comments & editing.

Publication 3: Improvement in airflow and temperature distribution with an in-rack UFAD system at a high-density data center

Xiaolei Yuan (Damon) is the principal author of the paper. The author put forward the concept of in-rack UFAD system by himself, and carried out the measurements in accordance with the guidelines, advice, coaching and supervision of Prof. Risto Kosonen from Aalto University and Prof. Jinxiang Liu from Nanjing Tech University. The author was responsible for the conceptualization, methodology, software, measurements, data curation, formal analysis, literature sortation, writing original draft, figure and table drawing, and discussion & summary. Prof. Yiqun Pan from Tongji University, Yang Gao and Xinjie Xu from Nanjing Tech University contributed to the comments and editing.
Publication 4: Airflow management and energy saving potentials at a high-density data center with stepped-like server placement

Xiaolei Yuan (Damon) is the principal author of the paper. The author put forward the concept of stepped-like server placement by himself, and carried out the measurements in accordance with the guidelines, advice, coaching and supervision of Prof. Risto Kosonen from Aalto University and Prof. Jinxiang Liu from Nanjing Tech University. The author was responsible for the conceptualization, methodology, software, measurements, data curation, formal analysis, literature sortation, writing original draft, figure and table drawing, and discussion & summary. Prof. Yiqun Pan, Yumin Liang from Tongji University, and Prof. Yu Wang from Nanjing Tech University contributed to the scientific discussion, comments and editing.

Publication 5: Phase change cooling in data centers: A review

Xiaolei Yuan (Damon) is the principal author of the paper. The author was responsible for the conceptualization, literature sortation, writing the original draft, and figure and table drawing. Xuetao Zhou contributed to the paper equally with the author, responsible for the literature sortation, writing the original draft, figure and table drawing and discussion & summary. The author finished this paper in accordance with the guidelines, advice, coaching and supervision of Prof. Risto Kosonen from Aalto University and Prof. Yiqun Pan from Tongji University. Prof. Hao Cai, Prof. Yu Wang, and Yang Gao from Nanjing Tech University contributed to the scientific discussion, comments and editing.
1. Introduction

1.1 Background

Indoor thermal environment and energy use within buildings have become increasingly significant and hot topics in building sectors, while the corresponding research on the thermal environment optimization and energy efficiency have been widely studied (Jin et al., 2019; Yuan et al., 2021). Actually, there are inherent mutual relationships between the thermal environment and energy use in buildings. Thermal environment optimization can affect the building energy use, while building energy use can also react to the thermal environment changes. Heating, ventilating and air-conditioning (HVAC) system is an important part of contemporary building sectors, which directly and significantly affects both indoor thermal environment and energy use (Dvorak et al., 2020). Building sectors are responsible for 20-40% of the total energy consumption in different countries, of which HVAC systems account for 40-60% (Alimohammadisagvand et al., 2016; Yuan et al., 2021). Thus, the operation efficiency of HVAC system could be decisive to both indoor thermal environment and energy use in buildings.

Data centers (DCs) are centralized repositories housing computer servers and associated data storage and telecommunication systems for the data storage as well as the data processing and transmitting, while they have the characteristics of large scale and working 24/24 hrs., 365 days/year, causing potentially very high energy consumption (Durand-Estebe et al., 2014). DCs belong to the special building sector, whose specific heat load could be more than 100 times larger than that of a typical similarly sized office building, while the energy use of DCs is usually 30-50 times that of office buildings with similar-size (Wang et al., 2016). Although DCs are just a small group of building sector, they are also significant energy consumers, accounting for approximate 2% of worldwide total electricity use, while this figure is predicted to reach 5% by 2024 (Cooling working group of data center of China Refrigeration Society, 2020). Information and communications technology (ICT) sector was responsible for more than 2% of the global carbon emissions, including 14% from DCs (Kubler et al., 2019; Whitehead et al., 2014). Thus, in order to reduce the energy consumption & cooling cost in DCs and mitigate related environmental problems (e.g., global warming and greenhouse effects), energy conservation strategies should be proposed and applied in DCs.
1.2 Data center energy breakdown

DCs are building synthesis equipped with many different instruments, mainly divided into four parts: IT equipment, cooling system, power supply equipment and safety & control systems. Figure 1 illustrates the composition of typical DCs (Joshi & Kumar, 2012; Patankar, 2010). IT equipment and cooling system are two main parts occupying more than 80% of the total energy consumption in DCs (Khalaj & Halgamuge, 2017). A huge amount of data processing and circulation results in tremendous electricity consumption, and a large number of heats are converted from the electricity, which needs to be dissipated immediately. Otherwise, the temperature of computer servers may rise dramatically threatening the operation safety of computer servers and causing malfunction (Ding et al., 2016; Siriwardana et al., 2012). Figure 2 shows the breakdown of the energy use within typical DCs, while IT equipment and the cooling requirements account for 44% and 40% of the total energy use, respectively (Almoli et al., 2012; H. Zhang et al., 2014).

Figure 1. Main equipment in a typical DC.

Figure 2. The breakdown of energy use in a typical DC.
Thus, energy conservation in DCs can be achieved by addressing IT load inefficiencies or improving cooling efficiency. It is reported that enhanced semiconductor technologies and server virtualisation could efficiently decrease the energy use of IT equipment (Cushing & Dohety, 2009). While, for the HVAC side, the focus should be on improving cooling efficiency & operation safety, and achieving energy saving through feasible methods (e.g., airflow management, liquid cooling and free cooling technologies).

1.3 Air-side cooling Technology

The air-side cooling system is generally preferred due to its high reliability of cooling, strong adaptability, and lower initial investment and maintenance costs. Among all the air-side cooling systems, computer room air-conditioning units (CRACs) are the most widely and commonly used one applied in DCs, while their systems are generally composed of chillers, water pumps, fans and cooling towers (Ni & Bai, 2017). The CRACs have the benefits of high applicability and reliable cooling conditions, but the cooling power has limits, especially in high-density DCs. Thus, a series of measures for optimizing the airflow organization and cooling efficiency based on the CRACs come into being, and they can be called airflow management in DCs.

Airflow management is devoted to changing the DC’s configuration and CRACs’ air supply state to improve the cooling efficiency and the thermal environment (Cho et al., 2014). In addition, it is aimed at consuming the minimum energy to keep intake conditions (IT equipment) within the recommended ranges (Lu et al., 2011), which is a promising and maneuverable energy-saving approach. High performing airflow distribution can not only decrease energy consumption in DCs, but it can also avoid the appearance of overheating servers bringing catastrophic data losses, and extend the service life of the servers (Almoli et al., 2012). Cho et al. (2014) evaluated an air distribution system’s airflow performance for cooling energy conservation in the high-density DCs, and they defined five criteria: supply air temperature (SAT), ceiling height, raised floor height, air distribution type (supply/return air location), and zoning regarding aisle containment configuration. These criteria can largely influence the DCs’ general thermal airflow and cooling efficiency, and can be considered as the room-level airflow management (Cho et al., 2014; Chu & Wang, 2019).

1.3.1 Room-level

The criteria defined by Cho et al. (2014) can be considered as the design factors for cooling system conditions and the available approaches to optimize airflow organisation from room-level, which have been comprehensively studied by many researchers. Ham et al. (2015) studied the optimum SAT ranges of various air-side economisers in a modular DC. The results demonstrated that cooling energy consumption was at a minimum when the SAT of the CRACs was between 18°C and 23°C. The lower the SAT conditions, the greater the cooling energy consumption. Oró et al. (2016) performed numerical and experimental
analyses of airflow management in a DC in Spain, and concluded that the cooling energy consumption could be reduced significantly by reducing the air flow rate and increasing the SAT at the same time.

Sorell et al. (2006) analysed the impacts of ceiling height on the air distribution in a DC. The results showed that increasing the space height improved the overall performance of the air distribution system, but the space height was no more than it needed to be. However, under-floor air distribution (UFAD) systems with ceiling plenum returns need to be considered since nothing indicated that this air distribution mode could improve the air distribution systems’ thermal performance. Subsequently, they used computational fluid dynamics (CFD) modelling to compare the underfloor and overhead air supply systems in a DC (Sorell et al., 2005). They found that the UFAD performed better when the airflow for the under-flow system was beyond 15% of the total racks’ airflow, while the overhead system was considered acceptable in case of a lesser airflow. Schmidt & Lyengar (2007) evaluated the suitability of a raised floor air supply and an overhead air supply design in very high-density DCs by CFD modelling. Conclusions were drawn that the UFAD design was appropriate for the higher chilled-air supply cases to produce cooler rack intake air, while the overhead air supply design was suitable for the lower chilled-air supply cases to yield a cooled rack intake temperature. Cho et al. (2009) simulated a high-density DC with six different air distribution modes to evaluate its thermal performance. The results demonstrated that the UFAD design performed better in optimizing the airflow and thermal performance in DCs.

S.A. Nada and his research team have made many efforts to study the thermal and energy impacts concerning aisle containment configuration. In 2015, they (Nada et al., 2016b) built CFD models to investigate the thermal performance of different CRAC configurations and aisle separation. One of the results showed that the thermal performance of DC was improved by the application of cold aisle containment (CACs). Then, they experimentally investigated the thermal management methods in DCs for different arrangements of CACs (Nada & Elfeky, 2016). The results indicated that the application of semi-enclosed cold aisles (CAS) enhanced the DC’s thermal performance, while the use of fully enclosed CASs exhibited an important thermal performance improvement with the rise in power density. In 2016, they performed a numerical and parametric study on the improvement of energy and thermal management in DCs (Nada et al., 2016a). They further found that the application of appropriate CACs enhanced the performance of cooling in DCs, especially in those with high power density. In addition, Tatchell-Evans et al. (2017) made both experimental and theoretical analysis of the extent of bypass air in DCs with the aisle containment and its impact on energy conservation. The results showed that about 20% of the supplied air may bypass servers at a typical CA, while up to 16% of energy would be saved by the application of the containment system.

To sum up, there were many research on the room-level airflow management, which has tended to be saturated and has begun to come up against a bottleneck (Yuan et al., 2018). Thus, more efficient airflow optimization methods should
be proposed and studied. With the development of the DC airflow management, smaller-scale airflow distribution optimization methods (e.g., row-and-rack level airflow management) get more attention, and become a new research focus.

1.3.2 Row-and-rack level

Although the room-level airflow management can improve the overall thermal distribution and reduce the overall energy use, it cannot solve the problem of thermal environment deterioration of a single rack or the local part. As the rack with highest server power will form the local hotspot, and cause too high server operating temperature, which affects the operation efficiency and the service life of the servers. Thus, the row-and-rack level airflow management gets more attention as it belongs to the local airflow optimization, which is closer to the heat source, and can further optimize the thermal environment and cooling efficiency on the basis of the computer room level.

In terms of the row-based method, many researchers have studied a lot mainly on the in-row air-conditioners. Cho & Woo, (2020) developed and experimentally studied a row-based cooling system to improve thermal performance in a DC. They concluded that the row-based cooling could improve the thermal environment within DC because it was closer to the hotspots and could avoid the heat losses in the delivery process from the room-based cooling system to the rack row. In addition, Cho & Kim (2021) evaluated the thermal performance, and compared the cooling provisioning of different configurations of row-based cooling systems to find the optimum placement of the in-row CRAC units. Nada & Abbas, (2021) investigated the possible airflow management solutions for the thermal management problems in DCs with the in-row cooling architectures. They found that the in-row CRACs could improve the thermal environment and energy utilization coefficient of the terminal racks by 45%. In addition, Wang et al., (2015) proposed a novel draw-type rack, while the layout of these racks was movable and flexible to reduce the bypass and recirculation air. They found the drawer-type rack could decrease the rack maximum inlet temperature by up to 13.3 °C compared with the traditional rack layout.

Compared with the row-level airflow management, the rack-level airflow optimization is closer to the heat source with better thermal optimization effect. Huang et al. (2017) established a numerical model to compare and analyze the airflow distribution in DCs with three different airflow patterns in 2017. The results showed that compared with the under-floor and row-level air supply patterns, the rack-level airflow pattern performed the best with the highest cooling efficiency. Almoli et al. (2012) attached the liquid loop heat exchangers and additional fans to the rack backdoor, and found that it could improve the thermal distribution. The results showed that the use of heat exchangers with additional fans at the racks’ rear door could improve the thermal environment and reduce the cooling energy use. In addition, Phan et al. (2019) introduced and applied a new composite performance index to evaluate the turbulence and tile models at the server-rack level in DC. Chen et al. (2014) applied a dynamic fan speed control algorithm to achieve the server-level thermal optimization,
while the control algorithm reacts to the CPU utilization by corresponding setting adjustment once the CPU temperature upper bound requirement was met, and finally improved the thermal environment and achieved the energy saving within DC.

As research on the room-level airflow management methods tend to be saturated, the smaller-scale local airflow management methods (row-and-rack level, or even server level) deserve more research. At present, there are some publications on airflow management at the row-and-rack level, but the related research deserves further development, and nowadays it is also a hotspot method for optimizing the cooling efficiency in DCs.

1.4 Liquid-side cooling technology

Apart from the air-side cooling, the liquid-side cooling is also applicable in DCs, which has the characteristics of high cooling capacity and efficiency with low operating costs (Khalaj & Halgamuge, 2017). The liquid-side cooling method is divided into direct and indirect liquid cooling. In the direct liquid cooling system, the liquid coolants directly contact the electronic device, otherwise it is the indirect cooling.

The direct liquid cooling can be achieved through the two-phase passive immersion cooling and spray cooling (Bar-Cohen et al., 2006; Kim, 2007). Some researchers have proved the applicability of two-phase passive immersion cooling in DCs (Qiu et al., 2015; Shojaeian & Koşar, 2015), while the applicability of spray cooling has also been proved (Johnston et al., 2008; Silk et al., 2008). Although the direct contact of the cooling medium, with the surface of the heat source and the phase transition phenomenon, can reduce the thermal resistance, the thermophysical properties of the dielectric fluid are significantly lower than those of water. Thus, the direct cooling solutions can hardly be considered as a heat transfer technology compared to the traditional indirect liquid cooling solutions (Kheirabad & Groulx, 2016).

The indirect liquid cooling methods can be divided into single phase cooling, two-phase cooling and heat pipe (HP) cooling (Garimella et al., 2012; Siedel et al., 2015; Tuma, 2008). The indirect liquid cooling in DCs has been investigated and implemented in IBM, CoolIT Systems, and Asetek (Asetek, 2017; Coolit Systems, 2022; Zimmermann et al., 2012).

Compared with the air-side cooling, the liquid-side cooling can greatly reduce the cooling power consumption in terms of setting environment and device cooling, and can achieve the high-power utilization efficiency. However, at present, the vast majority of DCs still maintain the traditional air-side cooling method. The advantages of the air-cooled systems lie in their cooling reliability and wide applicability. Especially in recent years, airflow management for DCs from the room-level to the server-level has continuously improved the efficiency of air-side cooling and heat dissipation in DCs. In addition, the matching design of the server substrate CPU and radiator has formed a normalized model; and the industrial characteristics of the large-scale device design and DC operation also determine the long cycle of introducing alternative technologies (Cooling
working group of data center of China Refrigeration Society, 2020; Neudorfer, 2017; Wei, 2019). Therefore, the current typical air-cooling mode in DCs is still the mainstream, and deserves further and in-depth study.

1.5 Objective

The main objectives of the thesis are:

- To compare and analyze the current cooling technologies in DCs, and highlight the benefits and researchability of the row-and-rack level airflow organization.
- To propose innovative and feasible row-level and rack-level airflow management methods.
- To obtain the optimized airflow distribution method to reduce the hotspot temperatures in the racks with experimental and simulation studies as well as energy saving potentials.
- To recommend another hotspot phase change cooling in DCs for future study.

This thesis is combined with five publications. Publications 1 & 3-4 belong to the rack-level airflow management method, while Publication 2 belongs to the row-level method. In addition, in Publication 5, phase change cooling in DCs is reviewed and summarized for future research recommendation.

1.6 Novelty and content of the study

The novelty of this thesis is to propose and apply four innovative row-or-rack level airflow management methods to improve the thermal environment and cooling efficiency in DCs. One of them (In-rack UFAD system) belongs to the row-level optimization method, while the rest are the rack-level methods (tilted server, lower-side server terminal baffles, and step-like server placement). Compared with the room-level method, the row-and-rack level airflow management is closer to the heat sources, and can enhance the local cooling with airflow and optimize the local thermal environment.

The content of this thesis is to investigate the thermal environment (e.g., air temperature and velocity distributions) and cooling efficiency in the studied DC under the application of four proposed row-or-rack level airflow management technologies through experimental tests, simulation research and model validations. Then, the optimum cases for each airflow management method can be obtained as well as the optimal thermal environment distribution and cooling efficiency improvement or energy saving. The current high-efficient cooling methods are reviewed and summarized in the recommendation part, while the phase change cooling (PCC) methods are recommended for further study in the future. Table 1 shows the schematic framework of this thesis.

In Publication 1, an innovative airflow distribution method applying tilted servers was introduced to improve the airflow management and cooling efficiency in DCs. This study mainly focused on simulation research, and validated the original DC simulation model by the experimental data in the
actual studied DC. Based on the validated model, the model was adjusted to study the optimization effect of tilted servers on airflow organization and cooling efficiency.

In Publication 2, the lower-side server terminal baffles were introduced and applied to optimize the temperature and airflow distributions in DCs. In this paper, both experimental tests and simulation studies were carried out to improve the local cooling of the racks with the terminal baffles. The simulation model was validated by the on-site experiments, while the numerical results were in good agreement with the experimental results.

In Publication 3, this paper introduced and analyzed a new concept where an UFAD system with cold aisle containment (CAC) was replaced by a new in-rack UFAD system called an in-rack cold aisle (IR-CA). This study mainly focused on simulation research, and validated the original DC simulation model by the experimental data in the actual studied DC. Based on the validated model, the model was adjusted to study the effect of IR-CA on thermal environment and cooling efficiency.

In Publication 4, an innovative concept of step-like server placement was proposed in DCs, replacing the traditional servers placed on the same vertical plane with step-like server placement. This study mainly focused on simulation research, while the feasibility and reliability of the DC simulation model was validated by the on-site experimental results. Based on the validated model, the DC model was simplified, and then adjusted to study the effect of step-like server placement on the thermal environment and cooling efficiency.

Publication 5 reviewed and summarized current available high-performance and effective cooling technologies in DCs, and mainly focused on the phase change cooling (PCC) application for further study recommendation.

Table 1. The schematic framework of this thesis.
1.7 Research questions

The overall aim of the thesis is to ensure the safe operation and improve the cooling efficiency in DCs. This topic is approached by means of three sub-questions. The three research questions of this thesis are presented below and summarized in Figure 3.

**RQ1: What are the current widely applied and hotspot technologies for cooling-efficiency improvement in DCs?**

This research question is addressed in all Publications 1-5, and is also addressed in the Introduction part of this thesis.

**RQ2: How the optimized row-and-rack level solutions can enhance the thermal environment and improve the cooling efficiency in DCs?**

This research question is addressed in Publications 1-4, which study the thermal environment (e.g., temperature and velocity distributions) and cooling efficiency in the studied DC under the application of four proposed row-or-rack level airflow management technologies through experimental tests, simulation research and model validations.

**RQ3: What are the recommended future research focus for high-efficient cooling in DCs?**

This research question is addressed by Publication 5, which summarizes available high-efficient cooling solutions in DCs, and recommends phase change cooling technologies as future and further study focus.

<table>
<thead>
<tr>
<th>RQ1</th>
<th>RQ2</th>
<th>RQ3</th>
</tr>
</thead>
<tbody>
<tr>
<td>What are the current widely applied and hotspot technologies for cooling-efficiency improvement in DCs?</td>
<td>How the optimized row-and-rack level solutions can enhance the thermal environment and improve the cooling efficiency in DCs?</td>
<td>What are the recommended future research focus for high-efficient cooling in DCs?</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Row-level AM</th>
<th>Publication 3</th>
<th>Publication 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rack-level AM</td>
<td>Publication 1, 2 &amp; 4</td>
<td>Publication 1, 2 &amp; 4</td>
</tr>
<tr>
<td>PCC</td>
<td>Publication 5</td>
<td>Publication 5</td>
</tr>
</tbody>
</table>

Notations: AM= air distribution; PCC= phase change cooling; DC= data center

Figure 3. Research questions and scope of publications.

1.8 Structure of the thesis

The thesis is in comprises of a summary and the appended research articles, which all discuss the topic of airflow management and energy efficiency
improvement in DCs. The introductory section depicted the background of the research and presented the research aim. The utilized research methods are introduced in the following Section 2. Section 3 presents the results of this study, and followed by a discussion part in Section 4. Finally, Section 5 presents the conclusions, limitations and recommendations for further research.
2. Methodology

2.1 Studied data center description

The studied data center is located on a university campus in Nanjing, China, which is comprised of three separate data rooms, namely basic, core and super-computing data rooms, while the core data room is selected as the research object and is specifically referred to as ‘data center (DC)’ in the following part of this thesis. Publications 1-4 share the same DC, which is a typical UFAD DC with a cold aisle containment (CAC) and two open hot aisles (OHAs), while it serves approximately 30 000 people on this campus by supplying network creation, information processing and communication support working 24/24 hrs., 365 days/year.

Figure 4 shows the studied DC and its layout and plan, while Figure 5 shows the air flow within the studied DC. The cool air generated by the CRAC units supplies the plenum chamber and flows into the CAC through the perforated tiles afterwards. The cool air within the CAC is drawn into the rack front doors and is exchanged with the hot air at the terminal of servers, and then the mixed air enters the OHAs through the rack rear doors. Finally, the mixed air in the OHAs returns directly to the CRACs. The heat loss of the plenum is assumed to be insignificant, and can be neglected.

Figure 4. The studied DC (left) and its layout & plan (right).
Methodology

Figure 5. The air flow within the studied DC.

Table 2. Parameters setting and involved equipment in the studied DC.

<table>
<thead>
<tr>
<th>Items</th>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>DC room</strong></td>
<td>Dimensions of DC</td>
<td>11 m (L) × 8 m (W) × 4 m (H)</td>
</tr>
<tr>
<td></td>
<td>Plenum height</td>
<td>0.45 m</td>
</tr>
<tr>
<td></td>
<td>Height above the plenum</td>
<td>3.55 m</td>
</tr>
<tr>
<td></td>
<td>Air supply mode</td>
<td>UFAD &amp; direct air return</td>
</tr>
<tr>
<td></td>
<td>Aisle configuration mode</td>
<td>1 CAC &amp; 2 OHA</td>
</tr>
<tr>
<td></td>
<td>Dimension of CACs</td>
<td>5.4 m × 1.8 m × 2.2 m (H)</td>
</tr>
<tr>
<td></td>
<td>Number of perforated tiles</td>
<td>Total of 27 in three rows</td>
</tr>
<tr>
<td></td>
<td>Dimension of perforated tiles</td>
<td>0.6 m × 0.6 m</td>
</tr>
<tr>
<td></td>
<td>Porosity of perforated tiles</td>
<td>45%</td>
</tr>
<tr>
<td><strong>CRACs</strong></td>
<td>Number of CRACs</td>
<td>2</td>
</tr>
<tr>
<td></td>
<td>Dimension of CRACs</td>
<td>1.8 m × 0.8 m × 2.25 m (H)</td>
</tr>
<tr>
<td></td>
<td>Distance between CAC and CRACs</td>
<td>1.6 m</td>
</tr>
<tr>
<td></td>
<td>Height of CRACs above the plenum</td>
<td>1.8 m</td>
</tr>
<tr>
<td></td>
<td>Height of CRACs under the plenum</td>
<td>0.45 m</td>
</tr>
<tr>
<td></td>
<td>Flow air velocity from each CRAC</td>
<td>5.33 m/s</td>
</tr>
<tr>
<td></td>
<td>Area of each air vent of CRACs</td>
<td>0.45 m²</td>
</tr>
<tr>
<td></td>
<td>SAT</td>
<td>22 °C</td>
</tr>
<tr>
<td></td>
<td>Operating SAT</td>
<td>22.2 °C &amp; 22.1 °C</td>
</tr>
<tr>
<td></td>
<td>Total power per CRAC</td>
<td>43.1 kW</td>
</tr>
<tr>
<td></td>
<td>Supply air cooling capacity</td>
<td>65.8 kW</td>
</tr>
<tr>
<td></td>
<td>COP</td>
<td>1.53</td>
</tr>
<tr>
<td></td>
<td>Operating time</td>
<td>24/24 hrs, 365 days/year</td>
</tr>
<tr>
<td><strong>Racks</strong></td>
<td>Number of racks</td>
<td>Altogether 18 in 2 rows (Rack A &amp; Rack B)</td>
</tr>
<tr>
<td></td>
<td>Dimensions of each rack</td>
<td>1.2 m (L) × 0.6 m (W) × 2.2 m (H)</td>
</tr>
<tr>
<td></td>
<td>Front door porosity</td>
<td>65%</td>
</tr>
<tr>
<td></td>
<td>Rear door porosity</td>
<td>65%</td>
</tr>
<tr>
<td></td>
<td>Rated power of racks</td>
<td>Varied between 0 kW and 16.98 kW</td>
</tr>
<tr>
<td></td>
<td>Dimensions of servers</td>
<td>0.8 m (L) × 0.46 m (W) × 0.09 m (H)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>0.7 m (L) × 0.46 m (W) × 0.09 m (H)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>0.8 m (L) × 0.46 m (W) × 0.045 m (H)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>0.8 m (L) × 0.46 m (W) × 0.18 m (H)</td>
</tr>
<tr>
<td></td>
<td>Rated power of serves</td>
<td>300 W × 2, 495 W × 2</td>
</tr>
<tr>
<td></td>
<td></td>
<td>600 W × 2, 750 W × 2</td>
</tr>
<tr>
<td></td>
<td></td>
<td>900 W × 2, 2000 W × 2</td>
</tr>
</tbody>
</table>
Table 2 shows and summarizes the parameters setting and involved equipment in the studied DC. There are two CRACs, and 2 rows (namely racks A and B) of altogether 18 racks serve in this DC. The conditions of each rack are different in terms of the server number, rated power and placement. In addition, the rated power of each sever varies from 0.6 kW to 4 kW, while the rated power for one rack varied from no power to a maximum of 16.98 kW. The dimensions of the servers are non-uniform, and there are four kinds of physical sizes for the servers, while the servers sized of 0.09 m × 0.46 m × 0.80 m (Height × Width × Length) have the largest proportion. The air supply temperatures to the racks in all tiles are assumed to be the same.

2.2 On-site experimental tests

2.2.1 On-site experimental conditions

The on-site experiments were carried out twice in 2017 and 2018 separately, which were applicable to Publication 1 and Publications 2-4, respectively.

Experiment in 2017 for Publication 1
The first on-site experiment was carried out in July and August 2017 in the studied DC, while its objective was to obtain the outlet air temperature distribution of the targeted racks. This experiment served the Publication 1. According to the server running information over the whole year operation condition from the facility management (FM) department, Rack A was selected for the study as the no-load rate of Rack A was lower than that of Rack B, and its number and power of servers were higher than Rack B. Figure 6 shows the heat load of each rack in the Rack A, and racks A1, 4 and 9 are not in operating condition. In the experiments, Rack A2-A8 were selected to be the measured objectives.

Figure 6. Heat load of each rack in Rack A.
To improve the reliability and obtain more accurate experimental results, there were carried out altogether 9 experiments between 15th July and 17th July, while each day was divided into three groups of the morning, afternoon and evening experiments. Each measurement case lasted 2 h. Thus, 9 groups of experimental results were obtained during three days. The results were the average value of all measured values (average of three days data, except of experimental data with large deviation).

**Experiment in 2018 for Publication 2**

In 2018, the idea of the lower-side server terminal baffles was put forward, and thus the experimental studies on setting flexible baffles at the server lower terminal were carried out. Compared with the previous experiments in July and August of 2017, the servers’ placement and rated power for each rack have changed. However, we fortunately found that the Rack A7 with the highest server power in the experiment in 2017 was totally the same to Rack B4 in 2018, whose server number, rated power and placement stayed unchanged.

Rack B4 was selected for the detailed measurements of the thermal environment monitoring and further optimization of the thermal behavior analysis. The scheme of the servers and the air movement through the Rack B4 are shown in Figure 7. There were only two kinds of the rated power of server, which are 0.75 kW × 2 and 0.595 kW × 2, while the total power of the Rack B4 is 16.98 kW. Compared with other racks, Rack B4 had the following advantages and appropriateness: (i) the power consumption was the highest, which met the requirement of high-density rack, (ii) the dimensions of the servers were the same, which was 0.09 m × 0.46 m × 0.80 m (Height × Width × Length), and (iii) the placement of the servers was relatively uniform.

![Side of rack B4](image)

**Figure 7.** The scheme of servers and the air movement through the Rack B4.

The experiments were divided into four Scenarios E0-3 as shown in Table 3. In Scenario E0, no baffle was applied, while 0.46 m × 0.04-m, 0.46 m × 0.08-m,
and 0.46 m × 0.12-m baffles were applied in Scenarios E1, E2 and E3, respectively. In the different scenarios, six angles (0°, 15°, 30°, 45°, 60°, 75° and 90°) were tested and altogether 19 cases during two test periods are measured. In Scenarios E1-3 with the baffles, the baffle angle was defined as the angle between the baffle and the underside of the server. In each scenario, the only two variables were the size and angle of the baffles, while all other conditions (e.g., CRACs, racks and servers) remained unchanged. In Cases 2, 8 and 14, the baffle angle was 0°, which represents the baffles were parallel to the servers instead of no baffles. Figure 8 shows the position of the baffles, while there was no baffle in the lowest server. The reason for no baffle in the lowest server was that there was no heat source below the lowest server.

Table 3. The measured sizes and angles of the baffles in different Scenarios.

<table>
<thead>
<tr>
<th>Case</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
<th>9</th>
</tr>
</thead>
<tbody>
<tr>
<td>Scenario E</td>
<td>0</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>2</td>
<td>2</td>
<td></td>
</tr>
<tr>
<td>Baffles width/cm</td>
<td>0</td>
<td>4</td>
<td>4</td>
<td>4</td>
<td>4</td>
<td>4</td>
<td>4</td>
<td>8</td>
<td>8</td>
</tr>
<tr>
<td>Angle/°</td>
<td>no</td>
<td>0</td>
<td>15</td>
<td>30</td>
<td>45</td>
<td>60</td>
<td>75</td>
<td>0</td>
<td>15</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Case</th>
<th>10</th>
<th>11</th>
<th>12</th>
<th>13</th>
<th>14</th>
<th>15</th>
<th>16</th>
<th>17</th>
<th>18</th>
<th>19</th>
</tr>
</thead>
<tbody>
<tr>
<td>Scenario E</td>
<td>2</td>
<td>2</td>
<td>2</td>
<td>2</td>
<td>3</td>
<td>3</td>
<td>3</td>
<td>3</td>
<td>3</td>
<td>3</td>
</tr>
<tr>
<td>Baffles width/cm</td>
<td>8</td>
<td>8</td>
<td>8</td>
<td>8</td>
<td>12</td>
<td>12</td>
<td>12</td>
<td>12</td>
<td>12</td>
<td>12</td>
</tr>
<tr>
<td>Angle/°</td>
<td>30</td>
<td>45</td>
<td>60</td>
<td>75</td>
<td>0</td>
<td>15</td>
<td>30</td>
<td>45</td>
<td>60</td>
<td>75</td>
</tr>
</tbody>
</table>

Figure 8. The actual baffle arrangement (Left) and scheme of the baffles’ positions (Right).

Experiment in 2018 for Publications 3 & 4
The on-site experiment results of Scenario E0 in 2018 can be used for Publications 3 & 4 to validate the original DC model, as the original DC models for Publications 2-4 were the same. In Scenario E0, the experimental results were the temperatures and velocities for the original DC.

2.2.2 Measuring instruments
The type-K thermocouples of nickel-chromium-alumel were used as the temperature sensors, which were connected to the Agilent 34972A data
acquisition logger. This Agilent data acquisition logger recorded the rack outlet air temperature of each sensor and transferred the data to a personal computer. The specific parameters of the thermocouples and handheld air flow anemometer (a hot-wire thermal anemometry system) are shown in Table 4.

All measurement instruments were calibrated before the tests. The temperature data was received by the Agilent data acquisition logger every 10 seconds, which meant there were approximately 720 temperature data for each test point after 2-h measurement time and 0.5 h adaptive time (data during adaptation was excluded). The adaptive time included the time for changing the baffles and adjusting the angles and recovery time, ensuring that the initial thermal environment could reach steady state conditions. The temperatures of each test point during the 2-h measurement time were the arithmetic mean of 720 data avoiding the interference of the transient temperature fluctuation on the actual rack outlet air temperature distribution. The measured air temperature was 22.1 °C, and the air velocity was 5.33 m/s in the middle area of the air vent of the CRACs.

Table 4. Specific parameters of thermocouples and air flow anemometer.

<table>
<thead>
<tr>
<th>Items</th>
<th>Type</th>
<th>Measuring range</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thermocouple</td>
<td>Type-K</td>
<td>0 °C~400 °C</td>
<td>±0.4%</td>
</tr>
<tr>
<td>Handheld air flow Anemometer</td>
<td>KIMO VT200/FC300</td>
<td>Temperature: -20 °C~80 °C</td>
<td>±0.4%</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Velocity: 0.15~30 m/s</td>
<td>±2.0%</td>
</tr>
</tbody>
</table>

2.2.3 Arrangement of measuring points

For Publication 1

The location of the temperature sensors for Publication 1 is shown in Figure 9, describing the measurement arrangement and the locations of sensors. For each rack, there were installed 6 temperature sensors in the rear door of the racks. The first temperature sensor was set 0.2 m above the plenum, while the distance between the adjacent sensors was the same, which was set as 0.3 m.
For Publication 2-4

The temperature distributions of the outlet air in Rack B4 rear door were tested at the studied DC, and were presented by 6 measured temperature points (MTPs). The arrangement of the MTPs is as shown in Figure 10, while the distance between the adjacent MTPs was 0.3 m. The MTPs were the same in Publication 2-4. The velocities across the slots between the adjacent servers in Rack B4 were also measured by the handheld airflow anemometer (KIMO VT200/FC300), while there was one measured velocity point (MVP) for each slot between the servers, and altogether 13 MVPs.


2.3 Numerical simulation

2.3.1 Data center model

A commercial CFD software Airpak 3.0 was used to establish the numerical DC model, while the original model is applicable for all Publications 1-4. Figure 11 shows the numerical model, which is based on the actual configuration and the specification of DC shown in Figure 4. The locations and heat load of the servers in the numerical model were in accordance with the actual DC. The air conditioning system applied the UFAD mode, while the complete air circulation process was as follows: (1) 22 °C cool air was generated in the CRACs and was transported with an air supply speed of 5.33 m/s via the underfloor plenum; (2) the cool air rise through the floor vent and entered into the CAC between Rack A and Rack B; (3) the cool air was drawn via the servers and heated by the IT loads of the servers; (4) Finally, the hot air entered the hot aisles behind the two rows of racks and returned to the CRACs. The air supply amount of the CRACs was equal to the total airflow required by the server fans.

![Figure 11. Simulation model of the studied DC.](image)

2.3.2 Governing equation

The main research object was the outlet air temperature of the racks, and thus both the steady-state and unsteady-state flow fields were needed to be calculated in the simulation process. The turbulent convection flow regime of the racks and server conditions was used in the simulation model considering the airflow conditions, scale of racks and server conditions. Many researchers have analysed and compared the effects of different turbulent models on the DC.
simulation accuracy and convergence time. The RSM model was recommended to be used in the DC simulation model due to more precise simulation results compared with the DES and k–ε turbulent models (Wibron et al., 2018), while other researchers claimed that the zero-equation turbulent model can achieve time-targeting and well-balanced models, thus was more suitable than standard k–ε model to be applied in DCs (Phan et al., 2019).

However, the standard k–ε turbulent is a currently widely used and accepted turbulent model used in the DC numerical simulations (Yuan et al., 2018), while its reliability and feasibility have been validated in many studies (Alkharabsheh et al., 2015; Cruz & Joshi, 2015; Ham & Jeong, 2016; Phan & Lin, 2014; Song, 2016). Some assumptions were taken into consideration in the simulation process with the standard k–ε model: (i) the ignorance of radiation effects, (ii) no air leakage, (iii) the air was regarded as an incompressible fluid, and the flow was considered to be steady turbulent flow, and (iv) the temperature of air flow supplied from all the tiles were the same.

The governing equations for the incompressible fluid are as shown in Equation (1)-(3), which represent continuity, momentum and energy conservation equation, respectively.

\[ \nabla \cdot \bar{u} = 0 \] (1)

\[ \frac{\partial \bar{u}}{\partial t} + \bar{u} \cdot \nabla \bar{u} = \nabla \cdot \left( v_{eff} \nabla \bar{u} \right) - \frac{1}{\rho} \nabla p + \bar{g} \] (2)

\[ \rho c_p \left[ \frac{\partial T}{\partial t} + (\bar{u} \cdot \nabla) T \right] = \nabla \cdot \left( k_{eff} \nabla T \right) + S \] (3)

where \( \bar{u} \) is the average velocity vector, \( p \) is the static pressure, \( T \) is the static temperature, \( \bar{g} \) is the gravitational acceleration vector, \( v_{eff} \) and \( k_{eff} \) are the effective fluid viscosity and thermal conductivity, \( \rho \) is density, \( S \) denotes the volumetric heat sources and \( c_p \) is the specific heat capacity.

### 2.3.3 Boundary conditions

The finite volume method was used to calculate the CFD and transform the differential equations into discrete equations. Since the simulation’s success or failure depended strongly on the grid generation of the CFD calculations, some criteria were formulated for grid generation, including smooth element changes in the entire solution domain and that changes in density in the grid domain should be based on the variables (Ni et al., 2017). The specifications of the simulation cases are shown in Table 5, while the parameters for the boundary conditions are shown in Table 6. The convergence criteria were set as follows: the residuals for both the velocity in the directions of X, Y, and Z and the continuity were \( 10^{-3} \), and that for the energy was set at \( 10^{-6} \) (Alkharabsheh et al., 2015; Zhang et al., 2014).
Table 5. Simulation model’s specifications and boundary conditions.

<table>
<thead>
<tr>
<th>Objects</th>
<th>Type</th>
<th>Dimension/cm × cm (× cm)</th>
<th>Quantity</th>
<th>Remark</th>
</tr>
</thead>
<tbody>
<tr>
<td>Walls</td>
<td>Adiabatic</td>
<td>1100cm × 400cm</td>
<td>2</td>
<td>External walls</td>
</tr>
<tr>
<td></td>
<td></td>
<td>800cm × 400cm</td>
<td>2</td>
<td></td>
</tr>
<tr>
<td>Floor</td>
<td>Adiabatic</td>
<td>1100cm × 800cm</td>
<td>1</td>
<td>External walls</td>
</tr>
<tr>
<td>Ceiling</td>
<td>Adiabatic</td>
<td>1100cm × 800cm</td>
<td>1</td>
<td>External walls</td>
</tr>
<tr>
<td>Plenum</td>
<td>Adiabatic</td>
<td>1100cm × 800cm</td>
<td>1</td>
<td>Exclude CAC</td>
</tr>
<tr>
<td>Top of CAC</td>
<td>Adiabatic</td>
<td>540cm × 180cm</td>
<td>1</td>
<td></td>
</tr>
<tr>
<td>Perforated tiles</td>
<td>Vent</td>
<td>60cm × 60cm</td>
<td>27</td>
<td>45% free air ratio</td>
</tr>
<tr>
<td>Front door of racks</td>
<td>Vent</td>
<td>220cm × 60cm</td>
<td>18</td>
<td>65% free air ratio</td>
</tr>
<tr>
<td>Rear door of racks</td>
<td>Vent</td>
<td>220cm × 60cm</td>
<td>18</td>
<td>65% free air ratio</td>
</tr>
<tr>
<td>CRACs</td>
<td>Block</td>
<td>180cm × 85cm × 180cm</td>
<td>1</td>
<td>N/a</td>
</tr>
<tr>
<td></td>
<td>Fan</td>
<td>180cm × 15cm</td>
<td>1</td>
<td>Air outlet</td>
</tr>
<tr>
<td></td>
<td></td>
<td>180cm × 85cm</td>
<td>1</td>
<td>Air inlet</td>
</tr>
<tr>
<td>Servers</td>
<td>Partition</td>
<td>80cm × 46cm × 2</td>
<td>12</td>
<td>Upper and lower</td>
</tr>
<tr>
<td></td>
<td></td>
<td>80cm × 9cm × 2</td>
<td>12</td>
<td>Sides</td>
</tr>
<tr>
<td></td>
<td></td>
<td>46cm × 8.5cm</td>
<td>12</td>
<td>Front</td>
</tr>
<tr>
<td></td>
<td>Source</td>
<td>79cm × 45cm × 4</td>
<td>12</td>
<td>Uniform layout</td>
</tr>
</tbody>
</table>

Table 6. Parameter settings for boundary conditions.

<table>
<thead>
<tr>
<th>Boundary conditions</th>
<th>Symbol</th>
<th>Equation</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet airflow velocity</td>
<td>$U_0$</td>
<td></td>
<td>5.33 ms$^{-1}$</td>
</tr>
<tr>
<td>Tile flow rate</td>
<td>$Q_t$</td>
<td>$Q_t = U_0 A_{tile}$</td>
<td>1.92 m$^3$s$^{-1}$</td>
</tr>
<tr>
<td>Inlet air temperature</td>
<td>$T_{in}$</td>
<td></td>
<td>22 °C</td>
</tr>
<tr>
<td>Server flow rate</td>
<td>$Q_s$</td>
<td>$Q_s = Q_t/4$</td>
<td>0.48 m$^3$s$^{-1}$</td>
</tr>
<tr>
<td>Server heat generation</td>
<td>$P_s$</td>
<td>$P_s = p c \Delta T$</td>
<td>5 040 w</td>
</tr>
<tr>
<td>Perforated tiles porosity</td>
<td>$\eta_t$</td>
<td></td>
<td>45%</td>
</tr>
<tr>
<td>Rack porosity</td>
<td>$\eta_r$</td>
<td></td>
<td>65%</td>
</tr>
</tbody>
</table>

2.3.4 Simulation conditions

In order to easily distinguish the simulation scenario numbers involved in Publications 1-4, the simulation scenarios are numbered in Publications 1-4 sequentially in Roman numerals, namely Scenarios I, II, III and IV, respectively. In all Publications 1-4, the trial simulations were done in the model with all the parameters of the racks and servers totally consistent with the actual DC. The mesh number reached the 28,000,000+ level, while the convergence time for the trial simulation took more than 7 hours, and even beyond 13 hours. Thus, the model was simplified to save the simulation time. In the following numerical simulations, only racks A2-8, and only racks B3-5 were equipped with the servers based on the actual servers’ status in Publications 1 and Publications 2-4, respectively, while other racks were represented by the blocks with no heat load for the simplification of the simulation model. The reliability of the simplified models has been validated by a previous study (Yuan et al., 2018).

For Scenario I (Tilted server placement)

The numerical simulation was divided into three Scenarios I-1-3. In Scenario I-1, all the parameters and specifications were set according to the actual DC. After getting the simulation results, the numerical model was validated by the experimental results. In Scenario I-2, once the numerical model and actual model was validated to be consistent, the numerical model was made some
changes in servers’ tilted angles and heat load, but kept other DC’s parameters and specification unchanged.

To obtain general results instead of the results from the special working conditioning, 8 servers were placed evenly for each rack. In avoidance of too many grid numbers causing a very long convergence time, 2 racks (racks A7 & 8) were selected for the detailed analysis. In this condition, the convergence time was still more than 2 h. The heat load of each server was set to be the same 1 kW, which can be considered as the high-density racks. The distance between the adjacent servers was also the same, which was set as 0.09 m. In addition, the first server was 0.09 m above the plenum. In Scenario I-2, the tilted servers were applied with 4 different tilting angles (15°, 30°, 45° and 60°) and the effect of the tilted servers on the improvement of the thermal environment and cooling efficiency were studied. Figure 12 shows the sketch map of the servers’ position in one rack with different tilted angles. As a result, the optimal tilted angle recorded as X° was obtained as well as the optimal exhaust temperature distribution. Based on Scenario I-2, different IT loads were introduced in Scenario I-3. The exhaust temperature distributions were compared. All the parameters setting of each scenario are shown in Table 7.

Table 7. Parameters setting of each Scenario I.

<table>
<thead>
<tr>
<th>Scenario I</th>
<th>Heat load per rack (kW)</th>
<th>Tilted angle (°)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>8</td>
<td>0</td>
</tr>
<tr>
<td>2</td>
<td>8</td>
<td>Case I 0 15 30 45 60</td>
</tr>
<tr>
<td>3</td>
<td>Case1 8</td>
<td>0 X 20 24 32</td>
</tr>
<tr>
<td></td>
<td>Case2 12</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Case3 16</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Case4 20</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Case5 24</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Case6 28</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Case7 32</td>
<td></td>
</tr>
</tbody>
</table>
For Scenario II (Lower-side server terminal baffle)
The effects of applying the server lower-side terminal baffles on the thermal environment were studied. The numerical simulation was divided into similar four scenarios (Scenarios II-0-3) and altogether 19 cases were measured. Table 8 shows Scenario II and the cases setting for numerical simulation.

Table 8. The Scenarios II and cases setting for numerical simulation.

<table>
<thead>
<tr>
<th>Case</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
<th>9</th>
</tr>
</thead>
<tbody>
<tr>
<td>Scenario II</td>
<td>0</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>2</td>
<td>2</td>
</tr>
<tr>
<td>Baffle width/cm</td>
<td>0</td>
<td>4</td>
<td>4</td>
<td>4</td>
<td>4</td>
<td>4</td>
<td>4</td>
<td>8</td>
<td>8</td>
</tr>
<tr>
<td>Angle/°</td>
<td>no</td>
<td>0</td>
<td>15</td>
<td>30</td>
<td>45</td>
<td>60</td>
<td>75</td>
<td>0</td>
<td>15</td>
</tr>
</tbody>
</table>

For Scenario III (In-rack UFAD system)
Table 9 shows the specific case description in Scenario III, including the CA mode and geometry dimensions. Case 1 adapted a cold aisle containment (CAC), while Cases 2–7 replaced the CAC by the in-rack cold aisle (IR-CA). In Cases 2–7, the only independent variables were D₀ and Lₑ. In addition, except in Case 1, D₂, and Lₑ are the same parameter in Cases 2–7. The value of D₁, Lₑ, and D₃ were the same in Cases 2–7, respectively.

The layout and specification of Rack B4 in seven cases can be seen in Table 9 and Figure 13. The schematic map of these cases in Figure 13 is based on the...
central section of Rack A4, CA, and Rack B4 in Figure 11. The red dotted lines and blue solid lines in Figure 13 represent the rack air outlet and inlet, respectively.

Table 9. Case categories and descriptions in Scenario III.

<table>
<thead>
<tr>
<th>Case</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>CA mode</td>
<td>CAC</td>
<td>IR-CA</td>
<td>IR-CA</td>
<td>IR-CA</td>
<td>IR-CA</td>
<td>IR-CA</td>
<td></td>
<td></td>
</tr>
<tr>
<td>( D_1 / m )</td>
<td>( \text{Constant value: 0.3} )</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>( L_s / m )</td>
<td>( \text{Constant value: 0.8} )</td>
<td>( 0.1 )</td>
<td>( 0.2 )</td>
<td>( 0.3 )</td>
<td>( 0.4 )</td>
<td>( 0.5 )</td>
<td>( 0.6 )</td>
<td>( 0.7 )</td>
</tr>
<tr>
<td>( L_c / m )</td>
<td>( \text{Constant value: 0.6} )</td>
<td>( 2.2 )</td>
<td>( 2.2 )</td>
<td>( 2.2 )</td>
<td>( 2.2 )</td>
<td>( 2.2 )</td>
<td>( 2.2 )</td>
<td>( 2.2 )</td>
</tr>
<tr>
<td>( L_r / m )</td>
<td></td>
<td>( 1.2 )</td>
<td>( 1.3 )</td>
<td>( 1.4 )</td>
<td>( 1.5 )</td>
<td>( 1.6 )</td>
<td>( 1.7 )</td>
<td>( 1.8 )</td>
</tr>
<tr>
<td>( L_r = D_1 + L_s + D_2 )</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>( D_2 / m )</td>
<td></td>
<td>( 0.1 )</td>
<td>( 0.2 )</td>
<td>( 0.3 )</td>
<td>( 0.4 )</td>
<td>( 0.5 )</td>
<td>( 0.6 )</td>
<td>( 0.7 )</td>
</tr>
<tr>
<td>( D_3 / m )</td>
<td></td>
<td>( 1.8 )</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>( D_4 / m )</td>
<td></td>
<td>( 4.2 )</td>
<td>( 3.2 )</td>
<td>( 3.4 )</td>
<td>( 3.6 )</td>
<td>( 3.8 )</td>
<td>( 4 )</td>
<td>( 4.2 )</td>
</tr>
<tr>
<td>( D_4 = L_r + D_3 )</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

\( D_1 \) is the distance between the terminal of servers and the rack rear door. \( D_2 \) is the distance between the server and the rack front door. \( D_3 \) is the distance between two rack rows. \( D_4 \) is the distance between the rear doors of two rack rows. \( L_s \) is the length of the server. \( L_c \) is the length of a rack’s air inlet. \( L_r \) is the rack’s length.

For Scenario IV (Step-like server placement)

There were two scenarios (Scenarios IV-1&2) of altogether 9 cases in Publication 4. In Scenario IV-1, there were 5 cases including Cases 1-5 with 5 different horizontal spacings (0, 0.01, 0.02, 0.03 and 0.04 m) between the adjacent servers, while Scenario IV-2 was based on the optimal case obtained from Scenario IV-1 and was divided into Cases 6-9 with different SATs (22.5, 23, 23.5 and 24 °C). Figure 14 shows the schematic map of how horizontal spacings were arranged in Cases 1-5 in Scenario IV-1 from the side view of Rack B4, while
the corresponding geometric dimension differences between them were shown in Table 10.

Table 10. Dimensions in Cases 1-5 of Scenario IV-1.

<table>
<thead>
<tr>
<th>Case</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
</tr>
</thead>
<tbody>
<tr>
<td>W₁ [m]</td>
<td>0.8</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>C₁ [m]</td>
<td>0.3</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>C₂ [m]</td>
<td>0.1</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>D [m]</td>
<td>0.11</td>
<td>0.22</td>
<td>0.33</td>
<td>0.44</td>
<td>0.55</td>
</tr>
<tr>
<td>L₁ [m]</td>
<td>1.31</td>
<td>1.42</td>
<td>1.53</td>
<td>1.64</td>
<td>1.75</td>
</tr>
</tbody>
</table>

Where W₁ is the length of each server; C₁ is the distance between the rack rear door and the terminal of the bottom server; C₂ represents the distance between the rack front door and front vertical side of the top server; D represents the distance between the front vertical sides of the bottom and the top servers; L₁ is the length of each rack.

The corresponding distances between the adjacent servers on the vertical plane were totally the same among Cases 1-5, while the distances between the adjacent servers on the horizontal plane were even in any separate case of Cases 1-5, but were different among Cases 1-5. According to Figure 15, the placement of the servers in Cases 2-5 looks like a ladder or steps; thus, this innovative server placement was named as the step-like server placement in *Publication 4.*

![Figure 14](image-url)

**Figure 14.** Comparison of the horizontal spacing in Cases 1-5 of Scenario IV-1 from the side view of Rack B4.

2.3.5 Arrangement of simulated points

*For Scenario I*

The location of the simulated temperature points (STPs) was set totally consistent with that of the MTPs shown in Figure 9. For each rack, there were 6
STPs in the rear door of the racks to record the temperatures. The first STP was set 0.2 m above the plenum, while the distance between the adjacent STPs was the same, which was set as 0.3 m.

For Scenarios II-IV
The location of the STPs was set totally consistent with that of the MTPs shown in Figure 10. For each rack, there were 6 STPs in the rear door of the racks to record the temperatures. The first STP was set 0.2 m above the plenum, while the distance between the adjacent STPs was the same, which was set as 0.3 m. The STPs were the same in Publications 2-4. The location of the SVPs was also set totally consistent with that of the MVPs in Publications 2-3, while there were altogether 13 MVPs in Rack B4, which were set in the slots between the server terminal. There was a little difference between the location of the SVPs and the MVPs in Publication 4. Due to different horizontal spacings between the adjacent servers and the corresponding change of rack lengths, the locations of the velocity points in Cases 2-5 were changed compared with that in Case 1 in Publication 4. Thus, Figure 15 takes Case 5 as an example and shows the detailed locations of the SVPs of Rack B4 in Cases 2-5 from the rack side-section view. Here, the SVPs represented the simulated velocity points where the air flows out of the corresponding slot between the adjacent servers.

Figure 15. The locations of the SVP/STP of Rack B4 in Cases 2-5 of Scenario IV from the rack side-section view (Take Case 5 as an example).

2.3.6 Grid generation
The grid generation plays an important role in the calculation accuracy of the simulation results (Ni et al., 2017). The coarse grid and round-off errors caused by the tremendous increasing of the mesh number can increase the chance of the calculation errors, while too few meshes may lead to the inaccuracy of simulation (Yuan et al., 2018). Thus, a grid proper quality and number should be generated in this DC model to obtain accurate simulation results. For the DC model, the refined meshes were all applied near the vents and servers. The mesh type was Hexa unstructured, which could generate a non-uniform grid to
effectively reduce the mesh number under the same calculation accuracy as a homogeneous structured grid. According to the method of the independence test mentioned by the previous researchers (Nada et al., 2016; Siriwardana et al., 2012; Zhou et al., 2016), the independence tests of the grid were used in the DC model in all Publications 1-4.

In Publication 1, there were two sets of mesh numbers for Scenario I-1, and for Scenarios I-2-3, and thus two different grid independence tests were done.

In Publication 2, there were four Scenarios II-0-3 with no baffle, 0.46 m × 0.04 m, 0.46 m × 0.08 m and 0.46 m × 0.12 m baffles, respectively. Thus, the grid number for Scenarios II-0-3 should be different, while four grid independence tests were done. However, the optimum mesh numbers for Scenarios II-1-3 were almost the same (approximate 2 800 000), and thus one grid independence test was done for them.

In Publications 3 & 4, one grid independence test was done separately.

The refined meshes were all applied near the servers and the baffles. The maximum mesh size for axis X, Y and Z were all 0.1 m, while the minimum mesh size for axis X, Y and Z were 3.5 × 10⁻³ m, 1 × 10⁻³ m and 1 × 10⁻² m, respectively. As shown in Figure 16, the optimum mesh numbers for Scenarios I-1, I-2-3, II-0, II-1-3, III, and IV, were approximate 574 938, 1 098 659, 1 800 000, 2 800 000, 1 800 000, and 1 800 000, respectively. When the grid number reached the optimum level and the number of grids continued to increase, the maximum rack outlet air temperatures dropped to the minimum and began to stay stable. It is noted that the optimum mesh numbers for Scenarios II-0, III, and IV were the same, that was because the original DC models for Publications 2-4 were the same.
2.3.7 Model validation

For Scenario I

The temperature values of the Rack A2-A8 were validated between the measurements and the simulation results in Figure 17. In Figure 17, ER means the experimental results, while SR means the simulation results. The STPs were slightly higher than the MTPs except in the Rack A4. There are two reasons that may lead to the higher rack outlet air temperature than in the simulations. One reason is that the heat power of the servers in the simulation was set according to their full load condition, while the actual servers may not operate with the full load. Another reason could be that the internal structure of the servers modelled was not exactly the same as the actual one. In the Rack A4, the simulation results were slightly lower than the experimental results. The reason for this temperature difference is that there was no server placed in the real Rack A4. Thus, the outlet air temperature of the Rack A4 was equal to the CRACs’ SAT, while the CRACs’ actual SAT was slightly higher than the CRACs set temperature. The maximum relative error was around 4%. Thus, the rack outlet air temperature distribution with simulation results fitted well with the measurement data in Publication 1.
Methodology

Figure 17. Comparison of temperature value of six outlet measuring points in Rack A2-A8 between experimental (solid line) and simulation (dotted line) results in Publication 1.

For Scenarios II-IV

Figure 18 shows the temperature and velocity validation between the experimental results and the simulation results of the original DC model in Publications 2-4. The trend of the temperature points in the experiment and simulation were almost the same, while the STPs were slightly larger than the MTPs from Scenario E0 in all temperature points. In addition, the maximum deviation of the temperatures between corresponding MTPs and STPs was smaller than 3.5%. Thus, the numerical results were consistent with the experimental results in terms of the temperature of the MTPs/STPs.

Likewise, the trend of the velocity points in the experiment and simulation of the original DC model were almost the same, while the maximum deviation of the velocities between corresponding MVPs and SVPs was smaller than 11.5%, which was accepted. Thus, the numerical results were consistent with the experimental results in terms of the velocities of the MVPs/SVPs. In summary, the original DC model was strongly validated by the experimental results in terms of the temperature and velocity distributions. Figure 18(Left) could also explain why the exhaust temperatures of six measuring points in the experiment were higher than the corresponding temperatures in the numerical results in Figure 18(Right). The larger velocities resulted in lower temperature distribution.
2.3.8 Energy saving calculation

Under the circumstance that the proposed airflow management methods effectively improved the thermal environment within DC, the daily energy and electricity saving potentials of HVAC system could be compared and analyzed between the original and optimum cases in this thesis. The calculation process of the diurnal electricity saving of HVAC system can be calculated by Equations. (4)-(11).

\[
Q = cp \cdot m \cdot \Delta t \tag{4}
\]

\[
m = \rho \cdot V \tag{5}
\]

\[
V = v \cdot A \tag{6}
\]

\[
\Delta t = SAT_{cx} - SAT_{c1} \tag{7}
\]

\[
Q_1 = cp \cdot m \cdot \Delta t \cdot 24 \tag{5}
\]

\[
Q_2 = Q_1 / \text{COP}_{CRAC} \tag{9}
\]

\[
\text{COP}_{CRAC} = 1.53 \tag{10}
\]

\[
Q_2 = cp \cdot \rho \cdot v \cdot A \cdot (SAT_{cx} - SAT_{c1}) \cdot 24 / \text{COP}_{CRAC} \tag{11}
\]

Where \( Q_1 \) is the daily cooling energy saving, kWh; \( Q_2 \) represents the daily total electricity saving for CRAC units, kWh; \( cp \) is the specific heat capacity of air, kJ/(kg K); \( \rho \) is air density, kg/m\(^3\); \( V \) represents the volume of air from the CRAC units, m\(^3\); \( v \) is the air velocity through the air outlet of the CRAC units, m/s; \( A \) is the area of air vent of each CRAC, m\(^2\); \( \Delta t \) is the SAT difference between original case and optimal case, K; \( SAT_{c1} \) is the SAT of original case, °C; \( SAT_{cx} \) is the SAT of the optimum case, °C; \( x \) is the number of the optimum case; \( \text{COP}_{CRAC} \) represents the COP of CRAC units.
3. Results

3.1 Rack outlet air temperature and airflow distribution

3.1.1 Scenario I (Tilted server placement)

Scenarios I-1&2
Scenario I-1 in Publication 1 was set to validate the DC simulation model, and the specific results have been analyzed in the model validation section. In Scenario I-2, the total heat load for each rack was set to 8 kW. The simulation results in Scenario I-2 are shown in Figure 19. According to our previous work (Yuan et al., 2018), the rack hotspots were defined as the highest rack outlet air temperature points in each rack, which were the most possible locations that could damage the servers.

As shown in Figure 19, the rack hotspots of racks A7 & A8 were optimized in Cases II & III, but deteriorated in Cases IV & V. Figure 20 shows the velocity vector of Rack A8 in Cases I, III and V. In Case I, the cool air from the vents inclines upward into Rack A8 and a part of the cool air is blocked by the servers, while the rest of the cool air flows through the tunnel between the servers in a horizontal direction. A small part of the cold air flowing through the channel takes away the heat from the terminal of the servers, while the rest become the bypass cold air and flow out of the rack. Thus, the heat removal by the cool air is very limited in Case I. The heat accumulation zone always existed in the terminal of the servers (shown in Figure 20). In addition, as shown in Figure 19 of Case I, a high percentage of bypassing air flows directly from the upper side of the rack where no server is placed. Thus, the cool air supplied to the bottom of the rack is limited, causing the heat to accumulate at the rack bottom.

![Figure 19](image.png)

*Figure 19.* The cloud map of outlet air temperature of racks A7 and A8 in Scenario I-2 with different server tilted angles.
Figure 20. The air flow direction of Case I, Case III and Case IV.

However, the application of 30° tilted servers in Case III keeps the tilted angle of servers and the inlet cool air direction of the rack basically the same. Moreover, the occupied spaces of inclined servers inside the rack are larger than the servers of Case I to reduce the bypassing effect. In Figure 20, Case III depicts that although a part of cool air is still blocked by the servers, the cool air flows along the upper and lower sides of the servers, and then the cool air inclines downward and dissipates the heat at the terminal of the servers. The most direct effect of the application of 30° tilted servers is shown in Figure 19 of Case III, where the temperature of the rack hotspot in Rack A8 decreases from 28.4 °C (Case I) to 26.7 °C. The heat accumulation of Rack A8 is reduced and the highest temperature rises to the middle of the rack.

In Case V, when the tilted angle of the servers continues rising to 60° in Case V, the distance between neighbouring servers decreases greatly and almost sticks together, which blocks the flow pattern of the cool air extremely. Considering that the height of rack cannot be changed, the distance between the adjacent tilt servers cannot be increased. The cool air rises up to the top and crosses the top server, which can be regarded as a significant airflow bypass.

As a summary, 30° is regarded as the most optimal tilted angle, and the outlet air temperature distributions of both racks A7 & A8 are the best. Therefore, under the circumstance of this server distribution, the heat accumulation inside the rack can be significantly relieved.

Scenarios I-3
According to Scenario I-2, the best air temperature distribution is achieved at the application of 30° tilted servers. In Scenario 3, the heat load of each rack is changed from 8 kW to 32 kW shown in Case 1-7. In each case, the exhaust mean temperature drop of Rack A8 before and after applying 30° tilted servers is compared as well as the temperature drop of maximum temperature before and after using 30° tilted server.

As shown in Figure 21, exhaust mean temperature reduction of Rack A8 with 0° and 30° tilted servers steadily increases with the increase of the heat load, until the maximum value is about 1.4 °C in Case 7. In addition, the temperature
reduction of the maximum temperature before and after applying 30° tilted servers in Case 1-7 increases with the increases of the heat load. In Case 7, the temperature reduction of the maximum temperature before and after applying 30° tilted servers reaches almost 3.3 °C, which successfully lowers the maximum temperature of Rack A8 from around 45.3 °C to 42 °C. Thus, it can be concluded that the application of 30° tilted servers can improve the airflow distribution and cooling effects, when the total power of the rack changes and the servers are evenly placed.

To sum up, not all server tilt angles can optimize the thermal environment, and excessive server tilted angles can block the slots between servers. When the tilt angle is 30°, the thermal environment is optimized, and the optimization result is not only applicable to the original case, but also to the case where the power of the rack changes and the servers are placed evenly.

![Figure 21. The mean and maximum temperature reduction of Rack A8 with 0° and 30° tilted servers.](image)

### 3.1.2 Scenarios E0-3 & II (Low-side terminal server baffle)

For Scenarios Eo-3

In the experiment part, the rack hotspot was assumed to be the highest temperature point of 6 measuring points. Figure 22 shows the temperature drop of the rack hotspots in Cases 2-19 compared to Case 1. The temperature of the rack hotspot in Cases 2, 8 and 14 rose significantly. The same trend also exists in Cases 3 and 9. In Cases 3, 7, 10, 15 and 19-20, the improvement of the rack hotspots was not significant. The temperature drops were not more than 0.5 K, while the temperature of the rack hotspots dropped moderately (1 K) in Cases 5-6, 13 and 16. In addition, there were significant temperature drops (2 K) in Cases 11-12 and 17 compared to the reference Case 1. In Cases 11-12 and 17, the overall performance of the air distribution enables the most optimal performance.
Figure 22. Temperature drops of the rack hotspot in Cases 2-19 compared to the reference Case 1.

The results of the temperature differences at different distances were summarized in Table 11. In Table 11, the symbol of “√” represents that the temperature of the measuring point decreased compared to the reference Case 1, while the symbol of “×” represents that the temperature of the measuring point in this case increased compared to that in Case 1. As shown in the Table 11, the temperatures of 6 measuring points all decreased only in Case 11. In the case analyzed, Case 11 has the best performance of the measured cases.

Table 11. Comparison of temperatures for 6 MTPs in Cases 2-19 compared to the reference Case 1.

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Above all, synthetically considering the temperature drops of the rack hotspot and the temperatures of 6 MTPs, Case 11 could be selected as the best optimization scheme to optimize the airflow distribution. Figure 23 shows the comparison of the temperatures in the MTPs between Case 1 and Case 11. The temperature dropped significantly at the MTPs 1 & 2, which dropped by 1.9 K and 1.5 K, respectively. At other MTPs, the temperature drops were around 1 K.
According to the experimental results, Case 11 with 8 cm and 45° terminal baffles can improve the temperature distribution the best in the studied DC. The maximum temperature decreases of the MTPs reached 1.9 K.

**Scenario II**

Figure 24 shows the outlet air temperatures of 6 STPs in the Rack B4. It is obvious that Case 11 with 8-cm 45° baffles performed the best in terms of the rack outlet air temperatures of 6 different STPs. The temperatures of the STPs were all below 31 °C, while the maximum temperature drop achieved approximately 2 K.

Figure 25 shows the outlet air temperature profiles of the rear door in Scenarios II-0-3. In Scenario II-0, there is an obvious heat accumulation in the lower part...
of the Rack B4, while the application of 0° baffles in Cases 2, 8 and 14 could not improve the thermal environment. The area of the heat accumulation increased when the width size of baffles increased. The area of heat accumulation in Case 3-4, 5-9, 13, 15-16 and 18-19 slightly increased. In addition, a part of heat was transferred to the upper part of the Rack B4 in these cases. This increases the overall rack outlet air temperature. Thus, although the temperature of the rack hotspot decreased in Cases 3-4, 5-9, 13, 15-16 and 18-19, the heat accumulation problem and thermal environment were not improved.

Both of the heat accumulation problem and the thermal environment were improved in Cases 11, 12 and 17. The outlet air temperature distribution of the upper and lower parts of Rack B4 was relatively uniform in these cases. In terms of the temperature of the rack hotspot, Case 11 was the best, while next came Case 17. In addition, in terms of the temperature uniformity, Case 11 also ranked the first, better than that in Case 12 and 17. Thus, Case 11 performed the best in terms of the elimination of the heat accumulation and the thermal environment improvement.

**Figure 25.** Outlet air temperature distribution of the Rack B4 in Scenarios II-0-3.
To sum up, when 45° 8-cm lower-sider server terminal baffles were applied in Case 11, the rack outlet air temperature distribution and the thermal environment were optimized the most. It could realize the following optimization: (i) the temperatures of the STPs were all below 31 °C, while the maximum temperature drop achieved approximate 2 K, and (ii) heat accumulation was greatly diminished and the thermal environment was improved in Case 11.

Combining the experimental and simulation results, it can be concluded that the application of the lower-sider server terminal baffles can optimize the thermal environment, while the optimum thermal environment can be achieved by applying 45° 8-cm lower-sider server terminal baffles.

### 3.1.3 Scenario III (In-rack UFAD system)

**Optimization of temperature distribution**

As shown in Figure 26, an obvious heat accumulation exists in the lower part of Rack B4 in Case 1. The application of the in-rack cold aisle (IR-CA) can to varying degrees improve the outlet air distribution of the middle and top part of Rack B4 in all Cases 2–7. For the lower part of Rack B4, the obvious heat accumulation happens in Cases 2 and 3, which means that the application of IR-CA cannot improve the airflow and thermal distribution of B4. However, when the width of IC-RA increases from 0.4 m in Case 4 to 0.6 m in Case 6, the heat accumulation in the lower part of Rack B4 is significantly reduced, while the heat is never accumulated in the lower part in Case 6. When the width of IR-CA increases to 0.7 m, the heat accumulation phenomenon recovers in the lower part of Rack B4. Thus, combining the optimization effects of the whole rack, Case 6 performed best in terms of the heat accumulation mitigation and the temperature distribution uniformity.

**Figure 26.** Outlet air temperature profiles of Rack B4 in seven cases.

Figure 27 illustrates the temperature variation of the rack hotspot and the maximum temperature difference in seven cases. The temperature of the rack hotspot stands at 33.2 °C in Case 1 and then rapidly increases to 35.3 °C in Case 2, where there is an exponential decrease to an all-time low of 30.8 °C in Case 6.
In Case 7, the rack hotspot temperature recovers to 32.4 °C. Thus, the rack hotspot of Rack B4 reaches the minimum temperature in Case 6. Likewise, the maximum outlet air temperature difference of Rack B4 has the same change trend with the rack hotspot, while the maximum rack outlet air temperature difference is the lowest at 8.8 °C in Case 6. Thus, Case 6 can achieve the lowest rack hotspot and the lowest temperature difference.

![Figure 27. Temperature variation of rack hotspot and maximum temperature difference in seven cases.](image)

To sum up, when the width of IR-CA is larger than 0.3 m, the in-rack UFAD system can make the thermal distribution of Rack B4 more uniform. In addition, concerning the heat accumulation mitigation, rack outlet air temperature distribution, rack hotspot, and maximum temperature difference, the optimal thermal distribution can be obtained in Case 6 with 0.6 m IR-CA in width. The maximum temperature drop of the rack hotspot in Case 6 can reach 2.4 K compared with the original case.

**Velocity optimization**

Figure 28 shows the air velocity profiles of thirteen simulated velocity points (SVPs) and the maximum velocity difference of all the SVPs in seven cases separately. The larger the velocity difference, the less uniform the supply air in the upper and lower part of the rack. Case 1 ranks first in terms of the supply air velocity uniformity, which is followed by Case 6, then Case 5 and Case 7. The bar graph in Figure 28 shows the air velocities at thirteen SVPs in seven cases. The majority of velocities of the SVPs are relatively low (below 0.5 m/s) in Case 1, while the velocities of the most SVPs are greater than 0.5 m/s in Cases 2–7. There is only one SVP whose velocity is below 0.5 m/s in Cases 2–7. There is only one SVP whose velocity is below 0.5 m/s in Cases 5–6, while the number of VPs with a velocity below 0.5 m/s is at least three in the other cases. As a conclusion, Cases 5–6 have a relatively uniform air supply when there is a higher mean air velocity level.
Results

Advanced model analysis

Although Case 6 of Scenario III improves the airflow and temperature distribution of Rack B4 to a great extent, the airflow through the route between the upper side of the top server and rack top surface has a relatively high velocity, compared with some other flow paths between the neighboring servers in Rack B4. In that situation, the majority of the cold air bypassed and flowed directly out of the rack, causing the waste of the cold air. Thus, an additional partition plane is placed horizontally between the upper side of the top server and the rack front door in Case 8. The side description and the 3-D model of Rack B4 in Case 8 are shown in Figure 29. The dimension of the partition planes is 60 cm × 60 cm × 0.1 cm.

Figure 29. The side description (left) and 3-D model (right) of Rack B4 in Case 8 of Scenario III.

Figure 30 shows the outlet air temperature and airflow distribution of Rack B4 in Cases 1, 6, and 8. The lower air velocity contributed to higher rack outlet air temperature in the same horizontal plane. Compared to Case 1, the heat accumulation is reduced significantly in both Cases 6 and 8, while the rack outlet air temperature distribution is simultaneously improved. In addition, the velocities of the rack lower part are greatly enhanced in both Cases 6 and 8.
However, the application of a partition plane in Case 8 prevents the cold air from escaping through the gap of the upper server side. As shown in Figure 30, comparing Case 6 with Case 6, the air velocities of the middle and top part of Rack B4 are increased. The air velocities are the most uniform in Case 8. Thus, compared to Case 6, the heat accumulation in the top part is further mitigated in Case 8, while the heat accumulation in the lower part is almost the same.

![Figure 30. The outlet air temperature distribution and airflow distribution of Rack B4 in Cases 1, 6, and 8.](image)

The corresponding analysis of the velocity distributions in Cases 6 and 8 is shown in Figure 31. Except for the SVP 8 in Case 6 and the SVP 13 in Case 8, the air velocities of all the VPs are higher than the corresponding VPs’ velocities in Case 1. Thus, both Cases 6 and 8 have a higher air velocity in Rack B4 than Case 1. However, Case 6 has slightly higher velocities in the SVPs 1–3, while Case 8 has higher velocities than Case 6 in the SVPs 4–12 to varying degrees.

![Figure 31. Airflow velocities of thirteen SVPs in Cases 1, 6, and 8.](image)

Through comparing and analyzing the temperature and airflow distribution of Cases 1, 6, and 8, it can be concluded that the airflow and temperature distributions in Case 8 are more uniform than that in Case 6.
3.1.4 Scenario IV (Step-like server placement)

Air temperature distribution

Figure 32 shows the temperature distributions of the air outlet and side views of Rack B4 in the original Case 1. There is a serious heat accumulation in the bottom side of air outlet, where the rack hotspot is also located, while the temperature of rack hotspot in Rack B4 is about 33.2 °C. From the side view of Rack B4, there is an upward heat accumulation trend in the server terminals of zone 1, while the opposite trend exists in the server terminals zone 2. Thus, the heat accumulation of these two zones overlaps each other, bringing more serious heat accumulation in the rack bottom side. In that condition, the servers in zones 1 and 2 are overheated, whose excess heats from the server terminals are not removed effectively.

Figure 32. Temperature distributions in Case 1 of the air outlet and side views of Rack B4.

Figure 33 shows the comparison of the hotspots of Rack B4 in Cases 1-5. The server hotspot is defined as the highest temperature point of the parallel and congruent plane 30-cm away from the corresponding server terminal side, while there is one server hotspot to each corresponding server. Here, the maximum temperature of all the server hotspots in one rack can be defined as the rack hotspot. The application of the step-like server placement can relieve the rack hotspot in all Cases 2-5, while the temperature of the rack hotspot dropped the most in Case 2 by 2.5 K compared to Case 1. With the increase of spacing between the adjacent servers from Cases 2 to 5, the rack hotspot gradually increases. In other words, the temperature of the rack hotspot is inversely proportional to the spacing between the adjacent servers in Cases 2-5. In general, Case 2 performs the best in terms of the rack hotspot temperature.
Figure 33. Comparison of rack hotspots in Cases 1-5.

Figure 34 shows the temperature distributions of 12 vertical planes 30-cm from each server terminal in Cases 1-5. The application of the step-like server placement can improve all the temperature distributions in Cases 2-5 to varying degrees. The serious heat accumulation at the bottom part is relieved significantly in both Cases 2-4, while that is also moderately improved in Case 5. In addition, the temperature distribution uniformities in Cases 2-4 are also improved a lot. Among Cases 2-5, the smaller the horizontal spacing between adjacent servers, the more uniform the temperature distribution of the 12 planes, while the heat accumulation in the bottom side of rack is almost completely eliminated in Case 2, contributing to the most uniform thermal distribution. Thus, the heat accumulation mitigation effect is the best in Case 2, and so is the temperature distribution uniformity.

Figure 34. Temperature distributions of 12 planes 30-cm from each server terminals in Cases 1-5.

To sum up, Case 2 with 0.1-cm horizontal spacing between the adjacent servers can achieve the optimum temperature environment considering the rack
hotspot elimination, heat accumulation mitigation and temperature distribution uniformity.

Air velocity distribution

Figure 35 shows the air velocity difference of all the SVPs between Case 1 and Cases 2-5, and part 1 includes the four bottom SVPs in the targeted rack. Only in Case 2 all the SVPs are bigger than that in Case 1, while except the 1st SVP in Case 2 is not the highest. All the other SVPs have the highest velocities in Case 2 compared with that in Cases 1 and 3-5. The velocity increase level of the SVPs in part 1 is much more significant compared to the rest SVPs in all Cases 2-5. Compared to Case 1, almost all the velocity improvement of the SVPs in part 1 increase by at least 0.2 m/s in Cases 2-5, while that in the rest SVPs have a maximum increase of 0.2 m/s. Based on the analysis, it can be concluded that the application of all step-like server placement significantly improves the air velocities at the rack bottom part, while Case 2 can improve the overall air velocities of the SVPs the most.

![Figure 35](image)

**Figure 35.** Comparison of velocity difference of all the SVPs between Case 1 and Cases 2-5.

As a summary, the application of the step-like server placement can improve the air flow and further reduce the velocities through the slots between the adjacent servers. In Case 2 effectively improvement to the overall air flow and velocities through all the slots between adjacent servers is noticed. Also, it can increase the air flow and velocities through the slots in part 1 where the serious heat accumulation and the rack hotspot are.

## 3.2 Energy saving potentials

### 3.2.1 Scenario III (In-rack UFAD system)

Based on the thermal distribution analysis, the application of a 0.6 cm × 0.6 m IR-CA and 0.6 m × 0.6 m × 0.1-m partition plane in Case 8 can achieve optimal
thermal distribution and minimize the rack hotspot. In order to calculate the energy-saving potential of Case 8 compared to Case 1, the SAT of the CRACs is adjusted to 23 °C (Case 8-1) and 23.5 °C (Case 8-2) to simulate the temperature and airflow distributions.

Figure 36 compared the outlet air temperature distribution of Rack B4 in Cases 1, 8-1, and 8-2. Compared to Case 1, whether the SAT is set to 23 °C or 23.5 °C, the heat accumulation in the lower part of the rack is mitigated to varying degrees. The heat accumulation in the top part of the rack deteriorates slightly in Case 8-1, while it deteriorates moderately in Case 8-2. However, the deterioration of the heat accumulation in both Cases 8-1 and 8-2 is acceptable because the upper heat accumulation alleviates the lower heat accumulation. In addition, compared to the heat accumulation in the lower part of Rack B4 in Case 1, the heat accumulation in the upper and lower Rack B4 of Cases 8-1 and 8-2 is more moderate.

Both Case 8-1 and Case 8-2 have a better temperature distribution than Case 1. In addition, the heat accumulation phenomenon in Case 1 is also mitigated in Cases 8-1 and 8-2. However, the temperature distribution and heat accumulation mitigation effect in Case 8-1 is much better than that in Case 8-2.

The electricity use of CRACs should decrease due to the rise of SAT in Case 8-1. In the analysis, equations 4-11 in Section 2.4 are used to calculate the electricity saving of Case 8-1 compared to Case 1. Equation 11 shows the hourly cooling energy saving of 6.2 kWh in Case 8, which means approximately 150 kWh/day cooling energy will be saved in this DC. According to Table 2, the COP of each CRAC is around 1.53. Thus, the total electricity saving of CRACs is about 98 kWh/day, respectively.

To sum up, Case 8-1 performed the best in terms of the thermal distribution, heat accumulation mitigation effect and rack hotspot. In addition, it can also
improve the cooling efficiency and save the electricity use of 98 kWh/day with the improved solution.

3.2.2 Scenario IV (Step-like server placement)

According to the analysis in Scenario IV, the optimum thermal environment is achieved in Case 2, where the rack hotspot temperature and airflow distributions are improved the most. Thus, Case 2 was selected as the basic Case in Scenario 2 for Cases 6-9, where SAT were 22.5 °C, 23 °C, 23.5 °C and 24 °C, respectively.

The temperature distributions of 12 vertical planes 30-cm away from each server terminals in Cases 1 and 6-9 are compared as shown in Figure 37. In Cases 6-8, the temperature distributions of all the vertical planes are better than that in Case 1. In addition, the heat accumulation of the rack bottom side in Case 9 is slightly relieved compared with that in Case 1, and is the closest to that in Case 1 among all Cases 6-9. Considering the strict-level and lenient-level requirements of the temperature distributions compared with Case 1, in Cases 8 and 9 are accepted.

Figure 37. The temperature distribution of 12 planes 30-cm from each server terminals in Cases 1 and 6-9.

In short, considering the server hotspot and the temperature distributions, Cases 8 and 9 are more similar to Case 1 at the strict-level and lenient-level requirement, respectively. Compared with Case 2, the SATs of CRACs increase from 22 °C to 23.5 °C and 24 °C in Cases 8 and 9. Under this circumstance, the overall temperature distributions of Rack B4 in Cases 8-9 are still better than that in Case 1, and they can also save the electricity use of CRAC units. According to Equations 4-11 in Section 2.4, the electricity use savings for CRAC units are 146 kWh (strict level) and 195 kWh (lenient level) per day in Cases 8 and 9 with the same spacing of 0.01 m between the adjacent servers and different SATs of 23.5 °C and 24 °C, respectively.
3.3 Energy efficient improvement potentials with free cooling

Recently, the free cooling technology has undergone rapid development in DCs, which utilizes the natural cold sources to cool DCs differentiated from using conventional CRACs (ASHRAE, 2001; Pawlish & Varde, 2010). When the outdoor environment is cold enough, free cooling technology can use the outdoor cold air/water as the cooling medium in DCs, which is considered as the air-side and water-side economizer (Lee & Chen, 2013; Ma et al., 2016, Zhang et al., 2014).

Publication 5 is a literature review about the free cooling, especially the phase change cooling (PCC) application potentials in DCs for future study recommendations. The main cited references are the journal articles, conference proceedings, book sections, patents and technical reports, which are reviewed and downloaded in some common bibliographic databases (e.g., Web of Science, ScienceDirect Taylor and Francis, and Springer). In addition, there are also some documents from the following databases, including MDPI, Chinese patent database, Wiley online library, IEEE explore, ResearchGate and Aalto University online library. Except for the official publications, some sources from web pages are also adopted and cited as well as working papers and thesis.

3.3.1 Phase change cooling

Apart from outdoor cold air/water sources, the heat pipe (HP) system also belongs to the free cooling technology, which has been studied and applied in DCs for many years. Faghri (1995) defines the heat pipe as “a heat-transfer device that combines the principles of both thermal conductivity and phase transition to effectively transfer heat between two solid interfaces”. Thus, the cooling system based on the HP can be considered as the phase change cooling (PCC) system.

The PCC has become an innovative and promising cooling technology since 2004 (Marcinichen et al., 2010; Sundaram et al., 2010; Zalba et al., 2004), which can make up for the cooling lack of the CRACs and achieve better thermal environment and energy saving of the CRACs at the cost of little energy use in DCs (Liu et al., 2012). The PCC can realize the forward state change of a thermally conductive medium from the liquid to the gas and the corresponding reverse state change from the gas to the liquid (Joybari et al., 2015). When the operating powers and temperatures of computer parts are too high, the evaporating process of PCC can take away the heat and prevent them from overheating and damage (Khan, 2016). The PCC essentially utilizes renewable energy and free cooling sources, which has the following advantages:

- The latent heat used is hundreds of times higher than sensible heat (Fernández et al., 2010). Compared with other cooling methods, the PCC can achieve the decrease of the heated surface temperature variation and improvement of the heat transfer rate and heat dissipation of the racks;
- It reduces the risk of airflow leakage inside the servers and achieves a more reliable and flexible cooling form in DCs (Ding et al., 2016);
The reduction in the use frequency of the fan reduces the noise, and the noise level can be controlled below 45 dB (Garimella et al., 2016).

The evaporating and condensing processes are achieved by the phase change materials, absorbing and releasing much heat, while the evaporating process will improve the thermal environment and reduce the ambient temperature by 3-10 °C (Farid et al., 2004; Safari et al., 2017). The cooling capacity of PCC is more than 1 000 times higher than that of the air-side cooling technologies (Zhao, 2009). Compared with other cooling technologies, the PCC always undergoes an evaporation process, while various low-boiling electrolyte fluids and refrigerants can be used as coolants (Liu et al., 2012). Reasonable selection of refrigerants will greatly affect the cooling performance of the PCC (Dincer, 2017).

The PCC has been widely investigated and applied in DCs, and proved as high thermal performance and energy-saving method. The PCC technology applied in DCs is mainly divided into four categories, including independent HP cooling, integrated HP cooling, two-phase immersion cooling, and cold storage systems. Figure 38 shows the categories of detailed PCC methods in DCs (Publication 5).

![Figure 38. Categories of detailed PCC technologies in DCs.](image)

**Independent heat pipe system**

Independent HP system is divided into the separate system and the integral system. In the separate HP system, the only energy consumer is the fans of indoor and outdoor units; Thus, its operating energy use is approximately three quarters less than that of the vapor compression system with the same cooling capacity (Khalaj & Halgamuge, 2017). In DC application scenario, the phenomenon of air mixing and recirculation can be obviously alleviated by the separate heat pipe system.

Many studies are on the separate HP system applied in DCs, including typical separate system and micro-channel separate system (Chen et al., 2012; Ding et al., 2016; Guo et al., 2007). Compared with the separate HP system, the integral HP system has high heat transfer efficiency, and the fins can be used to enhance heat transfer on the cold and hot sides of the HP as required. The evaporating
and condensing sections of the integral HP are in the upper and lower spaces of
the same whole (Yang et al., 2009). Integral HP system is further divided into
the pulsating integrated HP, the integral HP system with air duct, and the loop
HP, whose reliability and feasibility have been proven separately (Lu & Jia, 2016;
Maydanik et al., 2009; Zhou et al., 2011; Zimbeck et al., 2008).

**Integrated HP with other cooling systems**

The application scope of the independent HP system described above is limited,
and requires an additional and separate air-conditioning system to meet the
cooling demand during the hot season. This is increasing the installation and
maintenance space requirements in DC as well as investment cost (Maydanik et
al., 2009). Apart from the independent HP system, the HP system can also be
integrated with the vapor compression and thermosyphon system to further
enhance its application range and energy-saving potentials, but auxiliary
mechanical cooling is required in the integrated HP system (Reay et al., 2013).
The integrated HP and other systems can be divided into two categories, which
are integrated HP system with a common flow channel and that with heat
exchangers, while both of them have been widely studied and applied in DCs
(Zhang et al., 2015; Han et al., 2013; Lee et al., 2006; Okazaki & Seshimo, 2011;
Wang et al., 2013; Wang et al., 2015; Yu & Wang, 2019). To sum up, most
integrated systems have three cooling modes:

- **When the ambient air temperature is sufficiently low,** the thermosyphon
circuit system can separately undertake the cooling work in DCs;
- **When the ambient air temperature is moderate,** the thermosyphon circuit
and the vapor compression loop operate simultaneously to ensure the
ambient temperature in DCs is stable;
- **When the outdoor temperature is high,** the thermosyphon circuit system
stops working and the vapor compression system operates separately to
offer the required refrigerating capacity in DCs.

**Two-phase immersion cooling**

Besides, the two-phase immersion cooling also belongs to the PCC technology,
which is an also effective method to eliminate the thermal generated by the
electronic equipment, and is also known as the pool boiling (Chu et al., 2004;
Rainey & You, 2000). It immerses the electronic equipment of the system in a
tank with the boiling volatile dielectric coolant (Mohammed, 2017; Shelnutt et
al., 2017). The heat generated by the electronic device is effectively absorbed in
the form of the saturated vapor, and can be efficiently transferred by the
condensation to an external cooling medium. The two-phase immersion cooling
has been also widely investigated in DCs (Almaneea et al., 2014; Kuncoro et al.,
2019; Sarangi et al., 2015; Tuma, 2010; Wu et al., 2019).

The heat flow characteristics of the two-phase immersion cooling make it a
major contender for the air-cooling solutions. The whole-process energy
consumption of heat dissipation is almost zero, and the overall energy efficiency
is much high. This system is not affected by the natural environment, which
greatly increases its application potential in DC. In addition, it not only ensures
efficient heat exchange but also eliminates the potential threat of water.
However, this cooling method requires a high demand for refrigerant (Bock et al., 2019). Firstly, the refrigerant should have a low boiling point at low pressure and be close to the normal temperature. At the same time, it must have no impact on the electronic components. The performance and heat flow of the system are decided by different parameters, which have been thoroughly studied under different surface roughness and coolant conditions.

**Cold storage system**

Furthermore, as the cooling demand must be met continuously in DCs throughout the year due to its characteristic of 24/24 hrs., 365 days operation (Pawlish et al., 2014), the cold storage system can make up for the cooling demand in case of the emergency power failure, whose purpose is to overcome the mismatch in the energy supply and demand in time (Ibrahim et al., 2008). When the energy price fluctuates greatly, it can reduce the peak load demand in a certain period, which brings benefits to the smart grid. The phase change materials (PCMs) are usually combined with the cold storage technology. The application effect of the PCMs in buildings is better than that of the sensitive energy storage materials, and high energy storage density can be obtained (Zalba et al., 2004). The cold storage system has also been investigated and applied in DCs (Chen et al., 2017; Singh et al., 2011; Wang et al., 2015).

### 3.3.2 Comparison of PCC

To sum up, an independent HP system can meet the cooling demand only through the HP without any other mechanical cooling. The only trigger of this system is the temperature difference. Thus, no power transmission is needed for the cold source to take away the heat, causing low system energy consumption. It dramatically reduces the energy use of current general CRAC in DCs, and makes full use of the outdoor natural cold sources to dissipate the heat, achieving good indoor cooling effects. In addition, this system uses fewer moving parts and wearing parts, and has high stability.

However, when limited by the external environment temperature conditions, the independent HP unit cannot undertake the cooling task in DCs. Under this circumstance, an additional cooling system must be used to meet the cooling demand in hot season. In order to avoid using two sets of cooling equipment, an integrated system, combining the HP and other cooling systems, was proposed. The system can reduce the initial investment cost to a certain extent, and flexibly switch the operating mode according to the ambient temperature. When the outdoor environment temperature is too high to satisfy natural heat dissipation, the system switches to VC mode (VC system working time is reduced). The system combined with vapor compression and thermosyphon system operate alternately and back up each other, which not only greatly saves the energy consumption, but also effectively guarantees the safety and reliability of the DC operation. It can be used as the first choice for the environmental control of the electronic equipment operating throughout the year such as DCs, especially in low temperature areas.
In addition, the two-phase immersion cooling technology doubles the heat transfer efficiency by boiling and condensing the coolant. This method hardly includes moving parts, but the control is relatively complicated. The pressure will change during the phase change, which places high requirements on the container, and the coolant is easily contaminated during use. The cold storage technology is suitable for newly built instead of in-use DCs, which use the phase change latent heat (much higher than sensible heat) to passively store and release energy. It is expected that the application market for cold storage system will develop rapidly in the future. Table 12 summarizes the characteristics, pros and cons of these PCC technologies, and summarizes future research needs (Publication 5).

Table 12. The characteristics, pros, and cons of four common PCC technologies.

<table>
<thead>
<tr>
<th>Category</th>
<th>Independent heat pipe system</th>
<th>Integrated vapor compression and thermosyphon system</th>
<th>Two-phase immersion system</th>
<th>Cold storage system</th>
</tr>
</thead>
<tbody>
<tr>
<td>Characteristics</td>
<td>-Cooling demand is met totally by HP; -DCs can be cooled without other mechanical cooling.</td>
<td>-Realize simultaneous operation of vapor compression and HP cooling</td>
<td>-Two-phase coolant can produce two states: liquid and gas.</td>
<td>-The latent heat of phase change stores cold energy and releases cold energy when needed.</td>
</tr>
<tr>
<td>Pros</td>
<td>- No effect on the indoor air quality and humidity; - Relatively high heat transfer effect and utilization rate of natural cold sources</td>
<td>-Higher heat transfer effect and utilization rate of cold free cooling sources. -Independent heat pipe system can meet the cooling requirements during hot seasons.</td>
<td>-The heat transfer efficiency is doubled compared to typical HP cooling; -System temperature is stable; -Operating environment is quiet.</td>
<td>-The AC system can achieve the continuous cooling and has high economy.</td>
</tr>
<tr>
<td>Cons</td>
<td>-Limited by the ambient temperature conditions</td>
<td>-Most ISVT systems rely on solenoid valves for mode switching, which has high requirements for solenoid valves. -The reliability of the system for long-term operation cannot be guaranteed</td>
<td>-Compared with single-phase coolant, the cost of two-phase coolant is high.</td>
<td>-The cold storage device requires large space and increases the cost of cold storage equipment</td>
</tr>
</tbody>
</table>

Notations: HP= heat pipe, DC= data center, ISVT= integrated vapor compression and thermosyphon.
4. Discussion

This thesis analyzed the performance of several innovative airflow management methods on the thermal environment and energy saving potentials in a high-density data center (DC). The main goal of this thesis was to ensure the safe operation and improve the cooling efficiency in DCs through the row-and-rack level airflow management and potential high-efficient phase change cooling. The model validation based on the experimental data, simulation studies and optimization analysis were carried out in Publications 1-4 to analyze the performance of the proposed methods on the thermal environment improvement. Finally, the current high-efficient cooling technologies, mainly phase change cooling, was reviewed and recommended for further study (Publication 5).

4.1 Practical implications

DCs are centralized repositories housing computer servers and associated data storage and telecommunication systems for data storage as well as data processing and transmitting. Thus, the safe operation of DCs is a prerequisite for the stable development of the information society and greatly affects the economic development. With the rapid development of ICT industry, the servers become more highly integrated with much high power, which greatly increase the power consumption and power density. When the high-density servers are operating at full capacity, if the large amount of heat caused by them is not eliminated in time, it will cause server downtime and network instability, and even devastatingly damage the servers. To a certain extent, the safe and efficient operation of DCs is one of the important factors for maintaining human, economic and social stability. To support that goal, the airflow management methods studied in this thesis have very strong practical significance.

For the DC owners and operators, the proposed methods can be applied to newly-built DCs (Publications1-4), as well as to the reconstruction of existing DCs (Publications 1&2), and can achieve a certain degree of the thermal environment improvement. The rack hotspots can be reduced by 1.5 to 2.5 K in the studied DC, which is conducive to the safe operation of the server and prolongs its service life. In general, the physical price (selling price) of a typical 2U server usually ranges from 3500-8500 euros, while the virtual value (operational value) of the server is immeasurable. Apart from equipment maintenance cost savings, applying the proposed method can achieve
Discussion

approximate 98-150 kWh electricity savings per day in a DC of the similar size (Publications 3-4). Thus, the proposed methods have strong scalability potential, and can reduce the equipment and energy costs on the premise of ensuring the safe operation for DC owners and operators.

For DC equipment manufacturers and solution providers, the proposed methods are technically easy to implement to the existing facilities (Publications 1-4). Also, the proposed methods are highly applicable to all DCs with the under-floor air distribution and cold aisle containment, while these kinds of DCs could be used by the vast majority on the market. When these methods can achieve significant thermal environment improvement and energy savings, it could be assumed that those proposed solutions could more likely rapidly capture some role in the market.

For normal internet users, the proposed methods can guarantee the operating efficiency of the servers, and the network conditions (e.g., data storage, data transmission, conversion and exchange) (Publications 1-4). As the proposed methods can reduce the outlet temperature of targeted racks, which means cooling efficiency improvement and more heat removed from operating servers. The improvement of server heat removal efficiency can greatly increase its operating efficiency, and optimize the network conditions and the servers’ lifespan.

For the DC researchers, the proposed methods prove that airflow management (especially row-and-rack level methods) is an important area for research. Novel airflow management solutions can enhance the thermal environment and energy consumption to some extent (Publications 1-4). However, considering heat exchange efficiency and future development trends, phase change cooling is one important area for further study in DCs (Publication 5).

For sustainable development, the proposed methods can help to reach low carbon targets, which alleviate the environmental challenges (e.g., climate change) caused by the energy consumption (Publications 1-5). DCs are responsible for approximate 2% of the global total electricity use, whose figure is predicted to be 5% by 2024 (Cooling working group of data center of China Refrigeration Society, 2020). It has been estimated the ICT sector was responsible for more than 2% of the global carbon emissions (Kubler et al., 2019; Whitehead et al., 2014). The proposed methods (Publications 3-4) can save approximately 98-150 kWh electricity use in a small-scale DC per day, and if they were applied to more and larger DCs, the energy savings would be considerable as well as carbon emission reduction.

For social and economic development, the safe operation of servers and DCs can ensure the stability of information network and society, and information plays an important role in social and economic development. The proposed methods are aimed at ensuring guaranteed operation of servers and achieving energy savings (Publications 1-5), which will undoubtedly promote information storage, transmission, conversion efficiency, and then promote the social and economic development.
4.2 Limitation

There are some limitations in this thesis, which are listed as follows.

- The findings of this thesis can be only applicable for the rack with heat accumulation at the middle and bottom part, as the targeted racks with high heat load are placed with more servers or high-load serves at the middle and bottom part. Thus, more research works should be done to show the effects of proposed methods on the rack with heat accumulation at other parts;

- In this study, U is the thickness unit of servers, and 1 U is equal to 0.445 m. The servers within the studied rack are all 2 U and share the same dimension, while the heat loads of the servers are relatively even. However, the server placement, size, and power vary greatly in many DCs. Thus, further works should be done on the racks with complex conditions within racks (e.g., different server dimensions, various heat loads and combinations of 1 -, 2 -, 4 -, and 7-U servers). In addition, the power of a server is basically proportional to its thickness;

- The operating state of the servers is idealized for full power operation. However, the actual operation of the servers is irregular. Thus, the effects of irregular operating load of servers should be taken into consideration in future studies.

- The setting of parameters in the simulation will inevitably deviate from the actual situation, which leads to some deviation between the simulation results and the experimental results. However, more detailed and specific on-site experimental tests can be done to validate and correct the simulation model. In addition, more research on model parameter settings should be conducted and compared to obtain more accurate DC simulation models.

4.3 Recommendations for further research

After analyzing the experimental results and simulation results, all the proposed row-or-rack level airflow management methods can improve the thermal environment to varying degrees, and some of them can achieve considerable energy savings under the premise of safe operation. Apart from the analysis of proposed airflow management methods, phase change cooling (PCC) is proposed in the thesis for further study.

The comprehensive PCC technology application in DCs was reviewed and summarized in Publication 5, while the main findings and recommendations are as follows:

- Heat pipe (HP) system has been applied as independent cooling or integrated-supplementary cooling modes at DCs with high heat density, while it has become a research hotspot in the DC high-efficient cooling area. However, the application of this technology is limited by the outdoor ambient temperature.

- Both the independent HP system and integrated vapor compression and thermosyphon system can obtain a good natural cooling effect, while fewer moving parts and wearing parts are used, and the operation stability is high. However, independent HP system application is limited by the
outdoor ambient temperature, while integrated vapor compression and thermosyphon system can handle the insufficient cooling capacity in case of high ambient temperature. Most existing performance studies on the integrated system use experimental models, and the internal flow performance parameters and geometric optimization of the system are still missing, and further simulation studies are needed.

- The selection of working fluid for the HP system lacks a unified and applicable standard. Due to factors such as price and performance, the DC HP system uses R22 and other Freon refrigerants as the refrigerant, but it cannot meet the environmental requirements. Therefore, researchers should put more efforts on studying the green working fluid.

- Two-phase immersion cooling can offer lower server operating temperatures, and reduce fan noise. However, the control of the evaporation of the cooling fluid is relatively complicated, and it is susceptible to contamination. This technology requires adjusting the design of DC around the working characteristics of the coolant, which increases DC construction costs greatly. Therefore, the system is more suitable for in-built DCs instead of existing ones.

- The cold storage system can be used as an emergency backup cold source to achieve continuous cooling in DC. For areas with time-sharing peaks and valley electricity price differences, it can significantly reduce system operating costs and improve energy efficiency. In addition, the storage time for DCs is limited, and the cooling capacity provided by the system has certain limitations.
5. Conclusions

This thesis proposed and applied four innovative row-or-rack level airflow management methods in the studied DC as well as analyzing the promising cooling methods for future study. The main goal of the thesis was to ensure the safe operation and improve the thermal environment and cooling efficiency in DCs with the proposed methods. This thesis was conducted through the on-site experimental tests, simulation studies and model validations, while by comparing and analyzing the results, the optimum cases for each airflow management method can be obtained as well as the optimal thermal environment distribution and the cooling efficiency improvement or energy saving. Firstly, an innovative airflow optimization method applying tilted servers was introduced and analyzed (Publication 1) through model validation and simulation study, and then lower-side server terminal baffles were studied through both experimental and simulation studies (Publication 2). Furthermore, a new concept, where an UFAD system with CAC was replaced by a new in-rack UFAD system, called an IR-CA was introduced and analyzed through model validation and simulation study in Publication 3, while the similar research processes were also done in Publication 4 applying innovative step-like server placement. Finally, the potentials of PCC were comprehensively reviewed and summarized for further study in the future (Publication 5).

5.1 Tilted server placement application

With the tilted server placement, the results show that the simulation model is validated by an on-site measurement, and they are in good agreement. The airflow profiles and cooling efficiency are both improved with the server inclined angle of 15°and 30°, while the more uniform thermal environment is achieved when the angle of servers is set to be 30° in the horizontal direction. In addition, the maximum and mean outlet air temperature of rack is reduced to varying degrees with the application of 30° tilted servers. Especially when the heat load of the rack is 32 kW, the maximum rack outlet air temperature is significantly reduced from the operation risk temperature of 45.3 °C to the accepted temperature level of 42 °C.

5.2 Lower-side server terminal baffle application

With the lower-side server terminal baffles, the results show that the original DC simulation model was validated by on-site experiments, while the experimental results are in good agreement with numerical results. The use of
45° angle and 8 cm terminal baffles can effectively eliminate the heat accumulation and improve the airflow distribution and cooling efficiency in DC, and decrease the temperature of the rack hotspot can reach approximately 2 K, which reduces significantly the risk of the local hotspot damages to the servers. Under this circumstance, the outlet air temperature distribution of the upper and lower parts of the rack was relatively uniform, while the maximum temperature difference reduced from 2.2 K to 0.5 K.

5.3 In-rack UFAD application
With the in-rack UFAD system, the results show that the original DC simulation model is validated by an on-site measurement, and they are in good agreement. In-rack UFAD system can improve the thermal distribution when the width of IR-CA is larger than 0.3 m. In addition, although the thermal distribution in Case 6 with 0.6 m × 0.6 m IR-CA is much improved compared to Cases 1–5 and 7, the optimal thermal distribution is achieved in Case 8 with a 0.6 m × 0.6 m IR-CA and a partition plane. Under the circumstances of Case 8, the maximum temperature drop of a rack hotspot can reach 2.4 K. When the SAT is set to 23 °C and 23.5 °C in Case 8-1 and 8-2, respectively, the thermal distribution and rack hotspot are still better than those in original DC model. However, Case 8-1 performed much better than Case 8-2 in terms of thermal distribution and rack hotspot temperature. In addition, the application of a 0.6 m × 0.6 m IR-CA and a partition plane in Case 8 with SAT of 23 °C can achieve about 98 kWh/day electricity saving in this DC.

5.4 Step-like server placement application
With the step-like server placement, the results show that the original DC simulation model is validated by an on-site measurement, while the simplified DC model is validated by the original DC model and corresponding experimental results. The application of step-like server placement can improve the thermal environment in the studied DC, while it can also largely increase the air velocity across the slots between servers at the rack bottom part. In addition, the optimum thermal environment is achieved when 0.01-m horizontal difference between adjacent servers exists. Under this circumstance, the temperature of rack hotspot can reduce by 2.5 K, while the heat accumulation at the bottom part of rack can be alleviated significantly. Considering strict-level and lenient-level requirements of server hotspots and thermal distributions, the application of 0.1 cm step-like server placement can achieve a maximum of 146 kWh and 195 kW h electricity energy saving for the CRACs per day when the SATs are set to 23.5 and 24 °C, respectively.

5.5 PCC recommendation
The free cooling technology is currently a promising and hotspot cooling method in DCs, while the PCC technology plays an important role in the free cooling application. Nowadays, the reliability and feasibility of the PCC have been proven in many research. The PCC includes sole and integrated HP system, two-phase immersion cooling, and cold storage system. The PCC is recommended for future study focus as it can make up for the cooling lack of the CRACs and achieve better thermal environment and energy saving of the CRACs
at the cost of only little energy use. In addition, when the operating powers and temperatures of computer parts are too high, the evaporating process of the PCC can take away the heat and prevent them from overheating and damage. The PCC essentially utilizes renewable energy and free cooling sources, which have been widely investigated and applied in DCs, which have been proved to be high thermal performance and energy-saving.
References


References


