Experimental Study of Airflow Designs for Data Centers

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Abstract
Growth in ICT has led to increased demands on data centers. Highly reliable operations of data centers require an efficient and robust air-conditioning system. In most data centers, circulating air enters at the floor level, and returns at the ceiling level (floor supply design). However, the authors have developed an airflow design in which air is both supplied and returned at the ceiling level (ceiling supply design). To compare the cooling characteristics of both designs, temperatures were measured in a full-scale model of a data center. For both designs, changes in cooling characteristics were evaluated in response to changes in heat density, supply-air volume, and supply-air temperature. In addition, the authors studied the effects of separating cold and hot aisles by hanging walls from the ceiling between aisles. Dimensionless indices, dimensionless rack-temperatures, and values of the rack-cooling index (RCI) were used to compare the cooling characteristics of the two designs. The authors confirm that ceiling supply design is more robust regarding changes in major design parameters compared to floor supply design.

Keywords: data centers; airflow design; temperature; cooling characteristics

1. Introduction
With widespread developments in ICT, demands on data centers have increased. Because ICT machines generate huge amounts of heat, highly efficient and reliable air-conditioning systems are indispensable to conserve energy. Moreover, heat densities of ICT machines, such as blade servers, are increasing. Therefore, highly efficient air-conditioning systems are needed to control operating costs as well as to conserve energy (Chandrakant (2003); Hannaford (2006); Rasmussen (2006)).

Improvements in airflow designs contribute to energy conservation. Nakao et al. (1994) performed experiments using a full-size model of a data center with racks that had outlets on top of them. He concluded that the most efficient airflow design supplies air at the floor level and returns it at the ceiling level. However, Nakao's study focused on airflow designs that cool the entire space in a server room. Today, most data centers use the cold-aisle/hot-aisle airflow design.

Suwa et al. (2011) performed computational fluid dynamics studies and compared the performance of cold-aisle/hot-aisle systems with a configuration that supplies air at the floor level and returns it at the ceiling level (floor supply design) and with a configuration having both supply and return on the ceiling (ceiling supply design). Suwa et al. concluded that ceiling supply design is more efficient, and in addition carried out experiments on a shrink-scale model and found the same trend.

In this study, the authors built a full-size experimental model to compare the cooling characteristics of two airflow designs: ceiling supply design and floor supply design. In addition, they performed parametric studies to understand the cooling characteristics of both designs. The major parameters were heat density, supply-air volume, supply-air temperature, and length of walls hanging above racks.

Hayama et al. (2009) proposed some dimensionless numbers to express the cooling characteristics of an airflow design. For example, the dimensionless rack-temperature can be used to quantify the cooling characteristics of a design. Moreover, the rack-cooling index (RCI) proposed by Herrlin et al. (2005) is important because rack-inlet temperature is significant in evaluating an airflow design; specifically, the purpose of an air-conditioning system is to keep the rack-inlet temperature low.

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2. Evaluation Methods
2.1 Cooling Characteristics
The authors adopt the dimensionless rack-temperature that Hayama et al. (2009) proposed. Supply-air volume divided by rack-air volume gives the dimensionless supply-air volume,
\[
\kappa_m = \frac{V}{V_m} = \frac{\theta_{1m} - \theta_{0m}}{\theta_1 - \theta_0}, \quad \kappa_m \geq 0 \tag{1}
\]
Here, \(V\) is the volumetric flow rate of supply-air [\(\text{m}^3/\text{s}\)], \(V_m\) is the volumetric flow rate of air leaving the server machine [\(\text{m}^3/\text{s}\)], \(\theta_{1m}\) is the outlet air temperature of the server machine [\(^{\circ}\text{C}\)], \(\theta_{0m}\) is the inlet air temperature of the server machine [\(^{\circ}\text{C}\)], \(\theta_1\) is the return air temperature [\(^{\circ}\text{C}\)], and \(\theta_0\) is the supply-air temperature [\(^{\circ}\text{C}\)].

Often, hot air exhausted from a hot aisle enters a cold aisle and increases the rack-inlet temperature. To avoid this phenomenon, hot aisles and cold aisles are physically separated by walls hung above the racks. In this study, the ratio of the length of the hanging walls to the gap between racks and ceiling is defined to be the dimensionless length of hanging walls,
\[
r_{hw} = \frac{L_{hw}}{L_c - L_r} \tag{2}
\]
where \(L_{hw}\) is the length of the hanging walls, \(L_c\) is the height of the server room, and \(L_r\) is the height of the racks.

The dimensionless rack-temperature is defined as the ratio of the temperature difference between supply-air and rack-inlet to the temperature difference between supply-air and return air,
\[
m_{0m} = \frac{\theta_{0m} - \theta_0}{\theta_1 - \theta_0} \tag{3}
\]
This number corresponds to the ratio in (1), which is the ratio of rack-inlet volume to air volume that comes from the hot aisle as suggested by Suwa et al. When \(m_{0m} < 1\), it becomes the ratio of the quantity of heat used for cooling at the rack-inlet to the quantity of heat supplied from the air-conditioning system. This value is comparable to CR11, which was proposed by Kato et al. (1998) to understand the characteristics of a thermal environment.

2.2 Rack-Inlet Temperature
To maintain stable operations of servers, the rack-inlet temperature must be kept within the recommended temperature. In particular, it is desirable to control rack-inlet temperatures within the range proposed by ASHRAE and NEBS. Therefore, in this paper the authors use RCI—a dimensionless number used to evaluate the rack-inlet temperature based on acceptable and recommended temperatures. RCI takes two forms. One is RCI_{hi}, which is used for temperatures above the recommended temperature; the other is RCI_{lo}, which is for temperatures below the recommended temperature. In this paper, the authors focus on higher temperatures; thus, RCI_{hi} is adopted. The definition of RCI_{hi} is given by
\[
RCI_{hi} = 1 - \frac{\sum(T_x - T_{max-rec}) r_{>T_{max-rec}}}{(T_{max-all} - T_{max-rec}) n} \tag{4}
\]
where \(T_x\) is the inlet air temperature of server machines at point \(x\) [\(^{\circ}\text{C}\)], \(n\) is the number of points, \(T_{max-rec}\) is the maximum temperature recommended by guidelines or regulations [\(^{\circ}\text{C}\)], and \(T_{max-all}\) is the maximum temperature allowed by guidelines or regulations [\(^{\circ}\text{C}\)].

Fig.1. illustrates RCI_{hi}. As the value of RCI_{hi} approaches 1.0, the area under the recommended temperature in Fig.1. decreases. In this paper, the authors adopt the values from ASHRAE: the recommended temperature is 25°C and the maximum permitted temperature is 32.2°C.

3. Full-Size Model Experiment
3.1 Outline of Full-Size Model
(1) Outline
Fig.2. shows the full-size experimental model. Section and plan views are shown in Fig.3. The structural material of the model is steel, and structures are covered with plywood to make ceiling, walls, and floor. To avoid sunlight, this model is located inside...
an experimental facility. To guarantee high heat-insulation, walls are doubled. The size of the server room is 4,500 mm × 5,450 mm, and the height is 3,000 mm. The area of the room is 20.025 m², and the volume is 60.075 m³. Two rows of mock server racks were aligned. A cold aisle is located along the center of the room, and hot aisles are located at the sides. In ceiling supply design, air is supplied from the ceiling and returned to the ceiling. In floor supply design, air is supplied from the floor and returned to the ceiling.

(2) Air Conditioning System

Table 1. shows the properties of the air-handling unit, chiller, and pumps. The air-handling unit controls the supply-air temperature with an inverter. Its maximum supply-air volume is 23,000 m³/h and its heat quantity is 72.0 kW. Dampers are installed near the supply-air inlet to supply uniform air by controlling the air volume. A diffusely spread airflow is formed by louvers to prevent hot air entering through hot aisles. The supply-air inlet does not have fans; the air-handling unit only has a fan.

(3) Mock server and server racks

Ten server racks, which are used in actual data centers, are installed as mock server racks. Fig.4. shows the inside of the room. The server racks are aligned to face the cold aisle. The height of a rack is 2,000 mm, the width is 700 mm, and the depth is 1,000 mm. The gap between the top of the racks and the ceiling is 1,000 mm because the height of the room is 3,000 mm.

To replicate the amount of heat and air volume of general servers, hair dryers and DC-fans are used. A hair dryer and fan are installed on a step. Steel panels separate the inside space in a rack into five parts to make the steps. The air volume of the hair dryer and fan is 0.092 m³/s, and the amount of heat generated is 1,192 W. In this situation, the temperature difference between rack-inlet and rack-outlet is around 13°C.

3.2 Temperature Measurements

(1) Measurement Equipment

Type-T 0.2-mm thermocouples were used to measure temperatures. Under stable and steady conditions, instantaneous values at intervals of 10 s were measured for 5 min, and averages over that time were considered as the temperature.

(2) Measurement Points

Fig.5. shows measurement points. For stable operation of servers, it is important to avoid invasion of hot air into the cold aisle. Therefore, five points at each rack inlet and outlet were measured. Each measurement point corresponds to a heat source in the racks. Measurement points for air temperatures in the aisles correspond to measurement points in the racks. To obtain air temperatures above racks, thirty measurement points were set. As basic values in the experiments, the supply and return air temperatures were also measured. The total number of measurement points was 207.

3.3 Heat Balance on the Experimental System

To confirm the heat balance on this experimental system, the authors compared the amount of heat generated and the amount of cooling heat supplied by the air-conditioning system. The amount of cooling
heat was obtained from the amount of supply-air volume and the temperature difference between supply and return air. The amount of heat generated was obtained from the amount of power consumed by the servers.

The temperature outside the experimental model was 29°C. The average supply-air temperature was 17.6°C, the average return air temperature was 29.9°C, and the supply-air volume was 14,256 m³/h. The amount of cooling heat from the air-conditioning system was 60,379 W. At the same time, the power consumed by the servers was 59.611 W. This means the air-conditioning system supplied cooling heat at 101.3% of the power consumed by the servers. Therefore, this experimental model was sufficiently insulated and maintained a heat balance.

3.4 Experimental Cases

Table 2. shows the experimental cases. Major parameters are heat density, dimensionless supply-air volume, supply-air temperature, and dimensionless length of hanging walls. These parameters are important because they affect the rack-inlet temperature. The target parameter in Case 1 was heat density, while that for Case 2 was dimensionless supply-air volume, while that for Case 3 was supply-air temperature, and that for Case 4 was dimensionless length of the hanging walls.

4. Results

The authors used dimensionless rack-temperatures to evaluate rack-inlet temperatures. Focusing on the vertical and horizontal distributions of the dimensionless rack-inlet temperature, they evaluated the effects of the major parameters. Moreover, RCI was used to comprehensively evaluate rack-inlet temperatures.

4.1 Effects of Heat Density

Fig. 6 shows the vertical distributions of dimensionless rack-temperatures for (a) ceiling supply design and for (b) floor supply design. For ceiling supply design, higher heat densities correspond to lower dimensionless rack-temperatures. In contrast, for floor supply design, higher heat densities correspond to higher dimensionless rack-temperatures. Thus, the trends differ depending on airflow design. While changes in the slopes in Fig. 6.a are almost the same for ceiling supply design, changes in slopes in Fig. 6.b differ in floor supply design. The highest dimensionless rack-temperature is 0.57 for ceiling supply design and 0.70 for floor supply design. For floor supply design in Case 0 at 2.4 kW/rack, the dimensionless rack-temperature at the top of the racks is higher than 0.7, but is lower than 0.1 at the bottom of the racks. In Case 1-2 at 6.0 kW/rack, the dimensionless rack-temperature at the top of the racks decreases to 0.52, but increases to 0.33 at the bottom.

Fig. 7. shows the horizontal distributions of dimensionless rack-temperatures for both airflow designs. For ceiling supply design, higher heat densities correspond to lower dimensionless rack-temperatures. For floor supply design, higher heat densities correspond to higher dimensionless rack-temperatures. These trends in horizontal distributions

| Case  | Unit [kW/Rack] | Heat density | Dimensionless rack-temperature | Air temperature | Supply air temperature | Dimensionless length of hanging wall |
|-------|----------------|--------------|---------------------------------|-----------------|-----------------------|------------------------------------|
| Case 0| 2.4            | 3.0          | 0.57                            | 18              | 18                    | 0.0                                |
| Case 1-1| 4.8          | 3.0          | 0.57                            | 18              | 18                    | 0.0                                |
| Case 1-2| 6.0          | 3.0          | 0.57                            | 18              | 18                    | 0.0                                |
| Case 2-1| 2.4          | 3.0          | 0.57                            | 18              | 18                    | 0.0                                |
| Case 2-2| 2.4          | 3.0          | 0.57                            | 18              | 18                    | 0.0                                |
| Case 3-1| 2.4          | 3.0          | 0.57                            | 20              | 20                    | 0.0                                |
| Case 3-2| 2.4          | 3.0          | 0.57                            | 22              | 22                    | 0.0                                |
| Case 3-3| 2.4          | 3.0          | 0.57                            | 24              | 24                    | 0.0                                |
| Case 4-1| 2.4          | 3.0          | 0.57                            | 18              | 18                    | 0.2                                |
| Case 4-2| 2.4          | 3.0          | 0.57                            | 18              | 18                    | 0.4                                |
| Case 4-3| 2.4          | 3.0          | 0.57                            | 18              | 18                    | 0.6                                |
| Case 4-4| 2.4          | 3.0          | 0.57                            | 18              | 18                    | 0.8                                |
| Case 4-5| 2.4          | 3.0          | 0.57                            | 18              | 18                    | 1.0                                |
of dimensionless rack-temperatures are the same as those in the vertical distributions. Dimensionless rack-temperatures decrease as heat densities increase. However, dimensionless rack-temperatures increase at most points in floor supply design. Especially, dimensionless rack-temperatures increase more at the edges of rack rows than at the centers of rack rows.
Fig. 8. shows $RCl_h$ at selected values of the heat density. For all cases in ceiling supply design, values of $RCl_h$ are almost 1, even though the heat density changes. However, for all cases in floor supply design, the values of $RCl_h$ are approximately 0.9.

Therefore, in ceiling supply design, an increase in heat density improves cooling characteristics; however, in floor supply design, cooling characteristics deteriorate with the increases in heat density. The reason is probably that, in ceiling supply design, the directions of buoyancy of cold air and supply-air are the same, but in floor supply design, the directions differ.

4.2 Effects of Supply-air Volume

Fig. 9. shows the vertical distributions of dimensionless rack-temperatures. For ceiling supply design, dimensionless rack-temperatures decrease uniformly with the increases in dimensionless supply-air volume. However, for floor supply design, changes in dimensionless rack-temperatures are not uniform. In ceiling supply design, dimensionless rack-temperatures at the top of the racks are, in all cases, around 0.55, and those at the bottom, around 0.2. In floor supply design, dimensionless rack-temperatures decrease from 0.77 to 0.33 at the top of the racks as the dimensionless supply-air volume decreases. However, they increase from 0.10 to 0.28 at the bottom of the racks. This is believed to result from the increases in the flow rate of supply-air. For floor supply design in Case 2-2 at $κ_m = 1.4$, cold air from the floor-supply reaches the top of the cold aisle and decreases the rack-inlet temperatures at the top, but cold air supplied at high flow-rates draws hot air at the bottom of the racks. The highest dimensionless rack-temperatures were 0.60 for ceiling supply design and 0.77 for floor supply design.

Fig. 10. shows the horizontal distributions of dimensionless rack-temperatures. In both airflow designs, the dimensionless rack-temperatures decrease as the dimensionless supply-air volumes increase. However, the amount of change in ceiling supply design is less than 0.1, while that in floor supply design is more than 0.1.

Fig. 11. shows $RCl_h$ for selected values of the dimensionless supply-air volume. In ceiling supply design, $RCl_h$ is 1.0 in all cases, but in floor-ceiling design, it decreases from 1.00 to 0.61. This means that in floor-ceiling design, it is necessary to supply enough cool air to keep rack-inlet temperatures below the recommended temperature; however, this is not necessary in ceiling supply design.

As a result, in ceiling-floor design, reduction in dimensionless supply-air volume has little effect on dimensionless rack-temperatures and $RCl_h$; however, in floor supply design, reduction in dimensionless supply-air volume adversely affects cooling characteristics and decreases $RCl_h$.

4.3 Effects of Supply-air Temperature

Fig. 12. shows the vertical distributions of dimensionless rack-temperatures. In both airflow designs, there are no clear trends concerning changes. Even though there are some differences among the distributions, most cases have similar values. This may be because buoyancy does not change even though the supply-air temperature increases. Rise in supply-air temperature uniformly increases all temperatures in the experimental model. In other words, the difference between supply-air temperature and the average temperature of the experimental model does not change. Therefore, the dimensionless rack-temperatures of each airflow design are almost the same, even though the supply-air temperature changes.

Fig. 13. shows the horizontal distributions of dimensionless rack-temperatures. Along with vertical distributions of dimensionless rack-temperatures, the horizontal distributions in each airflow design are almost the same and there are no clear trends concerning change. This is probably explained by the same reasoning used above for the vertical distributions.

Fig. 14. shows $RCl_h$ at selected values of supply-air temperature. In both airflow designs, as the supply-air temperature increases, $RCl_h$ decreases. In the base case with $θ_s = 18$, the value of $RCl_h$ for floor supply design is smaller than that for ceiling supply design. This trend is the same for other cases with higher supply-air temperatures.

Therefore, supply-air temperature does not affect dimensionless rack-temperatures because buoyancy does not change even though the supply-air temperature changes. Values of $RCl_h$ decrease in both airflow designs as the supply-air temperature increases. In ceiling supply design, $RCl_h$ is higher than it is in floor supply design.

4.4 Effects of Length of Hanging Walls

Fig. 15. shows vertical distributions of dimensionless rack-temperatures. In ceiling supply design, dimensionless rack-temperatures decrease with the increases in the dimensionless length of hanging walls. In floor supply design, dimensionless rack-temperatures do not show a clear trend of change with the increases in the dimensionless length of hanging walls. In floor supply design, the difference in changes is smaller than those in ceiling supply design.

Fig. 16. shows horizontal distributions of dimensionless rack-temperatures. Just as for vertical distributions in ceiling supply design, as the dimensionless length of hanging walls increases, dimensionless rack-temperatures decrease. Dimensionless rack-temperatures at the centers of rack rows decrease more than at the edges of rack rows. For floor supply design, horizontal distributions of rack-temperatures do not show obvious trends with the increases in length of hanging walls. Fig. 17.
shows RCI_{hh} at selected values for the dimensionless length of hanging walls. For all cases in ceiling supply design, the values of RCI_{hh} are 1.0, which means that all rack-inlet temperatures are below 25.0°C. However, for floor supply design, no case reaches 1.0. The dimensionless length of hanging walls for Case 4-1 at rhw = 0.2 is the lowest at 0.72, and that for Case 4-5 at rhw = 1.0 is the largest at 0.92.
Therefore, in ceiling supply design, an increase in the dimensionless length of hanging walls reduces the dimensionless rack-inlet temperature; however, in floor supply design, there is no apparent trend with the increases in dimensionless length of hanging walls. In ceiling supply design, all cases have $RC_{hi} = 1.0$, but in floor supply design, the lowest $RC_{hi}$ is 0.72.

5. Conclusions
To evaluate the cooling characteristics of different airflow designs, the authors built a full-size experimental model and performed experiments to study major parameters. The parameters were heat density, dimensionless supply-air volume, supply-air temperature, and dimensionless length of hanging walls. The authors found the following behaviors.

- **Rise in heat density improves the cooling characteristics of ceiling supply design but degrades the cooling characteristics of floor supply design.**
- **Reduction in dimensionless supply-air volume degrades the performance of both airflow designs; however, the cooling characteristics of floor supply design are affected more than those of ceiling supply design.**
- **Changes in supply air temperature do not affect the cooling characteristics of either airflow design.**
- **Rise in length of hanging walls improves the cooling characteristics of ceiling supply design but shows no apparent trend regarding floor supply design.**

The authors also applied $RC_{hi}$ to evaluate rack-inlet temperatures.

- In all cases, the values of $RC_{hi}$ for ceiling supply design are larger than those for floor supply design.
- The difference between the values of $RC_{hi}$ for floor supply design and ceiling supply design is approximately 0.4 in Case 2-1 with $k_m = 1.0$.

As ICT continues to develop, heat densities can be expected to increase. We can also expect that supply-air volumes will decrease, supply-air temperatures will increase, and that hanging walls will be used in future efforts to conserve energy. The authors have confirmed that the cooling characteristics of ceiling supply design are better than those of floor supply design; in addition, ceiling supply design exhibits better robustness regarding changes in the major parameters.

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