Analysis of Cooling Characteristics in Datacenter Using Outdoor Air Cooling

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Abstract: Advancement of the information society has proceeded with the development of information and communication technology, and a demand on a data center has increased. In such a situation, the number of servers is increasing in a data center. Thus, the heat density in a data center is much higher than that of usual offices. And typically, almost 40% of the total power consumption is used for cooling servers in a data center. Thus, cooling effectiveness is one of the most important factors in evaluating the value of the data center. The data center taken up in this paper is located in Ishikari, where is a cold district in Japan. Using the cool outdoor air for cooling servers helps us to cut the power consumption for cooling. This paper first assesses the efficiency of Ishikari data center measuring the temperature of seven parts in a building where the cooling air flowing. Second, this paper describes the most efficient method for the operation and estimates 1.11 of PUE (power usage effectiveness).

Key words: Data center, cooling characteristics, outdoor air cooling, PUE.

Nomenclature

N	Scale (-)
Subscripts θ	Temperature (°C)
H_{fan}	Exhaust fan load (W)
H_m	Heat load (W)
H_w	Heat loss on wall (W)
V	Air conditioning supply-air volume (m ³ /s)
V_{fm}	Effective air conditioning supply-air volume in an equipment (m ³ /s)
V_{fr}	Effective air conditioning supply-air volume in a room (m ³ /s)
V_m	Equipment ventilation volume (m ³ /s)
V_{OA}	Outdoor air introducing volume (m ³ /s)
V_{um}	Ineffective air conditioning supply-air volume in an equipment (m ³ /s)
V_{ur}	Ineffective air conditioning supply-air volume in a room (m ³ /s)
Γ	Recirculation (-)
θ_0	Air conditioning supply-air temperature (°C)
θ_{0m}	Equipment inlet air temperature (°C)
θ_1	Air conditioning return-air temperature (°C)
θ_{1d}	Exhausted air temperature (°C)
θ_{1m}	Equipment outlet air temperature (°C)

θ_{mix}	Mixing air temperature (°C)
θ_{OA}	Outdoor air temperature (°C)

1. Introduction

With the rapid development in information technology in recent years, the energy consumption of datacenters follows the increasing proportion of total energy consumption in society as a whole [1-3]. In normal datacenters, air conditioning systems have to operate year round to treat a large sensible heat load. There are many efficient datacenters located in cold areas that use outdoor air cooling systems [4, 5]. These systems play an important role in reducing the processing loads on servers. Many researchers have studied indoor air conditioning systems. Furihata et al. [6] investigated the temperature of 100 points in sever rooms and found the characteristics which show a performance of datacenters. Hayama [7] proposed a method for estimating the energy consumption of air conditioning systems using the relation between the cooling characteristics and energy consumption.

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However, there is no specific control method for server rooms. It is important to determine the efficiency and characteristics of an outdoor air cooling system in detail. Therefore, we studied a datacenter located in Ishikari, a cold area in Hokkaido, Japan, which cools the electrical devices by outdoor air. First, we assessed the efficiency of this system by measuring the temperature of each area in the building. We then analyzed the cooling characteristics of this system by using a model for the system. Finally, we estimated the energy consumption and determined a proper management method for more efficient operation.

2. Ishikari Datacenter

2.1 Summary of Facilities

The datacenter we assessed is located in Ishikari, Hokkaido, Japan. Fig. 1 shows the building's facade, Table 1 lists the building specifications, and Fig. 2 illustrates a plan of a server room. In this datacenter, outdoor air comes into a server room from a wall. Fig. 3 shows a section of a server room. To stabilize the airflow volume, every rack has a fan on the surface of their ceiling to provide the necessary amount of cooling air.

2.2 Actual Measurements

We measured the temperatures at seven points around a server room from May 2012 to May 2013. Table 2 lists the details of the points and Fig. 4 illustrates where those points are located.

2.3 Temperature Distribution

Fig. 5 shows a boxplot of the temperature distribution obtained from the actual measurement. As a reference, there are recommended or allowable conditions in server rooms, as shown in Table 3, which were determined by ASHRAE (American Society of Heating Refrigerating and Airconditioning Engineer) (2011).

We can see that the blowout temperatures are relatively stable. They are within the limit of the ASHRAE (2011) [8] recommended temperature range

Fig. 1 The building facade.

Table 1 Specification of building.

	Items	Room A	Room B
Outline	Floor space	302.1 (m ² /room)	
	Setting condition	ASHRAE (2011)	
Air flow system	Supply air system	Wall	Ceiling
	Exhaust air system	Exhaust tower	
Air conditioner	Fan	FU × 4	AHU × 4
	Power	12.6 × 4 (kW)	33.0 × 4 (kW)
Exhaust fan	Power consumption	1.06 × 16 (kW/room)	
Chiller	Number of systems	1	
	Cooling capacity	1,617 (kW)	
	Power consumption	272 (kW) = 250 (kW) (Chiller) + 55 × 4 (kW) (Cooling tower)

Fig. 2 The plan of the server room.

Fig. 3 The section of the server rooms.

Table 2 Measurement detail.

Subject	Point	Place	Number	Time span	Term
Temperature	Outdoor	T1	4	1 day	2012/1/1-2013/7/31
	Mixing	T2	2		
	Supply air	T3	3		
	Inlet air	T4	100		
	Outlet air	T5	100		
	Exhausted air	T6	100		
	Return air	T7	4

Fig. 4 The section of the datacenter.

Fig. 5 Temperature distribution.

Table 3 Allowable and recommended condition (ASHRAE 2011).

	Recommended	Allowable
Low temperature	18 °C	15 °C
High temperature	27 °C	32 °C
Low humidity	DP 5.5°C	RH 20%
High humidity	RH 60%, DP15 °C	RH 80%, DP21 °C

of 18-27 °C (dew point). However, there is a drop in temperature between the exhaust ports of servers (T5) and return air (T7). This means that there is a large supply of cooling air that does not pass through servers but goes directly to the spaces between the roof and ceilings. We call this phenomenon “short circuit”. Because of the short circuit, there is a small gap in temperatures between supply air and return air. Moreover, there is a rise in temperature between supply air (T3) and the suction ports of servers (T4). This means that there is some cooling air passing through servers that comes back to the server room. We call this phenomenon “recirculation”.

3. Cooling Characteristics

3.1 Analytical Model

To understand the cooling characteristics, we constructed the model illustrated in Fig. 5. Using this model, we can see how large the short circuit and recirculation are during operation. The meaning of signs is listed in Nomenclature. Table 4 lists the definitional equations of cooling characteristics. These equations explain how cooling air is supplied properly, irrespective of the change in outdoor temperatures.

We now discuss the air flow. First, an air conditioner supplies cooling air (temperature = θ_0 (°C), volume = V (m³)) to server rooms. Then, the supplied air (V (m³)) separates into effective supplied air in room (V_{fr} (m³)), which goes above the server racks, and ineffective supplied air (V_{ur} (m³)), which exhausts before reaching the server racks. The temperature of V_{fr} (m³) rises to θ_{1d} , called the temperature of exhausted air, while V_{fr} passes through the heating servers (we define the heat load as H_m (m³) and heat loss as H_w (m³)). After that, V_{fr}

Fig. 5 Model for cooling characteristics.

Table 4 Cooling characteristics.

Ventilation flow ratio	Room exhaust efficiency
$\kappa_m = \frac{V}{V_m} = \frac{\theta_{1m} - \theta_{0m}}{\theta_1 - \theta_0}$	$\eta_{V_r} = \frac{V_{fr}}{V} = \frac{\theta_1 - \theta_0}{\theta_{1d} - \theta_0}$
Equipment exhaust heat efficiency	Exhaust heat efficiency
$\eta_{V_m} = \frac{V_{fm}}{V_{fr}} = \frac{\theta_{1d} - \theta_0}{\theta_{1m} - \theta_0}$	$\eta_V = \frac{V_{fm}}{V} = \eta_{V_r} \eta_{V_m}$
Equipment inlet air temperature ratio	Equipment outlet air temperature ratio
$m_{0m} = \frac{\theta_{0m} - \theta_0}{\theta_1 - \theta_0} = \frac{\gamma}{\eta_V}$	$m_{1m} = \frac{\theta_{1m} - \theta_0}{\theta_1 - \theta_0}$
Exhausted air temperature ratio	Recirculation ratio
$m_{1d} = \frac{\theta_{1d} - \theta_0}{\theta_1 - \theta_0}$	$\gamma = \eta_V m_{0m}$
Fresh air introduction ratio	Relation equation
$q = \frac{V_{OA}}{V} = \frac{\theta_1 - \theta_{MIX}}{\theta_1 - \theta_{OA}}$	$\gamma = -\eta_V \kappa_m + 1$

is mixed with V_{ur} (temperature = θ_0) and the temperature of V_{fr} rises to θ_1 —the temperature of return air. Moreover, the inlet temperature of servers (θ_{0m} (°C)) rises to θ_{1m} (°C), while the amount of ventilation in

servers (V_m) passes through the servers; their heat load is H_m (m³). Then the air flowing through the servers is exhausted. Next, we define the recirculation ratio (γ). This means how much air is coming back to the servers after passing through them. If the ratio is high, it is inefficient. The amount of air coming back to the servers can be expressed as γV_m .

The supplied air passing through a ceiling chamber (V (m³)) separates into the exhausted air and return air going to the mixing chamber. The return air is mixed with fresh air (temperature = θ_{OA} (°C), volume = V_{OA} (m³)), then the temperature becomes θ_{mix} (°C). Finally, the mixed air flows into the air conditioner.

3.2 Cooling Characteristics

Each characteristic calculated with the obtained temperature data is shown in Fig. 6. Basically, every temperature ratio changes in proper order. However, gaps among the temperature ratios are small, so a small amount of recirculation or short circuit may exist. Until November 2012, the exhausted air temperature ratio was lower than the equipment-exhausted air temperature ratio.

Thus, we put a baking sheet on top of the operational server racks so as not to let the air exhaust without passing through the equipment. After we placed the sheet in November 2012, the exhausted air temperature ratio exceeded the equipment-exhausted air temperature ratio. This means the exhausted air temperature became higher than the equipment-exhausted air temperature regardless of the outside temperature.

After all, the short circuit problem in equipment—the phenomenon in which supplied air is exhausted without passing through equipment—was addressed by using the baking sheet. Also, the recirculation ratio (γ) and equipment-inlet air temperature ratio decreased after November 2012. This

is because the placement rate in the server racks increased. Therefore, there is no relation between using a baking sheet and improvement in γ . Thus, we will have to consider capping the operational parts in server racks to reduce γ .

3.3 Cooling Characteristics Evaluation

To quantitatively verify the effect of setting a baking sheet on top of operating servers, we compared the cooling characteristics between June–July 2012 and June–July 2013, as shown in Table 5. The heat-exhaust efficiency (ηV) in 2013 (0.72), was much higher than that in 2012 (0.5). In particular, the room heat-exhaust efficiency (ηV_r) improved from 0.57 to 0.71, so the

Fig. 6 Change of the cooling characteristics.

Table 5 Results of cooling characteristics.

	June-July 2012		June-July 2013	
	Room A	Room B	Room A	Room B
Ventilation flow ratio	1.14	0.68	0.97	0.88
Room exhaust heat efficiency	0.57	0.59	0.71	0.67
Equipment exhaust heat efficiency	0.88	1.15	1.00	1.13
Exhaust heat efficiency	0.50	0.68	0.72	0.76
Equipment inlet air temperature ratio	2.01	1.49	1.41	1.50
Equipment outlet air temperature ratio	1.78	1.72	1.41	1.50
Exhausted air temperature ratio	0.43	0.55	0.30	0.34
Recirculation ratio	0.43	0.55	0.30	0.34

baking sheet seemed to work well. We can say that there seems to be a small amount of short circuit because the equipment heat-exhaust efficiency (ηV_m) also improved.

The equipment inlet temperature ratio in 2013 was lower than that in 2012 by 40%. This means that the gap in temperature between supply air (θ_0) and equipment-inlet air (θ_{0m}) decreased and the recirculation problem was mitigated. The ventilation flow ratio (κ_m) decreased from 1.14 to 0.97. The reason of this decrease is that the number of servers decreased and the amount of supply air needed also decreased.

4. Efficient Operation

4.1 Relation between Ventilation Flow Ratio and Power Consumption

We estimated power consumption in relation to κ_m with the method for estimating power consumption established by Futawatari [3]. We set parameters for the placement of a baking sheet, exhaust-air efficiency, and γ against κ_m and θ_0 and analyzed the characteristics of power consumption.

We determined the behavior of power consumption and found the best κ_m . Table 6 lists the setting conditions of the analysis.

4.2. Ideal Ventilation Flow Ratio

Fig. 7 shows the estimation of power consumption

Table 6 Analysis condition.

	Case A	Case B	Case C	Case D
Setting equipment inlet temperature (°C)	18.0	18.0	27.0	27.0
Baking sheet	Non-setting	Setting	Non-setting	Setting
Analysis time (h)	24 (h) $\times$ 365 (day) = 8,760 (h)			
Heat load (kWh/h)	188			
Equipment quantity of ventilation (m ³ /s)	48.2

Fig. 7 Estimation of power consumption against ventilation flow ratio in a year.

against κ_m in one year for each condition in Table 6.

This graph illustrates that there is an exponential relation between power consumption and κ_m .

Thus, to maximize efficiency, it seems best to reduce κ_m . However, a low κ_m is detrimental for proper operation. If κ_m is too low, the circulation ratio increases. A high circulation ratio causes rapid rise in θ_{0m} , which leads to server breakdown. Moreover, when κ_m is too low, θ_0 decreases, which is also detrimental for servers. In this section, we discuss the most efficient κ_m . To do this, we have to consider two factors: θ_0 and γ .

4.1.1 Supply Air Temperature

Fig. 8 illustrates the relation between estimated power consumption and θ_0 against κ_m . According to ASHRAE, the lowest θ_0 within the allowable range is 15 °C [1]. Thus, θ_0 should be higher than 15 °C. Then

Fig. 8 Power consumption and supply air temperature against ventilation flow ratio in a year.

the best κ_m is when θ_0 is 15 °C, which is 0.87 in pattern A and 0.67 in pattern B. However, we could not determine the best value, especially in patterns C and D, because even the lowest κ_m does not result in a θ_0 below 15 °C. The reason is that θ_{0m} is so high that θ_0 would not be outside the allowable range.

As a whole, we can say that it is better for efficient operation to set a high θ_{0m} . When κ_m is 1.0, the power consumption is 483 MWh/year in pattern A, 479 MWh/year in pattern B, 261 MWh/year in pattern C, and 249 MWh/year in pattern D. To maximize efficiency and determine the most appropriate κ_m in patterns C and D, we considered the allowable range of γ .

4.1.2 Consideration of Recirculation Ratio

We calculated and led the allowable range of γ by considering the defined equation for cooling characteristics and the allowable range of supply air defined by ASHRAE [1]:

$$18 \leq \frac{\theta_{0m} - \theta_{1m} \cdot \gamma}{1 - \gamma} \leq 27 \quad (1)$$

The yearly average difference between equipment-exhaust air temperature and θ_{0m} is 4.8 °C. With this value, we computed the allowable range of γ .

(1) θ_{0m} was set to 18 °C (for patterns A and B):

$$0 \leq \gamma \leq 2.2 \quad (2)$$

(2) θ_{0m} was set to 27 °C (for patterns A and B):

$$0 \leq \gamma \leq 0.6 \quad (3)$$

Considering these allowable ranges, we show the relation between power consumption and γ with κ_m in Fig. 9.

In patterns A and B, we already obtained the proper values of 0.87 and 0.67. We can confirm that these values are safe by the graph shown as Fig. 8.

Fig. 9 Power consumption and recirculation ratio against ventilation flow ratio in a year.

Table 7 Results of Analysis

Pattern	κ_m	Power consumption (MWh/year)	PUE	ICT power consumption (MWh/year)
A	0.87	483	1.29	$188 \text{ (kWh/h)} \times 8,760 \text{ (h)} \times 10^{-3} = 1,646 \text{ (MWh/year)}$
B	0.67	349	1.25	
C	0.67	184	1.12	
D	0.51	175	1.11	

We determined the best κ_m in patterns C and D, which is 0.67 in pattern C, and 0.51 in pattern D. The results of the analysis are listed in Table 7. We obtained the most effective operation in pattern D. The θ_{0m} was set to 27 °C, after placing a baking sheet, κ_m was 0.51, and the annual power consumption was 175 MWh/year.

Therefore, PUE (power usage effectiveness) can be estimated as 1.11, which is better than the measured

PUE of 1.20. The PUE is equal to the entire power consumption in the datacenter divided by the power consumption for cooling heated servers. Thus, the PUE 1.11 means that only 11% of all the power is used for cooling equipment.

5. Conclusions

We presented the cooling characteristics of a datacenter located in Ishikari, Hokkaido, Japan. This datacenter, which uses an outdoor air cooling system, can take in fresh air to cool servers. We assessed the cooling characteristics based on temperature measurements in the building and server rooms and determined a more effective operation method for cooling servers.

We first measured the temperature of seven points in

a building through which the cooling air from outside passes. We then found two problems in the server rooms: short circuit and recirculation. To solve these problems, we placed baking sheets on top of server racks that were not in operation. The baking sheets worked well to slow down the supplied air from leaking out into the ceiling chamber. We also constructed a model to determine the cooling characteristics in detail regardless of outdoor temperature. We defined the cooling characteristics as equations. From those equations, we found that there were still short circuit and recirculation.

We determined the most efficient operation method. We first set four patterns that differed in the temperature of equipment-inlet air and whether banking sheets were used. We estimated the power consumption in a year against the ventilation flow ratio. From that graph, we found there is an exponential relation between power consumption and ventilation flow ratio. It seemed to be the most appropriate to reduce the ventilation flow ratio to maximize efficiency.

However, too low a ventilation flow ratio causes equipment breakdown because the supply-air temperature decreased to below 15 °C, the lowest temperature of supply air in the allowable range. Also, low ventilation flow ratio increases the recirculation ratio. Under a high recirculation ratio, servers cannot be cooled enough to be safe.

Thus, we considered these two factors to determine the most effective operation method. In pattern D and with a ventilation flow of 0.51, we showed that this datacenter can be under control with an annual power consumption of 175 MWh/year for cooling servers and PUE 1.11. However, further efforts are still required to consider the relation between the failure rate of servers and supply-air temperature. As the effective way of

operation is presented in this paper, further efforts are still required to consider the relation between the failure rate of servers and the temperature of supply air.

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