PERFORMANCE OF RADIANT COOLING SYSTEM INTEGRATED WITH ICE STORAGE

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ABSTRACT

The purpose of this study is designing a hybrid system which is a combination of radiant cooling and low temperature air conditioning integrated with ice storage system. Also this evaluates the performance of the system.

We compared energy amount with annual operating costs of conventional air conditioning system and this radiant air conditioning system.

1. INTRODUCTION

Electricity consumption by air conditioning accounts for some 50% of the total power consumption in office building. It is a primary factor in the increase of the peak in power demand, particularly in summer. In this regard, aiming to level the load, both the power industry and the government are making efforts to promote thermal storage system for air conditioning. Generally due to the fact that an air conditioning system with thermal storage requires a higher initial investment, it stands at a disadvantage on an initial cost basis when compared with a conventional air conditioning system. Therefore, in order to promote thermal storage system, we must develop a better secondary air conditioning system which can make the most of the features of thermal storage. A comprehensive evaluation of such a system is also needed to verify both its advantages and disadvantages fairly.

In this study two systems are combined, namely, the ice storage system effective for load leveling, and the radiant cooling system for producing a better thermal environment. Thus, we propose an air conditioning system which incorporates the merits of both systems to the maximal extent. We attempt to comprehensively evaluate the system from the perspectives of load leveling, energy saving, and economy.

2. EXPERIMENTS OF PERFORMANCE EVALUATION

2.1. METHODS

2.1.1 Chamber

We designed a chamber in which the temperature and humidity could be controlled (fig.1). Measurements of the thermal environment and experiments with subjects were carried out in the chamber. Radiant cooling panel using chilled water as a refrigerant is set up, and the temperature of chilled water can be controlled. Room temperature and humidity are controlled by air conditioning.

Table 1 indicates environmental parameters from 8

Table 1 Environmental Parameters of 8 cases And Data of Measurements

		Parameters			Data of Measurements					
No.	System	Room Temperature [℃]	Humidity [%]	Ceiling Surface Temperature [℃]	Room Temperature [℃]	Humidity [%]	Ceiling Surface Temperature [℃]	MRT [℃]	OT [℃]	Air Velocity [m/s]
1	Convectional Air Conditioning	26	60	-	25.8	50.8	28.5	26.4	26.1	0.12
2		28	40		27.6	42.6	29.9	27.9	27.8	0.13
3		26	50		25.1	51.6	27.7	25.6	25.4	0.11
4	Radiant Cooling System	28	40	20	27.3	42.8	20.2	26.5	26.9	0.08
5				22	27.9	42.7	22.0	27.2	27.5	0.08
6				24	27.9	42.3	23.5	27.4	27.6	0.08
7		29	40	20	29.0	42.6	20.7	28.0	28.5	0.09
8		28	50	20	28.1	47.4	20.4	27.1	27.6	0.09

MRT: Mean Radiant Temperature
OT : Operative Temperature

cases. Test case from 1 to 3 are carried out under convectional air conditioning system(convectional AC), 4 to 8 are carried out under radiant cooling systems. In this report, convectional AC means conventional air conditioning system.

2.1.2 Experiments with subjects

Methods of experiments with subjects are as follows ;
1. 2 subjects per experiment enter the chamber
2. experiments begin 30 minutes after entering the chamber
3. subjects fill out questionnaires for 10 minutes, in total 7 times
4. Each case consists of 12 subjects

Table 2 presents scales of comfort and thermal sensation on the questionnaires. Thermal environments are measured during experiments with subjects.

2.2. RESULTS OF EXPERIMENTS

2.2.1. Room environments

Table 1 presents a list of the average conditions maintained during each of the 8 tests on radiant cooling systems or convectional AC.

Air velocity under radiant cooling is lower than under convectional air conditioning. Under convectional AC, mean radiant temperature (MRT) is higher than room temperature. This is because the ceiling surface temperature is higher. On the other hand, under radiant cooling systems, effect of the cooling panel makes MRT more than 0.5℃ lower than room temperature. Difference of operative temperature (OT) between case 1 and case 4 is 0.8℃.

Fig.2 presents the vertical temperature distribution. Top and bottom difference of air temperature is large, and ceiling surface temperature and floor surface temperature are high in Convection. Top and bottom difference of air temperature is small, and in addition to ceiling surface temperature, floor surface temperature is high, too.

Fig.3 presents the pictures of thermal camera in case of theraml environment.

In fig.3(a), there are many spots. In fig.3(b), tempareture distribution is stable, and Lighting Load is processed enough, and wall side temperature is lowered by effect of mutural radiation.

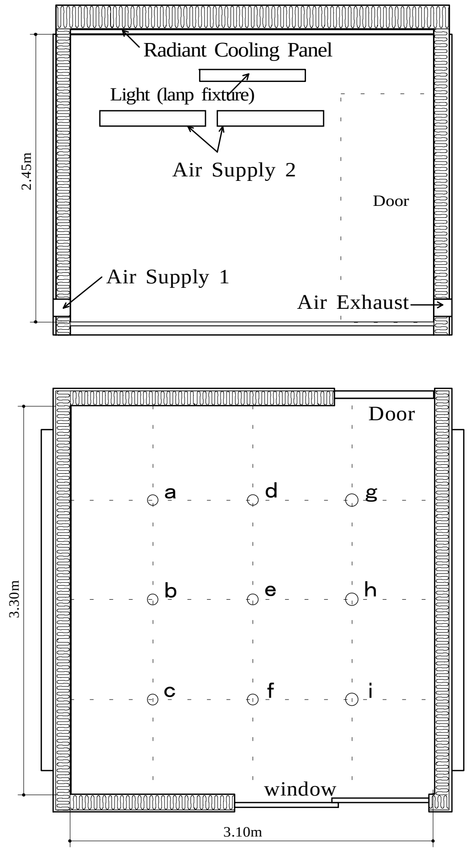


Fig. 1 Chamber

Table2 7-Points Scale of Comfort And Thermal Sensation

	Comfort	Thermal Sensation
+3	Very Comfortable	Hot
+2	Comfortable	Slight Hot
+1	Slight Comfortable	Warm
0	Neutral	Neutral
-1	Slight Uncomfortable	Cool
-2	Uncomfortable	Slight Cool
-3	Very Uncomfortable	Cold

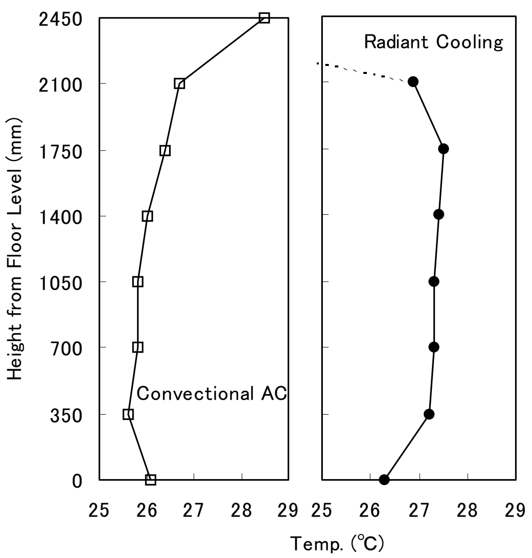
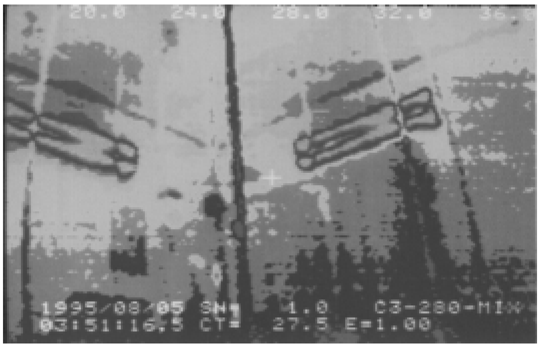
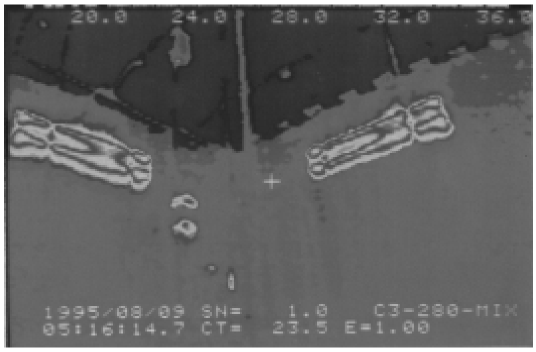


Fig.2 Vertical temperature distribution



(a) Convectional AC



(b) Radiant Cooling

Fig.3 Pictures of Thermal Camera

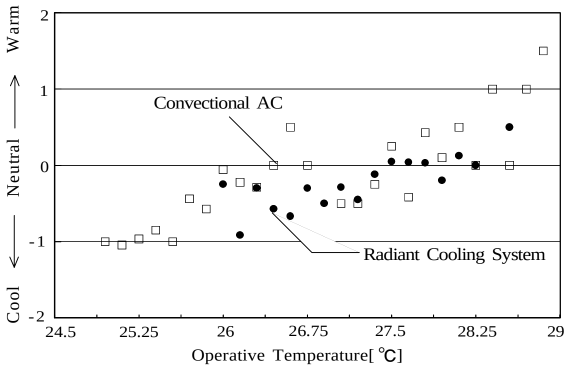


Fig. 4 Relation between OT and Thermal Sensation

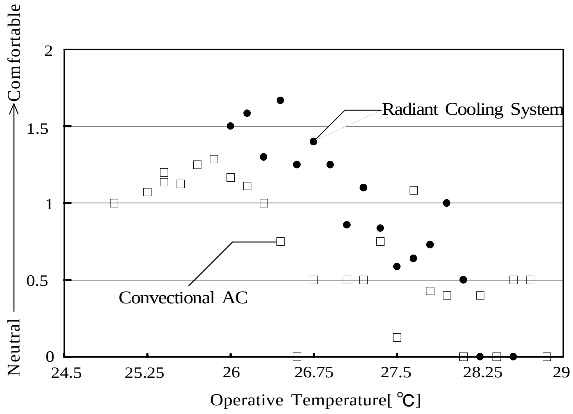


Fig. 5 Relation between OT and Comfort

2.2.2. Experiments with subjects

Fig.4 presents the relation between OT and Thermal Sensation Sense, and fig.5 presents the relation between OT and comfort. In fig.4, there is little difference between the cooling system in Thermal Sensation Sense. In fig.5, most comfortable temperature under convectional AC is 25.75～26.9℃.On the other hand, under radiant cooling system temperature ranging between 26.6～26.9℃ can produce the same comfort as convectional AC. It is shown that radiant cooling systems at OT 26.0～26.6℃ produce more comfort than any other system.

2.3. CONSIDERATION

The reason why radiant cooling systems can produce more comfort than convectional AC at equal OT is due to 1) the effect of radiation from the surface of the subject (skin-clothing) to the ceiling panel, and 2) the effect that low humidity makes human environmental temperature lower.

Equivalent OT to the most comfortable OT under convectional AC according to the experimental results is 27.0℃. We estimate the optimum OT is as follows; 26.5℃ if taking much of human comfort, 27.0℃ if taking into consideration energy-saving. Room condition at OT 27.0℃ is supplied at room temperature 27.5℃ if the ceiling surface temperature is 20～22℃.

3. SIMULATION

3.1. HYBRID SYSTEM

We suggested a hybrid system which is a combination of radiant cooling and low temperature air conditioning integrated with ice storage system (fig.6). A feasibility analysis determining whether this system can produce the optimum condition estimated in the experiments is simulated.

Ice storage system provides chilled water (1～2℃) to an air conditioner. The air conditioner supplies low temperature and low humidity air into the ceiling. The rectangular steel air-duct isn't installed in the ceiling, and conditioned air is supplied to ceiling plenum. The conditioned air cools the ceiling surface, then is blown into the room.

3.2. METHOD OF CALCULATONS

The heat transfer from the conditioned air blown by the air conditioning system to the ceiling surface should be expressed as

$$q_f = A \cdot K_f(\theta_o - \theta_c) \quad \cdots(1)$$

The heat flow when temperature of the conditioned air changes $\theta_i \rightarrow \theta_0$ should be expressed as

$$q_f = C_p \cdot \gamma \cdot V(\theta_i - \theta_o) \cdots(2)$$

adding (1) into(2):

$$q_f = K_{xc} \cdot \theta_o - K_{xc} \cdot \theta_c \cdots(3)$$

where $K_{xc} = \dfrac{1}{\left(\dfrac{1}{A \cdot K_f} + \dfrac{1}{C_p \cdot \gamma \cdot V}\right)}$

Calculations of sensible heat are carried out using the equation given below,

$$\begin{bmatrix} A_1\alpha_r(1-g_{11}) + A_1\alpha_c + A_1Z_0 + K_{xc} & -A_1\alpha_r g_{12} & \cdots\cdots & -A_1\alpha_r g_{16} & -A_1\alpha_c \\ -A_2\alpha_r g_{21} & A_2\alpha_r(1-g_{22}) - A_2\alpha_c + A_2Z_0 & \cdots\cdots & -A_2\alpha_r g_{26} & -A_2\alpha_c \\ \cdot & \cdot & & \cdot & \cdot \\ \cdot & \cdot & & \cdot & \cdot \\ \cdot & \cdot & & -A_5\alpha_r g_{56} & -A_5\alpha_c \\ -A_6\alpha_r g_{61} & -A_6\alpha_r g_{62} & & A_6\alpha_r(1-g_{66}) + A_6\alpha_c + A_6Z_0 & -A_6\alpha_c \\ A_1\alpha_c + K_{xc} & A_2\alpha_c & & A_6\alpha_c & \sum A_j\alpha_c - C_p\gamma V - K_{xc} \end{bmatrix}$$

$$\times \begin{bmatrix} \theta_1 \\ \theta_2 \\ \theta_3 \\ \theta_4 \\ \theta_5 \\ \theta_6 \\ \theta_r \end{bmatrix} = \begin{bmatrix} -A1\sum_{j=1}^{\infty} Z_j\theta_{1(k-j)} + K_{xc} \cdot t_i \\ -A_2\sum_{j=1}^{\infty} Z_j\theta_{2(k-j)} \\ \cdot \\ \cdot \\ -A_5\sum_{j=1}^{\infty} Z_j\theta_{5(k-j)} \\ -A_6\sum_{j=1}^{\infty} Z_j\theta_{6(k-j)} + A_6\sum_{j=1}^{\infty} W_j \cdot g_{k-j} \\ (C_p\gamma V - K_{xc})t_i - A_h \end{bmatrix} \cdots(4)$$

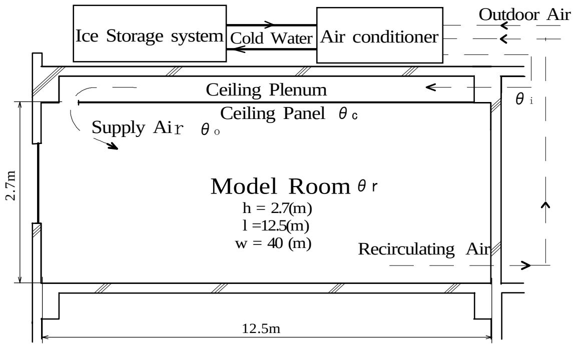


Fig.6 Hybrid System

Table 3 Nomenclature

θ_0	Temperature of supply Air
θ_i	Temperature of the Conditioned Air
θ_c	Temparature of Ceiling Panel
K_f	Thermal Transmittance
C_p	Specific Heat
V	Amount of Air
g_{ij}	Radiant Factor of Gebhart
Z_{ij}	Response Factor
W_j	Over Roll Thermal Transmittance
A_h	Interior Heat Load
α_r	Heat Transfer Rate of Radiation
α_c	Heat Transfer Rate of Convection

Calculations of latent heat and humidity are carried out using program on coil in the air conditioner.

3.3. RESULTS OF CALCULATIONS

Table 4 presents comparison of room environments between radiant cooling/heating system and convectional AC. Fig.7 present peak air conditioning load.

Fig.7(a) is under radiant cooling, and fig.7(b) is under convectional AC. It is apparent from table 4 that MRT in August is 1.6℃ lower than room temperature under radiant cooling. On the other hand, that MRT is 1.6℃ higher than room temperature under convectional AC, it is found that effects from radiant cooling with cooling panels are about 3.2℃. In June, room humidity can keep lower, so this system fits Japanese humid climate.

Fig.7(a) presents that cooling panel load is 68.7MJ, and accounts for 56% of sensible heat load (at 13:00). In fig.7, latent heat load under radiant cooling is 68.7MJ compared with that under convectional AC 53.5MJ. It

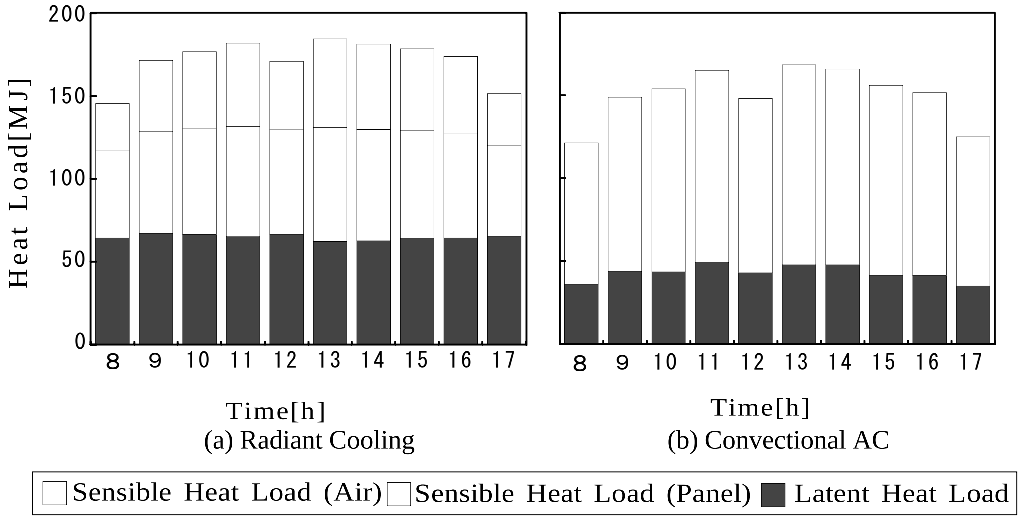


Fig. 7 Peak Air Conditioning Load

Table 4 Comparison of Room Environments

	System	Outdoor Air temp. (℃)	Outdoor Humidity (%)	Room temp. (℃)	Room Humidity (%)	MRT (℃)	Temp. of Coditioned Air (℃)	Ceiling Surface Temp. (℃)	OT (℃)
June	Radiant	23.3	69.8	28.0	37.0	26.4	11.0	24.9	27.2
	Convection			26.0	61.0	26.7	19.5	26.4	26.4
August	Radiant	28.5	63.0	28.0	37.0	26.4	11.0	24.9	27.2
	Convection			26.0	61.0	27.6	18.6	27.0	26.8
Peak Day of Cooling	Radiant	31.7	60.0	28.0	37.0	27.3	11.7	24.8	27.7
	Convection			26.0	58.0	28.2	117.7	27.4	27.1
January	Radiant	2.0	30.5	28.0	40.0	20.7	28.0	22.3	20.4
	Convection			26.0	40.0	19.7	25.6	20.5	20.9

is found that this system produces high efficiency of dehumidification. So this system is suit for warm and high humidity Japanese climate.

4. COMPREHENSIVE EVALUATION OF RADIANT COOLING SYSTEM

We study a model office building (in Tokyo, with the total floor area of 3,300㎡) in order to comprehesively evaluate the ice storage radiant cooling system, “radiant with ice storage”, by comparison with three other cooling systems, namely, “radiant without ice storage”, “convectional AC with ice storage”, and “convectional AC without ice storage”, as shown in Table 1. Simulations of the load leveling effect, energy saving possibility and operating cost were carried out for the four different systems. We then estimate the initial costs (construction costs) required to construct the model building with those systems.

4.1. OPERATION OF MODEL BUILDING

The room temperature and relative humidity settings for the office with the radiant system during the summer are 27℃ to 28℃ and 40%, as shown on the right in Table 5. An enivironment of those conditions is comparable to that of 26℃ and 50% relative humidity produced by the convection system in terms of thermal sensation.

4.2. CALCULATION OF THERMAL LOAD

In order to determine the capacity of the heat pump and air conditioners, the hourly thermal loads in summer and winter were calculated by the response factor method under the periodic steady state using the temperatures given by the Technical Advisory Committee (x = 5.0%) for Tokyo. The same procedure was applied for each month, using each hourly outdoor design temperature, to obtain each hourly thermal load.

Table 6 presents the calculated results, including the peak values of the cooling load and the circulating air volume, which are used to determine the capacities of the air conditioning facilities.

4.3. CALCULATION OF ELECTRICITY CONSUMPRION AND CHARGE

In the calculation of electricity consumption, the heat gain related to piping, pumps, and fans was assumed to be 10% of the cooling load. The heat loss, 3% of the

Table 5 Cases for Comparison and Evaluation

Case	Air Conditioning System	Ice Storage	Heat Source	Fan Control	Pump Control	Room Temperature & Humidity Setting (Target)
1	Radiant Cooling	N/A	Air Source Heat Pump	VAV	CWV	Summer (June - Sept.) 28℃ 40% Middle Seasons 24℃ 45% Winter (Dec.- Mar.) 22℃ 40%
2		Available	Air Source Heat Pump & Ice Storage Unit			
3	Convectional AC	N/A	Air Source Heat Pump			Summer (June - Sept.) 26℃ 50% Middle Seasons 24℃ 45% Winter (Dec.- Mar.) 22℃ 40%
4		Available	Air Source Heat Pump & Ice Storage Unit			

Table 6 Cooling Load and Circulating Air Volume

Air Conditioning System			Radiant Cooling			Convectional AC
Ice Storage	Room temperature Setting		28℃	27.5℃	27℃	26℃
N/A	Required Cooling Capacity (Peak Value)		289 kW	295 kW	301 kW	284 kW
Available	Required Stored Heat Capacity for Cooling (Peak Value)		1594 kWh	1635 kWh	1676 kWh	1702 kWh
Common	Required Circulating Air Volume	Office (7F)	3648 CMH	3823 CMH	4005 CMH	7464 CMH
		Office (2F-6F)	3049 CMH	3187 CMH	3333 CMH	5391 CMH
		Office (1F)	2962 CMH	3096 CMH	3238 CMH	5192 CMH
	Required Cooling Capacity for Sensible Heat (Peak Value)	Circulation Space (7F)	4.6 kW			
		Circulation Space (2F-6F)	2.8 kW			
		Circulation Space (1F)	2.3 kW			
		Entrance Hall	12.1 kW			

total thermal load, was also considered for the ice storage tank.

The electricity rates of Tokyo Electric Power Co. for a thermal storage system for commercial use, effective January 1, 1996, was used for the calculation. An 80% discount is applicable to the rates for the heat pump for thermal storage in the nighttime (10 P.M. to 8 A.M.).

4.4. RESULTS OF ENERGY CALCULATION

Fig.8 presents the mothly electricity consumption and breakdown for the radiant cooling systems. A comparison of fig.8 shows the reduction in the electricity consumption of the heat pump during the daytime due to the thermal storage.

4.4.1. Comparison between radiant and convection systems

Using the ice storage radaiant cooling system, the maximum power demand, the annual electricity consumprion and charges equivalent to , or slightly less than, those of the convection system are possible, as seen in table 7(1), (2),(3). Without the ice storage system, the maximum power demand and the annual electricity consumption for the radiant system is expected to increase approximately 10% over those of the convectional AC.

Table 7 Comprehensive Evaluation Results (￥:JPY)

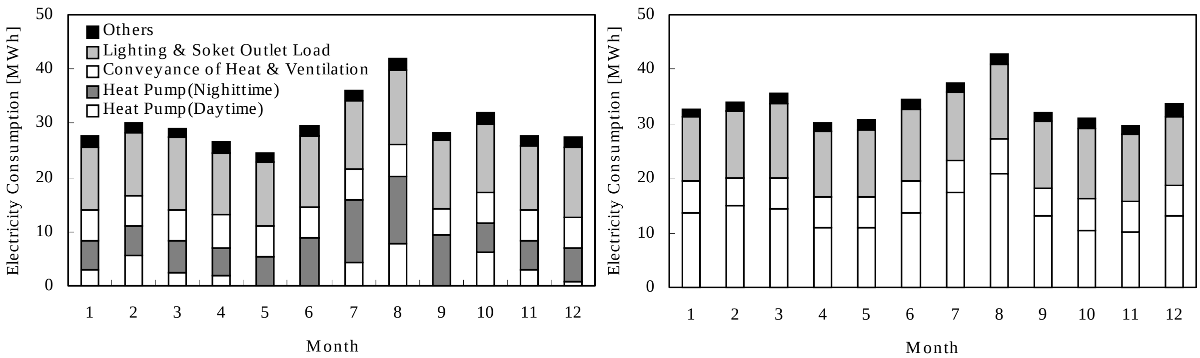
Air Conditioning System	Ice Storage N/A	Ice Storage Available	Difference
(1) Maximum Power Demand			
Radiant	219 kW	144 kW	-75 kW (-34%)
Convectional AC	194 kW	151 kW	-43 kW (-22%)
Difference	+25kW (+13%)	-7 kW (-5%)	
(2) Annual Electricity Consumption			
Radiant	401 MWh	362MWh	-39 MWh (-10%)
Convectional AC	370 MWh	371 MWh	+1 MWh (0%)
Difference	+31 MWh (+8%)	-9 MWh (-3%)	
(3) Running Cost (Annual Electricity Charges)			
Radiant	￥10.0 M	￥7.0 M	-￥2.9 M (-29%)
Convectional AC	￥9.1 M	￥7.2 M	-￥1.8 M (-20%)
Difference	+￥0.9 M (+10%)	-￥0.2 M (-3%)	
(4) Initial Cost (Construction Cost)			
Radiant	￥800.7 M	￥812.9 M	+￥12.1 M (+1.5%)
Convectional AC	￥794.0 M	￥810.4 M	+￥16.4 M (+2.1%)
Difference	+6.7 M (+0.8%)	+￥2.4 M (+0.3%)	

(5) Simple Years to Payback (Relative to "Convection without Ice Storage")

Air Conditioning System	Ice Storage	Difference: Initial Cost	Difference: Running Cosst	Simple Years to Payback
Radiant	N/A	+￥6.7 M	+￥0.9 M	—
	Available	+￥18.8 M	-￥2.0 M	9.3 years
Convectional AC	N/A	—	—	—
	Available	+￥16.4 M	-￥1.8 M	9.0 years

4.4.2. Comparison between cases with and without ice storage, for radiant system

With ice storage, the maximum power demand for radiant cooling can be reduced by 34% at maximum during the summertime as compared to the case without ice storage, as shown in table 7(1). The annual electricity consumption can also be reduced by 10%, as shown in table 6(2). At the current electricity rate, 29% of the charges can be saved, as shown in table 7(3). The ratio of the annual electricity consumption of the heat



(a) with Ice Storage (b) without Ice Storage

Fig.8 Monthly Electricity Consumption and Breakdown (Radiant System)

pump during the nighttime to the total consumption is expected to be 72%.

4.5. ESTIMATE RESULTS OF INITIAL COST

The initial cost for “radiant with ice storage”, ￥812.86M, proved to be ￥8.84M (2.4%) more expensive than that for “convection with ice storage”, ￥794.02M. However, comparing “radiant with ice storage”, with “convection with ice storage” (￥810.43M), the cost was only slightly increased, by ￥2.43M (0.3%), as also shown in table 7(4).

4.5.1. Interior finishing

The first row in table 8 presents the items of interior finishing related to the difference in the estimates. For the radiant system, aluminum panels are used for the ceiling radiant panels. For the convection system, sound - absorbing rock wool boards are used. Consequently, the cost for the radiant system is ￥14.00M more.

4.5.2. Electric equipment

The contract maximum power demand differs depending on the air conditioning system and whether or not ice storage is provided. This difference is reflected in the estimates of substation and main feeder. It produces a difference of ￥0.90M for the electric equipment between “radiant with ice storage” and “convection with ice storage”, as shown in the second row of table 8.

Table 8 Factors Affecting Construction Cost (￥:JPY)

		Items which Cause Difference: Radiant with Ice Storage	Convection with Ice Storage	Difference
(a) Interior Finishing	Ceiling	Ceiling Radinat Panel Aluminm Panel 2417 ㎡ Foamed Styrene Board 2077 ㎡	Sound - Absorbing Rock Wool Board 2075 ㎡	+￥14.00 M
(b) Electric Equipment	Substation & Main Feeder	Constract Maximum Power Demand 144kW	Constract Maximum Power Demand 151kW	-￥0.90 M
(c) Air Conditioning Facility	Air Conditioner & Installation	Air Conditioners 3600CMH × 6Units 4800CMH × 1Units Installation Expenses	Air Conditioners 6000CMH × 6Units 9000CMH × 1Units Installation Expenses	-￥5.20 M
	Duct & Installation	Duct 210㎡ Air Outlet 27Units Volume Damper 39 Other Material Costs Installation Expenses	Duct 644㎡ Air Outlet 54Units Volume Damper 66 Other Material Costs Installation Expenses	-￥5.40 M
* Total		￥262.80 M	￥260.40 M	+￥2.40 M

*Total = (a) Interior Finishing + (b) Electric Equipment + (c) Air Conditioning Facility

4.5.3. Air conditioning facility

The section area of the duct was calculated based on the circulating air volume in each case, and then the required area of the duct was determined. The number of volume dampers required was determined by adding 12 to the number of air outlets. These values are summarized in the third row of table 8 to illustrated the difference. In total, the air conditioning facility for “radiant with ice storage” proved to be ￥10.60M more expensive than that for “convection with ice storage”.

4.5.4. Years to payback

Table 7(5) presents the simple years to payback relative to “convection without ice storage”. Both “radiant with ice storage” and “convection with ice storage” are calculated to require nine years to payback. Under the assumption that ice storage is adopted for load leveling, the difference between the two systems with ice storage is negligible.

5. CONCLUSIONS

In this paper, we estimated the optimum room condition by mesurements of the thetmal environment and experiments with subjects, and suggested a hybrid system which is a radiant cooling system using conditioned air as a refrigerant integrated with ice storage system. This system can provide more comfortable room environment than convectional AC. The results of calculations show that ice storage system make the effedt of dehumidification higher, and prevent from the occurrence of condensation.

Using ice storage radiant cooling system, both the maximum power demand and the annual electricity consumption equivalent to or slightly less than those of the covection system can be achieved. Under the assumption that is exposed to be approximately ￥2.40M over that for the convection system.

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