SCALE MODEL EXPERIMENT OF AIR DISTRIBUTION IN THE LARGE SPACE OF THE SHINKOKUGIKAN SUMO WRESTLING ARENA

by Satoshi Togari and Shin Hayakawa

1. INTRODUCTION

The SHINKOKUGIKAN is a 11000-seat Sumo wrestling arena, which was constructed in January, 1986 in Tokyo, Japan.

Its arena is 80m x 80m with a maximum ceiling height of 40m. The main considerations in planning the air conditioning system for this large space are as follows:

a. The occupied zone only should be cooled in summer.
b. The temperature difference between the occupied zone and the upper zone should be small in winter.
c. The space surrounding the Dohyo ring should be kept cooler than elsewhere because it receives strong radiant heat from lighting, including spot-lights for television cameras.

To satisfy these objectives, an experiment was conducted with a 1/20 scale model, as a result, zoning of air conditioning systems and specifications of outlets, including dimensions, blow angle, supply air temperature and velocity, were decided.

In this experiment, a tracer gas (ethylene) was mixed into each duct line in succession and the concentration of the gas was measured at 63 points in the occupied zone. Consequently, the contribution of each duct line to the air conditioning of each zone was found. These results were very useful for designing the air temperature control system.

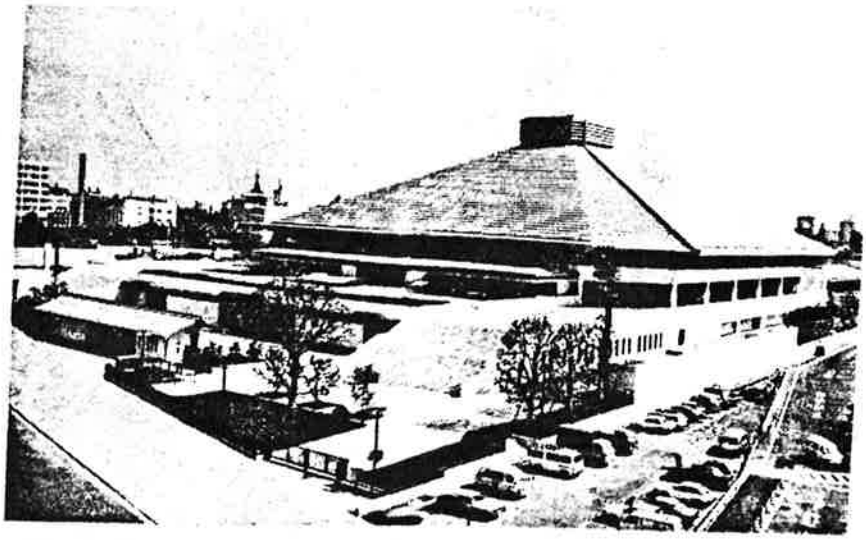


Photo 1 Outside View of the SHINKOKUGIKAN

* Kajima Institute of Construction Technology, JAPAN.

2. OUTLINE OF AIR-CONDITIONING SYSTEM FOR THE LARGE SPACE

2.1 AIR DISTRIBUTION SYSTEM

The air distribution system was determined by experiments using a model reduced by the scale of 1/20. Fig. 1 shows the layout of the supply outlets and return inlets.

Anemo-type outlets (A1) are mounted in the ceiling at the rear of the first floor. Slit-type outlets (S1) are at the tip of the seats of the 2nd floor.

Nozzle-type outlets (N2) are at the top of the rear wall of the 3rd floor (N2 and S2 constitute a single ductwork.)

Conditioned air is supplied to the large space through three ductworks (A1, S1 and N2) from the east and west sides, making six ductworks in total.

The air distribution system can be effectively operated when combined with the supply air temperature control system mentioned in Section 2.2.

The main features of the air-distribution system are as follows:

1) The area surrounding the "Dohyo" ring (C-zone, Fig. 1) receives strong radiant heat from lighting, including spot-lights for television cameras. To cope with this, it is necessary to keep air temperature lower and air velocity higher of this zone than other zones.

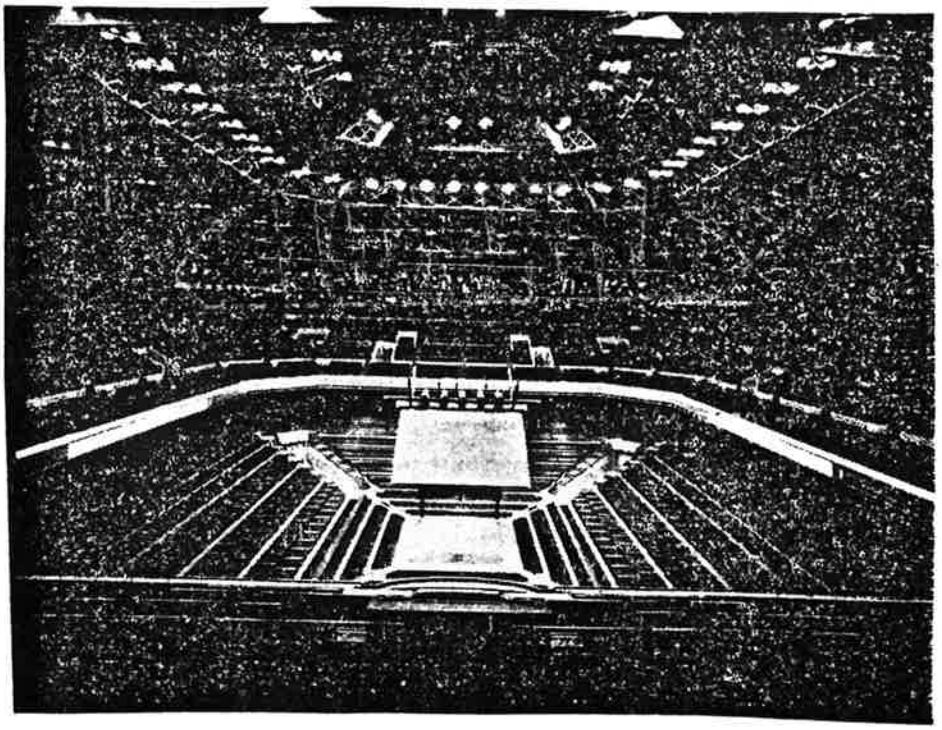
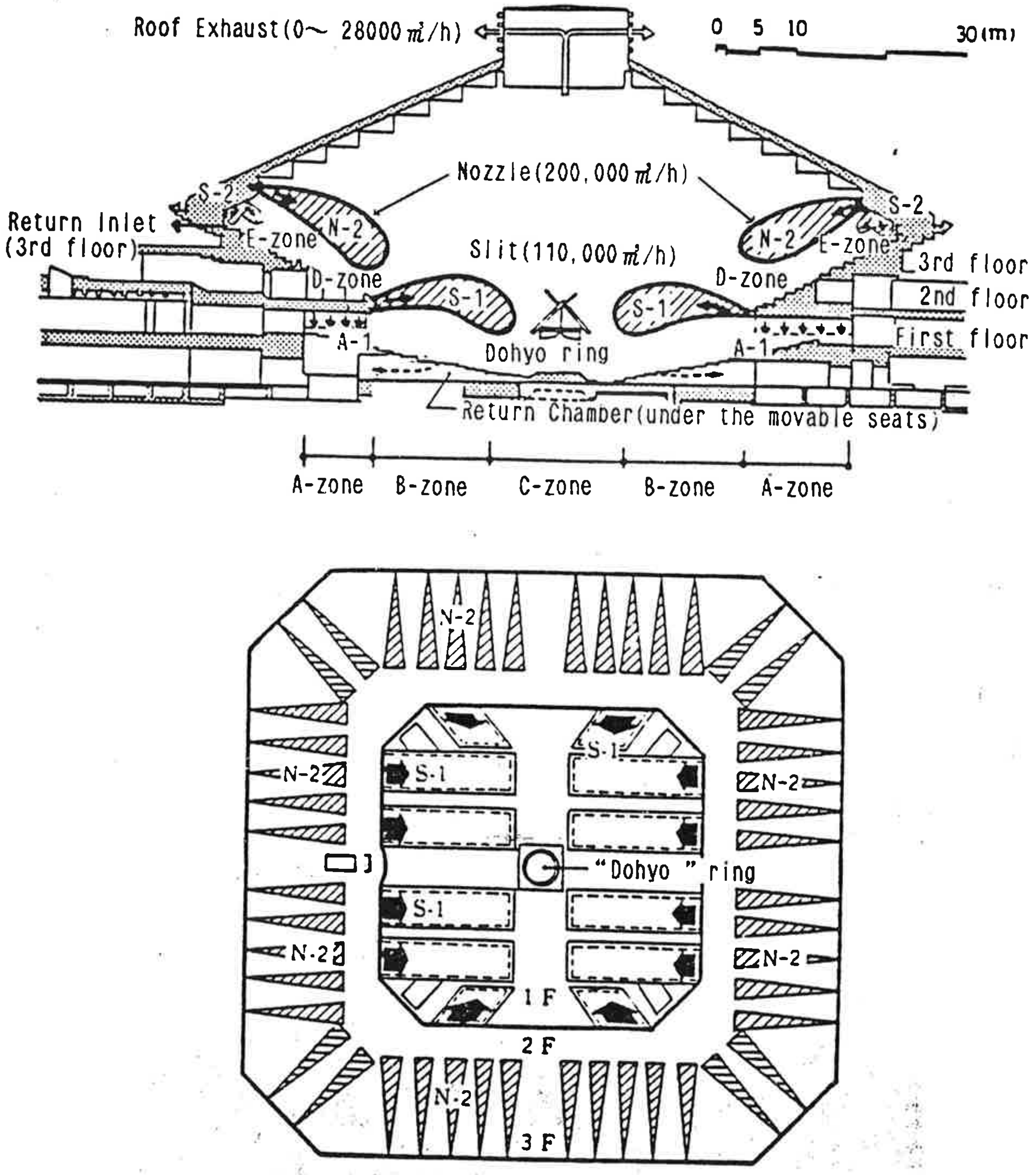


Photo 2 Sumo Wrestling Arena



A-zone : rear spectator seats on the first floor

B-zone : movable seats

C-zone : area surrounding the "Dohyo" ring

D-zone : spectator seats on the 2nd floor

E-zone : spectator seats on the 3nd floor

Fig. 1 Layout of Supply Outlets and Return Inlets

Table 1 Outlet Type and Supply Air Volume

Supply Air Channel	Outlet Type	Supply Air Volume (Velocity)
S1	Slit (Ho=150mm)	110,000 m³/h (3.5 m/s)
A1	Anemo-type Air Diffuser	150,000
N2	Nozzle (Do=400mm)	170,000 (6.0)
S2	Slit-type	30,000

To be concrete, it has been designed that cool air supplied from the slit-type outlet (SL) drops into the occupied zone around the "Dohyo" ring (C-zone).

2) Under the movable seats on the first floor (B-zone) is a return plenum chamber. Through extracting 90% of return air volume by this plenum chamber, the vertical temperature gradient has been minimized.

3) When necessary, the ventilating fan at the top is operated, so that hot air and contaminant in the upper part of the large space can be effectively removed.

4) Traditionally, smoking is allowed at Sumo wrestling shows. Therefore, a volume of fresh air intake amounting to 40 m^3/h per spectator is insured.

5) Since the lighting load and human-body sensible heat load are very large, room-cooling is necessary even during winter.
To cope with this situation, the amount of outdoor air intake is controlled in winter to save energy.

The air-distribution system can effectively operate when combined with the supply air temperature control system described in Section 2.2 (Next section).

2.2 SUPPLY AIR TEMPERATURE CONTROL SYSTEM

As an accessory unit for the central monitor unit, a microcomputer is installed with a program for supply air temperature control.

The concepts of the temperature control are described briefly below.

1) From the characteristics of the air conditioner, it takes about 5 min to achieve the designated supply air temperature.
To ensure stabilized control, adjustments to the supply air temperature are made at 15 minutes' interval.

2) For reasons mentioned below, the upper and lower limit values of the supply air temperatures are chosen as follows:

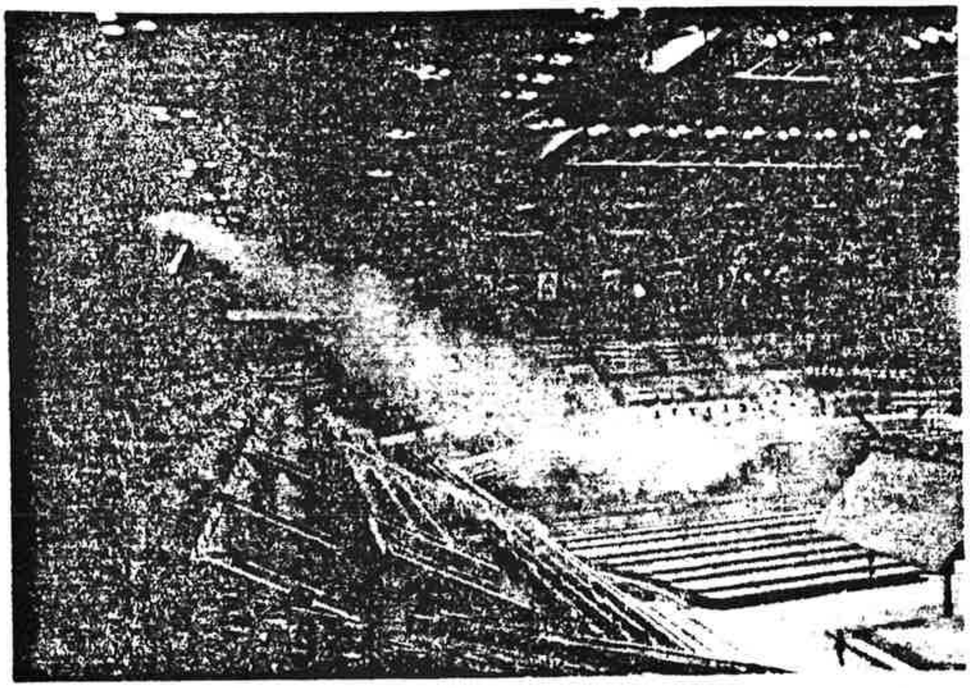
a. Cool air supplied from slit-type outlets (S1) drops into the occupied zone around the "Dohyo" ring when the outlet temperature difference is about 8°C as shown in Photo 3.
If the temperature difference becomes too large, the throw will become shorter.
To prevent this, a lower limit value of the supply air temperature has been chosen (S1 channel: 15°C).

b. The nozzle outlets (N2) for the seats on the 2nd and 3rd floors have been planned so that cool air will not fall into the occupied zone on the 2nd floor as shown in Photo 4. To achieve this, a lower limit value of the outlet temperature has been chosen (N2 channel: 16°C).

c. When preheating in winter, if the supply air temperatures are too high, the vertical temperature gradients will also become large; hence upper limits for the supply air temperatures have been chosen. (N2 and A1 channels: 30°C, S1 channel: 26°C)

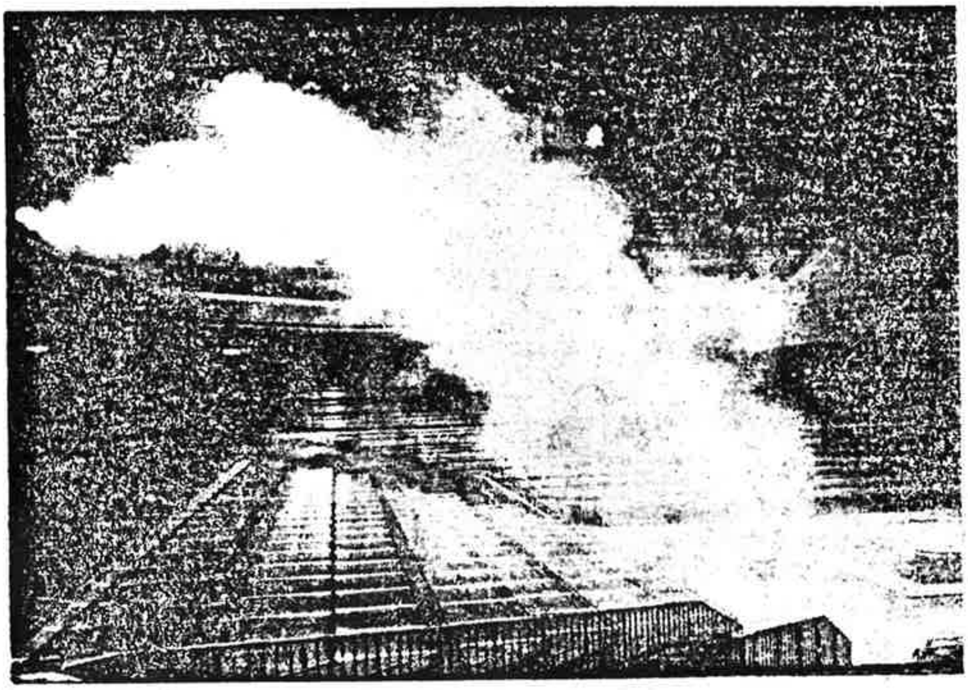
3) Heat and air discharged from the six duct channels, are mixed in the large space; hence the supply air temperatures of the respective channels are determined by taking into consideration the effect of this mixing.

At five points ambient air temperature is measured to determine the supply air temperatures; one point at the return air chamber on the first floor, two points at the rear spectator seats of the first floor (one point each at east and west) and two points at the return ducts from the third floor.



Temperature differential
between supply air and occupied zone: $\Delta\theta=7°C$

Photo 3 Cool Jet Trajectory Discharged from Nozzle (N2)



Temperature differential
between supply air and occupied zone: $\Delta\theta=7°C$

Photo 4 Cool Jet Trajectory Discharged from Slit (S1)

3. PREDICTION OF ROOM AIR DISTRIBUTION BY MODEL TEST

3.1 TEST METHOD

(1) Similarity Rule

Regarding the method of a scale model test, there have been studies by V.V. Vaturin,[1] T. Shoda and T. Tsuchiya,[2] etc. It has been clarified that as long as the following conditions are satisfied, similarity is obtained in the air distribution of the original room and its scale model.

1) Geometric similarity

2) Equality between Archimedes number (Ar) of incoming jet in the model and in actual conditions.

$$(Ar)_M = (Ar)_N \; : \; Ar = \frac{g\beta\Delta\theta\cdot L}{Uo^2}$$

where U_o: Supply air velocity at outlet [m/s]
$\Delta\theta$: Outlet temperature difference [°C]
L : Typical dimensions (outlet diameter) [m]
g : Gravitational acceleration [m/s^2]
β : Coefficient of cubical expansion (1/°C)
Suffix N: Original
" M: Scale model

3) Condition on the Reynolds number (Re) for incoming jet of the scale model

$$Re >> 2330 \text{ (in the turbulent region)}$$

The present study has adopted, this method as a rule. But the condition on Re is difficult to ensure in a reduced model, and in particular Re of incoming jet from the slit-type outlet (S1) cannot satisfy this condition (Re(S1) ≒ 1200).

(2) Test Facility

An overall scale model of the large space was made in a reduced scale of 1/20. Its approximate dimensions are 4 x 4 x 2 (H) m, and it is mostly made of plywood. An internal view of the model is shown in Photo 5. The Slit-type outlets (S1) and nozzle outlets (N2) are geometrically similar to the original. The anemo-type outlets have been replaced with simple openings. Also the spectator's seats have been simplified in the scale model. The scale model is placed in an artificial climate chamber. Fig. 2 gives an outline of the test facility.

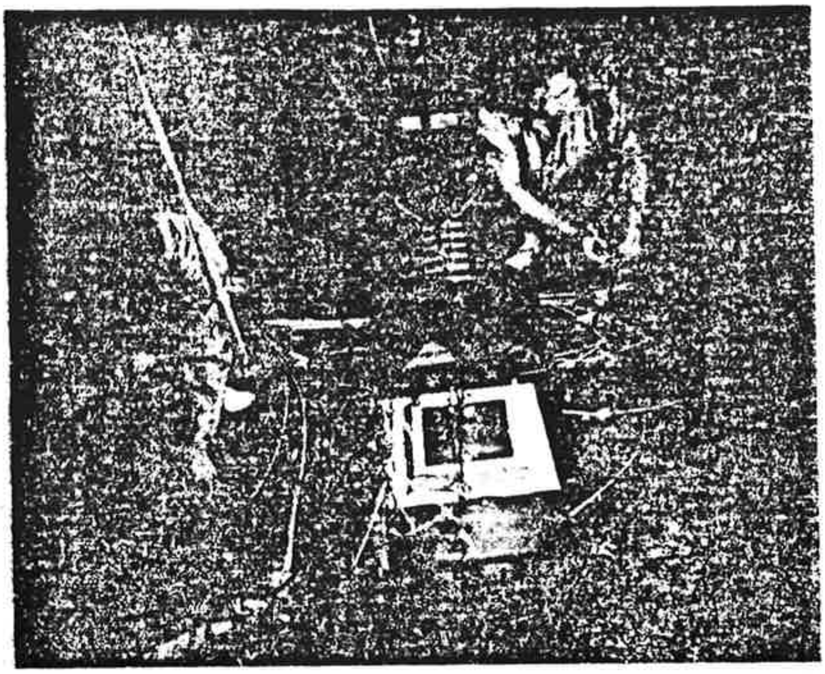


Photo 5. Internal View of the Test Model

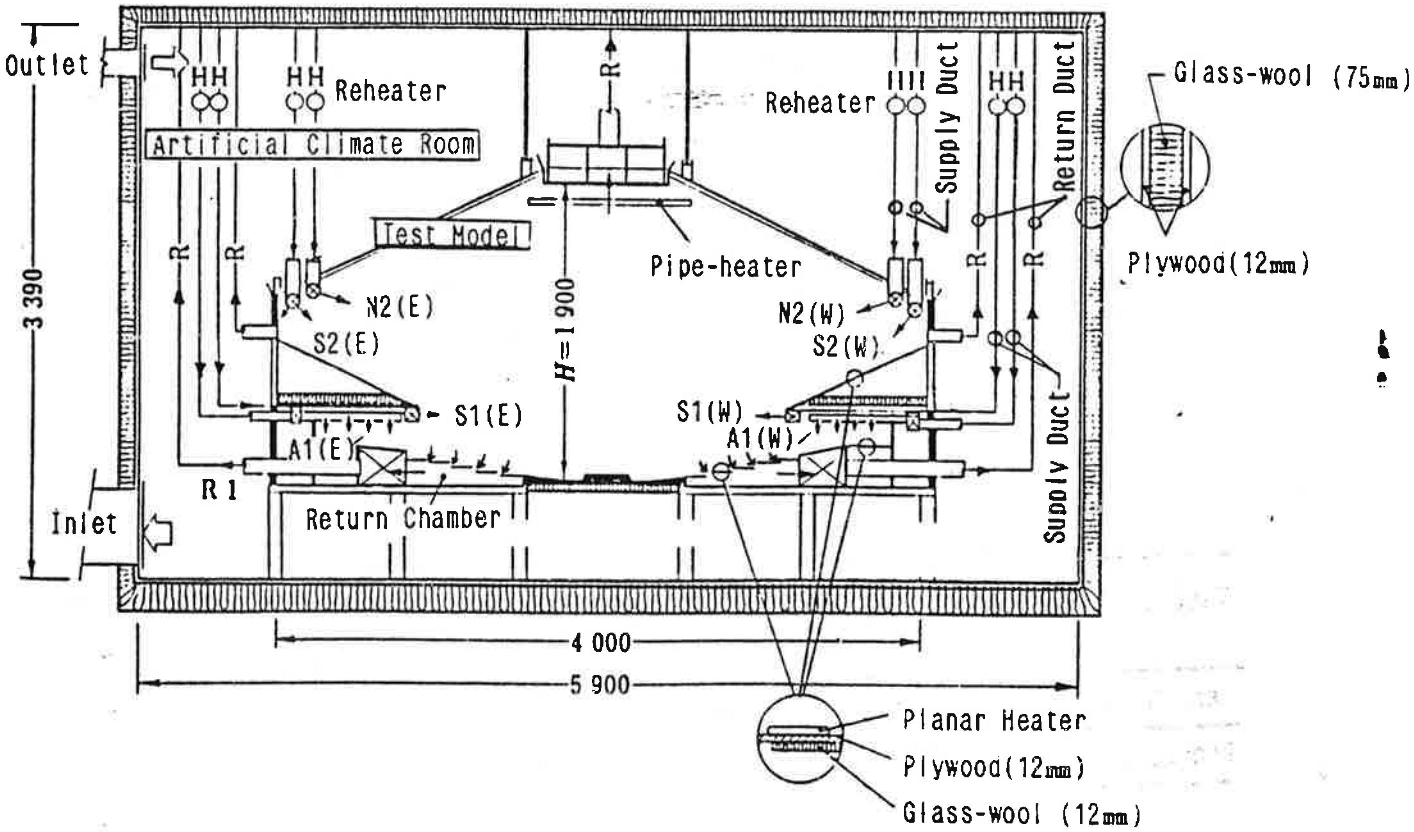


Fig. 2 Facility of the Model Test

(3) Test Conditions

Table 2 shows the air-conditioning load in the main conditions which has been made an object of the test. Most of the sensible cooling load comes from internal heat sources (lighting, occupants).

In the scale model test, human-body sensible heat and floor surface load (including radiant heat from lighting) were simulated by planar heaters installed on the floor surface. Also lighting heat (due to convection) in the upper region of the large space was given by a pipe heater (Fig. 2).

Heat gain through the roof was simulated by setting a temperature difference between the inside and outside of the scale model as shown in Table 3.

Table 2 Space Cooling/Heating Load (Sensible Heat Only)

		Peak cooling Load	Preheating Load
Outdoor Condition		Tokyo, Summer TAC 2.5%	Tokyo, Winter TAC 2.5%
Heat Gain * (Conduction)	Roof	97 kW	-225 kW
	Outside Wall	47	- 97
	Inner Wall	104	-150
	Floor	40	- 10
People (Sensible)		583	0
Lights (Covection)		168	0
Total		1039 kW	-482 kW

* Heat gain involves the effect of radiation from lights

Table 3 Temperature Conditions of the Model Test

Test Condition	Air Temperature		
	Supply Air	In the Model	Climate Room
Peak Cooling Load	25～30℃	≒40℃	60～65℃
Preheating Load	≒55℃	≒40℃	≒15℃

For room cooling, the heat load distribution according to respective locations in the original and the scale model have been set to the same values.

But for room heating, the temperature difference between the inside and outside only was used to control the entire heat loss.

Reduction ratios in the scale model test are as follows:

1) Size: $N\ell = 1/20$
2) Temperature difference: $N_\theta = 2.0$
3) Velocity: $Nu = 0.32$
4) Archimedes number; $N_{Ar} = 1$

The Reynold's number of the incoming jet in the scale model test at the nozzle outlet (N2), which is considered to have the most significant effect on air flow in the entire large space, is $(Re)_M = 2,700$. Table 4 shows measuring items and measuring instruments used.

Table 4 Measuring Items and Instruments

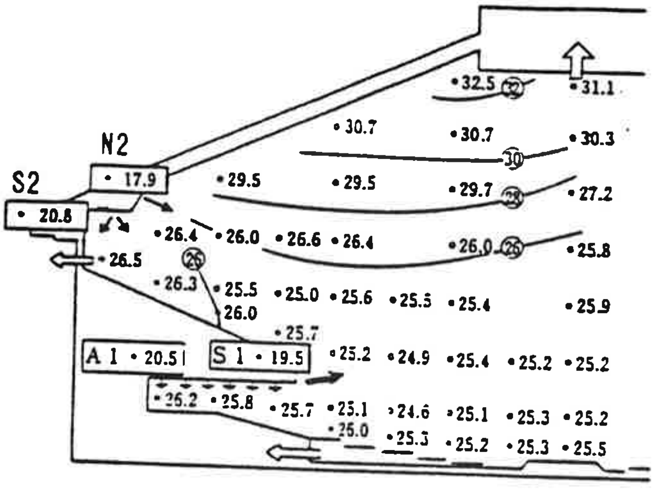
Measuring Item	Points	Measuring Instrument
Supply Air Volume	25	Orifice, Capacitance Manometer
Temperature	137	Cu-Co Thermo-Couple (Covered with Alminium Foil) Multi-channel Digital Temperature Recorder
Velocity	91	Anemometer with Omnidirectional Probe
Concentration	63	Hydro-Carbon Detector (FID Type)

3.2 TEMPERATURE AND VELOCITY DISTRIBUTION DURING COOLING

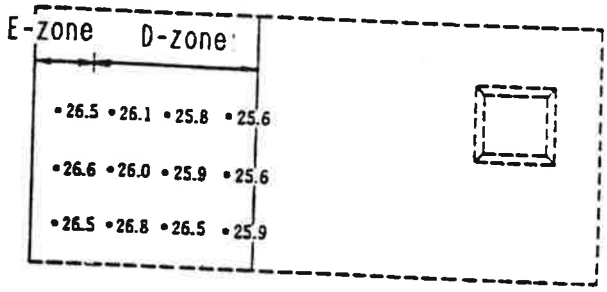
Temperatures and velocities of air obtained by the scale model test have been converted into values of the original and are shown below.

Fig. 3 shows the temperature distribution during the peak cooling load (17:00 in mid-summer, estimated number of spectators: 11,000).

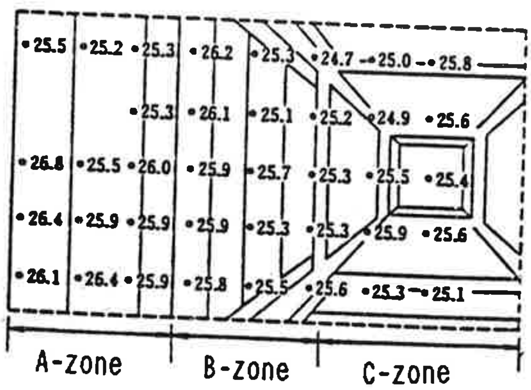
While the temperature in the greater part of the occupied zones remains within the range of about 25.5 to 26.5°C, the average temperature around the "Dohyo" ring (C-zone) is maintained at the lower temperature of 25.5°C or below. In the upper part of the large space, the temperature has risen to



— Main Vertical Section —



— 1.0 m above 2nd and 3rd Floor —



— 1.0 m above First Floor —

Fig-3 Temperature Distribution in Case of Peak Cooling Load

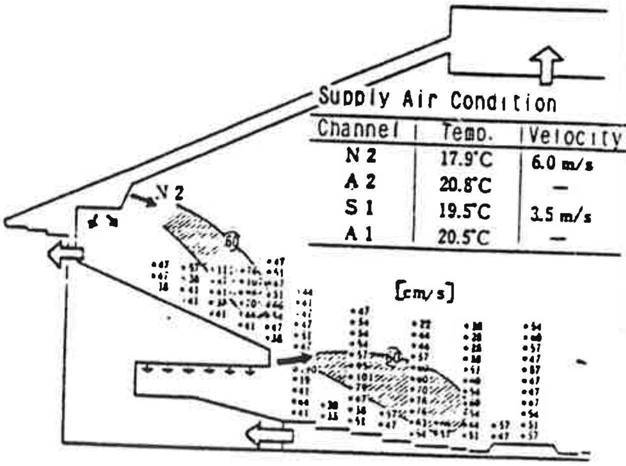


Fig-4 Air Velocity Distribution in Case of Peak Cooling Load

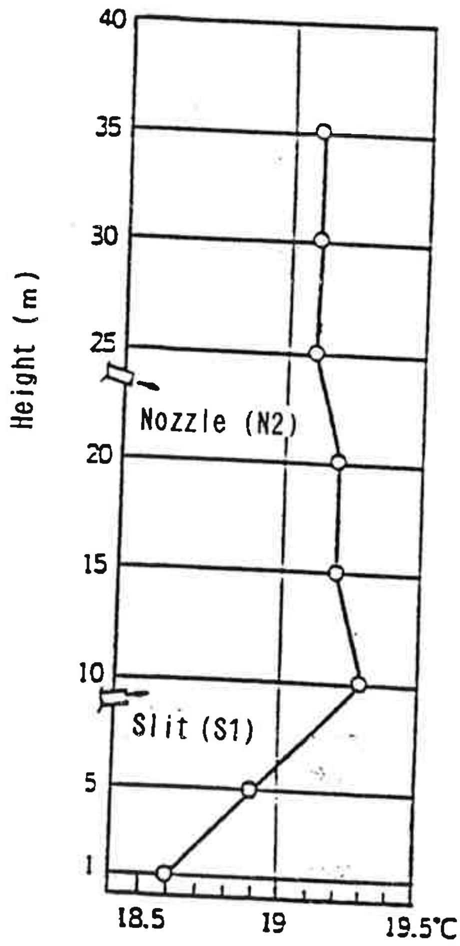


Fig-5 Temperature Gradient at Preheating Condition about 30°C. All these locations have satisfied the target values.

Fig. 4 indicates that cool air supplied from the slit-type outlet (S1) falls due to the temperature difference and reaches the occupied zone surrounding the "Dohyo" ring (C-zone). This is why the temperature surrounding the ring is maintained at a lower level. Velocity around this area is 0.5 m/s on average, higher than the other zones.

In a test where the supply air temperature of the S1 channel was further lowered, cool air fell near the outlet, and it was impossible to maintain the temperature around the ring at a lower level. As a result, it was decided to control the supply air temperature of the S1 channel by choosing a lower limit value.

For the S1 channel, a test was conducted using a nozzle outlet, but the result indicates that controllability of the temperature around the ring is poor and velocity in the occupied area is too fast. Consequently, this idea was discarded.

3.3 TEMPERATURE DISTRIBUTION DURING PREHEATING (WINTER)

When the temperature difference between supply air and occupied zone is about 7°C, the temperature difference from 1 m above the first floor to the upper part of the large space is as small as 1°C (Fig. 5).

At first, examination was made of a system in which the discharge air direction was changed depending upon supply air temperature, but in view of the above results, it was judged that the blow angle could be kept constant throughout the year.

This small temperature difference between the upper and lower sections is considered attributable to the following three reasons:

1) The temperature difference between the supply air and the occupied zone is comparatively small (within 7°C).

2) The nozzle of the N2 channel has an outlet velocity of 6 m/s (outlet nozzle diameter: 400 mm), and has a blow angle of 20° downward, which contributies the uniform temperature distribution in the upper part of the large space. Downward discharge through anemo-type outlet (A1 channel) is also effective.

3) 90% of return air (420,000 m^3/h) is taken from the movable seats on the first floor (B-zone).

3.4 UNDERSTANDING AIR MIXING CHARACTERISTICS IN THE LARGE SPACE USING TRACER GAS

To clarify the mixing characteristics of the air supplied from the six duct lines in the large space, a scale model test using a tracer gas was conducted.

It has been already described that room cooling is necessary even during winter, so the test was conducted under the cooling condition.

(1) Test Method

Concentration distribution was measured for the case of peak cooling load. The temperature distribution (value converted to that of the original) under this condition has already been shown in Fig. 3.

Tracer gas was added to each duct line in turn.

As a tracer gas, a ethylene standard gas (N_2 basis; 10% concentration) was used, and the ethylene gas concentration was measured using a hydro-carbon detector (Flame ionization detector). To ensure measurement accuracy, the quantity of the tracer gas was adjusted so that the concentration in the supply air was about 100 ppm.

Supply air volume is about 350 m^3/h in total (converted to the value for the original: 460,000 m^3/h).

(2) Test Results

The test results are shown in Fig. 6. The value "C*" (C* = C/Cs) in the figure is obtained by dividing the ethylene gas concentration, C [ppm], at the respective points by the supply air concentration, Cs [ppm], at the channel where the tracer gas was added. Numerical values (C*) at the respective points indicate the ratio (degree of contribution) of air supplied by the duct line to which the tracer gas has been mixed. The value at each point near the floor surface is an average of 2 to 4 points on the same plane.

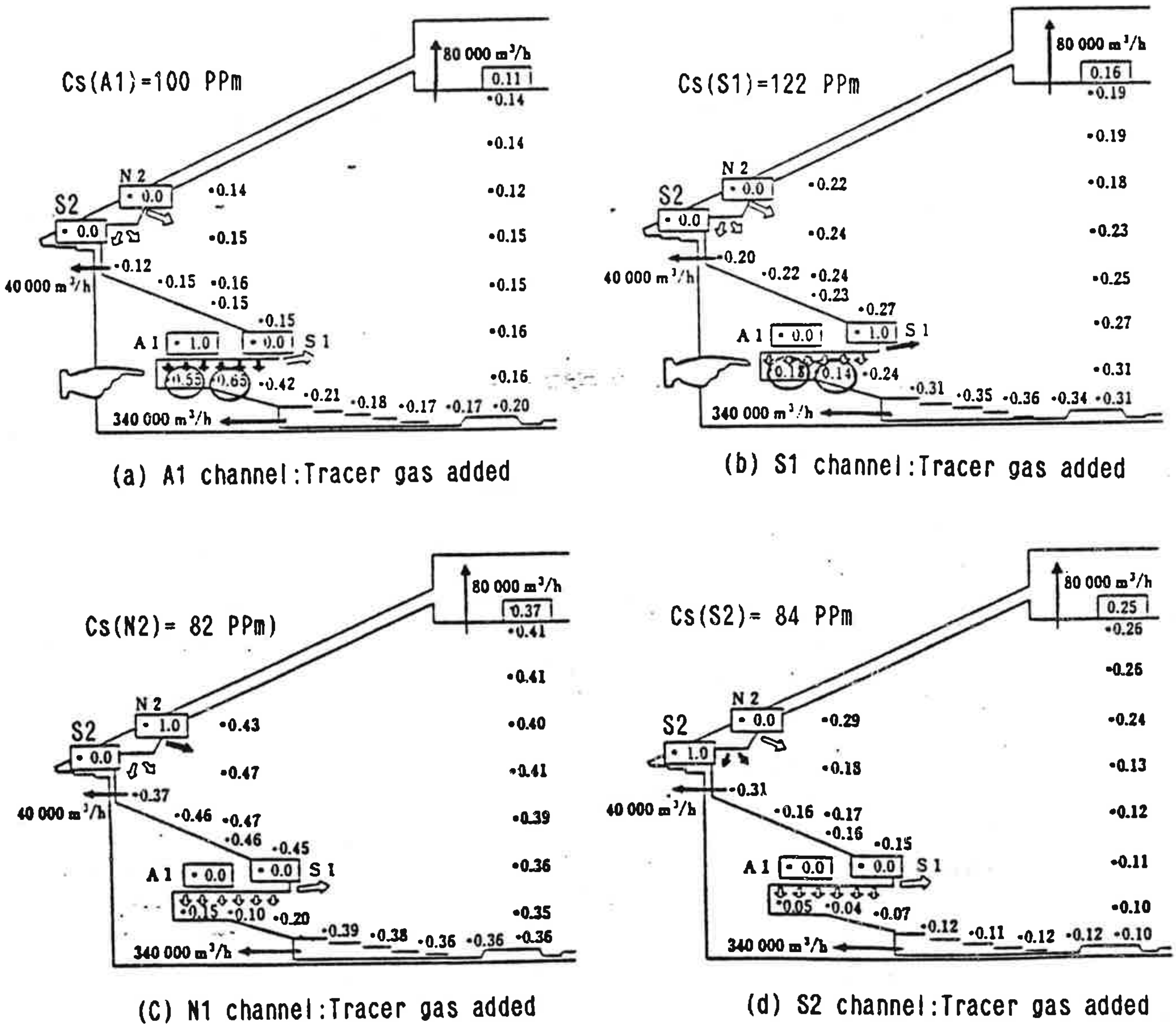
a) Relationship between Each Duct Channel and Occupied Zones in Large Space

Fig. (a) shows the relationship between A1 channel and each point of the large space. Non-dimensional concentration values (C*) of 0.55 and 0.65 are given at the rear of the first floor (A-zone), and it indicates that 55% or 65% of the air at that position is supplied from the anemo-type outlet (A1 channel).

b) Fig. (b) shows that at the same position, non-dimensional concentration values (C*) of 0.17 and 0.14 are given, thereby indicating that only 18% or 14% air has been supplied from the slit-type outlet (S1 channel).

When spectators' seats are broadly divided into five zones, Fig. 7 is obtained. For comparison's sake, supply air volume ratios of the respective duct lines are shown in Table 8. Fig. 7 indicates the following:

- o Seats at the rear of the 1st floor (A-zone) contains an overwhelming ratio of air supplied from the anemo-type outlet (A1 channel) as mentioned earlier (the supply air volume of A1 channel accounts for 30% of the total air volume).



(a) A1 channel:Tracer gas added

(b) S1 channel:Tracer gas added

(C) N1 channel:Tracer gas added

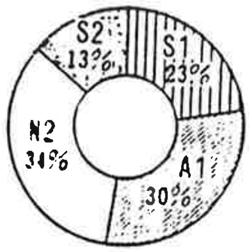
(d) S2 channel:Tracer gas added

Fig. 6 Concentration Distribution (C: Non-dimension)

Therefore, by controlling the supply air temperature of this channel (A1), it is considered possible to maintain A-zone temperature adequately.

- Distribution ratio of air around the ring (C-zone) closely resembles that at the movable seats (B-zone); and thus they may be assumed to be the same zone. In this zone, the air supplied from the slit-type outlet (S1 channel) at the front tip of the second-floor seats, which are nearest in distance, is slightly more than 30%, and that from the Nozzle-type outlet (N2 channel, including S2) is almost 50%. However air supply from the anemo-type outlet channel (A1) is as small as 19%.

The above indicates that in order to control temperatures in the areas surrounding the ring (C-zone) and movable seats (B-zone), it is necessary to control simultaneously supply air temperatures of both the S1 channel and N2 channels, but the effect of the A1 channel is small.



[S2 is larger in the test than in actual.]

Fig-8 Supply Air Volume Ratio

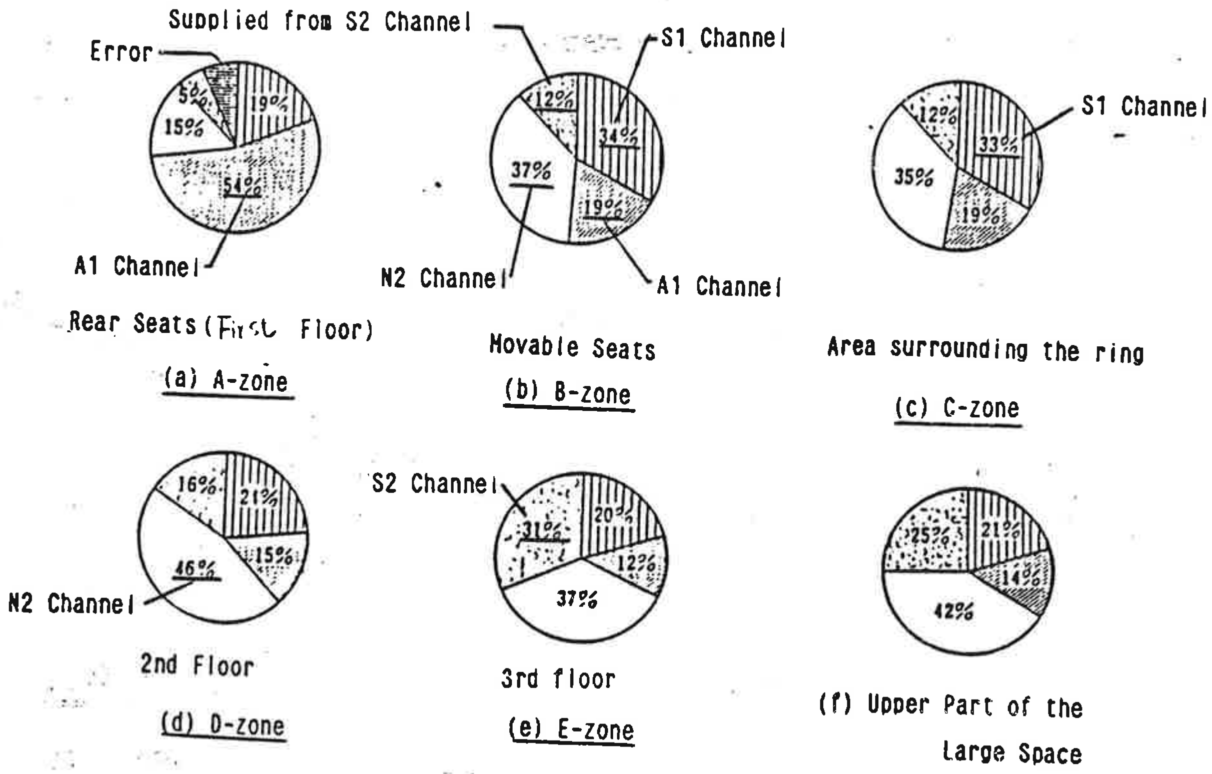


Fig-7 The Ratio of Air Supplied (Degree of Contribution) from the Tracer Gas Mixed Channel

- o At seats in the 2nd and 3rd floors, the effect of the N2 channel is strong, but in the section near the wall at the rear of the 3rd floor, the effect of A2 is also strong. But the range of influence of the A2 channel is narrow, so in the original, N2 channel and A2 channel are the same channel ((d) and (e) in Fig. 7).

(3) Determination of Constants for Mutual Influences of Various Air-conditioning Channels

From the results of temperature measurements in various combinations of supply air temperatures (not described in this paper) and concentration measurement results, the interaction between various air-conditioning channels has been taken into consideration and it was decided to control outlet temperature using the following equations:

$$\Delta\theta(S1) = \Delta\theta_{S1} - 0.5\Delta\theta_{N2}$$

$$\Delta\theta(A1) = 0.6\Delta\theta_{A1}$$

$$\Delta\theta(N2) = \Delta\theta_{N2} - 0.3\Delta\theta_{S1}$$

where $\Delta\theta(\ )$: Supply air temperature corrections of the S1, A1 and A2 channels

$\Delta\theta_{S1}$: Correction (PID control) of supply air temperature when the temperatures of B and C zone are assumed to be affected only by S1 channel.

$\Delta\theta_{A1}$: Correction of supply air temperature when the temperature of A zone is assumed to be affected only by A1 channel.

$\Delta\theta_{N2}$: Correction of supply air temperature when the temperature of D and E zone are assumed to be affected only by N2 channel.

4. CONCLUDING REMARKS

After completion of the construction, actual conditions were measured during winter and summer to confirm the performance.

Figs. 9 and 10 show variations in time of the supply air temperatures and return air temperatures of the various channels, which indicates a stable performance.

Fig. 11 shows the temperature distribution during summer cooling, indicating that the area surrounding the ring is maintained at a temperature lower than the other zones as planned. The upper section of the large space is as hot as 30°C, and only the occupied zone is cooled.

Fig. 12 shows the temperature distribution in the vertical direction during winter preheating. The vertical temperature difference is about 2°C, which is considered to be satisfactory, although it is higher than the predicted value (1°C) from the scale model test.

Although it is not clear how effective the supply air temperature control is, which has taken into consideration the interaction between the various air-conditioning channels, the authors are confident that the control has been successful.

ACKNOWLEDGEMENS:

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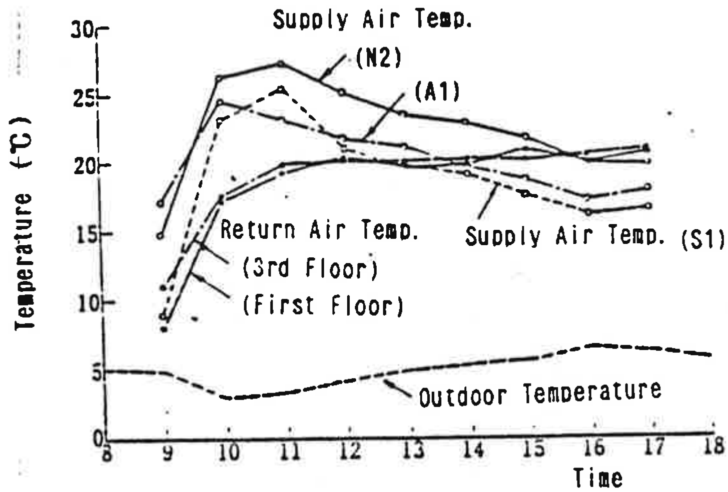

Fig-9 Supply Air and Return Air Temperatures Measured (in January, 1985)

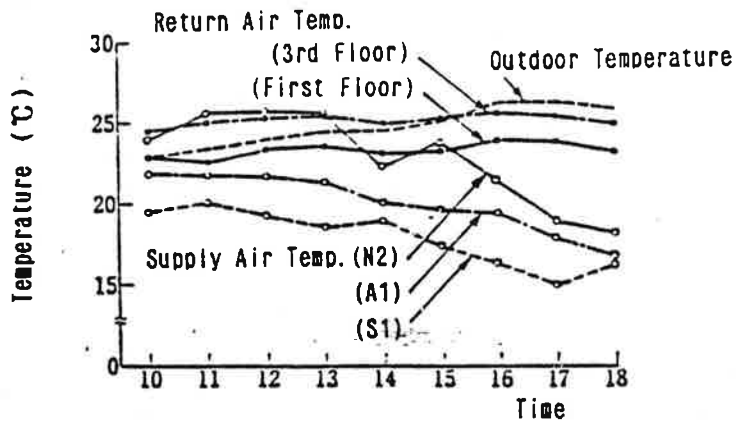

Fig-10 Supply Air and Return Air Temperatures Measured (in September, 1985)

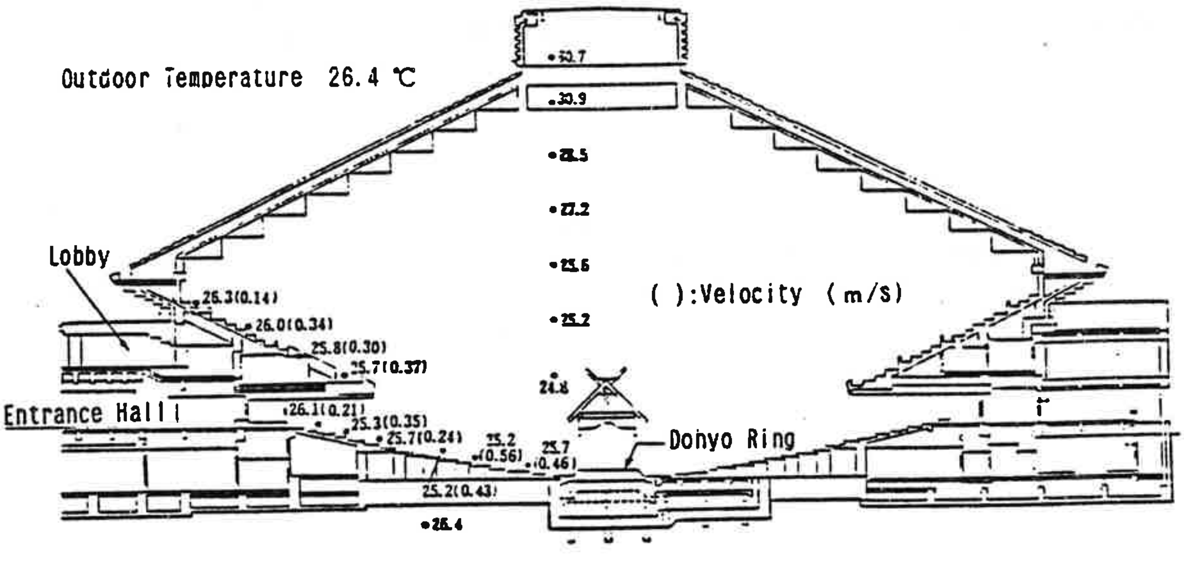

Fig-11 Temperature and Velocity Distribution Measured (at 5p.m, Sept.21, 1985)