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Applicability of Displacement Ventilation to Offices in Japan

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ABSTRACT

This paper aims to clarify the fundamental characteristics of displacement ventilation and examines its applicability to offices in Japan. Results of experiments performed on a 2.8 m × 2.6 m × 3 m test room with high heat loads (30-170 W/m²) during early summer and winter are presented. Results of computer simulation are also presented. Temperature, velocity, and contamination distributions in the room are also reported. A relation between height of clean zone and Archimedes number ($(Z - Z_h) / (bl)^{0.5} = 0.64\ Ar^{-0.2}$) is obtained from the experiments. Flow characteristics predicted by computer simulation agree well with experimental results.

INTRODUCTION

Displacement ventilation has been used in the Nordic countries during the past twenty years as a means of ventilation in industrial facilities requiring special indoor pollution control. Recently, its use has been extended to the ventilation of offices and, in Nordic Countries today, displacement ventilation has approximately 50% of the market share in industrial applications and 25% in office applications (Svensson 1989).

In Japan, a growing concern for the quality of indoor air makes displacement ventilation an attractive alternative method for ventilation of offices. To the authors' knowledge, however, displacement ventilation has not yet been applied to offices in Japan. There is doubt whether this method, which works well in Nordic countries where buildings are usually constructed tightly and are well insulated so the heat load is small, will function well in Japan where the ventilation requirement is different and heat loads are high. Recent trends in Japan for extensive use of electronic office equipment and efficient use of work space have pushed the heat load design value to as high as 150 W/m². In addition to high heat loads, there are differences between the Nordic and the Japanese climate and office worker acclimatization.

This paper is a first attempt to determine the applicability of displacement ventilation to offices in Japan. Results of experiments conducted on a full-size room during early summer and in winter (June and February) with especially high heat loads are reported. In addition, to test the applicability of computers to design ventilation systems, flow simulation was done using a workstation computer.

EXPERIMENT SETUP AND PROCEDURE

Figure 1 shows the experimental setup. All experiments were performed using a test room with 2.6 m by 2.8 m floor area and 3 m height that was housed inside a large laboratory building. The walls and roof of the test room are made of 50-mm-thick foampolystyrene board encased between 2-mm-thick aluminum plates. The floor is made of concrete slab covered with 5-mm-thick vinyl sheet. Ventilation air is introduced into the test room through a flat diffuser, which is connected to headers in order to realize uniform flow distribution over the diffuser. Air leaves the test room through a 2.5 m × 0.14 m exhaust open-

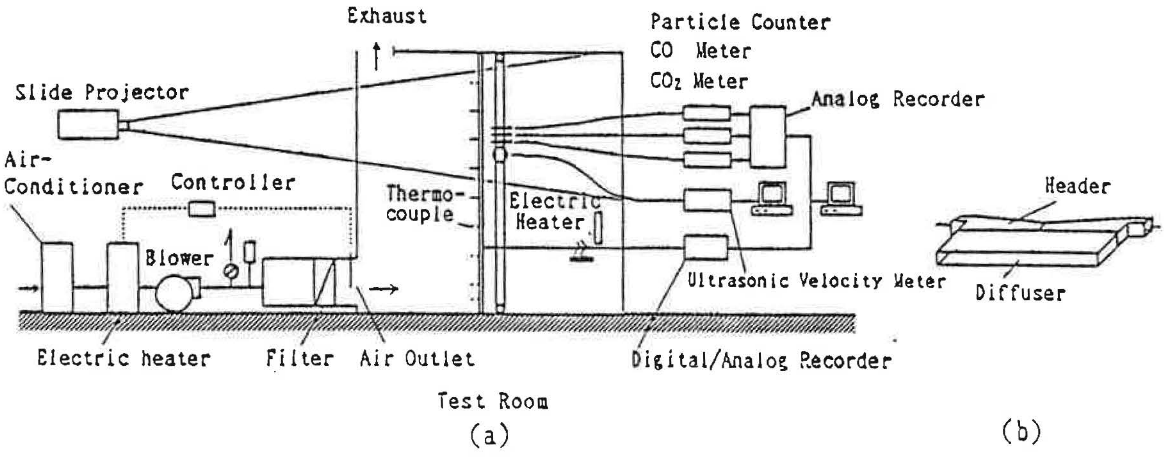

Figure 1 Experimental setup

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ing (fixed size and location) in the ceiling. The opening is located directly above the air outlet, with its long side (2.5 m) parallel to the wall containing the diffuser.

The heat load is supplied by a 2.4 m by 0.2 m (5-mm-thick) electrical plate heater suspended inside the room in such a manner that the long side (2.4 m) is parallel to the air outlet and the short side (0.2 m) is perpendicular to the floor. Its power is measured by a wattmeter.

The air temperature inside the test room is measured with eleven 100-μm copper-constantan thermocouples mounted to a vertical bar that can move across the test room. Also attached to this bar is a pulley-thread mechanism that carries the sampling tubes for measurement of the concentration distributions of particle, carbon monoxide, and carbon dioxide contaminants.

Experimental conditions are listed in Table 1. Experiments were carried out for various combinations of heater location, diffuser size and location, heater input, and ventilation rate. Heater locations I and II correspond to an office equipment place on top of an office desk, while heater location III corresponds to a standing office worker. The shape and dimensions of the diffuser were chosen to enable us to clarify the effect of diffuser location and dimensions b and l on the phenomena. Types 1 to 5 are as wide as the room and may represent a two-dimensional flow situation (the heater and the exhaust opening are also as wide as the room), which is relatively easy to analyze by computer simulation. Types 1 to 3 have equal areas but are located at different heights above the floor. Types 1, 4, and 5 have the same width but are of different opening heights. Types 6 to 8 represent narrow diffusers whose results, when compared with those of types 1, 4, and 5, allow the verification of the effect of width of diffuser. The three values (250 W, 750 W, and 1250 W) of heater input are possible heat load values for an office room. The 250 W and 1250 W inputs correspond, respectively, to heat generated by a lap-top microcomputer and a copying machine. The design value for cooling load in an ordinary office in Japan is about 150 W/m^2 or 1090 W for the present room, a value within the range of the heater input listed in Table 1. The ventilation flow rate ranges from 50 to 400 m^3/h or, for the present test apparatus, 2.3 to 18.1 air changes per hour.

Experiments were carried out in February and June, which are months representing winter and early summer in Japan. The temperature of the supply air for the experiments in June was about 19 °C, while that for the experiments in February was about 12 °C. From comfort considerations, 12 °C is rather low,

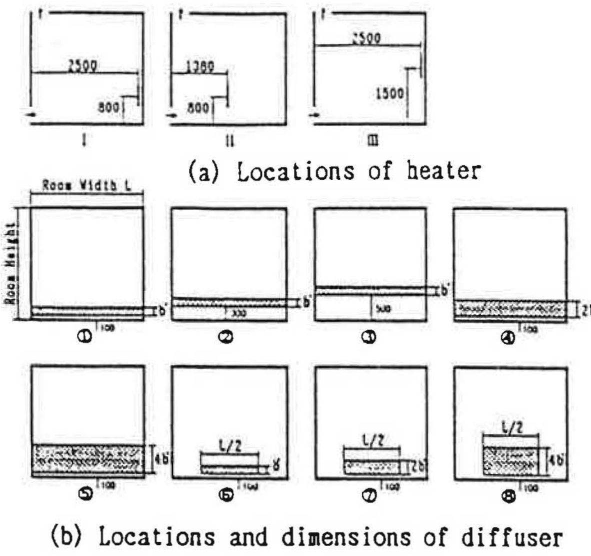


Figure 2

but when 19 °C was used for the experiments in February, heat loss through walls was large and strong downward natural convection was formed along the walls. This interacts with the ventilation supply air, causing mixing and disappearance of the clean zone.

The experiments were carried out at steady-state conditions. The flow rate and temperature of the supply air were maintained at desired values by the air-conditioner. After the air and wall temperatures of the test room attained steady state, measurements of temperature and velocity and observations of the height of the clean zone (or height of the smoke-free region) were made. For visualization of the clean zone height, the pollutant was smoke from burning insect repellent sticks located near the lower part of the heater. Visualization was done with all lights cut off, except for the thin plane of light coming from the projector. After adaptation, the contrast between smoke-free and smoke-filled regions could be detected with the naked eye. The observations were made through an acrylic window in the wall parallel to the plane of light. For experiments involving measurements of CO and CO_2 concentration distributions, pollutants were generated by a person sitting inside the test

TABLE 1
Experimental Conditions

Heater location	Air outlet location & dimension	Heater input W(W/m²)	Ventilation flow rate m³/h (1/h)	Supply air temperature °C
I	①	250(35)	50(2.3)	10 ~ 20
		750(104)	100(4.5)	
		1250(174)	200(9.1)	
	②	750(104)	300(13.6)	
	③		400(18.1)	
	④			
	⑤			
	⑥			
	⑦			
	⑧			
II	①			
III				

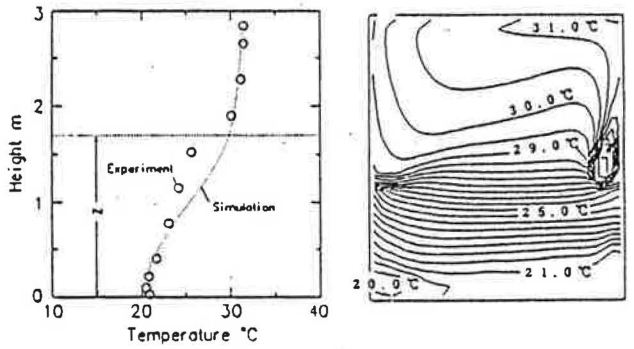
room and smoking 70-mm-long cigarettes at the rate of one every 20 minutes or a cycle of 5 minutes smoking and 15 minutes not smoking. Concentration measurements were made after a steady periodic variation of pollutant concentration was attained.

COMPUTER SIMULATION

To determine the applicability of the computer for designing displacement ventilation, the mass, momentum, and energy equations for two-dimensional flow inside the test room were solved using a control volume based, finite-difference discretization and algorithm of a software package. The computations were carried out on an engineering workstation computer. The boundary condition values for inlet flow rate and temperature and wall temperatures used in the simulation correspond to those of the experiments. The computer simulation did not include radiative heat transfer, and the values of heater input used in the computations were, therefore, only due to convective heat transfer, which is about half those in the experiments.

RESULTS AND DISCUSSIONS

Figures 3a and 3b show typical air temperature distributions in the test room. In Figure 3a, the measured temperature distribution along the central axis of the test room is compared with the results of simulation. In the figure, z is the observed height of the clean zone. Thermal stratification exists in the clean zone, while the temperature in the dirty zone is almost uniform, indicating the existence of almost motionless air in the clean zone and a strong horizontal flow toward the heater's plume in the dirty zone. The degree of stratification in the clean zone is an important parameter, as it limits the applicability range of displacement ventilation due to comfort restrictions (ISO 7730). In Figure 3b the isotherms in the clean zone are practically horizontal, showing the extent of floor area covered by the vertical temperature stratification.



(a) distribution along central axis of test room

(b) distribusion obtained by computer simulation

Figure 3 Temperature distributions

Figure 4 shows the temperature difference between 0.1 m and 1.1 m heights plotted against the heat load. According to the ISO 7730 recommendation, the temperature difference between these levels must not exceed 3°C. Based on this criterion, the figure shows that we have to select large flow rates when the heat load is large. In Japan, the design value for a mixed system usually ranges from 5 to 8 air changes per hour. Applying this range to the present system, the corresponding heat load limit that does not violate the above comfort criterion ranges from 70 to 100 W/m^2. The authors' results agree with Svensson's results (1989) of tests carried out in a room the same size as the present test room but equipped with a different type of air outlet. Due to the recent trend of increased automation and use of electrical equipment in offices, typical office cooling loads are now about 150 W/m^2, a load displacement with which ventilation alone may not be able to cope. Auxiliary means of cooling, such as radiation panels, should be considered.

Figures 5a and 5b show the velocity distributions at the middle cross section of the test room. In the upper zone, both the experimental and the simulation results indicate a strong horizontal flow caused by entrainment toward the heater's plume. In the clean zone, both results indicate practically motionless air (velocities below 0.1 m/s), except for the region occupied by the jet from the diffuser. The jet moves across the room, hits the opposite wall, and flows backward toward the diffuser before it diffuses. The relatively high velocity near the floor level combined with a low air temperature may cause a draft problem. For the present condition, the value of the draft parameter (Fanger et al. 1988) is approximately PD = 7%.

Figures 6a and 6b show the concentration distributions for particles and carbon monoxide pollutants, respectively. The pollutant concentration in the room fluctuated between C_{min} and C_{max} corresponding to a cycle of 5 minutes smoking and 15 minutes not smoking. The C_{max} for particles in the upper zone is about 20 times that of the clean zone, while the C_{max} for CO in the upper zone is about 5 times that of the clean zone. In the present example, the ventilation flow rate is 200 m^3/h, a typical design value for a mixed system of offices in Japan. The recommended limiting values for particle and CO concentrations for offices in Japan are 0.15 mg/m^3 and 10 ppm, respectively. The present results are well below these limiting values, and it can be inferred that the present system can easily cope with contamination control for two or three smoking persons in the room. The corresponding concentration distribution for CO_2 was also

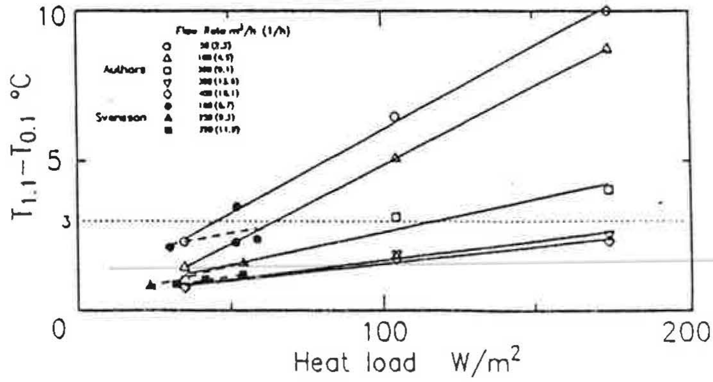


Figure 4 Temperature difference between 1.1 m and 0.1 m elevations above the floor as a function of the heat load

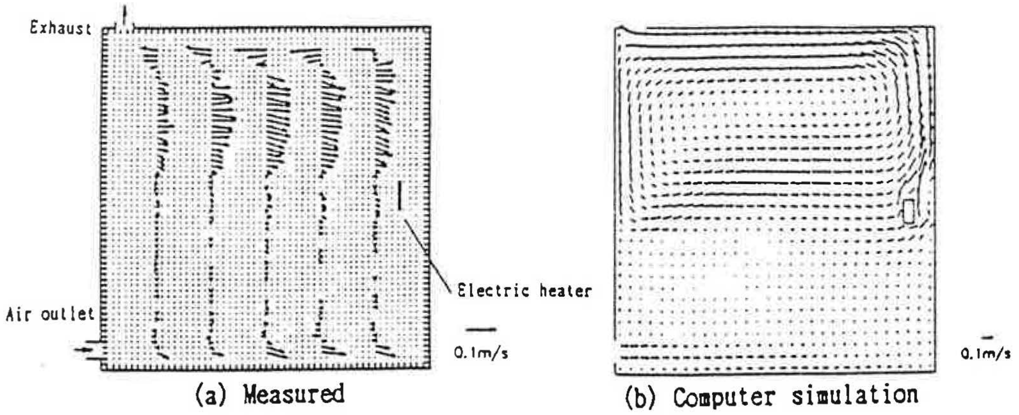


Figure 5 Velocity distribution in the middle cross section of the test room

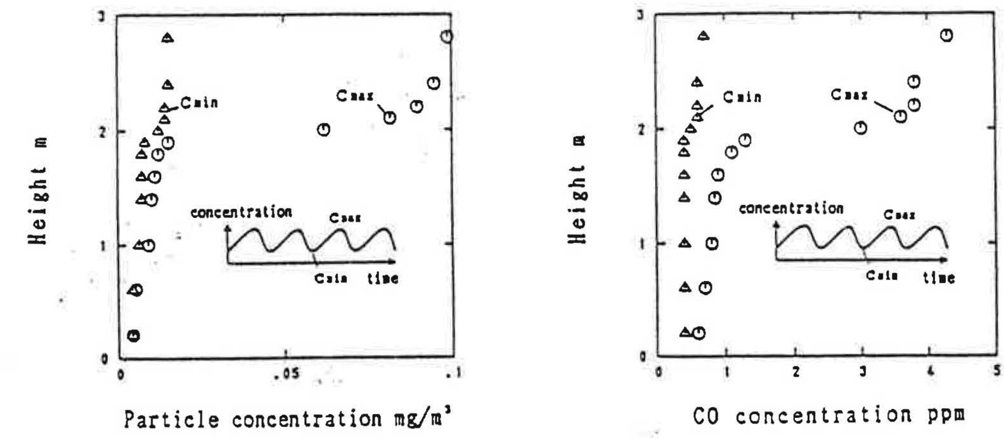


Figure 6 Pollutant distribution measured along the central axis of the test room

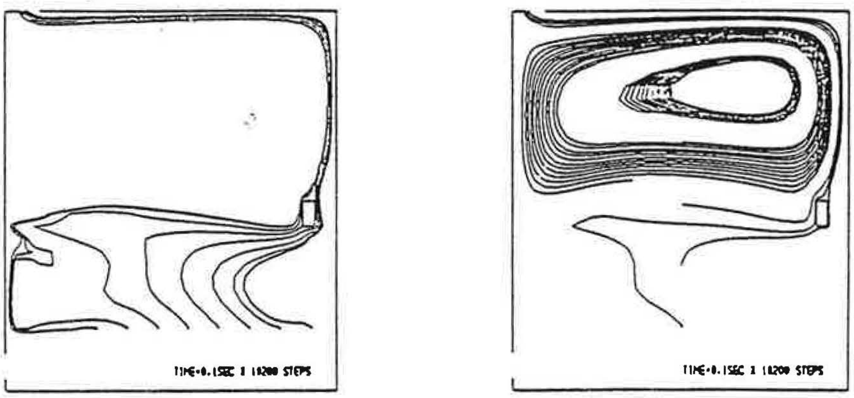


Figure 7 Streaklines or the path traveled by marker particles in the computer simulation

measured, but the results (1000 ppm) showed no distinct vertical distribution due perhaps to the high molecular weight of CO_2.

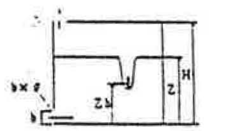
Figure 7 shows the streaklines calculated by the computer simulation. A streakline, which is a plot of the position of a massless marker particle variation with time, may indicate the path a particle travels. The left-hand side of the figure shows the paths traveled by ten marker particles from their respective starting points located 0.5 m above the floor. These particles first move upward, indicating a piston-like flow or displacement flow in the clean zone, before they bend toward the heater and proceed toward the exhaust. The total time for the particles to reach the exhaust is about 17 minutes. In the right-hand side of the figure, the particles initially released at locations higher than and fronting the heater recirculate and stay within the upper region until 17 minutes have elapsed.

The height of the clean zone is an important design parameter. While it is principally dependent on conditions of supply air (volume and temperature) and heat load conditions (location, temperature, or intensity), it is also affected by wall temperature level. Our experiments were performed in two different seasons (June and February), and clean zone heights differ slightly. For the experiments performed in June, the outdoor air was hot and wall (inner surfaces) temperature was about 1°C higher than room air temperature, while for those performed in February, the reverse situation holds and wall temperature is about 1°C lower. The clean zone height reported in this paper is a value corresponding to zero wall-to-air temperature difference obtained by interpolation using corresponding clean zone heights for June and February.

Figure 8 shows height of the clean zone plotted against ventilation flow rate. The height is strongly dependent on flow rate but more or less independent of the diffuser shape and location, heat input, and temperature of the supply air. The data of Sandberg and Blomqvist (1989) are higher than the present data due perhaps to differences in heat input, method for determining the clean zone height, and size of test room. At low ventilation rates, for example at 2.3 air changes per hour, the height of the clean zone is about equal to the height of the heater, a result that agrees with Sandberg's data. At ventilation rates greater than 18 air changes per hour, the height of the clean zone was not clearly detected. If the design values for mixing systems of 5 to 8 air changes per hour are applied to the displacement system, based on our results, the expected clean zone height is 0.3 m to 0.4 m greater than the heat source height. If the heat source is office equipment placed on top of a table, this clean zone height may be below the breathing level of a seated office worker. To alleviate this situation, it is suggested that heat-generating office equipment be located at higher levels.

In displacement ventilation, the supply air at temperatures lower than room temperature is introduced into the room at very low velocities. In this situation, the buoyancy force is comparable to inertia force and buoyancy has to be considered. The ratio of buoyancy force to inertia force is equal to an Archimedes number,

Symbol	Heater location	Inlet location & dimension	Heater input W(W/m²)
▽	I	①	250(35)
○	I	①	750(104)
△	I	①	1250(174)
○	I	②	750(104)
○	I	③	750(104)
○-	I	④	750(104)
-○	I	⑤	750(104)
●	I	⑥	750(104)
●-	I	⑦	750(104)
-●	I	⑧	750(104)
▢	II	①	750(104)
◇	III	①	750(104)



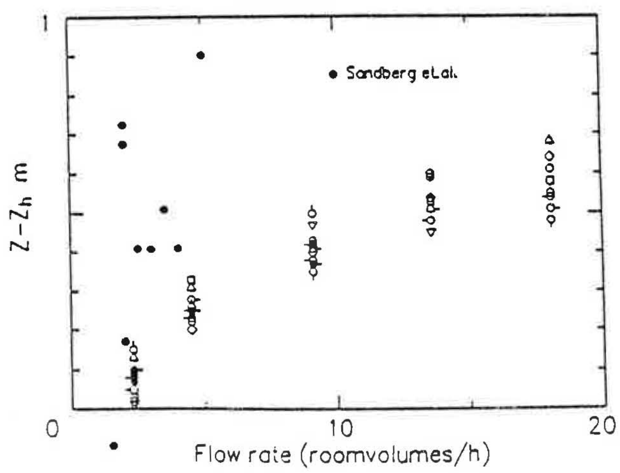


Figure 8 *Height of the clean zone as a function of the flow rate*

$$Ar = g\beta (T_a - T_{in})b/u_{in}^2$$

where

g = gravitational acceleration
β = coefficient of volumetric expansion
T_a = mean room air temperature
T_{in} = supply air temperature
b = height of diffuser opening
u_{in} = inlet velocity.

In Figure 9, clean zone height normalized with the square root of the area of the diffuser is plotted against an Archimedes number. The normalized height correlates with the Archimedes number as follows:

$$(Z - Z_h)/(bl)^{0.5} = 0.64\ Ar^{-0.2}$$

with a correlation coefficient of 0.94 and a standard error of 0.1.

CONCLUSIONS AND RECOMMENDATIONS

1. A clear boundary exists that separates the room into a polluted zone and a clean zone. Each zone has different velocity and temperature distributions.
2. Displacement phenomena transform to mixing phenomena at a ventilation flow rate of about 10 air changes per hour for narrow outlets and about 18 air changes for wide outlets. This transition is independent of heater location and supply air.
3. The height of the clean zone is strongly dependent on flow rate and the height of the heater. The relation between the height of the clean zone and the Archimedes number is correlated as follows:

$$(Z - Z_h)/(bl)^{0.5} = 0.64\ Ar^{-0.2}$$

4. From the comfort criterion of temperature stratification ($T_{1.1} - T_{0.1}$) < 3°C, the heat load should be below 70-100 W/m². For a high heat load, auxiliary cooling should be used.
5. To increase the height of the clean zone, the room should be insulated and heat-generating equipment should be elevated.
6. The applicability of computer simulation for analysis of displacement ventilation was made clear.

REFERENCES

Fanger, P.O., A.K. Melikov, H. Hanzawa, and J. Ring. 1988. "Air turbulence and sensation of draught." *Energy and Buildings*, Vol. 12, No. 1, pp. 21-39.

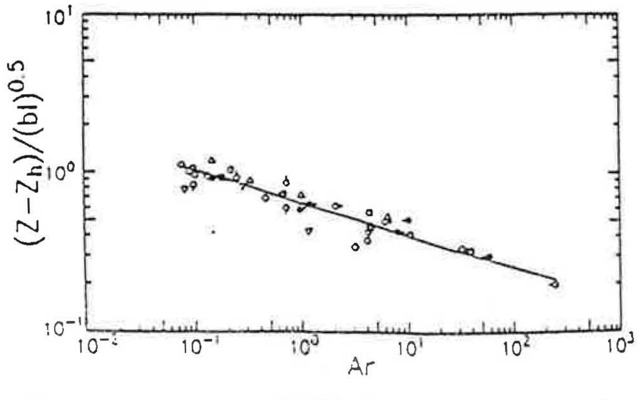


Figure 9 *Normalized height of the clean zone as a function of Archimedes number*

Miyatake, O., N. Nagafuchi, N. Suga. and I. Tanaka. 1986. "Studies on the internal characteristics of a temperature stratification-type thermal storage water tank." *Transactions of the Society of Heating, Air-conditioning, and Sanitary Engineers of Japan*, No. 32, pp. 35-43.

Sandberg, M., and C. Blomqvist. 1989. "Displacement ventilation systems in office rooms." *ASHRAE Transactions*, Vol. 95. Part 2, pp. 1041-1049.

Svensson, A. 1989. "Nordic experiences of displacement ventilation systems." *ASHRAE Transactions*, Vol. 95. Part 2, pp. 1013-1017.