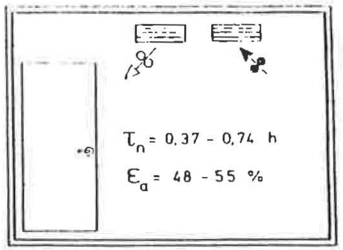
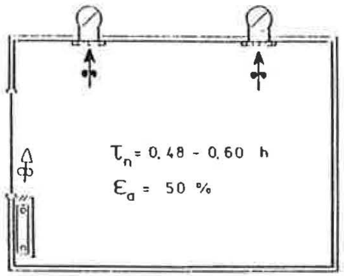
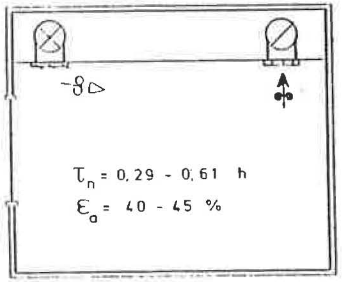


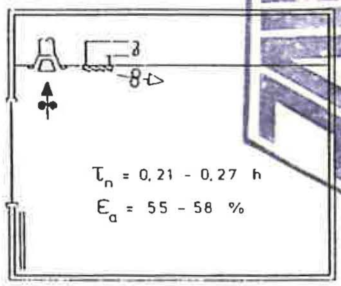
1h. Office, wall registers.



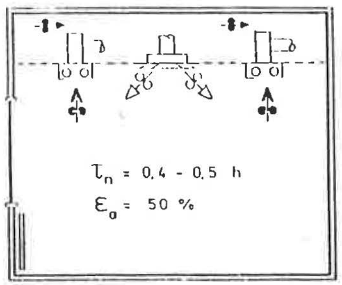
1i. Office, window induction units.



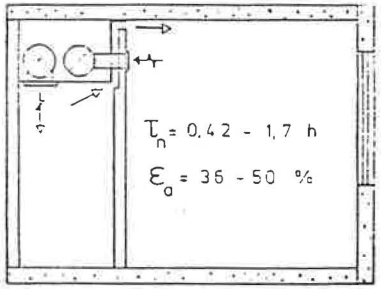
1j. Office, ceiling diffusers.



1k. Office, ceiling diffusers.



1l. Office, ceiling diffusers.



1m. Office, supply to hall.

Figures 1g-1m. Results of measurements with various ventilation patterns.

EXPERIMENTAL COMPARISON OF INDOOR CLIMATE USING VARIOUS AIR DISTRIBUTION METHODS

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Abstract

Several air distribution methods were analyzed on an experimental basis at Fläkt's Indoor Climate Center in Stockholm for the purpose of studying thermal comfort as well as air quality. The tests were conducted in a typical office module (15 m^2) during three cooling load situations: summer (45 W/m^2), spring/autumn (25 W/m^2) and winter (4 W/m^2). Climate data from the Stockholm area were used in all three cases. Test results were reported for jet-controlled air distribution with slot air diffusers, and for thermally controlled air distribution with wall-mounted as well as floor-mounted air supply devices.

Introduction

In the most common air distribution principle used to date for commercial and institutional ventilation, the air flow in the room is controlled by air jets from the supply air device. This method is designated jet-controlled air distribution. The air jets entrain ambient air, creating a steady circulating air stream with high degree of mixing air flow. This type of air flow is characterized by small pollution and temperature gradients in the room.

During the past few years, notably in the Nordic countries, a great deal of attention has been paid to an air distribution principle in which the air flow is controlled by thermal forces from heat sources in the room. The principle has therefore been designated thermally controlled air distribution. In this type of air distribution, undertempered air is supplied at low velocity and at floor level. The air is conveyed upwards to the ceiling by natural convection currents emanating from heat sources. At the ceiling, the air is exhausted. The resulting vertical temperature stratification provides a relatively stable displacement air flow in the lower part of the room, while the natural convection currents generate a mixing air flow in the upper part of the room. The relationship between the total strength of the heat source and the magnitude of the supply air flow will determine how far the mixing will penetrate down in the room and, consequently, the degree of mixing in the occupied zone. Displacement air flow is characterized by small spreading of heat and contaminants.

Thermal comfort conditions are stated in terms of temperature level, temperature gradient and air velocity, while air quality here is expressed in terms of air age and air pollution concentration. The air age indicates the rate at which room air is replaced with supply air, and thus can be considered as an indirekt measure of how quickly pollutants are removed from the room, provided that the contaminants are produced uniformly throughout the room from a homogenous source of pollution.

Experimental Facility and Procedure

The measurements were made in a typical office module with a floor area of 15 m^2 and a room volume of 40 m^3 (Fig. 1). The window is triple-glazed with an intermediate venetian blind. Other surfaces in the room are well insulated. Three load cases were selected - summer, spring/autumn and winter - using Fläkt's VENTAC computer program for calculation power and energy requirements in buildings. In new construction today, typical interior heat loads generally require the supply of cooling to a room throughout the year. Based on climate data from Stockholm, and loads in the form of lighting, people and computers (Table 1), the required cooling capacity was calculated at 45, 25 and 4 W per m^2 of floor area for the three load situations. Other design data are shown in Table 1.

Table 1. Design data for the three load cases

Load case	Loads: Ceiling lights (W)	Loads: Desk lamp (W)	Loads: Person (W)	Loads: Computer (W)	Loads: Window (W)	Cooling load (W)	Cooling load (W/m²)	Air flow (l/s)	Air flow (l/s.m²)	Temp-diff supply/exhaust (°C)	Temperatures: Room (°C)	Temperatures: Window (°C)
1a. Summer	290	60	75	100	160	685	45	57	3,8	10	25	33
1b. "	"	"	"	"	"	"	"	71	4,7	8	"	"
2. Spring/Autumn	240	60	75	0	0	375	25	39	2,6	8	23	23
3. Winter	105	10	75	0	-120	60	4	10	0,67	5	22	15

Test results are shown for one air distribution method providing a mixing air flow - jet-controlled air distribution with slot diffusers mounted in the ceiling by the facade wall - and for two methods that partly produce a displacement air flow - thermally controlled air distribution with low-velocity supply air devices mounted at the lower part of the corridor wall, and the same distribution method with floor air supply devices. Fig. 1 shows dimensions, positions and furnishings. In all three air distribution methods, the exhaust air device of the slot air diffuser type, is mounted in the ceiling by the corridor wall. The supply air device has been adjusted for an air flow of 3.8 l/s, m^2 floor area in the summer case. The reduction of the air flow to 68 % in the spring/autumn case, and to 18 % in the winter, simulates a VAV system. The slot diffusers used in test are Fläkt's type CTA(A,B) (2 for supply air and 3 for exhaust air). The low-velocity device is of the type Diff-A (2), while the floor device is Krantz type BR (9 placed in a square pattern at a distance of 1.2 m).

To analyze the thermal comfort provided by the various air distribution methods, the temperature and air velocity were determined at 50 and 30 measurement points, respectively, in the room. The test methodology is described more in detail in [1].

The air quality was analyzed by determining the air age in 4 measurement points at the workplace and in 6 points at places in the room that were more thermally neutral [1]. The local age of the air (at point p), which indicates the amount of time required for the supply air to reach the point in question, is designated $\bar{\tau}_p$ and was determined with tracer gas measurements. The mean age of the room air, $\langle\bar{\tau}\rangle$, which is proportional to

the time it takes to exchange the air in the room, was determined only through measurements in the exhaust air [2]. The air age is given in dimensionless form by normalizing the age with the room's nominal time constant $\tau_n = V/q$, with V = room volume and q = air flow.

The air quality was also analyzed by determining the contaminant concentration at equivalent places with a source of pollution in the vicinity of the workplace (Fig. 1). The stationary pollutant simulated cigarette smoke with a density of 1.14 kg/m^3. The local concentration of the contaminants is stated in dimensionless form by normalizing the equilibrium concentration (at point p), $C_p(\infty)$, with the corresponding concentration in the exhaust air, $C_e(\infty)$. The normalized age as well as normalized concentration can very between 0 and ∞ and is equal to 1, when total mixing air flow is obtained.

Results and discussion

Thermal comfort

For good thermal comfort, the desired air temperature has to be created without causing excessively high air movements or temperature gradients in the room. In accordance with ISO/DIS 7730, vertical temperature differences below 3°C between 1.1 and 0.1 m from the floor (head and ankle level when a person is seated) are required, as well air velocities in the occupied zone (up to 1.8 m above the floor) of below 0.25 m/s during the summer, and 0.15 m/s during the winter.

Jet-controlled distribution with slot diffusers provided a stable, circulating air stream with the highest velocities being 0.25 m/s in the summer, and 0.18 m/s in spring/autumn, at floor level. In contrast, the circulation air stream was not fully developed in the winter, due to the VAV function with a sharp reduction in air flow, and air velocities below 0.10 m/s were obtained throughout the room. The temperature gradients in the room were very small during all three load cases (see Fig. 2).

The thermally controlled air distribution method produced low air velocities throughout the room, except at floor level in the vicinity of the supply air device. With the low-velocity air supply device, the highest velocities measured 0.1 m above the floor were 0.22 m/s in the summer, 0.16 m/s in the spring/autumn, and less than 0.10 m/s in the winter. With the floor air supply devices, the highest air velocities were obtained in the areas immediately surrounding the supply inlets. The highest value, 0.12 m/s, was measured in the summer. In thermally controlled air distribution, the vertical temperature gradient limited the supplied cooling power. In the summer, the low velocity device produced a maximum temperature difference of approximately 5°C between 1.1 and 0.1 m above the floor (Fig. 2), which does not meet ISO standards. The corresponding temperature gradient obtained using the floor devices was a few degrees lower, approximately 3.5°C, since the devices produce some mixing air flow in the lower part of the room. To reduce the temperature gradient when low-velocity devices were used, the air flow in the summer was increased 25 %, and the difference in the supply and exhaust air temperature decreased to a corresponding extent (load case 1b in Table 1). The previously measured temperature difference of about 5°C was reduced to about 3.5°C as a result (Fig. 2). In the spring/autumn, slightly lower temperature gradients were obtained. The ISO standard of $3^{o}C$ was managed with the low-velocity device, while the value with the floor device was about $1^{o}C$ lower. In the winter, the temperature gradients with both air distribution methods were less than $2^{o}C$ throughout the occupied zone, since the natural convection currents from windows and heat sources - which were large in relation to the small air flow of 0.67 l/s, m^2 - caused a mixing effect in the room. Note that the difference in temperature level prevailing in the room during the various tests had a negligible effect on both the temperature gradient and air velocity.

Air quality

Fig. 3 shows the air quality as the age of the room air at the workplace normalized by the nominal time constant. As shown in the figure, the slot air diffusers provided a strong mixing air flow with values for the normalized age of just above 1 in the summer and spring/autumn cases, while a small temperature stratification in the winter (Fig. 2) caused a slight short-circuiting air flow in the upper part of the room (about 1.5 m above the floor). With both low-velocity and floor supply air devices, the larger temperature stratification clearly caused a displacement air flow in the lower part of the room (up to about 1.5 meters above the floor). In this zone the age of the air in the winter was approximately 25 % lower than in total mixing air flow, and when compared with the other load cases, was approximately half as large. Note, however, that the air flow in the summer and spring/autumn cases was respectively 5.7 and 3.9 times as high as in the winter case, why the air exchange rate also with mixing air flow was substantially higher in these load situations, compared with the winter case. The small age differences between the two thermally controlled air distribution alternatives can be explained by the fact that the floor devices provided some mixing effect in the lower part of the room. In the upper part of the room, even at standing height (1.8 m above the floor), strong mixing was obtained with normalized age values of more than 1. The differences concerning the entire room will not be as large as those in the lower zone, which is directly shown by the comparison in Fig. 3 between the mean age for the entire room and the local age in the lower zone. The uncertainty of the measured values was greatest in the winter case, since the impact of air leakage during the tests was greatest when the air flow was low.

The stationary concentration measurements, with the pollution source in the vicinity of the workplace, symbolized passive smoking without dynamic disturbances. As shown in Fig. 4, the convection currents produced by body heat attracted the pollutant, resulting in high concentrations in the breathing zone, as long as the air movements created by the supply air device did not dominate and remove the pollutant from the workplace. Slot diffusers were capable of producing a dominating air stream in the summer and spring/autumn cases, with air velocities of approximately 0.10 m/s at the workplace, while the convection current around the body dominated in the winter, due to the sharp reduction in supply air flow. However, in this load situation, the concentration in the breathing zone was again lower with the jet-controlled air distribution method, compared with the thermally controlled air distribution alternative, which caused very high concentrations in the breathing zone in all three load situations.

Conclusions

Good thermal comfort in accordance with the ISO standards was maintained under the circumstances with all three air distribution methods in both the winter and spring/autumn cases. However, in the summer case, the ISO standard sets the upper limit to cooling power supplied to the room. The jet-controlled air distribution method could supply a maximum cooling power of about 45 W/m^2, with a difference of $10^{o}C$ between the supply and exhaust air temperature, and without causing overly high air velocities. In contrast, the thermally controlled air distribution methods required a smaller temperature difference and consequently, a higher air flow, to supply equally high cooling power without causing an excessively high vertical temperature gradient in the room. With low-velocity devices, the difference between the supply and exhaust air temperature has to be kept below $7^{o}C$ approximately, while the floor devices permit a difference of $9^{o}C$.

The air quality, in terms of the air exchange rate, was very good with all three air distribution methods in both the summer and spring/autumn cases, due to the high air flows of 5.7 and 3.8 l/s, m^2. In winter with the air flow of 0.67 l/s, m^2 the differences in air age were too small to ascribe better air quality to the thermally controlled air distribution methods, especially since the concept is only an indirect measure of room air cleanliness. The simulation of passive smoking at the workplace showed that convection currents around a seated person have the ability to attract smoke, resulting in very high concentrations in the breathing zone when the thermally controlled methods were used. A higher concentration in the breathing zone was also caused by the jet-controlled method in the winter case.

In summary, it may be stated that thermal comfort in an office room is affected considerably by the air distribution technique, while the air quality is primarily determined by the magnitude of the air flow.

References

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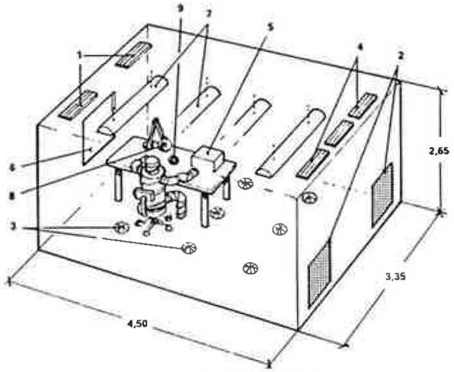

1. Slot diffusers
2. Low-velocity devices
3. Floor devices
4. Exhaust air device
5. Computer
6. Window (1,2 x 1,2 m)
7. Ceiling lights
8. Desk lamp
9. Contaminant source (0,1 m above the desk and 0,4 m from the vertical measurement line at the workplace)

Fig 1 Three different air distribution methods were tested in an experimental office module. Air was supplied to the room through slot diffusers, low-velocity devices and floor devices.

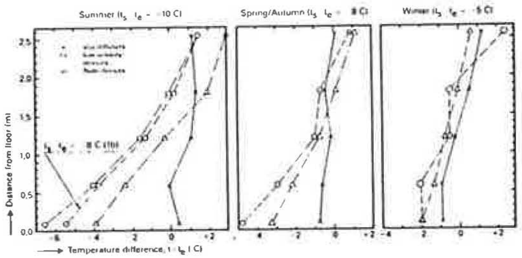

Fig 2 Maximum vertical temperature gradient for the three air distribution methods (t_e indicates exhaust air temperature)

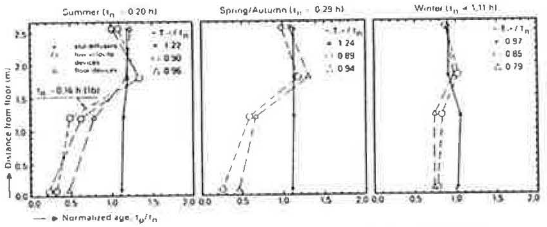

Fig 3 Normalized age of air at the workplace, τ_p/τ_n, and the normalized mean age for the entire room, $\langle\tau\rangle/\tau_n$

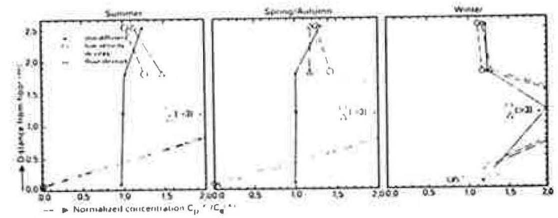

Fig 4 Normalized concentration of contaminant at workplace (C_e indicates exhaust air concentration in kg N_2O/kg air)

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VENTILATION EFFECTIVENESS AND ADPI MEASUREMENTS OF THREE SUPPLY/RETURN AIR CONFIGURATIONS

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Abstract

The two purposes of a building ventilation system are (1) to remove indoor contaminants and (2) to provide thermal comfort. Ventilation effectiveness is measured using tracer gas techniques and age distribution theory to compare the actual delivery rates of outside air at specific locations to those projected for the case of perfect mixing. ADPI is a thermal comfort parameter which compares the effective draft temperature as determined by dry bulb temperatures and air speeds to acceptability criteria established by ASHRAE. Three different supply/return configurations were evaluated for a recirculating constant volume system in a steady heating mode under standard conditions for an office space. Ventilation effectiveness ranged from 0.57 to 0.76 at points in the occupied zone. ADPI measurements ranged from 81 to 94 %. Short circuiting of supply air to the return air inlet was evident in tests of each configuration. More research is required to characterize the performance of existing systems and develop more effective ventilation strategies.

Introduction

Ventilation systems in the United States are designed to recirculate air at rates of 4 to 10 air changes per hour in order to satisfy internal cooling loads and provide acceptable thermal comfort. The flow rate of outside air required to remove indoor contaminants is often set at recommended minimums as a result of energy considerations. These recommendations follow from tests and calculations which are based on perfect mixing of indoor contaminants and outside air. The recommended minimum flow rate of outside air for an office space ranges from 2.5 to 10 l/s · person (1) which for an occupant density of 7 occupants/100 m^2 translates into ~ 0.2 to 0.8 air changes per hour or between 5 and 25 % of the total air circulation rate. While this relatively high rate of air circulation to outside air flow suggests that the delivered outside air should be well mixed with the indoor air the few tests conducted to date (3,4,5) indicate that the actual delivery rates of outside air to points in the occupied zone may be substantially less than those projected assuming perfect mixing. The effectivness of a ventilation system to deliver outside air to the occupied zone of a building is effected by a number of variables including the configuration of the supply/return air circuits, the rates of air circulation and outside air flow, the mode of operation (e.g. cooling, heating), and the presence of interior partitions. In this paper we compare measurements of ventilation effectiveness and thermal comfort for three supply/return configurations of a recirculating constant volume system in a steady heating mode under recommended minimum ventilation rates for an office space (1).