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Displacement ventilation in a lecture hall, a case study

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DISPLACEMENT VENTILATION IN A LECTURE HALL, A CASE STUDY

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Introduction

The lecture hall at Helsinki University and its ventilation system have been in use since 1988. The system is displacement type, the air being supplied to the lower part of the occupied zone at a temperature close to room air temperature. There have been some complaints concerning draughts due to low air temperature in the bottom of the hall. The HVAC-laboratory was asked to analyze the performance of the ventilation system in winter 1989.

The lecture hall. The lecture hall is shown in figure 1. It has 270 seats in 12 rows. The vertical height difference between two rows of seats is 0.32 m. The rear seats in the 12th row are 3.84 m above the bottom level. The floor is rectangular with an area of 320 m^2. The maximum ceiling height is 5.8 m at the front of the hall and the minimum ceiling height is only 2.6 m at the main entrance. The air volume is about 2000 m^3. In normal use the heat load from the ceiling lighting is about 3 kW. With 270 students (85 W) in the hall the total heat load is about 26 kW which corresponds to about 80 W/m^2. Three panel radiators at the both side walls of the hall were used to cover the heat losses. The height of the single radiator was 1,5 m and width 2,0 m. Their locations were at the levels of 2nd, 6th and 12th row.

Ventilation. The design value for the supply air flow rate was 10 l/s per seat. The total design value for the supply air rate was 2700 l/s. The measured value was 2630 l/s. The air exchange rate was about 5 ach. No return air was used. The supply air temperature was kept at 18°C with a sensor placed at a height of 1.5 m above the floor level on the rear wall of the hall. The air was supplied to the hall through four devices, two at the back and two at the front of the hall. The supply air rate at the front wall of the hall was 1800 l/s and at the rear wall 900 l/s. The exhaust air grille was in the ceiling at the rear of the hall. The lay-out of the ducting was very simple and low-cost. This was the main reason why the hall had only four supply air devices, and one exhaust opening close to the entrance.

Air supply devices. The supply air devices were normal, commercially available models. Their section was semicircular and they were placed near the corners of the hall. The free area was 85% of the total face area. The height of the supply air devices on the rear wall was 1.5 m and the diameter was 0.5 m. Values for the devices on the front wall being 2.0 m and 0.7 m. The total value of the free face area of the devices was 9.0 m^2. The mean face velocity of the devices at the front of the hall was 0.30 m/s and at the rear side 0.35 m/s.

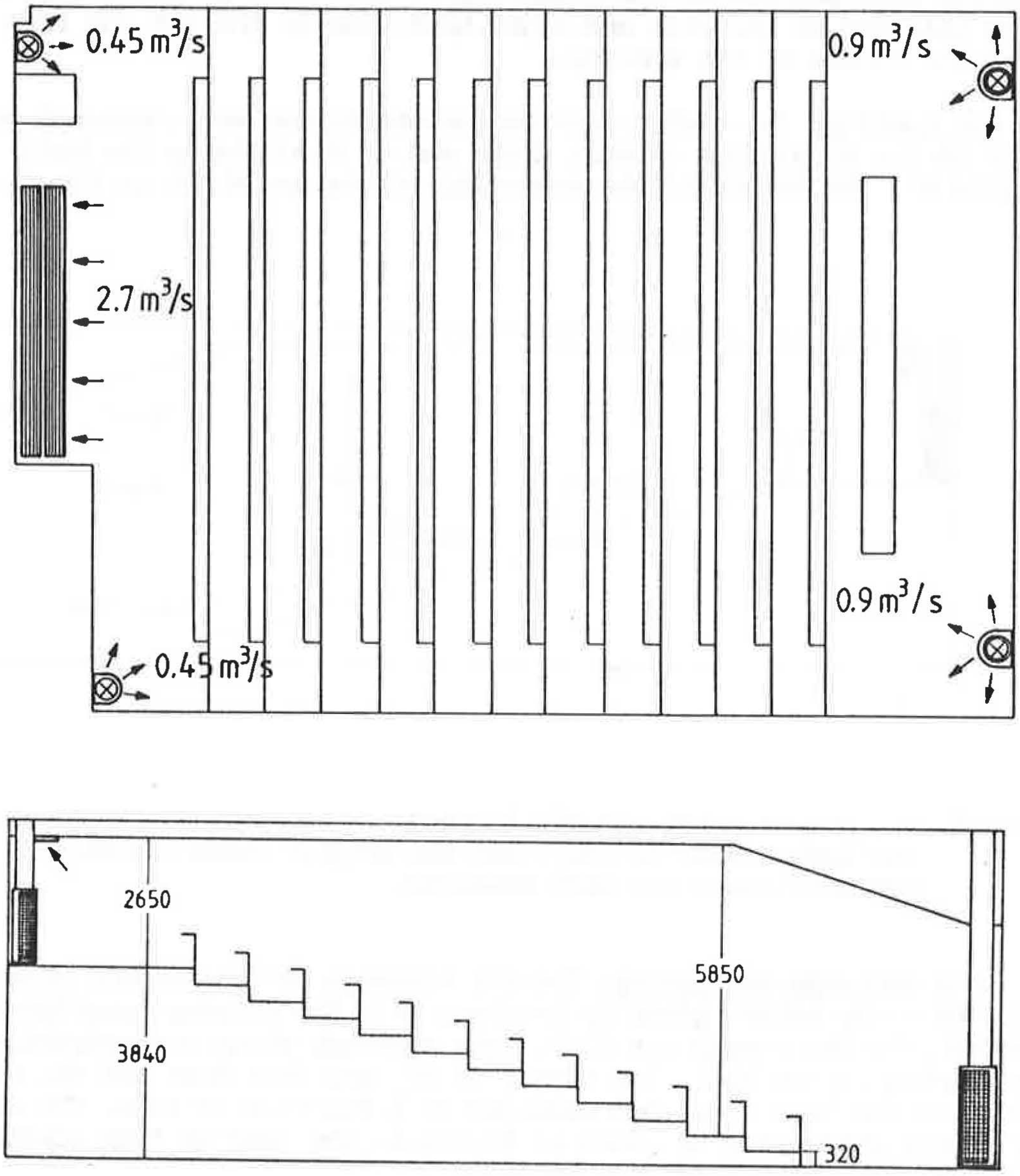

Figure 1. The lecture hall.

Measurements

Temperature. The air temperature was measured continuously in the supply and exhaust duct and at 9 locations in the hall. The placement and the height of the measuring points are shown in figure 2. The vertical temperature difference between the levels of 1.1 and 0.05 m above the floor was measured only while measuring the mean velocities.

Mean velocity. The mean velocity was measured with a multi-point thermistor anemometer (3) at heights of 0.05, 0.15, 0.6, 1.1 ja 1.7 m. The measuring time was 180 s. The measurements were made at the bottom level of the hall and at the left and right hand side of the 1st and 12th rows and in the middle of the 6th row.

Air quality. The carbon dioxide concentrations were measured continously in the supply and exhaust ducts and at 7 points in the hall. The location and the height of the measuring points are shown in figure 2.

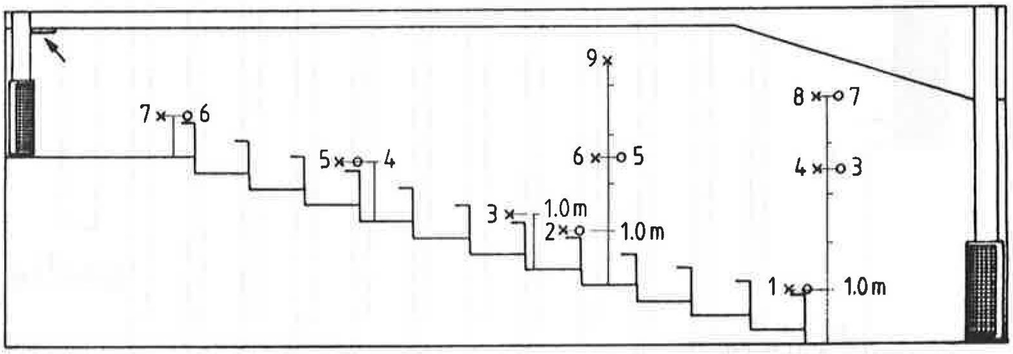

Figure 2. The points where the air temperature (x) was monitored and the carbon dioxide (CO_2) and the sulphur hexafluoride (SF_6) concentrations (o) were measured.

Air exchange efficiency. The air exchange efficiency was measured using the decay method given by Sandberg (4). The sulphur hexafluoride was injected into the supply air duct. Once a steady state concentration had been reached in the hall, the supply of SF_6 was shut down and the concentrations in the hall were then measured as a function of time. The measuring points where same as shown in figure 2. The nominal time constant of the air τ_n was calculated from equation 1.

$$\tau_n = \frac{\int_0^\infty C_e(t)dt}{C_o} \qquad (1)$$

where $C_e(t)$ is the concentration in the exhaust as a function of the time.

C_o is the steady state concentration at the beginning.

The mean age of the room air $\langle\tau\rangle$ was calculated from equation 2.

$$\langle\tau\rangle = \frac{\int_0^\infty t C_e(t)\,dt}{\int_0^\infty C_e(t)\,dt} \qquad (2)$$

Using the results of equations 1 and 2, it is possible to calculate the mean air exchange efficiency ϵ_a. This was obtained by dividing the nominal time constant by the average residence time of the room air which is twice its mean age.

$$\epsilon_a = \frac{\tau_n}{2\langle\tau\rangle} 100\% \qquad (3)$$

Pollutant removal effectiveness. The pollutant removal effectiveness shows whether the air quality in the occupied zone is better or poorer than in the exhaust. The local values of the pollutant removal effectiveness was calculated from equation (4).

$$\epsilon_p = \frac{C_e - C_s}{C_p - C_s} \qquad (4)$$

where C_e is the concentration of CO_2 in the exhaust

C_p is the concentration of CO_2 at the measurement point

C_s is the concentration of CO_2 in the supply

Loads. Lighting was constant during measurements. Small candles were used to simulate the occupancy. The heat production of a single candle was measured as 30 W and the CO_2 release was 3.4 l/h. 800 candles were needed to generate the same heat load as the full occupancy (270 students). The release of CO_2 corresponds to a load of 130 occupants. In the results the measured increases of the CO_2 concentration were multiplied by two to obtain concentrations close to real human occupancy.

Cases. The measurements were made with the simulated heat loads of 270 (100%) or 70 (25%) occupants and during lecture when there were 60 students in the hall. The supply air flow was 2700 l/s and 67% (1800 l/s) of it was supplied at the front and the rest (900 l/s) at the rear wall of the hall. Same tests were made when the supply air flow was reduced to 1600 l/s. The supply air flow was reduced to 2200 l/s during the tests when all air was supplied at the front of the hall. In that case no measurements were made during the lecture. The cases are shown in table 1.

Table 1. The supply air flow and occupancy in cases 1-8.

1) An air flow of 2700 l/s was supplied from the front and rear devices in the hall with a ratio of 2:1. The load was 270 persons.

2) An air flow of 2700 l/s was supplied from the front and rear devices in the hall with a ratio of 2:1. The load was 70 persons.

3) An air flow of 2200 l/s was supplied from the front devices in the hall. The load was 70 persons.

4) An air flow of 2200 l/s was supplied from the front devices in the hall. The load was 270 persons.

5) An air flow of 1600 l/s was supplied from the front and rear devices in the hall with a ratio of 2:1. The load was 270 persons.

6) An air flow of 1600 l/s was supplied from the front and rear devices in the hall with a ratio of 2:1. The load was 20-60 persons.

7) An air flow of 1600 l/s was supplied from the front and rear devices in the hall with a ratio of 2:1. The load was 70 persons.

8) An air flow of 2700 l/s was supplied from the front and rear devices in the hall with a ratio of 2:1. The load was 60 persons.

Results and discussion

Flow patterns

The air flow pattern in the hall was visualised with smoke, which was mixed with supply air in the duct or released in the zone of occupancy. Figure 3 shows the air movement in case 1. In the uppermost figure, smoke was supplied through the device in the front wall of the hall. In the next figure the smoke was supplied through the device in the rear wall of the hall. In the bottom figure emissions from occupants were simulated by releasing smoke in the middle of the 5th row.

In the first case, smoke moved rapidly and straight over the seats to the exhaust air opening at the rear side of the hall. Some of the air was circulated at the front of the hall, just over the supply air devices. There was a clear stagnation zone. Several minutes after the smoke had disappeared elsewhere in the hall there was still some stratified smoke above the lecturer.

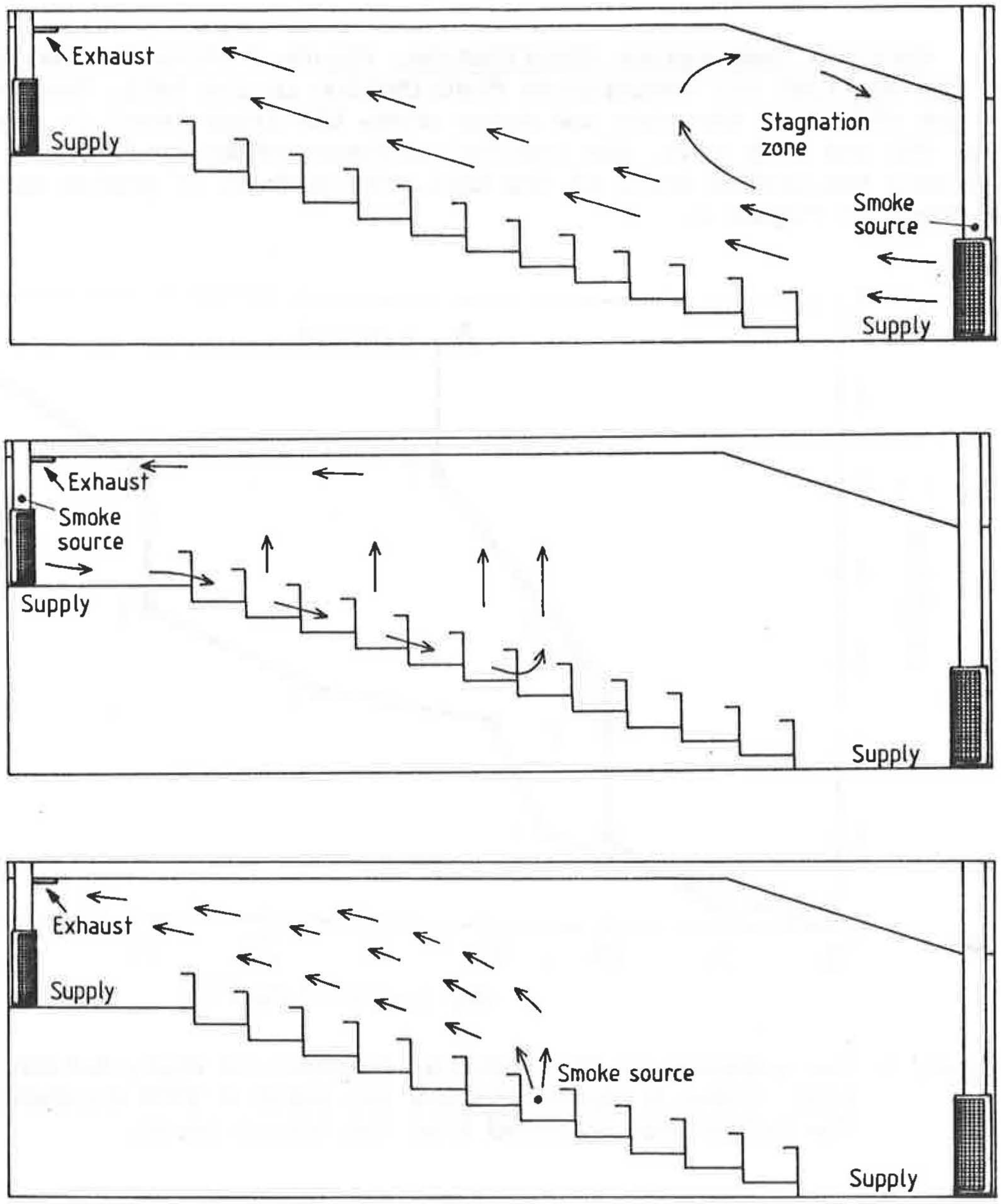

Figure 3. Air flow patterns in the hall. The smoke was supplied from the supply air devices at the front of the hall (upper case), at the rear of the hall (middle case) and in the 5th row (bottom case).

When the smoke was supplied through the supply air device in the rear wall, the smoke flowed down along the side balcony. The temperature difference between the air flow and the environment decreased as the air flowed towards the bottom of the hall. At the height of the 5th or 6th row the air flow was isothermal and stopped moving down. The radiators at the side walls were on in spite of the heat load in the hall. The convective flow caused by radiators rose the smoke from the side balconies up to the ceiling level. Smoke released from the 5th row moved straight and rapidly to the exhaust.

Thermal conditions

<u>**Vertical temperature distribution.**</u> Figure 4 shows the two basic cases of the vertical air temperature distribution in the hall. The points were in the middle of the rows one meter above the floor level in the 1st, 5th, 6th, 9th and 12th rows. The vertical distance shown in figure 4 is measured from the bottom level of the hall. The numbers of points are the same as shown in figure 2.

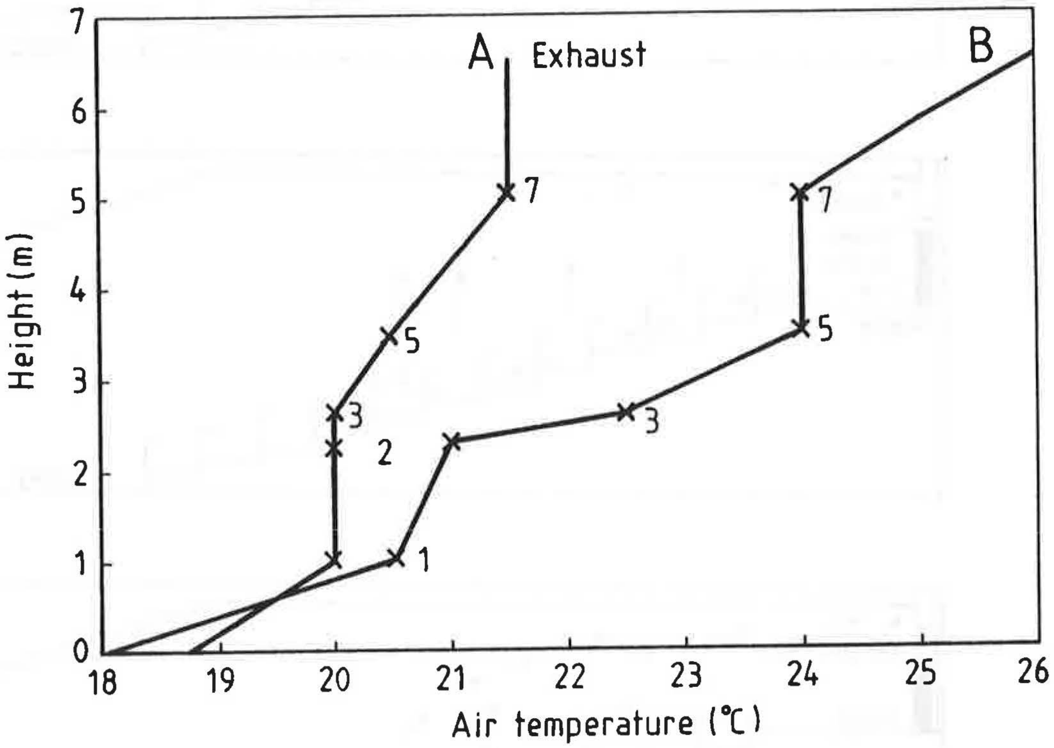

Figure 4. Two examples of the vertical temperature distribution in the hall. Curve A represents 25% and curve B 100% occupancy. The height was measured from the bottom level.

It is useful to study the results in two larger groups. The first group consists cases of 2,3,6,7 and 8 (curve A in figure 4). In these cases the occupancy and the heat load were low and the temperature difference between the exhaust and the supply air was between 2.5-4.0 K. The difference between air temperatures at the 12th and 1st rows was low, between 1.0-2.0 K. In the other cases 1,4 and 5 (curve B) the occupancy and the heat load were high and the temperature difference between the exhaust and the supply air was between 8.0 and 9.5 K. The difference between the air temperatures at the 12th and 1st row was high, between 3.5-5.0 K.

The supply air temperature was low, 18.0-18.5°C. At the front of the hall in the zone of occupancy, the air temperature was between 18 and 19°C. In the 1st row at the ankle level, the air temperature was less than 20°C. Even in the 6th row at the same level it was only 20°C. In the 11th and 12th rows the air temperature at the ankle level was usually over 20°C.

At the head level the air temperature was in the 1st row 20°C in all cases. With low occupancy the air temperature in the 6th row was about 20.5°C and with full occupancy between 22 and 24°C. In the 12th row the air temperature was about 22°C with low occupancy. With full occupancy the air temperature in the upper part of the hall (rows between 9th and 12th) at the head level was 24°C when the supply air rate was 2700 l/s (10 l/s per occupant). With supply air flow of 2200 l/s (8 l/s per occupant) the air temperature was between 24 and 25°C. It increased to 26°C when the supply air flow was reduced to 1600 l/s (6 l/s per occupant).

The vertical temperature difference between 1.1 and .05 m in the 1st row did not exceed 3 K. In the 6th row it was 4 K during full occupancy. In the 11th and 12th rows, if some of the air was supplied from the rear wall, the vertical temperature difference was highly dependent on the temperature difference between the exhaust and the supply. Thus the highest measured value was 6 K with full occupancy. The recommended (ISO) limits for vertical temperature difference were exceeded and the conditions were thus unacceptable.

To improve the thermal conditions in the lower part of the hall the supply air temperature should be increased to 19-20°C. This will increase the air temperature in the upper part of the hall to 25-26°C if the supply air rate per occupant is 10 l/s or less. This requires a supply air rate of 10-15 l/s per person. The design air flow of 2700 l/s is enough only for 80% occupancy. With full occupancy it is not possible to obtain satisfactory thermal conditions in the whole hall.

The location of the exhaust air grille forces the air to flow trough the occupied zone of 12th row where the vertical distance between the ceiling and desk is only 2.0 m. This may be one reason for the high air temperatures in the upper part of the auditorium. The air temperature near the ceiling over the lecture was 1-2K higher than in the exhaust. Removing the exhaust air grille there the air temperature in the upper part of the auditorium will probably decrease.

Draught

The predicted percentage of dissatisfied (PPD) was estimated from equation 5 (1).

$$PPD = (3.143+0.37\bar{v}Tu)(34-t_a)(\bar{v}-0.05)^{0.622} \qquad (5)$$

where Tu = the degree of turbulence

$$Tu = \frac{s}{\bar{v}} \cdot 100\%$$

s = standard deviation of the mean velocity, m/s

$\bar{v}$ = mean velocity, m/s

t_a = air temperature, °C

The degree of turbulence was between 20 and 55% when the mean velocity was between 0.1 and 0.2 m/s. At the bottom of the hall the mean velocity was less than 0.22 m/s. At the left and right sides of the 1st row the mean velocity was 0.15 m/s at ankle level (0.05 m) when the supply air flow through the devices at the front side of the hall was 1800 l/s (the cases 1,2 and 8).

It increased to 0.25 m/s when the air flow from the front wall was raised to 2200 l/s. The corresponding velocities at neck level (1.1 m) were below 0.1 and 0.30 m/s. In the middle of the 6th row, at ankle level the mean velocities were less than 0.1 m/s but at the neck level 0.2 m/s with full load. The most draughty places were on the left and right hand sides of the 12th row due to the supply air devices near the seats. At the ankle level the mean velocity was 0.30 m/s and at neck level 0.15 m/s.

The predicted percentage of dissatisfied based on equation (5) was 10% at the ankle level and at the neck level less than 10% in the 1st row when the supply air flow through the devices at the front of the hall was 1800 l/s. With an air flow of 2200 l/s supplied from the front devices the PPD value at the neck level in the 1st row increased to 40% and to 25% at the ankle level. In the 6th row at the neck level the PPD value was 20%. When the supply air devices in the rear wall were used, the PPD value in the 12th row at the ankle level was 50% and at the neck level less than 10% due to high air temperature (26°C).

The CO_2-concentration in the hall

Figure 5 shows the three basic cases of the carbon dioxide concentration at the middle of the 1st, 5th, 9th and 12th rows one meter above the floor. The vertical distance is measured from the bottom of the hall.

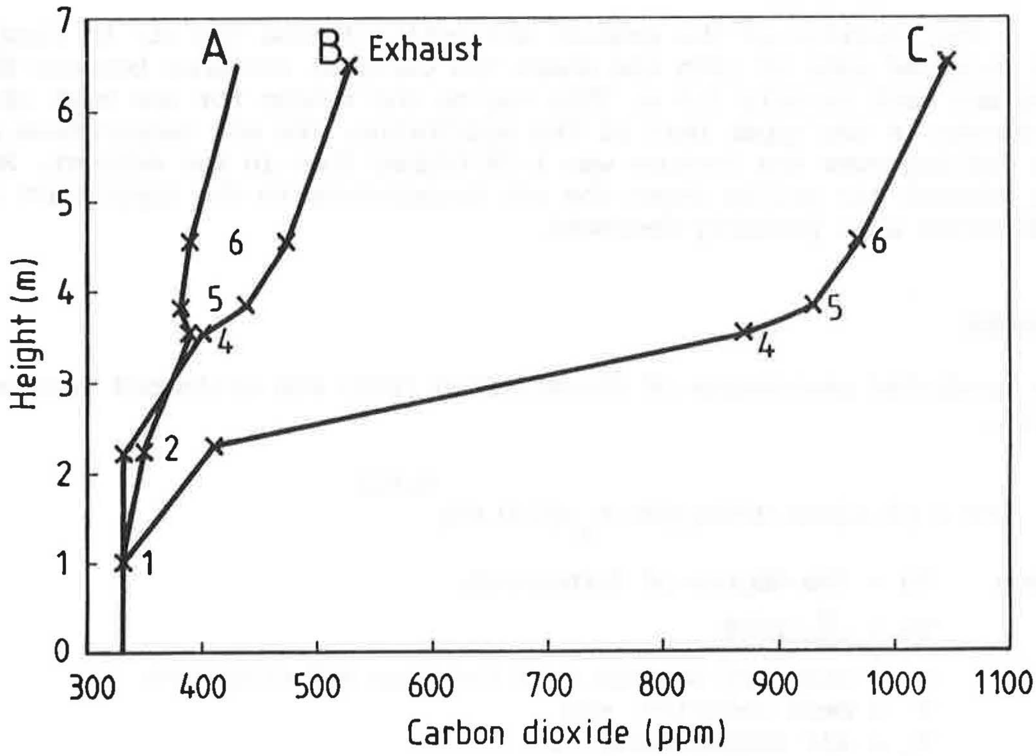

Figure 5. The vertical distribution of the CO_2-concentrations in three basic cases in the hall. A=case 6 and 8, B=cases 2, 3 and 7 and C=cases 1, 4 and 5. The height was measured from the bottom level.

Curve A shows a typical distribution in cases 6 and 8, where the real occupancy was 20 or 60 students during the lecture. Curve B represents cases 2, 3 and 7, where the simulated occupancy was 25%. In curve C (cases 1, 4 and 5) the simulated occupancy was 100%.

With a full load (curve C) there was a sharp increase in the measured concentrations of carbon dioxide between the 5th and 9th rows (between 2.3 and 3.5 m above the bottom of the hall). Up to the 5th row the concentration was close to the background concentration. In the 9th row the concentration was already the background concentration plus half of the difference between the exhaust and the supply. In the cases where the supply air rate was same, the intensity of the heat load did not seem to have any effect on the vertical concentration distributions. The highest values in the occupied zone were in the 12th row (point nr 6). The maximum concentration in the occupied zone was in any case lower than in the exhaust duct.

Ventilation effectiveness

Table 2 shows the local values of the pollutant removal effectiveness calculated by equation (4) from the CO_2-concentrations. The mean air exchange efficiency measured simultaneously in the exhaust is shown on the right side of the table. The locations of the points are the same as in figure 2. The vertical distances from the bottom of the hall are in parentheses.

Table 2. The local pollutant removal effectiveness in the hall and the mean air exchange efficiency in the exhaust.

Case	Pollutant removal effectiveness Location of the measuring points 1(1.0)	2(2.3)	4(3.5)	5(3.8)	6(4.5) *	Mean air exchange efficiency (%)
1	>100	13	2.0	1.3	1.4	45
2	90	90	3.0	1.8	2.3	47
3	>100	110	3.7	2.2	1.6	54
4	>100	9.5	1.4	1.3	1.2	31
5	52	4.3	1.5	1.2	1.1	66
6	>100	3.0	1.2	1.2	3.0	-- **Students
7	34	34	4.3	1.3	1.3	60
8	100	5.0	1.7	2.1	1.7	-- **Students

(*) In the brackets the height from the bottom level in meters
(**) The measurements were made during the lecture, no data available

The mean air exchange efficiency. In the basic cases 1 and 2 the air exchange efficiency was 47% with 25% occupancy and 45% with full occupancy. There was some short-circuiting in the hall. A reduction in the supply air rate from 2700 to 1600 l/s increased the mean air exchange efficiency to 60% with 25% occupancy (case 7) and to 66% with full occupancy (case 5). In these conditions the air flow pattern was displacing. When the supply air rate was decreased to 2200 l/s and the air was supplied only from the front side of the hall, the mean air exchange efficiency increased from 47% to 54% with 25% occupancy (case 3). With full occupancy (case 4) the system was strongly short-circuiting and the mean air exchange efficiency decreased to 31%. The difference between high and low heat load in values of the air exchange efficiency was not great in those cases where air was supplied through four devices.

Pollutant removal effectiveness. The results show a sharp change in the pollutant removal effectiveness between heights of 2.3 and 3.5 m above the bottom level. The pollutant removal effectiveness decreased when the supply air rate was reduced from 2700 to 1600 l/s. The same happened when the load was raised from 25 to 100%.

The values of the air exchange efficiency do not to explain the values of the pollutant removal effectiveness. Among cases where the heat load was high, case 5 had the best value of the air exchange efficiency but the local values of the pollutant removal effectiveness generally were better in cases 1 and 4. With low heat load, case 7 had the best air exchange efficiency but the local values of the pollutant removal effectiveness were poorer than in cases 2 and 3 except in point 4.

The location of the supply devices

Two supply air devices were placed in the rear wall of the hall. No such cases where the vertical height difference between the supply air devices is so high as 4.0 m have been reported earlier. The most common solutions are the supply air devices in the front wall or in the side walls. In several cases the supply air devices are mounted close to the floor under each seat.

Vertical air temperature distribution. The vertical temperature distribution in the hall was mainly controlled by the temperature difference between the exhaust and the supply air. Whether 70% or 100% of the air was supplied from the front of the hall did not seen to have any effect on the vertical temperature difference in the hall. If part of the supply air is introduced through the devices at the rear of the hall, excessively high vertical temperature differences may occur between the ankle and neck levels in the 11th and 12th rows.

Mean velocities and draught. The air flow from the devices in the rear wall easily cause a feeling of draught at the right and left hand sides of the 11th and 12th rows. On the other hand, if all the air is supplied from the front of the hall, draught problems ensue in the 1st row even if the total supply air flow is decreased from about 2600 to 2200 l/s.

Air quality. There was no significant difference in the pollutant removal effectiveness regardless of whether or not some of the air was supplied from the rear wall. With low temperature difference between the exhaust and the supply there might be some risk of short circuit between the supply and exhaust air devices because the exhaust air grille is only 1.2 m above the supply air device. In smoke tests, there was some short circuit at the rear of the hall. It is not visible in the results between the low and high heat load cases.

Conclusions

The minimum value for the supply air temperature is 19-20°C. The temperature difference between the exhaust and the supply must not exceed 6 K in this case. The design air flow did not create satisfactory thermal conditions with full occupancy. With the design air flow the mean velocities were over 0.2 m/s at several points in the hall and the air temperatures in the upper part of the hall were over 24°C. The vertical temperature difference between the neck and the ankle was over 3 K in seats situated between the 6th and 12th rows.

The mean velocities in the occupied zone were too high even in the middle of the hall. The main reasons for this were the high air exchange rate and the strong air flow over the seats from the supply to the exhaust. The full supply air capacity is very rarely needed. A typical load is about 50 students during a lecture. Because the temperature difference between the exhaust and the supply must not exceed 6 K, the air flow which is required per occupant is between 10 and 15 l/s.

The CO_2 concentrations in the occupied zone were less than they would be using conventional mixing air distribution. The use of demand controlled ventilation based on CO_2 concentration in the exhaust or at least 2-3 different supply air rates according to the occupancy of the hall would help to create draughtless conditions during low occupancy.

During the full occupancy it is possible to improve the thermal conditions in the hall in two ways. Firstly, changing the exhaust air grille to a place where the ceiling is highest (over the lecture) could decreased the air temperatures in the upper part of the auditorium. Secondly, replacing the supply air devices to devices which have larger face area and lower face velocities, could lead to lower mean velocities when full supply air flow is used. That allows increasing of supply air flow over 2600 l/s which will decrease the air temperature in the upper part of the auditorium. To avoid unneccessary draught when full air flow is not needed a supply air flow of 1000 l/s would be sufficient during lectures when the occupancy is not more than 50 students.

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Summary

The performance of displacement ventilation in a lecture hall was studied with eight air flow and load combinations. The measurements consisted of the vertical temperature distribution, velocity, CO_2 concentration, mean air exchange efficiency and pollutant removal effectiveness. The vertical difference between the 1st and 12th rows was 3.5 m. The measurements were made both with 100% and 25% occupancy of the hall. The supply air rates were between 1600 and 2700 l/s. During those cases four supply air devices were used. When the air was supplied only from the front of the hall the supply air flow was 2200 l/s. The pollutant removal effectiveness in the occupied zone varied from 1.1 to over 100. The mean air exchange efficiency was between 31 and 66%. With a full load the air temperature was 20°C or less in the lower part of the auditorium and 24-26°C in the upper part. With a partial load the thermal conditions were satisfactory if the temperature difference between the exhaust and the supply was less than 6K. The supply air temperature must be kept at 19-20°C to ensure thermal comfort.