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Case Studies of Displacement Ventilation in Public Halls

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ABSTRACT

In this paper four cases where displacement ventilation has been used in relatively large spaces are reviewed. In the first three cases measurements have been made of the temperatures, CO_2 concentrations (as an indicator of the air quality), efficiencies, and thermal comfort. In the fourth case numerical calculations have been made. In all the buildings studied the air is supplied directly to the occupied zone with low velocity either from inlets under the chairs or along the walls, and extracted through openings close to the ceiling. All the premises have a similar geometry to an amphitheater. The results show that the air quality and temperature efficiency for most of the space are better than that obtainable with a complete mixing system. The only exception is the CO_2 concentration and temperature at the back part of the premises, which are higher than the rest of the room. Each displacement ventilation system also seems to give satisfactory thermal comfort. Since the premises are relatively large, with the heat sources distributed across the floor, the results should be used with care when discussing other kinds of rooms.

INTRODUCTION

During the last few years there has been much work in the Scandinavian countries on developing efficient ventilating systems (Mathisen 1988; Skaret and Sandberg 1985; Mathisen and Skaret 1984), i.e., systems that remove pollution and excess heat with optimum use of outside air and energy. In this work the displacement ventilation system has proved to be the most efficient system for the removal of most kinds of pollution and excess heat, both in industrial and comfort ventilation. In premises like cinemas the main pollution and heat sources are the occupants. In assembly halls and auditoriums the ceiling lights and perhaps windows are additional sources. This means that these types of rooms often have a few well-defined heat and pollution sources that cause rising, polluted convection currents. One result is that the displacement ventilation system with air supplied directly to the occupied space and exhaust outlets close to the ceiling should work well in these kinds of rooms. However, the main problem with displacement ventilation in general is the thermal comfort when the heat load is high (Mathisen 1988). In this paper some case studies of the use with displacement ventilation in relatively large spaces are reviewed.

THEORETICAL CONSIDERATIONS

Ventilation of Large Spaces

In large premises, such as cinemas or assembly halls, there will usually be a good deal of space above the occupied zone. If the ventilation is based on a complete mixing system, the ventilation air is usually blown into the space through diffusers mounted close to the ceiling. To compensate for heat released from the occupants, this air has to be cooler than room air to maintain an acceptable temperature in the room. To ensure acceptable air quality, the fresh air should be evenly distributed in the occupied space. However, there will also be relatively strong rising convection currents caused by the heat released from the occupants. To establish complete mixing, the air should be supplied with high momentum. However, this might cause draft. On the other hand, if the momentum is too weak rising convection currents will flow in the opposite direction to the supplied fresh air, causing a possible unstable airflow situation. One possible airflow pattern is that the supplied fresh air will flow down along the walls and may cause a draft for the occupants sitting along the walls. At the same time, the occupants in the middle of the room will be sitting in a zone with bad air quality and high temperatures. In very large spaces the airflow pattern might be unpredictable, with cold supply air falling down with high velocity at random spots.

How Does a Displacement Ventilation System Work?

Displacement ventilation (Figure 1) is secured by supplying the ventilation air at a temperature that is always lower than the air temperature in the zone of occupation. Due to buoyancy, air will rise upward above a heat source and ambient air will be entrained into this flow, i.e., the airflow rate increases with the height. In a displacement ventilation system, where the ventilation air is supplied in the lower part of the room, this air is entrained into the convection flow. At a certain height above the heat source, the flow rate in the convection flow equals the supplied ventila-

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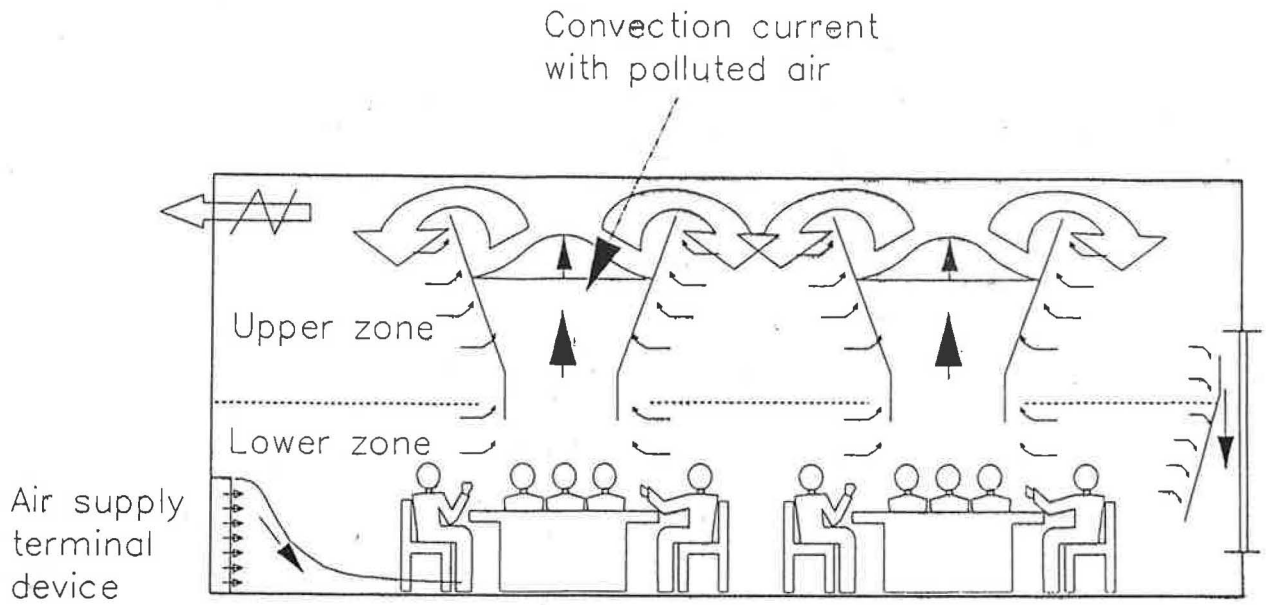

Figure 1 *Displacement ventilation in principle*

tion airflow rate. To feed the convection flow above that height, the air in the upper part of the room must recirculate. Pollution is often released from these heat sources, which include people. The pollution will therefore be transported toward the ceiling, where the exhaust opening is placed. In this way the air in the space will be stratified with a lower zone with fresh air and an upper zone with contaminated air.

The height of the lower zone depends on how much air is supplied. More air means that the rising warm air can be fed with fresh air to a higher level before it must recirculate and feed itself.

If it is necessary, heating of the premises is usually provided by the use of panel heaters under the windows.

A two-zone mixing model is often used to describe the concept of ventilation and define its effectiveness. This simple stratified model has been experimentally verified by earlier work (Mathisen and Skaret 1984). The model and the measurements generally predict high ventilation effectiveness for ventilation systems using the displacement principle.

Another consequence of the displacement system is that the supply air temperature required for cooling is higher than that for complete mixing because the supply air is supplied directly to the occupied part. The improved effectiveness compared to complete mixing also means that the fresh air supply to the space can be decreased without reducing the air quality. The concept of displacement also means that the airflow near the floor is driven by the buoyancy forces acting on the supply air.

Thermal Comfort

According to NKB (1981) it seems to be a reasonable assumption that the temperature difference between neck and ankles of a sitting person and the velocity close to the ankles should not exceed 3 K and 0.15 m/s, respectively. Also the turbulence intensity plays an important role. However, in this work, that parameter has not been measured.

The thermal comfort has been measured at a single position. However, this position has been chosen with care, to try to give a representative picture of the thermal comfort in the premises.

Ventilation Efficiency

It has been known for a long time that the efficiency of a ventilation system can vary, i.e., ventilating air has a positive effect on the occupied area and the efficiency varies among buildings and ventilation systems. When concepts such as age of air are introduced, it is possible to record ventilation efficiency quantitatively. This means that ventilation efficiency can be calculated based on measurements in buildings or predicted for new buildings. Ventilation efficiency can also be measured when a diagnosis is to be taken for plants where there are problems with air quality.

Some of the definitions that are possible to use are briefly defined in this section. The background for the definitions is given in Skaret and Sandberg (1985). The measurement techniques that are utilized to determine the various efficiencies are reviewed in detail in Mathisen and Skaret (1988).

Air Exchange Efficiency. The air exchange efficiency is the exchange time for the airflow through the premises in relationship to the exchange time for the entire airflow rate in the room.

$$\epsilon_a = \frac{\tau_n}{2\langle\bar{\tau}\rangle} \tag{1}$$

where

$\langle\bar{\tau}\rangle$ = mean age for all air in the room, s

$\tau_n = \frac{V}{\dot{V}}$ = nominal exchange time, s

V = volume of the room, m^3

$\dot{V}$ = airflow rate of ventilation air, $\frac{m^3}{s}$

The exchange time for the airflow through the premises is solely dependent on the volume of the airflow rate and the room volume, while the exchange time for the entire airflow rate in the room is also dependent on the airflow pattern in the room.

The efficiency rating reveals if there is a tendency toward displacement flow or stagnation. A possible contamination does not influence this efficiency unless the source of contamination contributes with kinetic energy and thereby changes the airflow pattern in the room.

Complete mixing will give an efficiency rating of 0.5. Piston flow (ideal displacement) will show an efficiency rating of 1.0. In other words, with complete mixing all the air is exchanged, on average, at half the speed as in a room with piston flow. Another way to state this is that the air in a room with complete mixing has twice the mean age as in a room with ideal displacement ventilation.

If a two-zone model is used for spaces where the supply air enters the lower zone and the outlet is placed in the upper zone, it can be stated that the best obtainable efficiency is 0.67. This presupposes that the two zones are equal in size and that the recirculation from the upper zone to the lower zone is close to zero. This means that for practical purposes one can expect efficiencies between 0.5 and 0.67 in rooms with displacement ventilation (Mathisen and Skret 1988).

In the cases reviewed in this paper the efficiency is measured by supplying a constant flow of tracer gas in the supply duct and measuring the response in the exhaust duct. It is then possible to calculate the efficiency from the recorded response.

Local Air Exchange Indicator. The mean age of the total airflow rate in relation to local mean age is obtained by:

$$\epsilon_i = \frac{\tau_n}{\tau_p} \tag{2}$$

where

τ_p = local mean age, s, i.e., age at any given point

In a room with complete mixing this efficiency rate will be 1.0 for any given point.

The procedure used for measuring the efficiency is the same as for the air exchange efficiency except that the response is measured at the point of interest.

Local Ventilation Index. The local ventilation index is measured in an actual contamination situation. This means that contaminations from natural sources can be used. Alternatively, it can be supplied tracer gas to simulate the natural contamination. The index shows the relative concentrations of pollution with ideal mixing (concentrations measured in the exhaust) in relation to actual measured concentrations, where both are compensated for any concentration of the same contaminants in the supply air.

$$\epsilon_v = \frac{C_e(\infty) - C_s(\infty)}{C_p(\infty) - C_s(\infty)} \tag{3}$$

where

$C_e(\infty)$ = concentration of pollution in exhaust, ppm

$C_s(\infty)$ = concentration of pollution in supply air, ppm

$C_p(\infty)$ = concentration of pollution at key measurement points, ppm

$C_p(\infty)$ = may also be mean value in an area of the occupied zone, ppm

The same definition is used for the temperature efficiency, replacing concentrations with temperature.

CASE STUDIES

Measurements were done during autumn 1988 to study the thermal comfort and air quality in some typical relatively large ventilated spaces with displacement ventilation systems. In addition a case is discussed where numerical calculation has been used to find the airflow pattern.

The cases reviewed in this paper are:

1. An auditorium with air inlets under the seats.
2. A theater with air inlets along the walls.
3. A cinema with air inlets under the seats.
4. A large assembly hall with air inlets under the seats where the airflow pattern has been numerically calculated.

Cases 1 through 3 are from buildings in the city of Trondheim, Norway. Case 4 is from a building under construction in Oslo.

The temperature was measured by means of thermocouples and a data collector. The tracer gas concentrations were measured by means of a gas analyzer based on infrared absorption. Its measurement column could be extended to an absorption length of ca. 24 m. N_2O was used as a tracer gas with the concentration ranging from 0 to 10 ppm. The CO_2 concentration was measured with a similar instrument. Additional temperature and velocity measurements were done by means of an indoor climate analyzer having probes for velocity, temperature, humidity, and radiation. For surface temperature measurements an infrared radiation meter was used. Air velocities were measured with a hot film anemometer. A smoke generator producing oil aerosol was used for the smoke tests.

Case 1: Auditorium with Air Inlets under the Seats

Description and Design Values

Structure and Dimensions. The premises are shown in Figure 2. The auditorium has 320 seats in 13 rows and is used for lectures at the Norwegian Institute of Technology. Each row has the shape of a bow, together forming part of a bowl. The seats are placed so that the lowest seats are at the same level as the lecturer while the rear seats are 3.56 m above that level. The floor area is rectangular, 15.8 by 21.5 m. The maximum ceiling height is 5.7 m. The auditorium has a wooden floor and the steps which form the amphitheater are constructed from a steel framework. Furthermore, the auditorium has a suspended ceiling and painted walls made of lightweight concrete blocks. The front wall has a large glazed area toward a glazed atrium. The ceiling light gives 3.8 kW in addition to the heat released from the students.

Air Supply Terminal Devices and Exhaust Air Outlets. The air supply terminal devices are mounted close to the floor under each seat, facing the front of the room. Each device is 0.45 m wide and 0.15 m high. Altogether there are 300 units, which provide the outflow-

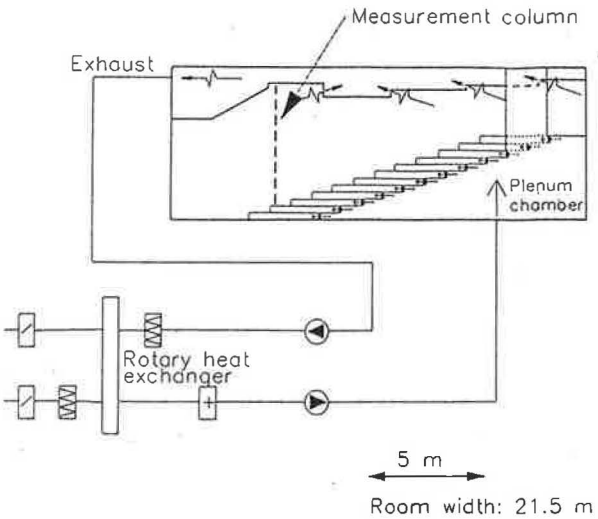

Figure 2 *Case 1. Auditorium with air supply under seats*

ing air a mean velocity of 0.176 m/s. At the outlet to the room each device is a coarse fibrous filter.

There is no elevation or other kind of obstacle for the air at the edge of the steps which can prevent the air from falling down from one step to another.

The exhaust outlet is situated over the suspended ceiling. The air is drawn toward the outlet through slots across the ceiling.

Ventilation Plant. The room air temperature is controlled by the airflow rate by changing the speed of the two-speed fans. The sensor for the temperature controller is placed in the front of the room, 2 m above the floor.

— The airflow rate is designed to be 12,800 m³/h at full speed.

— The plant is equipped with a heat recovery wheel.

— There are no cooling or humidifier units in the system.

— The temperature of the supply air is kept constant at 19°C.

Measured Results. The position of the column for the measurement of temperatures and CO_2 is shown in Figure 2. The column was placed 3 m from the middle of the auditorium, close to the second row.

Airflow Rate. The supply airflow rate was measured to be 10,700 m³/h, giving an airflow rate of 33.4 m³/h per person. This was measured by supplying a known flow rate of tracer gas in the supply air duct and measuring the concentration downstream. Tracer gas was also injected into the exhaust duct to check if there was any leakage between exhaust and supply air. This leakage was found to be 10% of the total airflow rate.

Air Exchange Efficiency. The air exchange mean efficiency was found to be 0.56. This number is influenced by the leakage in the heat recovery wheel noted in the previous section, so that the efficiency for the room itself should be somewhat better than indicated.

Local Air Exchange Indicator. The local air exchange efficiency was measured at the rearmost seat at the right-hand side of the auditorium. It showed a value of 1.13.

Temperature and CO_2 Concentration. The temperatures and CO_2 concentrations given in Tables 1 and 2 were measured at the column and some single points in the room. The values were measured at the end of the lectures and with a fully occupied auditorium.

Additional Temperature and Velocity Measurements. Measurements of temperature and velocity were also made at the seventh row 0.1 m and 1.1 m above the floor (see Table 3).

TABLE 1
Measured Temperature, Temperature Efficiency, CO_2 Concentration, and Ventilation Index at the Measurement Column in the Auditorium. October 5 and 6

Height above the floor	Temperature °C October 5	Temperature efficiency		CO_2 concentration, ppm October 6		Ventilation index	
	11:50 a.m.	Relative to exhaust duct	Relative to upper point on column	9:55 a.m.	11:55 a.m.		
Exhaust duct	22.4	1.00		890	845	1.00	1.00
5.5 m	23.9	0.53	1.00				
4.5 m	24.0	0.52	0.97		845		1.0
3.5 m	23.6	0.59	1.1		880		0.91
2.5 m	23.3	0.65	1.23				
1.7 m	23.0	0.74	1.39	740		1.47	
1.1 m	22.7	0.85	1.60	700		1.68	
0.6 m	22.2	1.13	2.13				
0.1 m	20.2	−3.4	−6.4		420		29.33
Supply air	20.7			420	405		
1.1 m above the floor							
Seventh row:							
—Right hand side	23.8	0.55	1.03	970		0.85	
—Left hand side	23.8	0.55	1.03	920	825	0.94	1.05
Rearmost row:							
—Right hand side				1260	1060	0.56	0.67
—Left hand side	24.8	0.41	0.78	1150	940	0.64	0.82

TABLE 2
Measured Temperature, Temperature Efficiency, CO_2 Concentration, and Ventilation Index at the Measurement Column in the Auditorium. October 10

Height above the floor	Temperature, °C 11:50 a.m.	Temperature efficiency		CO_2 concentration, ppm, 9:50 a.m.	Ventilation index
Exhaust duct	21.1	1.0		890	
5.5 m	22.9	0.36	1.0	930	0.92
4.5 m	23.1	0.33	0.93	900	0.98
3.5 m	22.7	0.38	1.08		
2.5 m	22.4	0.43	1.22	895	0.99
1.7 m	22.0	0.53	1.47	920	0.93
1.1 m	21.6	0.67	1.87	850	1.1
0.6 m	21.2	0.91	2.55		
0.1 m	18.9	−0.83	−2.33		
Supply air	20.1			445	

The surface temperatures in the auditorium were measured to be: ceiling, 25°C; walls, 23°C; and floor (not the floor at the lowest level), 22° - 23°C.

For most of the supply air terminals the measured supply air velocity ranged from 0.17 to 0.23 m/s, but some units in the middle of the auditorium had velocities of 0.35 m/s.

Smoke Test. The airflow pattern close to the inlet was studied in detail by supplying smoke. It was observed that some of the air flowed from one step to the next step down.

Discussion of Results. The air exchange efficiency shows that there is displacement ventilation in the auditorium. From the local air exchange indicator the fresh air seems to be well distributed at the back of the room, too. Nevertheless, Tables 1 and 2 indicate that the air quality is not as good at the back of the room as in the front. This might be due to the airflow pattern close to the floor, where the air seems to be streaming downstairs.

It is also evident from Tables 1 and 2 that the temperature in the exhaust duct is lower than the temperature at the upper point on the column. This might be due to the fact that there is concrete above the suspended ceiling. The air is cooled down by the concrete, which has a large thermal capacity and is cooled during periods when the auditorium is not in use. Therefore, it is more correct to use the temperature at the upper point on the column to evaluate the efficiency.

The measurements in Table 3 indicate poor thermal comfort since the temperature difference between the ankles and the head of a sitting person is as much as 3 K. Additionally the velocity at the floor seems to be too high. Tables 1 and 2 reveal temperature differences of 2.5° - 2.7°C between neck and ankle.

Case 2: Theater Hall with Air Inlets along the Walls

This is an old building where a displacement ventilation system was installed during 1987. The building used to have no mechanical ventilation, and there has been little emphasis on making the building airtight.

TABLE 3
Local Temperatures and Velocities

Height	Velocity, m/s	Temperature, °C
1.1 m	0.07-0.11	24.0
0.1 m	0.14-0.18	21.0

Description and Design Values

Structure and Dimensions. The theater hall is shown in Figure 3. The balcony and gallery form parts of a bow and stretch along the sidewalls. The hall has 237 seats in 12 rows in the lowest area (the stalls). The balcony has 68 seats and the gallery has 50 seats, making 355 seats in all. The floor area of the premises is rectangular, 9.44 by 11.9 m. The maximum ceiling height is 7.6 m. The building is constructed of brick. Inside, one of the sidewalls is brick, while the other surrounding surfaces are constructed of wood.

Air Supply Terminal Devices and Exhaust Air Outlets. The air supply terminal devices are mounted close to the floor along parts of the sidewalls, as shown in Figure 3. These devices are about 0.85 m high. There is also a supply of air along the back wall in the stalls. These devices are 0.4 m high and about 8 m long. On the balcony and the gallery there is supply along parts of the back wall. The total front area of the devices is 27.7 m². This gives a mean velocity of the outflowing air of 0.14 m/s. At the front, the devices have a perforated sheet of steel with a coarse filter inside to even out the flow. The exhaust outlet is situated in the middle of the ceiling.

Ventilation Plant. The theater premises are heated by convectors placed on top of the air supply devices. The

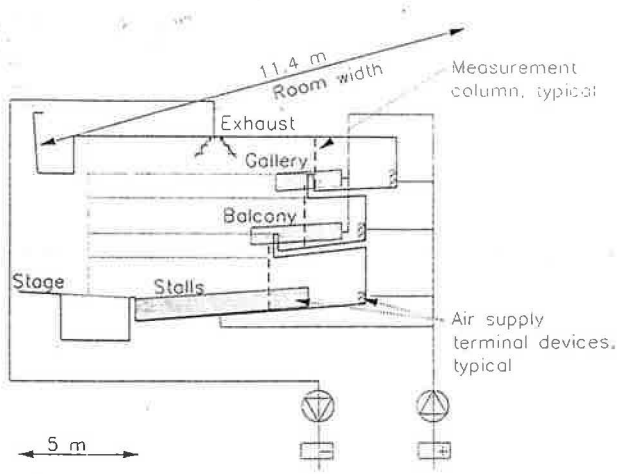

Figure 3 *Case 2. Theatre with air supply along walls*

TABLE 4
Measured Temperature, Temperature Efficiency, CO_2 Concentration, and Ventilation Index at the Measurement Column in the Theater

Height above the floor	Temperature, °C October 20 10:00 p.m.	Temperature efficiency	CO_2 concentration, ppm October 15 8:40 p.m.	Ventilation index 8:40 p.m.	Ventilation index Mean value for performance
Exhaust duct	22.4			1.00	1.00
Gallery					
1.7 m	23.1	0.86			
1.5 m			860	0.99	1.04
1.1 m	22.8	0.92			
0.1 m	20.2	2.0			
In the back, 1.7 m above floor	23.9	0.74			
Balcony					
2.5 m	23.2	0.85			
1.7 m	23.1	0.86	755	1.16	1.28
1.1 m	22.5	0.98	700	1.79	1.75
0.1 m	20.1	2.1			
1.1 m above the floor					
—Left side close to pillar	22.6	0.96			
—In the back	23.4	0.81			
Stalls					
2.5 m	22.7	0.94	940	0.88	0.91
1.7 m	22.2	1.05	830	1.01	1.16
1.1 m	22.4	1.0	790	1.16	1.25
0.1 m	20.0	2.2			
1.1 m above the floor					
—Left side close to pillar	22.8	0.92			
—Right hand side close to pillar	18.6	7.3			
Supply air	18.0		390		

room air temperature is controlled by the airflow rate due to changing the speed of the two-speed fans. The sensor for the temperature controller is placed in the lower part of the room 2 m above the floor at the back.

- The airflow rate is designed to be 13,940 m³/h at full speed.
- The plant is equipped with a glycol heat recovery system.
- The temperature of the supply air is kept constant at 19°C.

Measured Results. The position of the column for the measurement of temperatures and CO_2 is shown in Figure 3. The columns were placed along pillars 3.6 m from the side wall.

Airflow Rate. The supply airflow rate was measured to be 12,000 m³/h, using a Prandtl tube. This gives an airflow rate of 33.8 m³/h per person.

Air Exchange Efficiency. The mean air exchange efficiency was found to be 0.54. However, the infiltration of air from other parts of the building and the exterior is quite large. The contribution of this air is not included in the number above. If it had been, the number would have been a little larger since the infiltration air probably enters into the lower part of the room.

Local Air Exchange Indicator. The local air exchange efficiency was measured at the balcony 1.9 m above the floor close to the measurement column. It showed a value of 0.71.

Temperature and CO_2 Concentration. The temperatures and CO_2 concentrations in Table 4 were measured at the column. The values were measured at the end of performances with a fully occupied theater. In addition mean values are shown for the ventilation index for the last hour of the performance.

Additional Temperature and Velocity Measurements. Measurements of temperature and velocity were also done at the different parts of the theater 0.1 m and 1.1 m above the floor (see Table 5).

The surface temperatures in the theatre were measured to be: ceiling, 24°C; walls, 23°C; and floor at the balcony, 22°C.

Discussion of Results. The air exchange efficiency shows that there is displacement ventilation in the theater. The ventilation indexes shown in Table 4 are quite satisfactory, indicating that the air quality should be good. However, the temperature efficiency on the balcony and gallery is somewhat poorer but still within a reasonable discrepancy of what could be expected taking the height above the stalls into consideration. This must mean that the air supplied in these areas manages to create locally climatized zones.

The measurements related to thermal comfort given in Table 5 are rather incomplete but it seems that the thermal comfort is acceptable. Also, the temperature differences, which can be seen from Table 4, between points 0.1 and 1.1 m above the floor are within acceptable limits. The surface temperatures do not indicate radiation asymmetry.

TABLE 5
Local Temperatures and Velocities

	In the back right corner of the stalls, close to non-airtight door		At the balcony on the left hand side, first row halfway back in the theater		At the gallery on the right hand side (close to the air supply terminal device)	
Height	Velocity	Temperature	Velocity	Temperature	Velocity	Temperature
1.1 m	< 0.1 m/s	19°C	< 0.1 m/s	22.0 °C	0.2-0.28 m/s	21°C
0.1 m			0.15-0.18 m/s	19.7°C		

TABLE 6
Measured Temperature, Temperature Efficiency, CO_2 Concentration, and Ventilation Index at the Measurement Column in the Cinema. November 4, 1988.

Height above the floor	Temperature, °C 10:20 p.m.	Temperature efficiency	CO_2 concentration, ppm, 10:20 p.m.	Ventilation index, 10:20 p.m.	Mean
Exhaust air	22.8	1.00	1250	1.00	1.00
4.5 m	23.1	0.91	1220	1.05	1.04
3.5 m	22.8	1.0	1205	1.08	1.07
2.5 m	22.7	1.03	1230	1.04	1.06
1.7 m	22.2	1.24	1215	1.06	1.10
1.1 m	22.0	1.35	1235	1.03	1.09
0.6 m	21.2	2.07	830	3.63	3.84
0.1 m	20.6	3.44			
Supply air	19.7		670		
1.1 m above the floor					
Rearmost row in the middle	24.7	0.62	1287	0.94	0.99
In the middle of the audience	23.5	0.81			

Case 3: Cinema with Air Inlets under the Seats

Description and Design Values

Structure and Dimensions. The cinema is shown in Figure 4. It has 182 seats in 13 rows. The rows are parallel with the end walls. The floor area of the cinema is rectangular, 15.7 by 9.4 m. The maximum ceiling height is 5.4 m. At the back the floor is 3.15 m above the floor in the front. The premises are constructed of concrete except for the floor, where the steps that form the floor are constructed of a steel framework and a wooden carpeted floor.

Air Supply Terminal Devices and Exhaust Air Outlets. The air supply terminal devices are mounted close to the floor under each seat, blowing horizontally toward the front of the cinema. The inlet area of each device is 0.5 m wide and 0.17 m high. Altogether there are 182 units, which gives a mean velocity of 0.13 m/s for the outflowing air. The grilles at the front have a perforation degree of 33%. At the edge of the steps, shown in Figure 4, there is a board forming an elevation of 0.04 m. The exhaust air outlets are evenly distributed across the ceiling.

Ventilation Plant. The room air temperature is controlled by the supply air temperature. The sensor for the temperature controller is placed in the exhaust duct.

- The airflow rate is designed to be 7280 m³/h at full speed.
- The plant is equipped with a glycol runaround heat recovery system.
- There are no cooling or humidifier units in the system.
- The minimum temperature of the supply air is 16°C.

Measured Results. The position of the column for the measurement of temperatures and CO_2 is shown in Figure 4. The column was placed 1 m from the sidewall.

Airflow Rate. The supply airflow rate was measured to be 7000 m³/h, using tracer gas. This gives an airflow rate of 38.5 m³/h/person. Tracer gas was also supplied in the exhaust duct to check if there was any leakage in the recirculation damper between the exhaust and the supply air. This leakage was found to be 30% of the total airflow rate.

Air Exchange Efficiency. Due to the leakage in the recirculation damper it was impossible to measure the efficiency in a proper, straightforward way.

Local Air Exchange Indicator. Measuring the indicator gave the same problems as measuring the efficiency.

Temperature and CO_2 Concentration. The temperatures and CO_2 concentrations in Table 6 were measured at the column. The values were measured toward the end of a movie with a fully occupied room.

Additional Temperature and Velocity Measurements. Measurements of temperature and velocity, given in Table 7, were also made at the seventh row close to the middle of the row, 0.1 m and 1.1 m above the floor. During these measurements only 111 of a maximum of 182 people were present.

Smoke Test. Smoke supplied to the entering air revealed that little of the air flowed from one step to the other. At the edge of the steps the air was deflected and

TABLE 7
Local Temperatures and Velocities

Height	Velocity	Temperature
1.1 m	0.09 m/s	21.5°C
0.1 m	0.08-0.09 m/s	20.0°C

The surface temperatures in the premises were measured to be: ceiling, 22.5°C; walls, 23°C; floor (not the floor at the lowest level), 20.5°C.

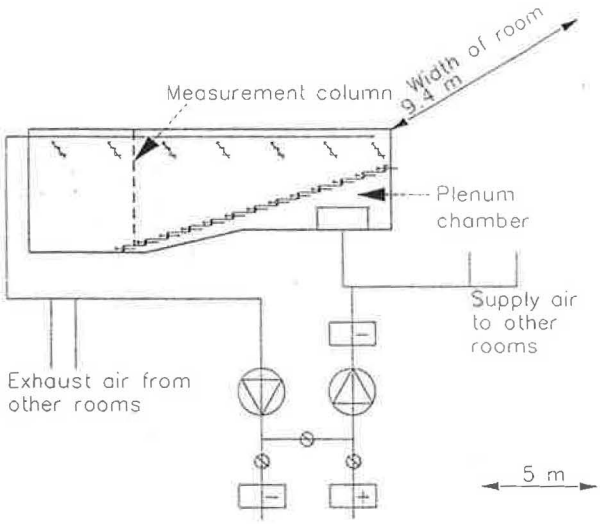

Figure 4 *Case 3. Cinema with air supply under seats*

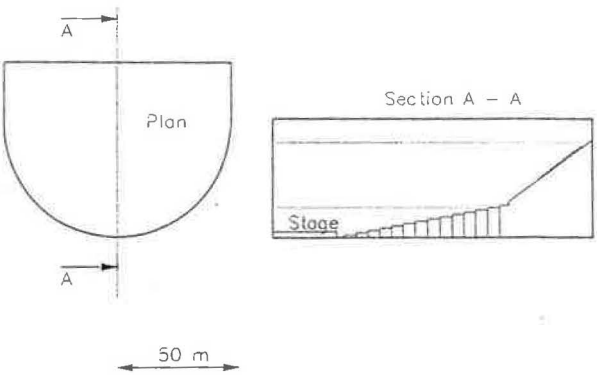

Figure 5 *Case 4. Assembly hall*

forced into an eddy by the elevation. Additionally the legs of the chairs also helped prevent the air from flowing down from one step to another.

Discussion of Results. The ventilation index and the temperature efficiencies in Table 6 indicate good efficiency. However, also in this case the temperature at the back of the cinema is much higher than in the front, where the column is placed. According to Table 7 the thermal comfort seems to be quite good; this might be due to the sparse audience. However, the measurements at the column with a fully occupied room indicate modest temperature differences between neck and ankle. The apparently good results might be partly explained by the boards forming the elevation at the edge of the steps, based upon what was seen during the smoke tests. The poor air quality indicated by Table 6 is due to the large recirculation of air.

Case 4: Numerical Calculations of the Airflow Pattern in a Large Space

In the results just reviewed, measurements have only been possible at a few points in the space. In the results that will be shown in this section the airflow pattern has been calculated by means of a computer code based on the finite difference solution of the general flow equations (Sorlie and Gangso 1988). The intention of this paper is not to discuss numerical calculations, but the calculations reviewed here provide the opportunity to obtain a better understanding of the airflow pattern in a large ventilated space.

Even with the coarse mesh size—2 by 2 by 2 m (total of 12,000 calculation points)—the calculations give a lot of information about how the mean air velocity, temperature, and humidity vary from point to point in the space. However, there are limitations to the degree of accuracy with which the parameters can be studied close to the walls. For turbulence calculations the code uses a k-ϵ model. The equations are solved by the SIMPLE algorithm. The computer code also takes buoyancy into consideration. The surrounding surfaces were kept at a constant temperature. Many different cases were calculated, but in this paper only one typical situation with a fully occupied space and concert orchestra will be reviewed.

Description and Design Values

Structure and Dimensions. The premises are shown in Figure 5. It has 10,000 seats and is used for several purposes such as ice hockey, rock concerts, and horse jumping. The seats in the lower part of the premises might be arranged in different ways depending on the use. The building is constructed from concrete. The ceiling has an open framework of steel. In the front of the premises there is a light grid with spot lamps which have a maximum heat release of 170 kW. The total floor area is 6500 m² and the volume is 90,000 m³. The maximum ceiling height is 26 m. In the calculations it has been assumed that there is a heat release of 100 W per person.

Air Supply Terminal Devices and Exhaust Air Outlets. The air is supplied in various ways to the different parts of the premises. At the lowest flexible rows of seats, the air is supplied through slots under each seat. In successively higher parts of the occupied space, the air is let in through large openings leading out to the lobby area. In the upper part, the air enters through openings under each seat. The exhaust openings are evenly distributed in the ceiling.

Ventilation Plant. To tackle different loads the ventilation plant is divided into eight separate units which can be used independently. The total airflow rate is 360,000 m³/h. The supply air temperature is kept constant at 20°C.

Calculated Results. The reviewed case is a concert situation with an audience of 10,000. The heat from the grid with spot lamps is 170 kW.

The airflow pattern is illustrated in Figure 6 by the components of the velocity vectors in a section through the middle of the premises. The length of the arrows is proportional

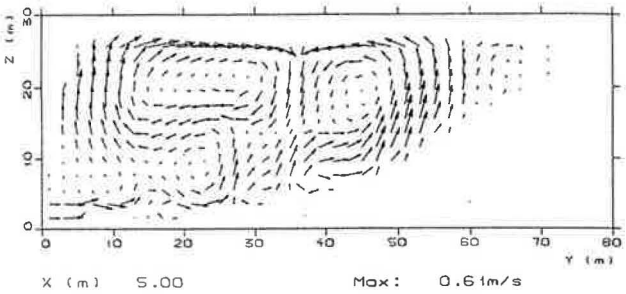

Figure 6 *Airflow pattern in assembly hall. The length of the vectors are proportional to the velocity. Section close to the middle of the room*

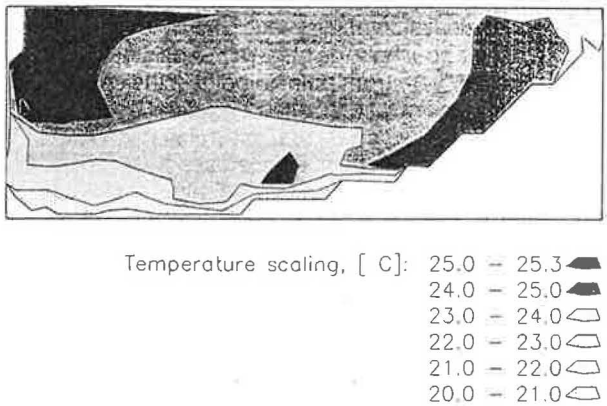

Figure 7 *Temperature distribution in assembly hall. Same section as in Figure 6*

to the velocity. The temperature distribution is shown in Figure 7.

Discussion of the Results. The results in Figure 7 indicate that the temperature at the upper back seats is somewhat higher than the temperatures in the front. The airflow pattern in Figure 6 shows that it is the convection currents that drive the air. There is a strong current above the light grid. The heat from the audience also causes an airflow along the seat rows. This causes the temperature to increase with the height. It is interesting to observe the complex airflow pattern and temperature distribution. There is no stratification of the air due to the strong mixing caused by the convective currents.

It is not possible to see the local variations of temperature and velocity close to the audience because of the coarse mesh, but both the air velocity and the temperature seem to be within the limits of thermal comfort.

DISCUSSION

The measurements and numerical calculations shown in Section 3 reveal that both the temperature and the concentration of CO_2 increase with the height. The strongest gradient is within the occupied space. Above the volume used by the audience the gradient evens out more, though there are still some irregularities. The cause of these irregularities might be explained from what can be seen from the vector and temperature plot in case four (see Figures 6 and 7). If measurements were done in that room, the shape of the temperature profile would depend on where the measurement column was placed.

Since the level of the floor in case 2, the theater, is not much different in front and back the temperature differences are smaller there than in the other premises. Also, at the balcony and the gallery the temperature is acceptable, which means that the supply air in these areas manages to create locally climatized zones.

The thermal comfort, i.e., the temperature difference and velocity close to the ankles, varies between case 1, the auditorium, and case 3, the cinema, even though they are quite similar with regard to the design of the spatial geometry and the air supply terminal devices. This might be due to a coincidence, but there are several observations in each case which confirm the differences. However, some details are different between the two premises. First, there is an elevation at the edge of the steps in the cinema. Second, the auditorium has no perforated grilles at the supply air terminal. Third, the air distribution in the auditorium is poor and, finally, the airflow rate per person is greater in the cinema. The smoke tests indicated that the supply air in the auditorium was more likely to fall down from one step to the next than in the cinema. It would probably be an advantage to close the openings between the rows and attempt to promote more local mixing of the air; for instance, by using coarse perforated grilles at the front of the inlets.

The airflow rate per person has two consequences on the thermal comfort:

1. The rising airflow along and above a convection source (for instance, a person) increases with the height. In a displacement system the supply air feeds this flow up to the level where the convection flow equals the supply airflow. Above that level the air is recirculated. The height of the lower zone is therefore a function of the airflow rate.
2. The temperature difference between the two zones will be larger when the airflow rate increases.

It seems from the measurements that the necessary airflow rate is a minimum of 40 m^3/h/person to ensure satisfactory conditions. This presupposes that there are no other heat sources than people. However, heat released in the upper zone will not affect the temperature in the lower zone.

In a complete mixing system the exhaust temperature is equal to the room air temperature. The measurements shown in this paper reveal that the exhaust temperature or the temperature at the upper point on the column is higher than the temperature in the occupied part of the room. It can be seen that an air supply temperature of 19.5°-20.5°C will produce a suitable temperature for the occupied part of the room. This means that the supply air temperature has a higher temperature in displacement ventilation systems than in complete mixing systems. Therefore, natural cooling is sufficient for a larger part of the year. On the other hand, displacement ventilation might require larger airflow rates than complete mixing.

In cases with heat load in addition to people, the necessary airflow rate can become too large. A possibility is to combine displacement ventilation with cooled panels

in the ceiling. This has been done with satisfactory results in some Norwegian office buildings.

In a complete mixing system, the space outside the occupied zone evens out temperature fluctuations in the supplied air. In a displacement system this is impossible since the distance from the inlet to occupants is small, which means that the temperature control must be more accurate. It is also important that the supply air temperature is kept at the design value.

The following items point out questions that must be resolved and comparisons that should be made:

- Can local mixing even out strong temperature gradients?
- Long-term measurement of efficiency under different conditions
- Details in airflow pattern with regard to thermal comfort
- Comparative measurements between displacement ventilation and complete mixing

CONCLUSIONS

The main conclusion from the four large spaces studied is that if the displacement system is carefully designed, both the air quality and the thermal comfort will be good. Since the temperature in the occupied part of the room is lower than the exhaust temperature, the cooling ability of the supply air is well used, which means that excess heat is removed in an energy-efficient way.

1. Thermal comfort. Thermal comfort is perhaps one of the must difficult problems to tackle with a simple displacement system, where all the air is supplied directly to the occupied part of the room. However, in the case of a cinema with air supplied under each chair, the thermal comfort is satisfactory. In the cinema premises the air is carefully distributed and there is a board forming an elevation at the edge of the steps that form the amphitheater, which might stop the supplied air from falling down from step to step. This is not the case in the auditorium. Furthermore, the only heat source is the occupants, which means that the heat load is lower in the cinema than in the similar auditorium.

2. Air quality. The CO_2 concentration has been measured at several points as an indicator of air quality. These measurements reveal that the air quality in the occupied part of the room seems to be satisfactory. However, in the back of the room the air quality does not to seem to be as good as in the front.

3. Temperature distribution. Temperatures that have been measured at the same points as the CO_2 concentration show that, in most cases, the temperature is lower in the occupied zone than in the space above. However, at points in the back of the rooms the temperature is higher than in the front.

4. Efficiency. The measurements show that the mean age of the air in the rooms is lower with displacement ventilation than complete mixing systems. Seen together with the two points above this means that the occupants are sitting in the supply air zone where the air quality ought to be good.

5. Optimum airflow rate per person. If the airflow rate is small, the temperature gradient will be strong and the temperature too high. A value of 40 m^3/h seems to be the minimum of what could be recommended in rooms where people are the main heat source.

6. High heat loads. In cases with heat load in addition to persons, the necessary airflow rate can become very large. A possibility is to combine displacement ventilation with cooled panels in the ceiling.

7. Recirculated air. It is not recommended to use recirculated air in displacement ventilation systems due to the air quality. A leaking damper will impair the air quality. The same could be said about rotary heat exchangers.

8. In a displacement system the supply air temperature control must be more accurate than in a complete mixing system since the distance from the inlet opening to the ankles is short.

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