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Displacement ventilation – The influence of the characteristics of the supply air terminal deviceon the airflow pattern

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DISPLACEMENT VENTILATION - THE INFLUENCE OF THE CHARACTERISTICS OF THE SUPPLY AIR TERMINAL DEVICE ON THE AIRFLOW PATTERN

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Introduction

Displacement ventilation systems have proved to be efficient means of removing contaminants and excess heat. In this system air is supplied to the room with low velocity through openings close to the floor at a temperature lower than the room air temperature. Usually the buoyancy forces will influence the flow. Here it is important to know the relation between the air flow rate/supply air temperature/design and arrangement of the air terminal device and the temperature/velocity of the flow along the floor.

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. The necessary heating of the room is usually provided by the use of panel heaters under the windows. Close to the heat sources in the room the air will rise upwards due to buoyancy. Often contaminants are released from these heat sources, for instance people. Then the contaminants will be transported towards the ceiling, where the exhaust opening is placed. 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 recirculates and feeds itself. In this way the air in the room will be stratified into a lower zone with fresh air, and a upper zone with contaminated air.

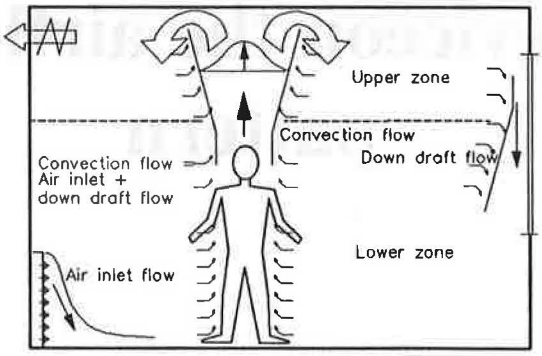

Fig. 1. The principle of displacement ventilation.

A two zone mixing model is often used to describe both the concept of ventilation and define its effectiveness. This simple stratified model has been experimentally verified by earlier work, Mathisen [1]. The model and the measurements generally predicted high ventilation effectiveness for ventilation systems using the displacement principle.

Another consistency of the displacement system is that the supply air temperature required for cooling is higher than for complete mixing i.e. free cooling could be used for longer periods of the year. However, in some types of rooms relatively large airflow rates are needed to obtain this improved efficiency. This is because the lower zone has to be "lifted" above the height of the people in the room, see Figure 1.

The concept of displacement also means that the air near the floor is driven by the buoyancy forces acting on the supply air due to the low velocity. This paper reviews the influence of the height and the width of the supply air inlet, the temperature difference between the room air and the supply air, and the influence from supply air velocity on the temperature and the velocity close to the floor. The results could be used for designing air supply terminal devices and deciding the necessary airflow rates and the supply air temperature to meet thermal comfort demands. In other work by the author some parts of this paper are discussed in more detail, [4], [5] and [6].

Test Equipment and the Tests

Test Room, Air Supply and Measurement Equipment

Tests have been carried out in a room of approximately 45 m^3 with a floor area of 16 m^2. Air was supplied through an adjustable opening on the short wall. This wall was 3.45 m wide. The opening consisted of foamed plastic.

The height and the width of the supply opening was varied, as was the supply air flow rate and the temperature. Data was collected and processed by a datalogging system.

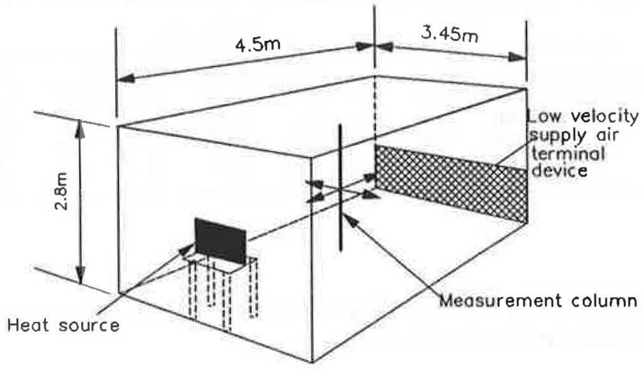

Fig.2. Test room. Temperatures were measured at 13 levels and velocities at 8 levels above the floor in 4 different positions, i.e. 52 temperature and 32 velocity measurement points in the room.

In the experiments two different widths were used in the air supply terminal device, 0.54 m and 3.15 m.

Outdoor air was used in the experiments. The air was filtered before it was heated in an electric heating element and blown into the room. The airflow rate was measured with an orifice plate and an inclined tube manometer. A damper was used to adjust the air flow rate. The exhaust airflow rate was set by the pressure difference between the laboratory and the test room to minimize the infiltration of air in the test-room. (This difference should be zero). The difference was measured with an electronic micromanometer.

The temperature of the inlet air was set by an electric heating element and a PID-controller.

The anemometers and temperature meters were located on a measurement column. The column could be moved in two directions by two electric motors. The velocities and temperatures were recorded at distances of 0.6, 1.2, 1.7 and 2.35 m from the opening along the middle of the room. Velocities and temperatures were measured at levels 0.025, 0.04, 0.65, 0.11, 0.15, 0.20, 0.24, and 0.30 m above the floor. In addition the temperature were measured at levels 1.1, 1.53, 1.95, 3.38 and 2.79 m.

The anemometers were of the 1620-12 and 1610-12 TSI-types. They were calibrated in a TSI-1125-calibration unit. As the anemometers of the 1620-12 type proved to be temperature dependent, they were calibrated at several temperature levels. The air temperature were measured with thermocouples type T. The mean value and sample standard deviation for each level and position were calculated. The air velocity and temperature was measured for a period of 2 minutes.

The Experiments

The experiments for the wide supply opening were planned as a fraction of an orthogonal first order design with three additional observations at the centre point. However, due to practical limitations in combining high air flow rates and low supply air temperatures it was not possible to realise the plan completely. Since the models were unknown a priori the chosen values of the parameters were not statistically optimal considering the final models.

Table 1 presents all the tests which have been carried out for a wide supply opening while Table 2 gives the tests for the narrow opening. The wide opening was 3.15 m wide while the narrow was 0.54 m wide.

Table 1. The tests which have been carried out with a wide supply opening. Height is the height of the supply air opening in metres. Power is the electrical power which was supplied to the room in watts. The airflow rate is measured in m^3/h. For each test room air velocities and temperatures were measured at several levels and positions.

	Test number											
	1	2	3	4	5	6	7	8	9	10	13	16
Height	1.05	1.05	0.05	0.05	0.10	0.10	0.10	0.38	0.38	0.38	1.05	0.10
Air flow rate	580	580	118	118	48	235	242	392	400	390	499	53
Power	2700	670	670	620	256	1200	320	1312	1320	1340	2674	65
Supply air temp.	10.8	18.4	17.6	13.2	17.5	12.3	18.0	17.2	17.2	16.8	10.8	17.8
Exhaust air temp.	26.3	22.3	29.6	27.7	27.3	29.4	24.5	27.6	27.4	26.8	28.4	21.9

Table 2. The tests which have been carried out with a narrow supply opening. (Details of parameters are listed under Table 1).

	Test number			
	11	12	14	15
Height	0.38	1.05	1.05	0.10
Air flow rate	65	383	85	36
Power	221	2049	440	53
Supply air temperature	16.9	11.0	13.3	17.6
Exhaust air temperature	24.4	29.4	24.1	22.7

Results

The flow from the wide opening is closely two-dimensional. That means that the flow is uniform across the flow direction. Fig. 3 shows a typical airflow pattern. As it can be seen there is a counterflow above the inflowing air. That means that we are discussing a compound flow where the velocity close to the floor should be calculated relative to the ambient flow. However, due to limitations in the test facilities (fixed position of the measuring points) the counterflow air velocity has only been measured for some of the thinnest flows. Therefore only absolute velocities have been used in the calculations.

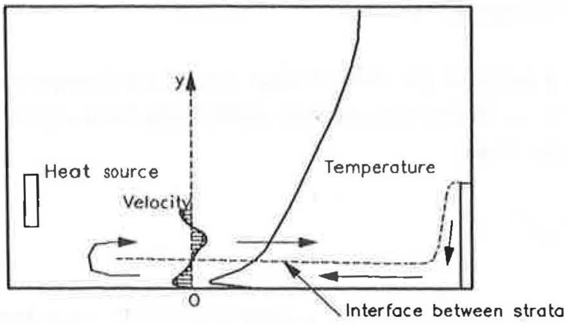

Fig. 3. Air flow pattern in the space close to the floor, wide supply opening.

Fig. 4 shows a typical air flow pattern for a narrow supply opening.

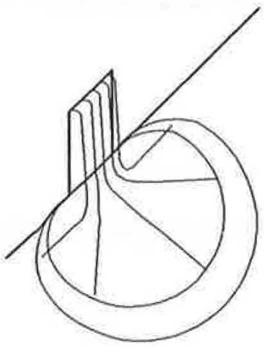

Fig. 4. Air flow pattern for a narrow air supply terminal device. The air falls down and spreads radially.

Air Velocity and Temperature along the Floor

It can be shown that the velocity of the air flowing close to the floor can be expressed as a function of the Archimedes number. The Archimedes number is the ratio between the buoyancy forces in the flow and the inertia forces. The relation can be expressed as:

$$\frac{u}{u_0} = k_1 + k_2 \cdot Ar_0^{1/2} \tag{1}$$

Where:

u – the velocity in the flow close to the floor, m/s
u_0 – the supply air velocity, m/s
$Ar_0 = \dfrac{g\beta\Delta T_0 h}{u_0^2}$
g – gravitational acceleration, m/s^2
β – volumetric expansion factor, 1/K
ΔT – temperature difference between room air and supplied air, K
h – height of the supply opening, m

It was harder to find a relation for the evening out of the temperature. However, an empirical relation was found where ΔT_m is the temperature difference between a point 1.1 m above the floor in the room and a point in the flow:

$$\Delta T_m = k_3 \cdot (\dot V_0 \cdot \Delta T_0)^{k_4} \cdot h^{k_5} \tag{2}$$

Where: $\dot V_0$ = air flow rate in m^3/s, ΔT_0 = temperature difference between room air and supplied air in degrees K.

However, as can be seen in the next section, the correlation for this model is unsatisfactory, so there is need for more research on this detail. Since there is little apparent variation in the ratio between $\Delta T_m / \Delta T_0$ a fixed ratio may also be used.

Using Eq. (1) and the heat balance equation, the excess heat removed from the zone of occupation is determined from:

$$Q = \frac{\rho C_p}{g\beta} \cdot \frac{(u \cdot h \cdot B - k_1 \cdot \dot V_0)^2}{B^2 h^3 k_2^2} \cdot \dot V_0 \tag{3}$$

Where: ρ – density in kg/m^3, C_p – specific heat capacity of air in J/kg K and B – width of supply air opening in m.

Wide Supply Opening In Figure 5 the velocity ratio u/u_0, for some heights above the floor at a distance of 1.2 m from the air supply device, is plotted against the Archimedes number of the supply air. The regression lines were found using a linear regression based on the least squares method. The regression coefficients k_1 and k_2 for all measured values are stated in Table 3. The standard errors of the coefficients and the coefficient of determination, R^2 are also given.

Using displacement ventilation vertically up, it is reasonable to assume that the highest air velocities will be just above the floor near the air supply. During the experiments we found the highest velocities 0.04 - 0.11 m above the floor for the wide opening.

A closer look at the figures in Table 3 reveals that the velocity does not change much as the air flows along the floor.

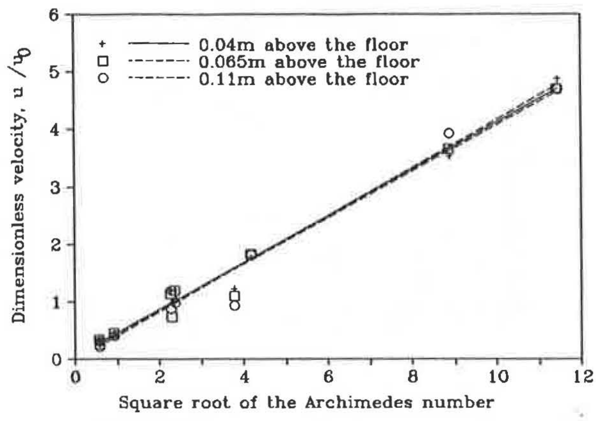

Fig. 5. Examples of plots of dimensionless velocities vs the supply air Archimedes number, measured 1.2 m from the opening. The width of the opening was 3.15 m.

Tab. 3. k_1 and k_2 for different levels and distances from the supply opening. Starred values are shown in Figure 5.

Height above the floor [m]	Distance from inlet [m]	Intercept k(1)	Coeff k(2)	Std Err of y	Std Err of Coeff	R Squared	Degrees of Freedom
0.025	0.6	0.146	0.335	0.289	0.026	0.948	9
0.025	1.2	0.054	0.404	0.169	0.015	0.986	10
0.025	1.7	0.034	0.511	0.002	0.002	0.999	1
0.025	2.35	0.29	0.342	0.145	0.013	0.988	8
0.04	0.6	0.043	0.3782	0.249	0.022	0.969	9
*0.04	1.2	0.059	0.407	0.165	0.015	0.989	8
0.04	1.7	0	0.526	0.003	0.002	0.999	1
0.04	2.35	0.337	0.32057	0.198	0.018	0.975	8
0.065	0.6	0	0.365	0.259	0.023	0.964	9
*0.065	1.2	0.047	0.403	0.197	0.017	0.981	10
0.065	1.7	0.021	0.518	0.005	0.004	0.999	1
0.065	2.35	0.197	0.3596	0.165	0.015	0.986	8
0.11	0.6	-0.05	0.3896	0.244	0.022	0.972	9
*0.11	1.2	-0.02	0.4184	0.256	0.023	0.975	8
0.11	1.7	-0.13	0.5252	0.007	0.005	0.999	1
0.11	2.35	0.227	0.2974	0.299	0.027	0.938	8
0.15	0.6	-0.02	0.291	0.080	0.007	0.994	9
0.15	1.2	0.022	0.3061	0.262	0.023	0.946	10
0.15	1.7	-0.13	0.4281	0.120	0.084	0.962	1

Figure 6 shows the temperature evening out 0.04 m above the floor 1.2 m from the inlet vs. the height of the supply opening and the supply air velocity.

The graph shown was found by linearizing Eq. (3) and using multiple linear regression. Results from other calculations are shown in Table 2, as are also standard errors of the coefficients (for the linearized equation), and the coefficient of multiple determination, R^2. R^2 varies between 0.7 and 0.8. That means that 70-80% of the measured values could be explained by the model.

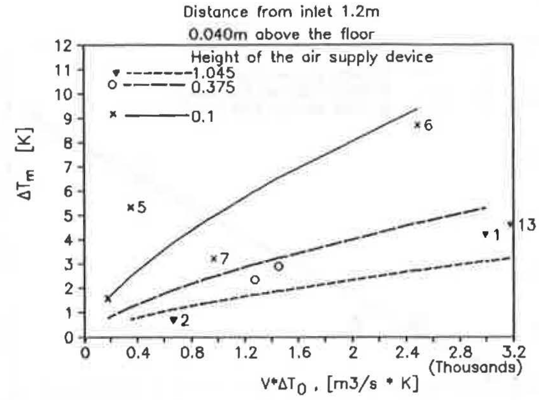

Fig. 6. ΔT_m against supply air flow rate times ΔT_0 , and height of supply opening. ΔT_m is the temperature difference between a point 1.1 m and 0.04 m above the floor. The width of the opening was 3.15 m.

Table 4. k_3 (value for the linearized model), k_4 and k_5 with statistics for different distances from their inlet and the floor. The starred value is shown in Fig. 6.

Height above the floor [m]	Distance from inlet [m]	Intercept k(3)	Coeff k(4)	Coeff k(5)	Std Err of y	Std Err of Coeff k(4)	Std Err of Coeff k(5)	R Sqrd	Degrees of Freedom
0.025	0.60	-2.200	0.395	-0.57	0.741	0.292	0.21	0.452	9
0.025	1.20	-3.840	0.622	-0.49	0.407	0.166	0.132	0.727	7
0.025	2.35	-3.940	0.622	-0.53	0.377	0.194	0.119	0.765	7
0.040	0.60	-3.920	0.634	-0.53	0.413	0.162	0.117	0.740	9
* 0.040	1.20	-4.210	0.670	-0.52	0.429	0.175	0.139	0.735	7
0.040	2.35	-4.390	0.645	-0.54	0.392	0.202	0.124	0.759	7
0.065	0.60	-3.160	0.542	-0.51	0.787	0.310	0.224	0.401	9
0.065	1.20	-4.590	0.729	-0.48	0.363	0.148	0.118	0.800	7
0.065	2.35	-4.110	0.656	-0.49	0.309	0.159	0.097	0.816	7
0.110	0.60	-4.020	0.655	-0.39	0.326	0.128	0.092	0.777	9
0.110	1.20	-4.800	0.748	-0.49	0.314	0.128	0.101	0.848	7
0.110	2.35	-5.060	0.774	-0.5	0.280	0.144	0.088	0.863	7

Narrow Supply Opening While the wide supply opening indicated two-dimensional flow, the narrow supply opening caused three-dimensional flow. Observations with smoke in the test room indicate that the flow is nearly radial when the Archimedes number is relatively high, see Figure 4. More results are given in Tables 5 and 6. These results show that the velocity decrease when the distance from the inlet increase. As mentioned this is due to the radial flow pattern.

Table 5. k_1 and k_2 for some levels and distances from the inlet.

Height above the floor [m]	Distance from inlet [m]	Intercept k(1)	Coeff k(2)	Std Err of y	Std Err of Coeff	R Squared	Degrees of Freedom
0.025	0.60	-0.250	0.535	1.088	0.125	0.948	1
0.025	1.20	0.255	0.300	0.219	0.025	0.986	2
0.025	2.35	0.495	0.139	0.240	0.026	0.933	2
0.040	0.60	-0.110	0.429	0.848	0.097	0.951	1
0.040	1.20	0.402	0.286	0.308	0.035	0.970	2
0.040	2.35	0.289	0.164	0.177	0.019	0.972	2
0.065	0.60	0.158	0.276	0.087	0.010	0.998	1
0.065	1.20	0.420	0.162	0.288	0.033	0.924	2
0.065	2.35	0.392	0.129	0.289	0.032	0.891	2
0.110	0.60	0.518	0.081	0.661	0.076	0.534	1
0.110	1.20	0.174	0.132	0.145	0.017	0.969	2
0.110	2.35	0.204	0.111	0.162	0.018	0.951	2

Table 6. k_3 (values for the linearized model), k_4 and k_5 for some levels and distances from the inlet.

Height above the floor [m]	Distance from inlet [m]	Intercept k(3)	Coeff k(4)	Coeff k(5)	Std Err of y	Std Err of Coeff k(4)	Std Err of Coeff k(5)	R Sqrd	Degrees of Freedom
0.025	1.20	-2.290	0.522	-0.310	0.170	0.135	0.158	0.951	1
0.025	2.35	-0.379	0.672	-0.138	0.829	0.677	0.794	0.790	1
0.040	1.20	-2.320	0.534	-0.240	0.070	0.056	0.066	0.993	1
0.040	2.35	-3.730	0.686	-0.370	0.136	0.134	0.157	0.975	1
0.065	1.20	-2.010	0.465	-0.110	0.279	0.223	0.261	0.907	1
0.065	2.35	-3.960	0.699	-1.370	0.922	0.756	0.887	0.745	1
0.110	1.20	-2.090	0.481	0.143	0.276	0.220	0.258	0.958	1
0.110	2.35	-3.170	0.615	-0.010	0.240	0.197	0.231	0.970	1

Cooling Capacity for the Wide and Narrow Supply Opening

By comparing the maximum heat loads it is now possible to compare the wide and narrow supply opening.

Figures 7 and 8 are based on Eqs. (2) and (3), with values for k_1, k_2, k_3, k_4, and k_5 for a point 0.04 m above the floor 1.2 m from the supply air inlet. u is set to 0.15 m/s. If, for instance, we allow a temperature difference of a maximum of 2.5 K between the floor level and a point 1.1 m above it, and we need to supply 0.035 m^3/s, we see that the maximum heat load for the wide supply opening becomes 200 W/m while it becomes 260 W/m for the narrow opening.

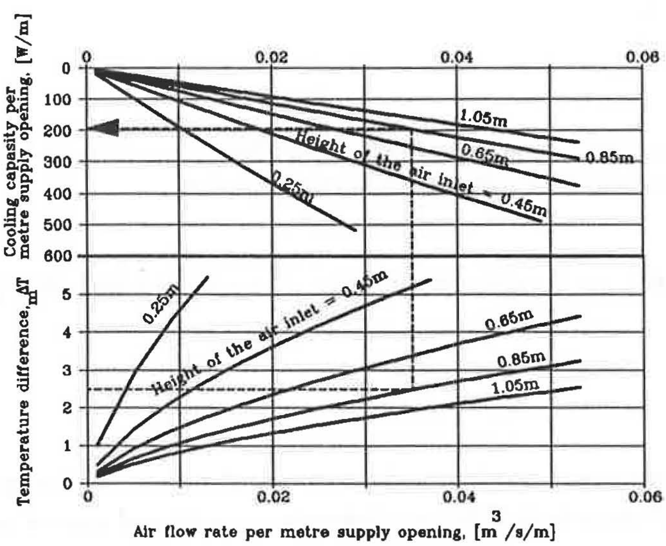

Fig. 7. Cooling capacity for a wide supply opening.

This informs us that the air is better mixed and the velocity more evened out if a narrow supply opening is used. However, if the excess heat exceeds what can be covered in one "narrow" supply opening, more openings have to be used. These must be mounted in the room with a certain distance, so that they do not influence each other, otherwise the results obtained for wide openings must be used.

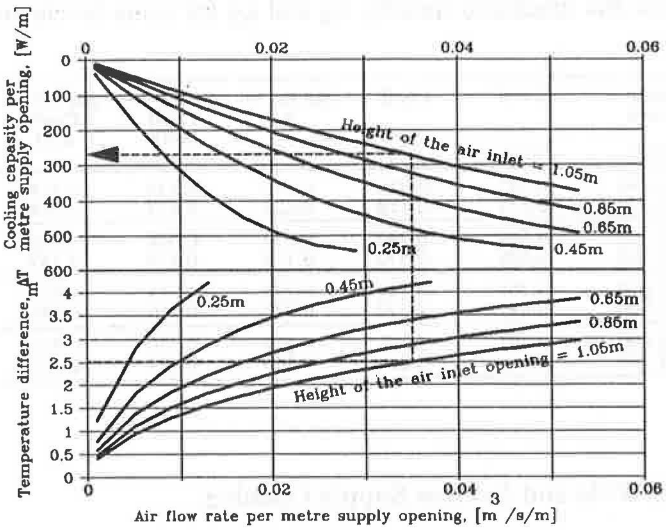

Fig. 8. Cooling capacity for a narrow supply opening.

Profiles for Velocity and Temperature

Abramovic [3] found that the profile for the velocity of a free turbulent jet could be described by:

$$\frac{u}{u_{max}} = \left(1 - \left(\frac{y}{b}\right)^{1.5}\right)^2 \tag{4}$$

where u is the velocity in distance y from the centre line, u_{max} is the velocity at the centre line, y is the distance from the centre line and b is the width of the jet.

It is convenient to express the ratio between the velocity and temperature profile as:

$$\frac{u}{u_{max}} = \left(\frac{\Delta T}{\Delta T_{max}}\right)^2 \tag{5}$$

where ΔT is the temperature difference between room air and a point at distance y from the centre line. ΔT_{max} is the temperature difference between room air and a point on the centre line.

The velocity and temperature profiles have been compared for all distances. To do this comparison the temperature ratio $\Delta T / \Delta T_{max}$ was squared. The conclusion is that the two curves are closely similar. This is in accordance with what can be seen for turbulent jets.

In Figs. 9 and 10 all measurements are combined for distances 1.2 m and 2.35 m to give a visualization of the deviation from the theoretical model. The velocities which deviate considerably from the model for $y/b > 0.7$ are actually negative due to the counterflow described in Fig. 3.

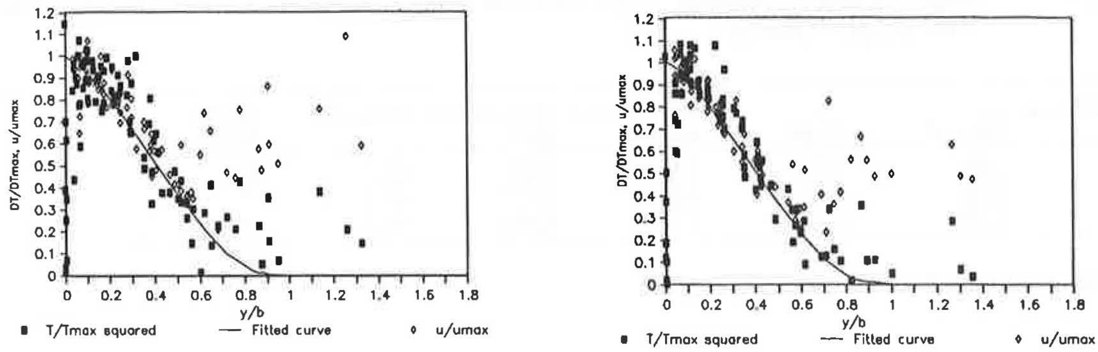

Fig. 9. Dimensionless velocities and temperatures for all measurements for distances 1.2 m and 2.35 m from the supply opening. Wide supply opening

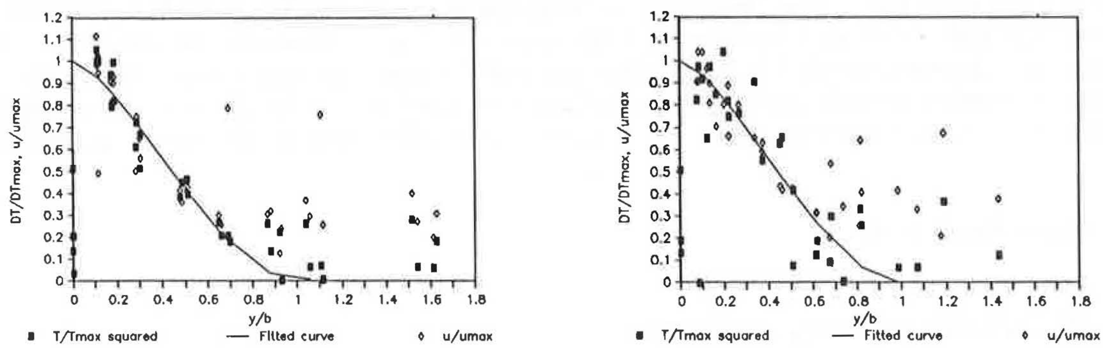

Fig. 10. Dimensionless velocities and temperatures for all measurements for distances 1.2 m and 2.35 m from the supply opening. Narrow supply opening.

The Width of the Flow

In a turbulent jet the width of the flow grows proportionally to the distance from the virtual origin of the flow, $b = C_b(x + x_p)$

In an isothermal jet one of the characteristics is that the impulse is constant along the flow. This means that a decrease in the velocity must be balanced by an increase in the mass flow rate.

A model for the width of the flow has been tested:

$$\frac{h-b}{b} = k_{1b} + k_{2b} \cdot Ar_0^{1/2} \tag{6}$$

In Table 7 the intercept and the slope of the model for the wide supply air opening are presented together with the statistics. Hypothesis testing showed that the coefficient for the slope was not zero, which means that the model gives an explanation of the data. Due to few experiments it was difficult to draw any exact conclusions for this model for the narrow opening.

As one can see the thickness do not grow with the distance from the inlet as it could expect from theory for turbulent jets. The reason is probably that the shape of the velocity profiles are influenced from the density differences.

Table 7. Height of supply opening minus width of flow vs square root of Archimedes number. Wide supply opening.

Distance from inlet	k_{lb} (Intercept)	k_{lb} (Slope)	Standard error intercept	Standard error slope	R squared	Degrees of freedom
0.6	-1.06	0.25	0.144	0.031	86.7	11
1.2	-0.93	0.17	0.066	0.014	93.9	10
1.7	-0.98	0.26	0.030	0.015	99.7	2
2.35	-0.81	0.19	0.135	0.029	80.7	11

Thermal Comfort

Thermal comfort depends on several parameters. Some of these are influenced by the principle of ventilation which is used. Parameters which are influenced by differences between ventilation systems are the temperature difference between the ankles and head, local mean air velocity and turbulence intensity. Other parameters like humidity and radiant asymmetry are more or less dependent on the design of the room and its use. The room air temperature is also dependent on the system because different ventilation systems have different temperature efficiencies. However, the temperature in the exhaust air is always controlled by the air flow rate and the supply air temperature. In the discussion of thermal comfort through displacement ventilation we need knowledge about the first-mentioned parameters. This section discusses the risk of drafts on the ankles from displacement ventilation.

Definitions, Draft Risk

The turbulence intensity is defined as:

$$Tu = \frac{\sqrt{\overline{v'^2}}}{\bar{v}} \cdot 100\% \tag{7}$$

where the mean air velocity in the main direction of the flow is:

$$\bar{v} = \frac{1}{\Delta t}\int_{t_0}^{t_0+\Delta t} v \cdot dt \tag{8}$$

and $\sqrt{\overline{v'^2}}$ is the Root Mean Square (RMS) of the velocity fluctuation. RMS is equal to the standard deviation. According to Fanger et al. [7] and [8] the percentage of dissatisfied occupants can be calculated as:

$$PD = (34 - t_a)(\bar{v} - 0.05)^{0.6223}(0.3696 \cdot \bar{v} \cdot Tu + 3.143) \tag{9}$$

t_a is the local air temperature. In this equation it is supposed that the skin temperature is 34 °C. This takes into account the heat transport due to forced convection and the dynamic response of the thermoreceptors in the skin. At velocities lower than 0.05 m/s the free convection boundary layer is supposed to remain undisturbed.

The equation is worked out for naked skin in the head region. It is also reported that at the extremities of the human body the sensitivity to draft seems to be a little less. Of course clothes also decrease the sensitivity to draft. This equation for the ankles should then give a conservative estimate of the draft risk.

Percentage of Dissatisfied Persons

In Table 8 mean air velocity is listed as a function of PD, Tu and local air temperature after Equation (9). Since the head region seems to be most sensitive to draft, the values in Table 8 probably overestimates the risk when used for displacement ventilation.

Table 8. Mean air velocity for different turbulence intensities, local air temperature and percentage of dissatisfied persons.

	Tu = 20%		Tu = 30%		Tu = 40%	
PD	T = 20°C	T = 22°C	T = 20°C	T = 22°C	T = 20°C	T = 22°C
10%	11.0 cm/s	12.5 cm/s	10.5 cm/s	11.5 cm/s	10.0 cm/s	10.8 cm/s
20%	20.0 cm/s	22.5 cm/s	18.0 cm/s	20.0 cm/s	16.5 cm/s	18.7 cm/s

If less than 10% of the occupants feel draft, the mean air velocity should be kept very low. A practical rule for winter conditions could be proposed, that the temperature close to the floor should be above 20 °C and the velocity should be kept below 11 cm/s. Respective values are 22 °C and 13 cm/s for summer conditions. The temperature difference between ankle and head level of a sitting person should not be more that 3 °K (according to NKB). Within a limited area close to the air terminal device lower temperatures and higher velocities could be accepted. A displacement ventilation system contributes little to the air motion outside the space close to the floor. That means that draft should be no problem at the head region of the occupants.

Equipment As mentioned, the velocity data were measured by means of type TSI 1610-12 anemometers. These have a 90% time constant of 0.25 to 0.30 seconds which means that the fluctuations which are relevant for thermal comfort could be measured. The frequency which seems to give strongest response on man ranges from 0.1 to 1 Hz.

How well the velocity fluctuations of the flow is described depends on the time constant of the anemometer. The calculated Tu depends on how well the fluctuations are described. Since different makes of anemometers have different time constants, Tu should depend on which equipment is used. However, since small fast fluctuations add little to the sum of squares, the calculated STD should be quite near the correct value. (Another problem is that a slow anemometer is cooled by fast fluctuations, leading to a measured mean velocity which is bigger than the real one since the anemometers are usually calibrated in near laminar flow.)

Measured Values

The velocity was measured every second. The duration of each measurement period was two minutes. This is less than what is recommended by ISO 7726 (i.e., three minutes). When several measurements are taken in the room, and an overall measure is the goal, then the mean will approach the correct value.

In Fig. 3, Tu is plotted vs the mean air velocity. According to Melikov [10] Tu should decrease when the mean air velocity increases. In these measurements the lowest velocities also seem to give the lowest Tu values. In the results shown above the air was supplied through a porous media giving a closely laminar airflow. This might lead to a stratification of the air giving the flow along the floor a laminar character.

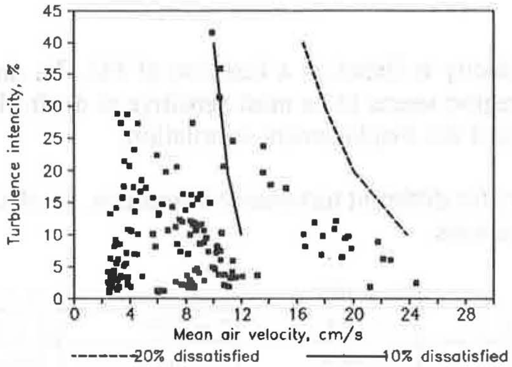

Fig. 11. Turbulence intensity vs mean air velocity for points 0.065 to 0.24m above the floor. The distance from the supply opening ranges from 0.6 to 2.35m.

However, if one looks at the highest Tu values as the mean air velocity increases, these values decrease. The curves plotted in the figure represent turbulence intensity vs mean air velocity at 20 ^{o}C for 10% respective 20% dissatisfied persons. These curves indicate that a reduction in the mean air velocities does not necessarily lead to fewer dissatisfied persons. The values listed in Table 8 show that only a slightly higher mean air velocity can be tolerated if the temperature is increased to 22 ^{o}C.

In most commercial air terminal devices the air is supplied through perforated sheets. Sørensen [11] has done measurements using commercial air terminal devices. These measurements were done in the same test chamber with the same equipment as the measurements in this paper. These measurements show a turbulence intensity ranging from 10 to 60 % with most values about 40%. This indicates that the design of the air terminal device influences the quality of the airflow and hence the thermal comfort. However, this does not mean that the thermal comfort is poorer for the commercial devices since the entrainment of ambient air might be better leading to less temperature differences.

Temperature Profile in the Room

In Fig. 12 the dimensionless temperature $(T_{actual} - T_{supply})/(T_{exhaust} - T_{supply})$ is plotted against the height above the floor. The temperatures were measured from the surface of the floor to a point 2.8 m above the floor.

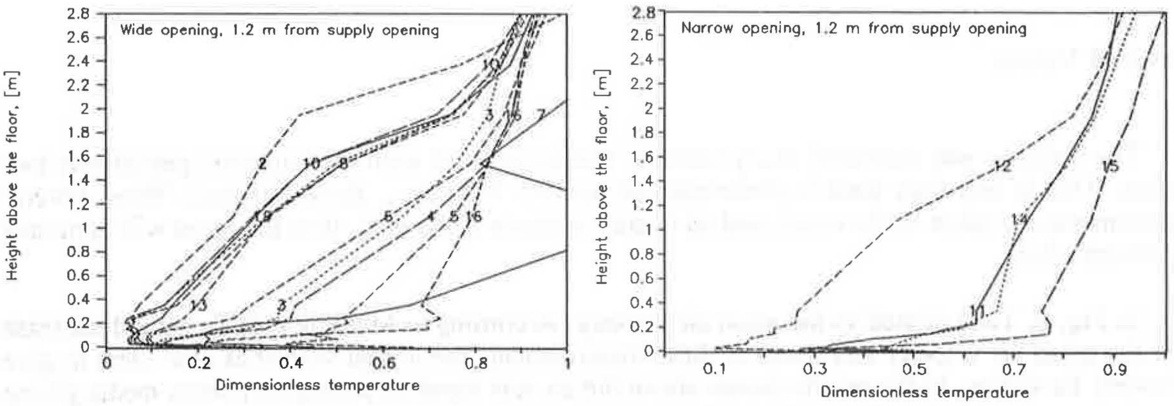

Fig. 12. Dimensionless temperatures plotted against the height above the floor. Wide opening to the left and narrow to the right. The numbers on the graphs refer to the test numbers in Tables 1 and 2.

It can be clearly seen from Fig. 12 that the shape of profile varies from test to test. If the curves are related to the air flow rates as shown in Figure 13 it is obvious that the shape of the curves depends on the air flow rate. What actually happens is that when the supply air flow rate is reduced the height of the lower zone decreases when the entrainment to the convection flow is constant.

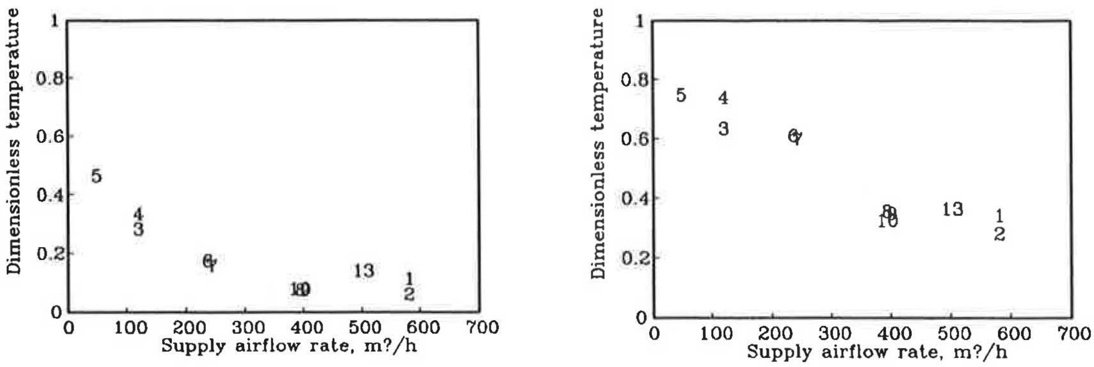

Fig. 13. Dimensionless temperatures plotted against the supply airflow rate. 0.15 and 1.1 m above the floor to left and wright respective. Wide opening. The numbers on the graphs refer to the test numbers in Tables 1 and 2.

Another interesting observation which can be made from the profiles in Fig. 12 is that if the profile line in the lower part is extended the intercept will be close to origin.

It can also be seen that the shape of the profiles is the same in all positions. This is due to the poor entrainment of ambient air in the flow discussed. However, the quality and the position of the heat sources probably plays an important role for the shape. One should therefore avoid generalizing too much from this data. However, there is an obvious limitation in the maximum temperature difference. To ensure thermal comfort, high heat loads must be met by high airflow rates rather than low supply air temperatures.

Conclusion

The work has revealed that the velocity ratio between the velocity close to the floor and the supply air velocity is a simple function of the inlet air's Archimedes number. The Archimedes number is the ratio between buoyancy and inertia forces.

The model which is used for temperature evening out gives relatively large residual values. However, this model also explains most of the variations in the measured variables.

With correct design it is possible to remove considerable excess heat from the room, without exceeding the limits of thermal comfort. For instance, with an air supply terminal device, which is 0.85 metre high, it is possible to remove 200 W per metre width of the opening, with an air flow rate of 0.035 m^3/s (126 m^3/h) if a mean air velocity of 0.15 m/s is allowed. However, the airflow rate may be larger than for complete mixing ventilation.

Further the results in this paper show that flows from air terminal devices used for displacement ventilation could be described with respect to width and form of the profiles for temperature and velocity. A complete description of the flow can be given. Data for turbulence intensity and temperature profiles are necessary to evaluate the thermal comfort. The air flow from air terminal devices for displacement ventilation does not behave like a turbulent jet since the entrainment of air into the flow is very small. However, narrow air terminal devices act in a different fashion. They seem to have larger entrainment and thus behave more like turbulent jets. However, for narrow openings more data are necessary to draw any exact conclusions.

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SUMMARY

Displacement ventilation is acknowledged to be an efficient system for the removal of contaminants and excess heat from occupied parts of rooms. However, air flow rates, temperature and the design of the air supply device strongly influence the parameters which decide the thermal comfort. This paper reviews experiments and theoretical models which show the connection between these parameters. The width and shape of the air supply device has also been varied, and a porous media has been used on the inlet area of the air supply device. The velocity and temperature profiles have been measured. The results presented also show that the flow could be described with respect to width and form of the profiles for temperature and velocity. The flow does not behave like a turbulent jet due to thermal stratification. It is shown that the Archimedes number of the supply air is the parameter which decides the air velocity in the area close to the floor. (The Archimedes number is here defined as the ratio between buoyancy and inertia forces).

The results show that it is possible to remove considerable amounts of excess heat from a room without exceeding the limits for thermal comfort, however, this requires relatively high airflow rates.