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AIR MOTION IN THE VICINITY OF AIR-SUPPLY DEVICES FOR DISPLACEMENT VENTILATION

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ABSTRACT

In displacement ventilation systems, 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. It is indicated 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 the ratio between buoyancy and inertia forces). 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 results show that it is possible to remove considerable amounts of excess heat from a room without exceeding the limits for thermal comfort. Comparisons with commercial elements indicate that the design of the device is of minor importance as long as the air flow is horizontal and the perforation ratio of holes is high enough.

1. INTRODUCTION

During the last few years there has been a lot of work done on developing efficient ventilating systems, i.e. systems which remove contaminants and excess heat with a minimum use of 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 contaminants and excess heat, both in industrial and comfort ventilation.

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 suplied. More air means that the rising warm air can be feeded with fresh air to a higher level before it must recirculate and feed itself. In this way the air in the room will be stratified with a lower zone with fresh air, and a upper zone with contaminated air.

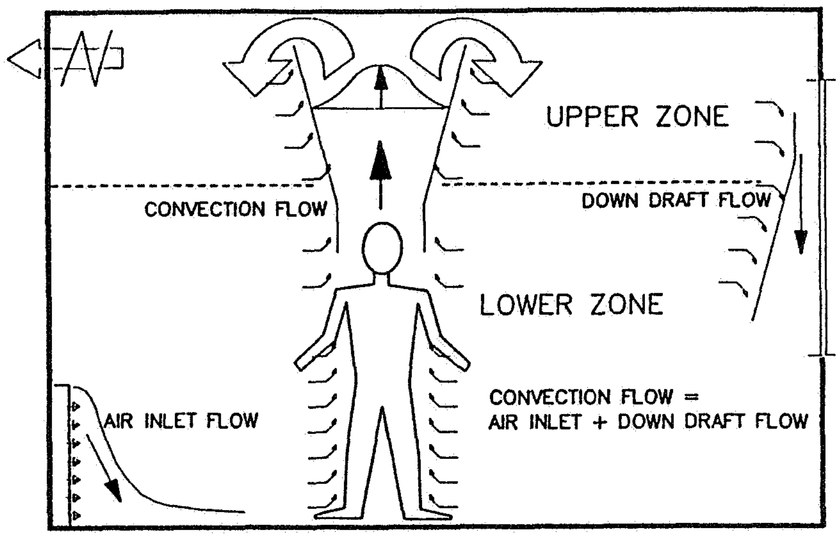


Figure 1. The principle of displacement ventilation

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. This improved effectiveness means that the fresh air supply to the room can be decreased without reducing the air quality found with complete mixing.

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 dimensioning air supply terminal devices and to decide necessary air flow rates and the supply air temperature to fullfil claims for thermal comfort.

2. TEST EQUIPMENT

2.1 Test room and air supply

Tests have been carried out in a room of approximately 45 m^2 with a floor area of 16 m^2, Figure 2. Air was supplied through an adjustable opening in the short wall. 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 microprocessor-based datalogging system.

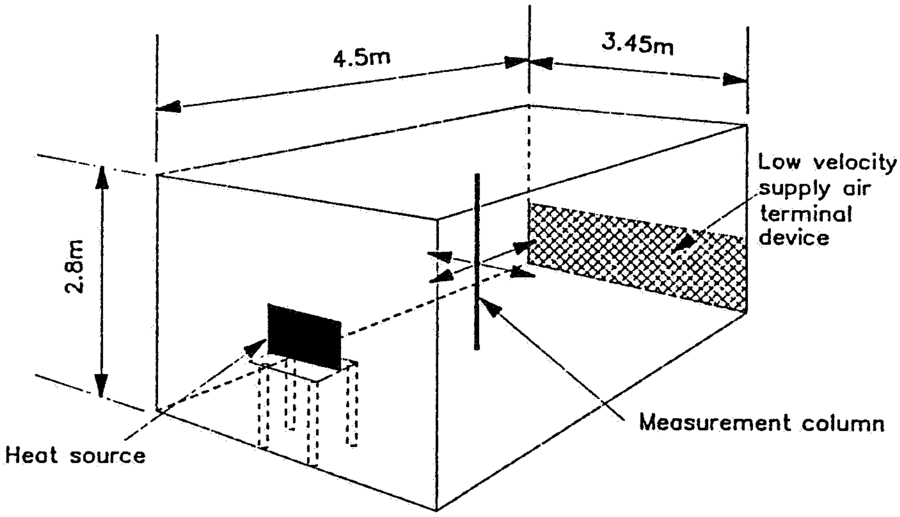


Fig. 2. Test room.

The air supply terminal device is shown in Figure 3. In the experiments two different widths were used, 0.54 m and 3.15 m.

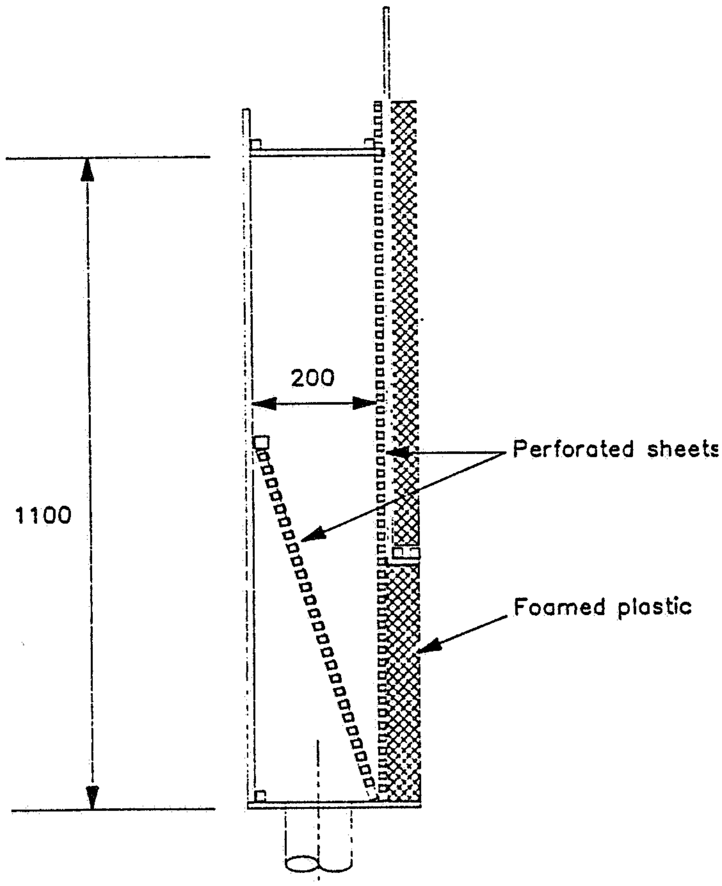


Fig. 3. Air supply terminal device

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 air flow rate was measured with an orifice plate and an inclined tube manometer. A damper was used to adjust the air flow rate. The evacuation flow 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.

2.2 Air velocity and temperatures

The anemometers and temperature meters were located on a measurement column as shown in Figure 4.

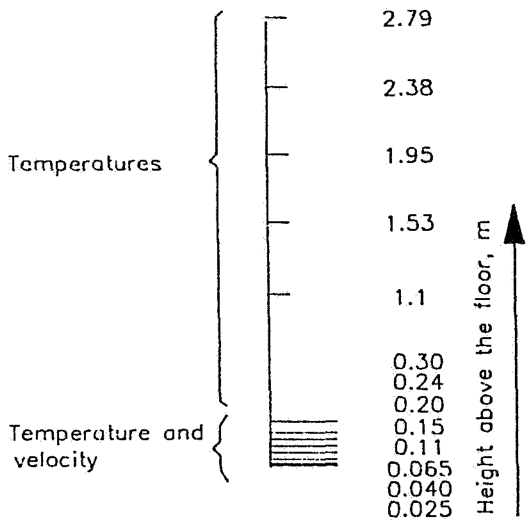


Fig. 4. Column for temperature and air velocity measurements

The column could be moved in two directions by two electric motors, controlled by the microprocessor. The position of the column was monitored by two potentiometers located on the shaft of the motors.

Velocities and temperatures were recorded at distances of 0.6, 1.2, 1.7 and 2.35 m from the opening. The anemometers were of the 1620-12 and 1610-12 TSI-type.

The anemometers were calibrated in a TSI-1125-calibration unit. As the anemometers of the 1620-12 type proved to be a temperature dependent, they were calibrated at several temperature levels.

The air temperatures 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.

3. RESULTS

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

$$\frac{u}{u_o} = k_1 + k_2 \, Ar_o^{1/2} \tag{1}$$

Where:

u - the velocity in the flow close to the floor, m/s

u_0 - the supply air velocity, m/s

$$Ar_0 = \frac{g \, \beta \, \Delta T_0 h}{u_0^2} \tag{2}$$

g = gravitational acceleration, m/s^2

β = volumetric exponsion factor

ΔT_0 = temperature difference between room air and supplied air, K

h = height of the supply opening

It was harder to find a relation for the evening out of the temperature. However, a relationship was found from a formula often used for calculating plumes rising from heat sources. In this relation the temperature difference between the room air and the maximum temperature in the plume is a function of the convective heat load supplied and the height above the source. The suggested formula is:

$$\Delta T_m = k_3 \cdot (\dot{V}_0 \cdot \Delta T_0)^{k_4} \cdot (h)^{k_5} \tag{3}$$

Where:

$\dot{V}_0$ - air flow rate, $[m^3/s]$

ΔT_0 - Temperature difference between room air and supplied air, K

h - height of the supply opening, m

However, as can be seen in the next section, the correlation for this model is not too good, so there is a need for more research on this detail.

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

$$Q = \frac{\varrho \cdot C_p}{g \cdot \beta} \frac{\left(u \cdot \frac{h \cdot B}{\dot{V}_0} - k_1\right)^2}{B^2 \cdot h^3 \cdot k_2{}^2} \dot{V}_0{}^3 \tag{4}$$

where

β = volumetric expansion factor, $\frac{1}{T}$

g = gravitational acceleration, m/s^2

ϱ = density, kg/m^3

C_p = specific heat capacity of air, $\frac{J}{kgK}$

B = width of supply air opening, m

$\dot{V}o$ = inlet air flow rate, m^3/s

The height, h, of the opening was varied from 0.05 m to 1.047 m. ΔT was varied from 3 to 14 K. The supply air velocity was varied from 0.045 to 0.2 m/s. Altogether 16 experiments were done, 12 with the wide supply opening and 4 with the narrow.

3.1 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 1. 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 numbers in Table 1 reveals that the velocity does not change much as the air flows along the floor. Neither do the thickness of the flow increase as one could expect from theory for turbulent jets. This means that the shape of the velocity profiles are influenced from the density differences.

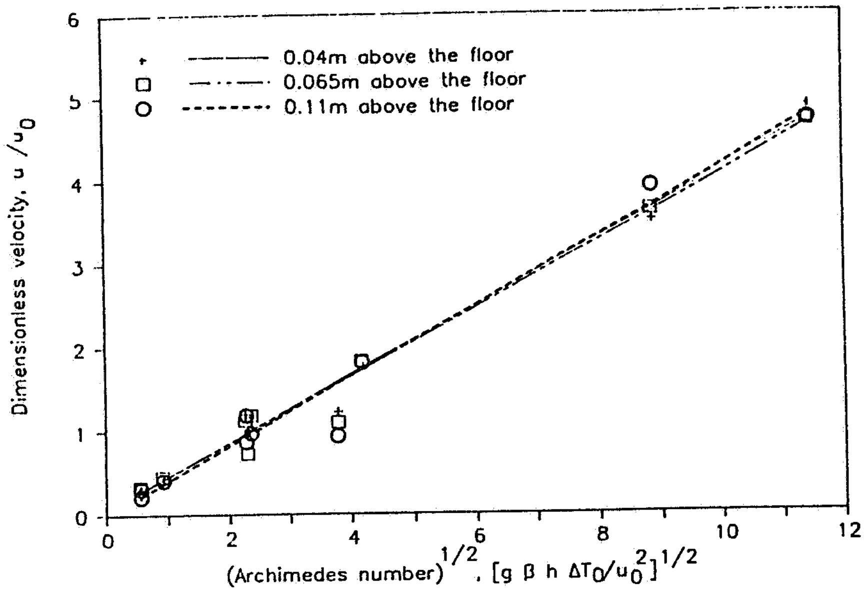


Fig. 5. Air velocity 1.2 m from the opening vs. supply air's Archimedes number and the velocity of supply air. The width of the opening was 3.15 m.

Tab. 1. k_1 and k_2 for different levels and distances from the supply opening. Starred graphs are shown in Fig. 5.

	Distance above the floor [m]	Distance from inlet [m]	Intercept	Coeff	Std Err of y	Std Err of Coeff	R Squared	Degrees of Freedom
			k(1)	k(2)				
	0.025	0.5	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	0.3296	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

Fig. 6 shows the temperature evening out 0.04 m above the floor 1.2 m from the inlet vs. 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 coeffisients (for the linearized equation), and the coeffisient 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.

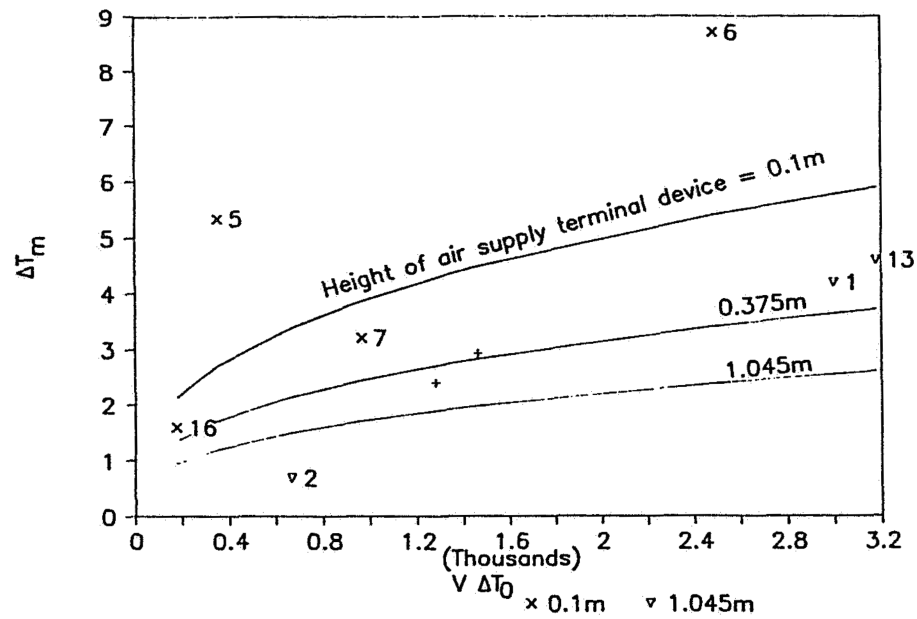


Fig. 6. Temperature difference against supply air flow rate times ΔT_0, and height of supply opening. ΔTm-temperature difference between a point 1.1 m and 0.040 m above the floor. The width of the opening was 3.15 m.

Table 2. 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 graphed in Fig. 6.

	Distance 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	0.134	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

3.2 Narrow supply opening

While the wide supply opening indicated a two dimensional flow, the narrow supply opening caused a three dimensional flow. From Figures 7 and 8 it can be seen that the narrow supply opening gives results that differ from the wide opening. Observations with smoke in the test room indicate that the flow is nearly radial when the Archimedes number is relatively high. More results are given in Tables 3 and 4. This results show that the velocity decrease when the distance from the inlet increase. As mentioned this is due to the radial flow pattern.

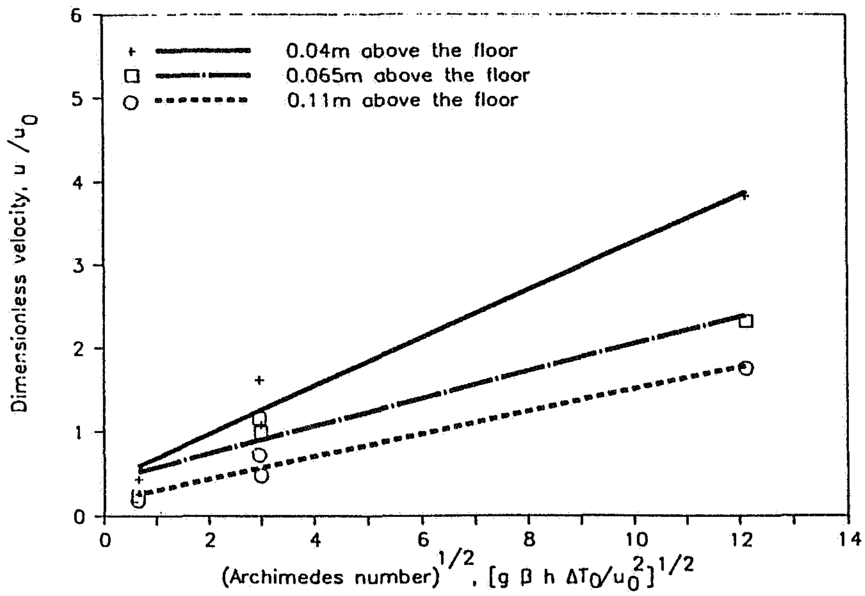


Fig. 7. Air velocity, 1.2 m from the opening vs. supply air Archimedes number and supply air velocity. The width of the opening was 0.54 m.

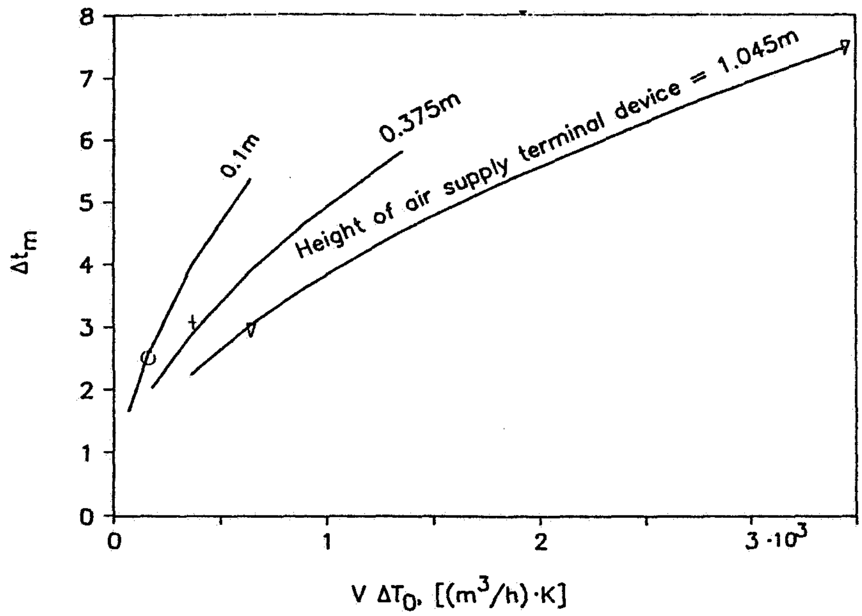


Fig. 8. Temperature difference vs. supply air flow rate times ΔT_0 and the height of the supply opening

Table 3. k_1 and k_2 for different levels and distances from the inlet. Starred values are graphed in Fig. 7.

	Distance 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 4. k_3 (values for the linearized model), k_4 and k_5 for different levels distances from the inlet. The starred value is graphed in Fig. 8.

	Distance above the floor	Distance from inlet	Intercept	Coeff	Coeff	Std Err of y	Std Err of Coeff k(4)	Std Err of Coeff k(5)	R Sqrd	Degrees of Freedom
			k(3)	k(4)	k(5)					
	[m]	[m]								
	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

3.3 Comparison between wide and narrow supply opening

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

Figures 9 and 10 are based on Eqs. 4 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

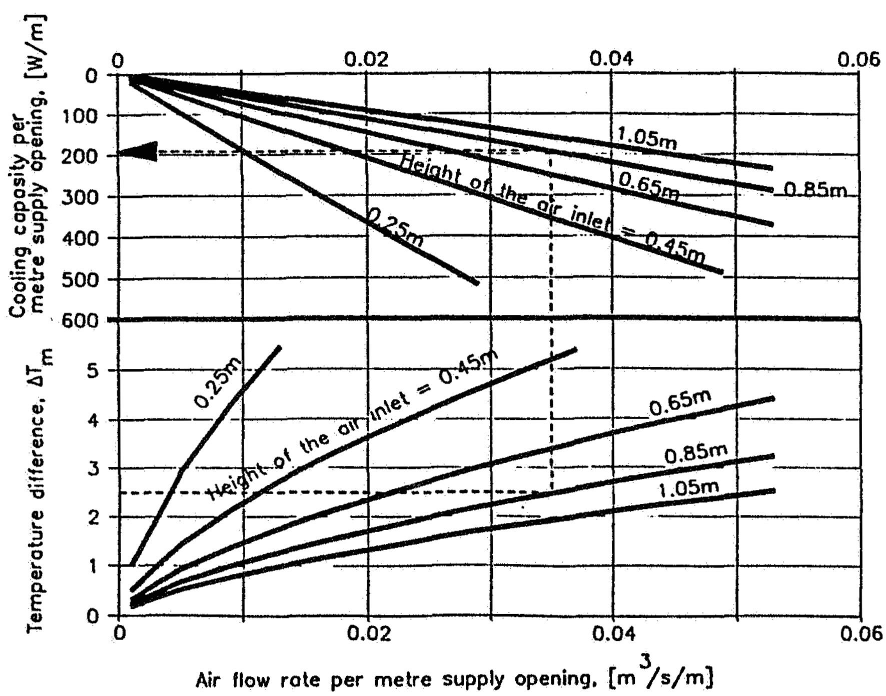


Fig. 9. Cooling capasity for a wide supply opening

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.

This informs us that the air is better mixed and the temperature 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.

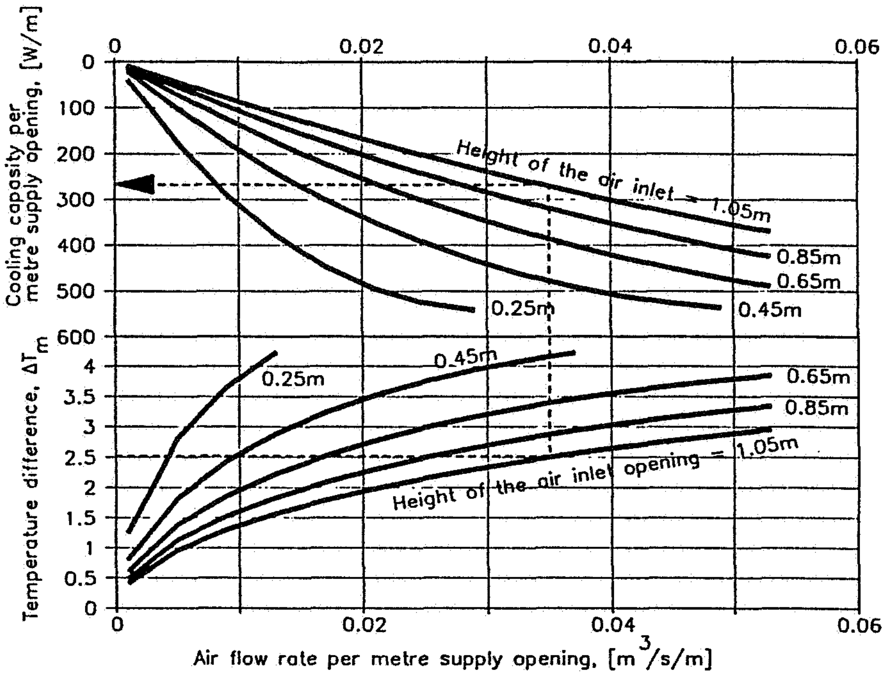


Fig. 10. Cooling capasity for a narrow supply opening

3.4 Commercial air supply devices

In Scandinavia there are several manufactorers of air supply devices for displacement ventilation. The narrow supply opening described in this paper has the same width tath some of the commercial devices use. It is therefore possible to compare this to the experimental results. The comparison has been done with the Sili IAC-03-05, IAC-05-05 and IAR-07-05 from GÄVLE VÄRKEN A/B. Their documentation is dated December 1984. The documentation is recalculated and plotted in Figure 11. Data for temperature evening out was not found in the documentation.

It can be seen that these commercial inlets cause higher velocities than the elements used during the tests. The reason might be that these elements have a greater entrainment of room air into the inflowing air, so that the flow rate in the room is larger. This should cause a better smoothing out of the temperature.

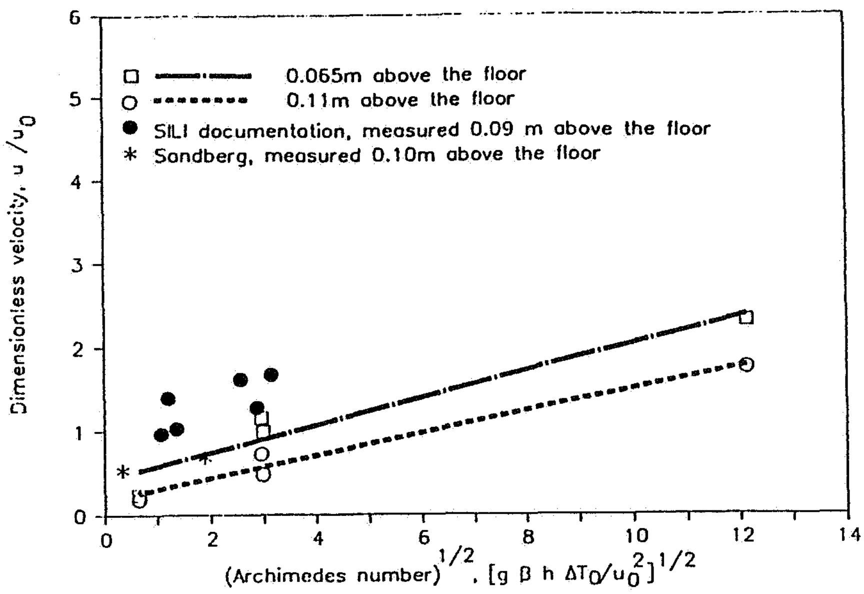


Fig. 11. Comparison with commercial air supply terminal dvicer

Sandberg[2], has tested one of these commercial elements (SILI IAC-03-05) in a test room simulating an office. These results are also shown in Figure 11. Strangely enough these results show lower velocities than the manufacturer's documentation.

4.0 CONCLUSION

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

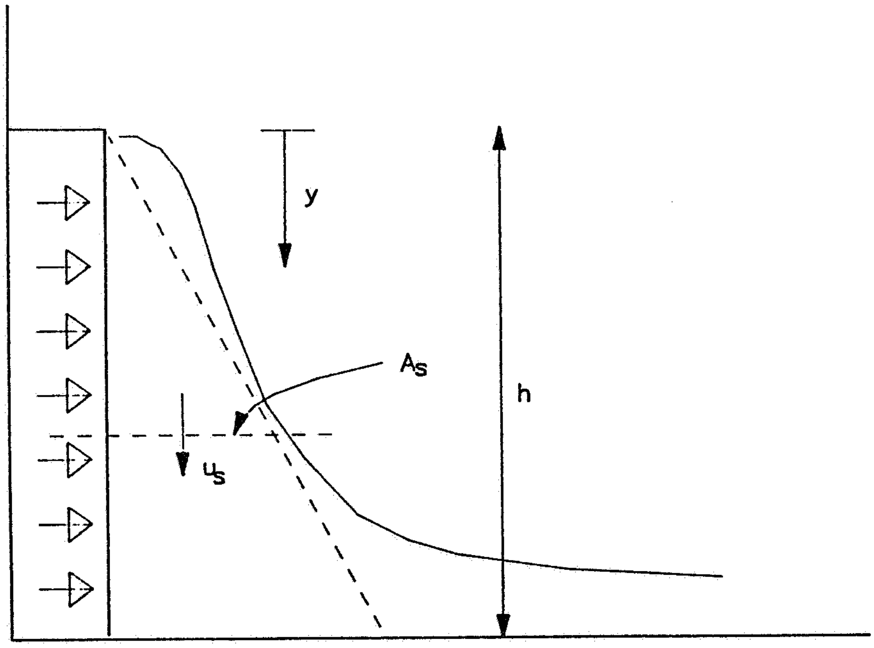
For the temperature evening out, the model which is used, gives relatively large residual values. However, this model also explains most of the variations in the measured variables.

With correct dimensioning 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 ($\sim$ 126 m^3/h).

The inlet opening consisted of porous foamed plastic. A simple comparison with commercial devices with perforated sheets do not reveal any advantages of this solution. Probably the design of the inlet is of minor importance as long as a horizontal outflow is secured. A low rate of holes will probably increase the velocity close to the floor, due to the increased induction of room air. However, this should lead to the temperature being more evened out.

APPENDIX

Equation for the dimensionless velocity in the downflowing air close to the inlet. The following deduction is done after an idea of PhD Eimund Skåret.



$$\dot{m} = \frac{M_0}{h} \cdot y \tag{I.1}$$

$$u_s = \frac{\dot{m}}{\varrho_0 \cdot A_s} \tag{I.2}$$

where

- M_0 - mass flow
- h - height of the supply opening
- y - distance from the top of the inlet
- A_s - section area of the flow
- ϱ_0 - density of the inlet air

The momentum flux can in the vertical direction be written as:

$$B_y = \dot{m} \cdot u_s \tag{I.3}$$

and the buoyancy as:

$$O = \varrho_r\ g\ \beta\ \Delta T_0\ A_s \cdot dy \qquad (I.4)$$

where

ϱ_r - room air density
g - gravitational acceleration
β - volumetric expansion factor
ΔT_0 - temperature difference between room air and inlet air

$$\varrho_r\ g\ \beta\ \Delta T_0\ A \cdot dy = \frac{d}{dy} B_y\ dy = \frac{d}{dy} (\dot{m} \cdot u_s) dy \qquad (I.5)$$

Using Eq. I.1 and I.2 this can be written as:

$$y\ u_s \frac{du_s}{dy} + u_s^2 - \frac{\varrho_r}{\varrho_0} g\ \beta\ \Delta T_0\ y = 0 \qquad (I.6)$$

Now we let

$$y\ u_s^2 = z \rightarrow u_s^2 = \frac{z}{y} \qquad (I.7)$$

Derivates

$$u_s^2 + y \cdot 2\ u_s \frac{du_s}{dy} = \frac{dz}{dy} \qquad (I.8)$$

Puts this into I.6

$$\frac{z}{y} + \frac{dz}{dy} = 2 \frac{\varrho_r}{\varrho_0}\ g\ \beta\ \Delta T_0\ y = 0 \qquad (I.9)$$

Solving this differential equation leads to:

$$u_s = (\frac{2}{3} g\ \beta\ \Delta T_0\ y)^{1/2} = (\frac{2}{3} \frac{g\ \beta\ \Delta T_0\ h\ u_o^2}{h \cdot u_o^2} \cdot y)^{1/2}$$

$$u_{s_0} \quad (\frac{2}{3} A_{r_0})^{1/2} \; (\frac{u0^2 \cdot y_0}{h})^{1/2} \qquad (I.10)$$

The increased momentum flux due to the density difference can now be written as:

$$B_{\Delta\varrho} = \dot{M}_0 \; u_{s_0} = \varrho_0 \; u_0 \; A_0 \; u_{s_0}$$

However, there will be an additional increase to the momentum flux due to the inflowing air's horizontal velocity. If we assume that this fluxes can be added, we have:

$$B_{tot} = B_0 + B_{\Delta\varrho}$$

$$I_4 \; A_s \cdot u_m^2 \cdot \varrho_0 = \varrho_0 \; A_0 \cdot u_0^2 \cdot \frac{i}{\varepsilon} + \varrho_0 \; u_0^2 \; A_0$$

$$\cdot \; (\frac{y_0}{h})^{1/2} \; (\frac{2}{3} A_{r_0})^{1/2} \qquad (I.11)$$

where

$$A_s = b \cdot B$$

B - width of the opening

b - C_b $(y + y_p)$ - thickness of the flow

$$A_s = \frac{\dot{V}}{u_m \; I_2}$$

$$I_n = \int_0^1 (1 - (\frac{y}{b})^{1.5})^n \frac{dA}{A_s}$$

$$u_m = \frac{I_2}{I_4} \cdot \frac{I}{\dot{V}} \; (A_0 \; u_0^2 \; \frac{i}{\varepsilon} + \varrho_0 \; u_0^2 \; A_0 \; (\frac{y_0}{H})^{1/2} \; (\frac{2}{3} \; A_{r_0})^{1/2})$$

$$\frac{u_m}{u_0} = \frac{I_2}{I_4} \; \frac{\dot{V}_0}{\dot{V}} \; \frac{i}{\varepsilon} \; (1 + \frac{\varepsilon}{i} \; (\frac{y_0}{H})^{1/2} \; (\frac{2}{3} A_{r_0})^{1/2}) \qquad (I.12)$$

$$\frac{u_m}{u_0} = k_1 + k_2 \; A_{r_0}^{1/2} \qquad (I.13)$$

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Discussion

Paper 7

W. De Gids (TNO Division of Technology for Society, Holland) Did you study the influence of moving room occupants?

H.M. Mathisen (SINTEF, Trondheim, Norway) The effect of moving people has been studied in earlier work by others, we have not published anything on this. It can be observed by using smoke visualisation or tracer gas that when a person enters a room some mixing of the room air occurs, but after a few minutes the situation becomes stable once more. However with displacement ventilation it is not possible for the ventilation efficiency to be lower than for a complete mixing situation.

J. Van Der Maas (Ecole Polytechnique Federale de Lausanne, Switzerland) Equation (3) which deals with temperature distribution in your paper confuses me because of (a) the units and (b) the absence of the heat load and the presence of the height of the inlet (h). Could you provide some information on the origin of this equation?

H.M. Mathisen (SINTEF, Trondheim, Norway) Equation (3) is empirically derived. Several models have been tried and, up to now, this is the one which gives the best fit to measured values. However in the measurements taken there has been found to be a quite strong correlation between h and V0. A simpler model which gives nearly the same fit to the measured values is to set the ratio DTm to DTo to be a constant.

P. Appleby (Paul Appleby Chartered Engineer, Norwich, UK) Have you measured the volume flow rate of air above each occupant and related this to the supply air volume flow rate at the displacement panels?

H.M. Mathisen (SINTEF, Trondheim, Norway) No. This is a very important question, because this is needed in order to size the panels. A typical supply flow rate of about 50 m3/h per person is suggested.