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Convection flows above common heat sources in rooms with displacement ventilation

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Elisabeth Mundt
Department of Building Services Engineering
Royal Institute of Technology
Stockholm, Sweden

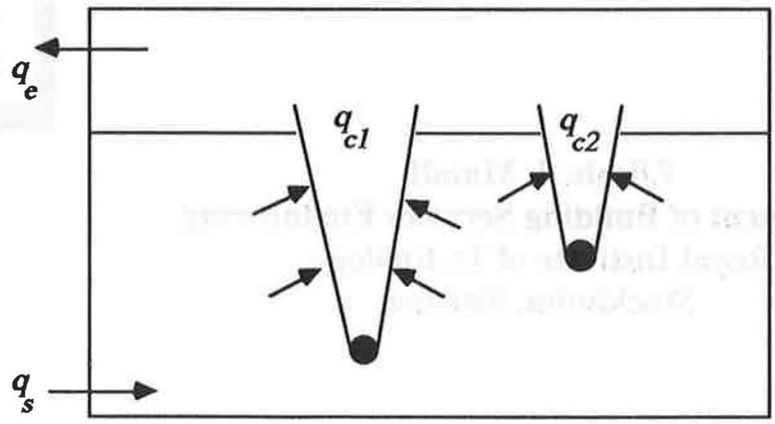
CONVECTION FLOWS ABOVE COMMON HEAT SOURCES IN ROOMS WITH DISPLACEMENT VENTILATION.

Elisabeth Mundt
Department of Building Services Engineering
Royal Institute of Technology
Stockholm, SWEDEN

Introduction.

The design of displacement ventilation has so far mostly been based on experience and the guidelines provided by the manufacturers of supply air devices. The base for the displacement ventilation is the convection flows in the room, transporting the warm and polluted air into the upper zone where the exhaust device is situated. Convection flows from different heat sources have earlier been investigated in surroundings with no temperature gradients, however the fundamental principle of displacement ventilation is the temperature gradient and consequently the convection flows in the presence of temperature gradients are of great interest.

The principle of displacement ventilation is shown in figur 1. The supply air is introduced in the floor area at low impulse and spreading out above the floor. The convection sources transport the air into the upper part of the room. The level where the supply air flow is equal to the air flow in the convection plumes will ideal be the border between the lower clean zone and the upper polluted zone. The amount of air transported within the convection flows at different levels is therefore influencing the position of the border between the zones.



q_s = supply air flow
q_{c1} = convection air flow
q_{c2} = convection air flow
q_e = exhaust air flow

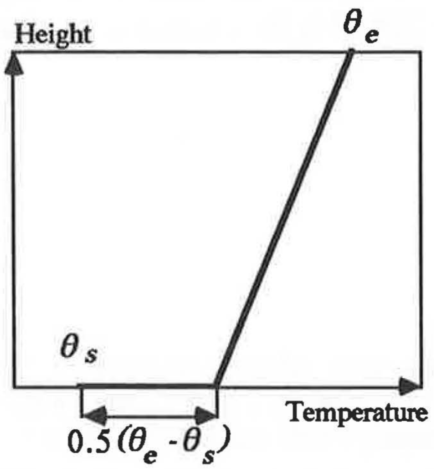
Fig. 1. Displacement ventilation

The initial flow from a heat source is depending on the area and geometry of the source and the temperature of the source. Once the flow has left the source, the driving forces are the momentum and the buoyancy of the plume, which is dependant on the temperature of the surrounding air. If the temperature increases with height the buoyancy of the plume will decrease with height. When the flows from different sources are known as function of gradients and height, the border between the zones can be calculated. This paper presents some resluts obtained in a research project at The Royal Institute of Technology in cooperation with some of the manufacturers of displacement supply devices in Sweden, Norway and Finland. The project consists of determining the convection air flows above common sources in offices with displacement ventilation.

Temperature gradients.

Background.

One part of the problem is to determine the gradients obtained in rooms with displacement ventilation. The gradient is of course depending of the temperature difference between the supply and exhaust air, but how is it distributed in the room? It is often said (1,2) that in offices half of the temperature difference is evened out in the floor area, and the rest of the difference is linear from the floor to the ceiling, see fig 2.



θ_e = exhaust air temperature
θ_s = supply air temperature

Fig. 2. Simplified calculation method of temperature gradient in offices (1,2).

Measurements.

In order to get some more information about this, some tests were made in a room (L x W x H, 3.6 x 3.6 x 2.4 m) made of fiber board. The floor is insulated with 10 cm mineralwool and the rest of the room is uninsulated. The room is situated in a basement floor with rather steady temperature conditions.The supply device (Stifab DAC 16) is placed in the middle of one wall close to the floor and the exhaust device is on the same wall close to the ceiling.

The heat source was a simulator of a man consisting of 1 m painted ventilation duct with a diameter 0.4 m. The top was covered and inside four 25 W bulbs were evenly placed. The simulator was placed in the centre of the room and tests were made with three different ventilation flows (75, 150 and 200 m^3/h). The temperature differences between supply and exhaust air were varied so that the 100 W was ventilated away.

Figure 3 shows the nondimensional gradients in the room, 1.8 m away from the supply device. The temperatures were recorded at each 2 cm close to the floor and the ceiling and each 20 cm inbetween. Three different rods were used for the measurements and there were no big differences between them. The nondimensional gradients are given as recorded temperature minus supply temperature divided by the difference between the supply and exhaust temperature. The relative raise of the temperature in the floor area seems to be very dependent on the ventilation rate.

The simulator was also placed 0.5 m above the floor and the the nondimensional gradient for this case is also showed in figure 3 for the lowest air flow. To increase the gradients an extra heat source (a small radiator) was placed on one wall at different heights and the gradients recorded, see fig 4, (airflow 150 m^3/h). As can be seen from these figures the gradients are rather linear in the room, except when great heat sources are placed on the floor. The most interesting is the relation floor area air temperature minus supply air temperature divided by the total temperature difference. This quotient varies between 0.3 to 0.5, and seems to be rather independant of the amount of heat in the room and the placing of the heat sources.

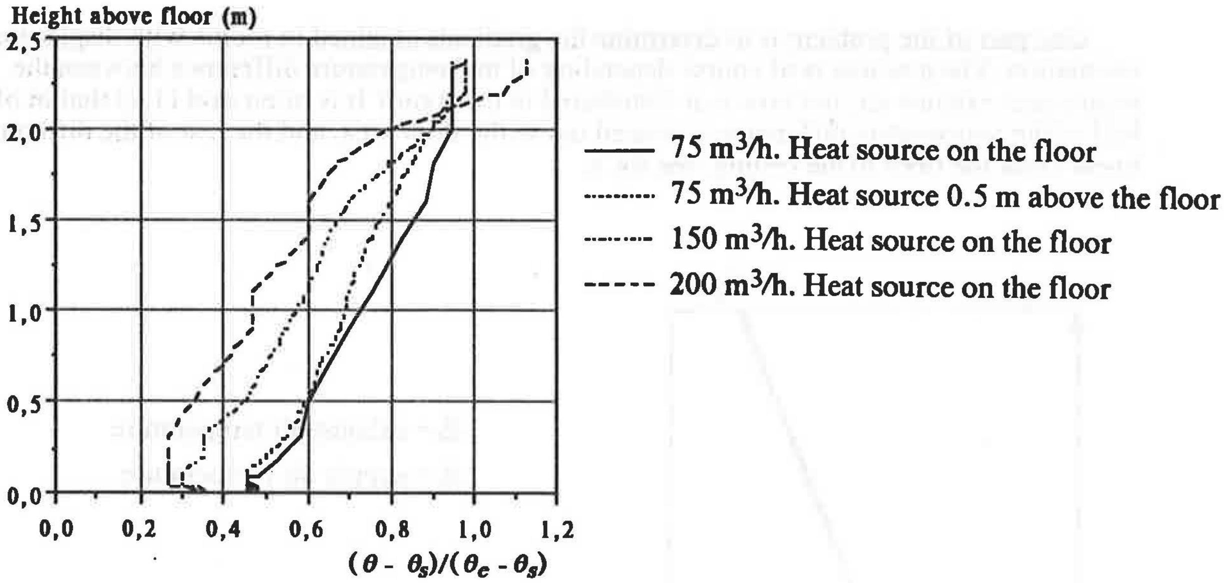


Fig 3. Nondimensional temperature gradients in a room with different air flows 75, 150 and 200 m^3/h. Heat source 1 m high, 100 W placed in the middle of the room on the floor and lifted up 0.5 m. Temperature difference supply - exhaust 4.0, 2.0 and 1.5 K.

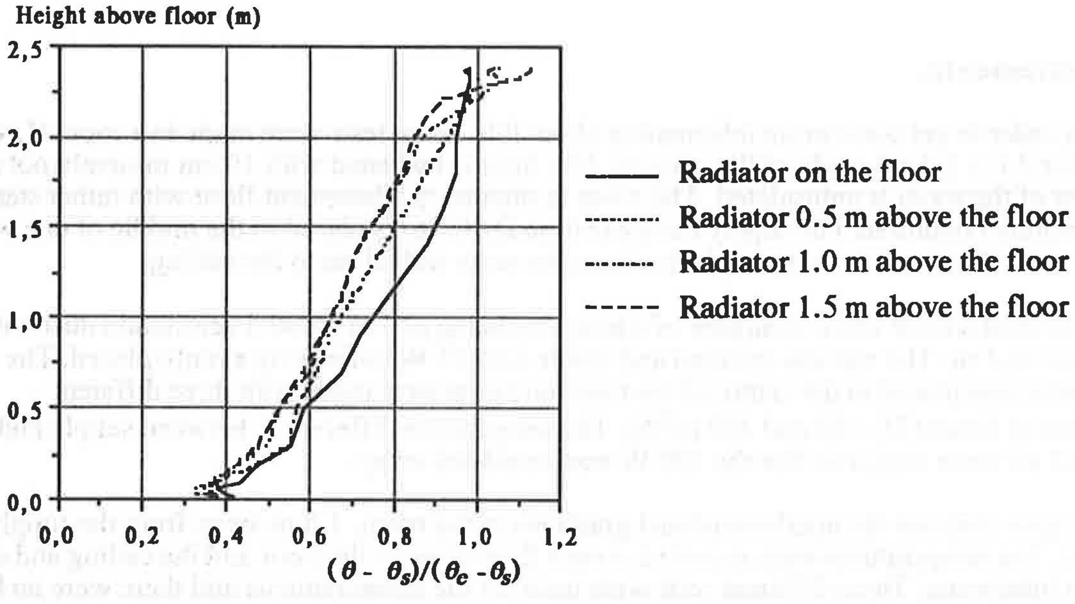


Fig 4. Nondimensional temperature gradients in a room with air flow 150 m^3/h. Heat source 1 m high, 100 W placed in the middle of the room on the floor and a radiator 200 W placed on one wall at different heights. Temperature difference supply - exhaust 6 K.

A calculation model.

The raise of the air temperature in the floor area is due to the radiation from the ceiling and the walls to the floor and the following convective transport to the supply air in the floor area. Part of the temperature raise is also due to the induction of room air into the supply air.

An approach to to this problem is the following very simplified equations

$$\alpha_r \cdot A \cdot (\theta_e - \theta_f) = \alpha_{cf} \cdot A \cdot (\theta_f - \theta_{af}) \qquad (1)$$

$$q \cdot \rho \cdot c_p \cdot (\theta_{af} - \theta_s) = \alpha_{cf} \cdot A \cdot (\theta_f - \theta_{af}) \qquad (2)$$

$$\frac{\theta_{af} - \theta_s}{\theta_e - \theta_s} = \frac{1}{\frac{q \cdot \rho \cdot c_p}{A} \left[\frac{1}{\alpha_r} + \frac{1}{\alpha_{cf}} \right] + 1} \qquad (3)$$

Where
α_r = Radiative heat transfer coefficient (W/m^2 K)
α_{cf} = Convection heat transfer coefficient at the floor (W/m^2 K)
A = Floor area (m^2)
θ_e = Exhaust air temperature (°C)
θ_f = Floor temperature (°C)
θ_{af} = Air temperature at the floor (°C)
θ_s = Supply air temperature (°C)
q = Air flow rate (m^3/s)
ρ = Air density (kg/m^3)
c_p = Specific heat of air (J/kg K)

This quotient as a function of the air flow per floorarea is shown in figure 5 with $\alpha_r = 5$ W/m^2 K, and varying convection heat transfer coefficients at the floor. In fig 5 is also shown the same relation from different literature references. These references include office rooms (H<3 m) and industrial buildings with higher roomheights.

As can be seen from fig 5 the relative raise of the air temperature in the floorarea is very depending on the air flow. The equations (1)-(3) are of course very simplified, they should also include the induction to the supply air and the heat transfer at the ceiling. Some other factors influencing this relation are direct radiant heating from lamps and sunlight as well as heat storage effects in the building construction.

However this simplified equation seems to fit rather well with the measured values and can be used until a further knowledge about the temperature conditions is achieved.

From this relation an approximation of the median temperature gradient in the room can be obtained, supposing that the gradient is linear in the room. This means that one can approximate the gradients influencing the plumes.

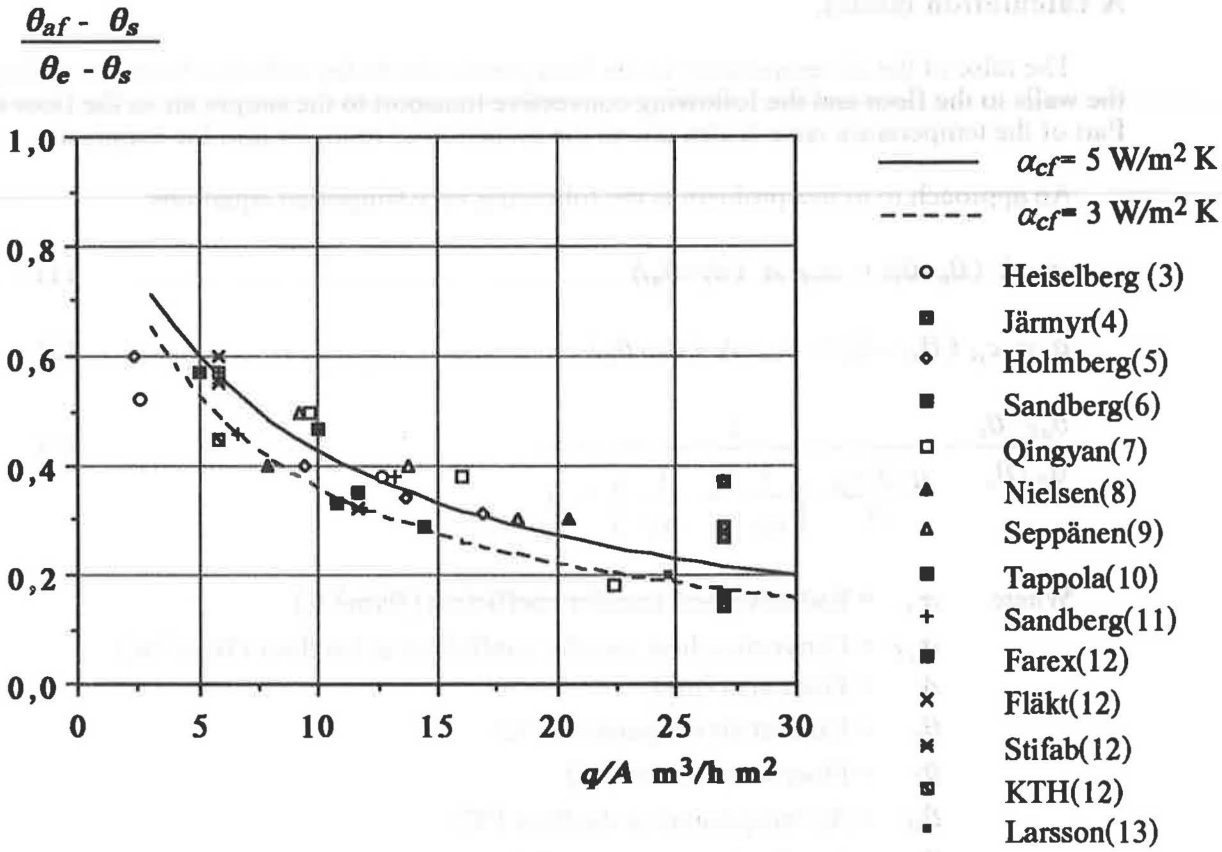


Fig. 5. Nondimensional temperature difference in the floor area for different air flows.

Convection plumes.

Background.

Convection plumes from people have been investigated by Mierzwinski (14) and Danielson (15) among others. Mierzwinskis measurements were made in rooms with a temperature gradient of 0.5 - 0.6 °C/m and he concluded that the airflows from human beings at the height 0.75 m above a standing or sitting person is about 30-60 l/s, and the maximum velocity 0.17- 0.23 m/s.

Danielssons results are around 40 l/s at the height of 1.75 m above the floor when the temperature gradient is 0.5°C/m and 20 l/s at the same height when the gradient is 1.5°C/m. This means a very strong influence of the gradient in the room on the convection flow. The same values are also reported by Fitzner (16).

In our measurements we tried to simulate a person instead of using a real person. The reason for this was that a person is changing the convection heat output depending on the surroundings and as we were doing the same measurements at different places and times, the correlation would be more difficult. The above mentioned cylinder was used for this purpose, the convection heat output from this simulator would be around 50 W when the air temperature and the temperature of the surroudings were in balance, that is when all the heat generated in the room was ventilated away. Of course a transport of heat through the walls cannot be avoided if not the surroundings have the same gradient as the room. There is always some heat coming in at the lower parts and leaving at the upper parts.

Description of the measurements.

The measurements reported here were conducted at different laboratories called lab. 1, 2 and 3 but with the same conditions. They were all made in an office room (L x B x H, 3.6 x 3.6 x 2.4-2.7 m). The rooms were ventilated with 75 or 150 m^3/h. In the first two series (serie 1 and serie 2) of measurements the simulator (100 W) was the only heat source in the room, this means that the temperature difference between supply and exhaust air was 4 or 2°C. According to the simplified calculation method, fig 2, we ought to get gradients around 0.8 and 0.4 °C/m for these two series. This was however not the case, the mean gradients in the room were about 0.8 and 0.6 °C/m, which can be explained with the above mentioned dependance of the air flow per floorarea, fig 5. The gradients in the upper part of the room above the cylinder was the same for the two series, 0.6 °C/m. In the third serie an extra heat source was used to obtain a greater gradient. A radiator (200 W) was placed on one wall 1 m above he floor. The air flow for this serie (serie 3) was 150 m^3/h, and the temperature difference between supply and exhaust air was 6 °C.The gradient above the cylinder was 1.5 °C/m.

The velocities in the plumes were measured with anemometers, at station 1 with Lambrecht L642, at station 2 with Dantec 54 R10 and at station 3 with TNO.The measuring time was in series 1 and 2, 60 seconds and in series 3, 200 seconds. The temperatures in the plumes were measured with thermocouples at station 1 and 3 and at station 2 with a termistor included in the anemometer.

The measurements were done at three different levels, 0.4, 0.8 and 1.2 m above the cylinder in two directions perpendicular to each other. The surface temperatures of the surrounding walls as well as of the heat source were measured.

Results

The flow in the plume was calculated in the same way as by Popiolek (17). The velocity and temperature profiles are assumed to be Gaussian. They can then be described and the volume flow calculated by the following equations.

$$w = w_{max} \cdot e^{-\left(\frac{r}{R_w}\right)^2} \tag{4}$$

$$\Delta\theta = \Delta\theta_{max} \cdot e^{-\left(\frac{r}{R_\theta}\right)^2} \tag{5}$$

$$q_g = 10\,\pi \cdot w_{max} \cdot R_w{}^2 \tag{6}$$

where w_{max} = maximum velocity in the plume (cm/s)
$\Delta\theta_{max}$ = maximum temperature difference between the plume and the surroundings at the same level (°C)
R_w and R_θ = the radii where the velocity and temperature have decreased to e^{-1} of the maximum value (m)
q_g = volume flow in the plume (l/s)

The measured velocities were fitted to the Gaussian curves so that w_{max} and $R_w$were found and the volume flow calculated.

The volume flows in the convection plumes at different laboratories and different cases are presented in table 1.

Table 1. Air volume flows q_g (l/s) in convection plumes at different heights measured at different laboratories and different cases (series).

	Flow (l/s)	Lab 1	Lab 2	Lab 3
Serie 1 Ventilation flow rate 75 m³/h (20.8 l/s) $\theta_e - \theta_s$ = 4.0 °C gradient above cylinder 0.6 °C/m				
H = 0.4	q_g	32	40	30
H = 0.8	q_g	35	52	40
H = 1.2	q_g	46	66	48
Serie 2 Ventilation flow rate 150 m³/h (41.6 l/s) $\theta_e - \theta_s$ = 2.0 °C gradient above cylinder 0.6 °C/m				
H = 0.4	q_g	35	49	40
H = 0.8	q_g	43	62	-
H = 1.2	q_g	60	76	68
Serie 3 Ventilation flow rate 150 m³/h (41.6 l/s) $\theta_e - \theta_s$ = 6.0 °C gradient above cylinder 1.5 °C/m				
H = 0.4	q_g	-	37	37
H = 0.8	q_g	-	43	46
H = 1.2	q_g	-	34	37

From Table 1 one can get some interesting information. First, the flows are very much increasing from serie 1 to serie 2, although the gradients in the upper area are the same. Fig 6 shows the temperature gradients in the room for series 1 and 2 at lab 3. The gradients calulated from fig. 5 or with equation (3) are also shown in this figure. These calculated mean gradients fit well with the measured mean gradients.

Second, the deviation between the different laboratories are rather big. All the conditions were supposed to be the same and the anemometers calibrated.

Third, when the gradient is of the size common in office rooms, the flows are desintegrating before they reach the ceiling.

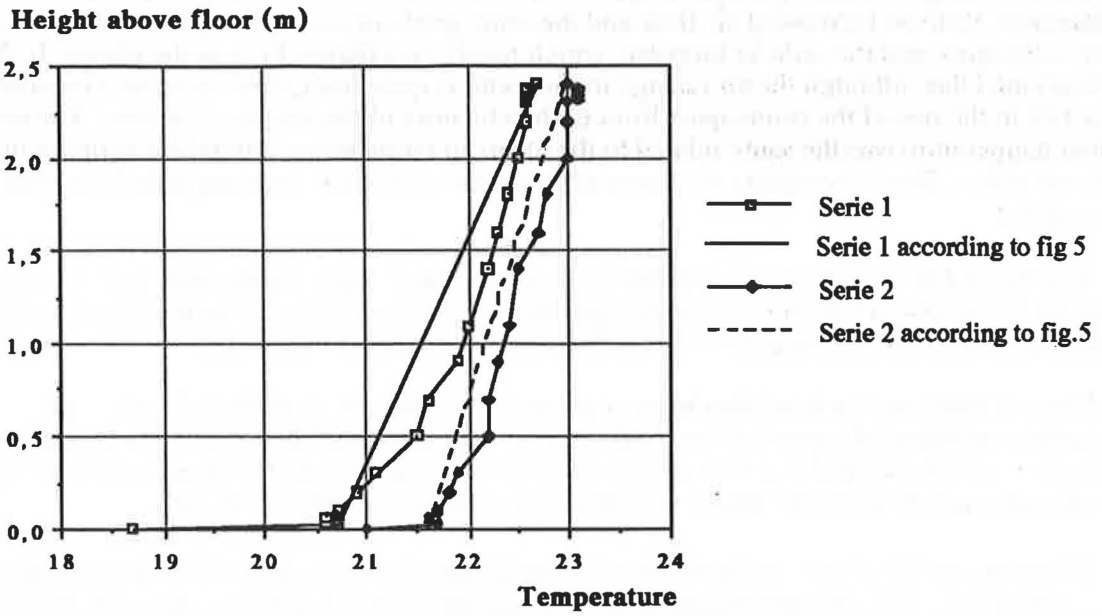


Fig 6. Temperature gradients for series 1 and 2.

In table 2 are shown the calculated maximum velocities and radies of the plumes

Table 2. Calculated maximum velocities w_{max} (cm/s) and radii R_w(m)

		Lab 1	Lab 2	Lab 3
Serie 1 Ventilation flow rate 75 m^3/h (20.8 l/s) $\theta_e - \theta_s$ = 4.0 °C gradient above cylinder 0.6 °C/m				
H = 0.4	w_{max}/R_w	19.5/0,23	21.8/0.24	24.4/0.20
H = 0.8	w_{max}/R_w	20.7/0.23	23.8/0.26	23.4/0.23
H = 1.2	w_{max}/R_w	19.4/0.27	23.5/0.30	18.9/0.29
Serie 2 Ventilation flow rate 150 m^3/h (41.6 l/s) $\theta_e - \theta_s$ = 2.0 °C gradient above cylinder 0.6 °C/m				
H = 0.4	w_{max}/R_w	16.4/0.26	22.2/0.27	18.6/0.26
H = 0.8	w_{max}/R_w	19.2/0.26	22.8/0.30	-
H = 1.2	w_{max}/R_w	16.6/0.34	21.1/0.34	17.9/0.35
Serie 3 Ventilation flow rate 150 m^3/h (41.6 l/s) $\theta_e - \theta_s$ = 6.0 °C gradient above cylinder 1.5 °C/m				
H = 0.4	w_{max}/R_w	-	19.4/0.25	21.6/0.23
H = 0.8	w_{max}/R_w	-	18.8/0.27	19.5/0.28
H = 1.2	w_{max}/R_w	-	11.5/0.31	7.1/0.40

Table 2 can give some clearence to the observations from table 1, but unfortunatly no explanation. With an increasing air flow and the same gradient above the cylinder the velocities seem to decrease and the radii to increase, which results in a bigger flow in the plume. It should be mentioned that although the air change in the room is quite high, there were no measurable velocities in the rest of the room apart from in the vincinity of the supply air device. The mean radiant temperature was the same related to the mean air temperature around the cylinder in the different series. These increasing air flows with higher ventilation rates ought to be further investigated.

The flows and the maximum velocities for our simulator agrees quite well with the values given by Mierzwinski (14) for human beings. However the influence of an increased gradient do not agree with the results presented by Danielsson (15) and Fitzner (16).

Some measurements have also been made on other subjects. A model of a personal computer consisting of a painted box with the side 0.3 m. Half of the top and the bottom were perforated to 50% and inside a heat source of 75 W was placed in a blackpainted smaller tight box. Measurements were also made on a ordinary desk lamp with a 60 W bulb.

These two subjects were only tested with the higher air flow 150 m^3/h with and without the extra heat source. The PC was placed on a table and the lamp 0.3 m above the table, which was placed near the wall opposite to the inlet device. The tests were made with one object at the time. The results are presented in table 3.

Table 3. Air volume flows q_g (l/s), maximum velocities w_{max} (cm/s) and radii R_w(m) of the convection plumes above a model of a personal computor (PC) and a lamp at different heights above the objects.

Serie A	Ventilation flow rate 150 m^3/h (41.6 l/s) gradient above object 0.6 °C/m	PC	Lamp
H = 0.4	q_g	15.5	4.7
	w_{max}/R_w	44.1/0.11	31.1/0.07
H = 0.8	q_g	29.3	11.4
	w_{max}/R_w	37.9/0.16	32.9/0.11
H = 1.2	q_g	53.8	12.2
	w_{max}/R_w	35.8/0.22	29.4/0.12

Serie A	Ventilation flow rate 150 m^3/h (41.6 l/s) gradient above object 1.5 °C/m	PC	Lamp
H = 0.4	q_g	19.7	4.6
	w_{max}/R_w	36.1/0.13	34.5/0.07
H = 0.8	q_g	30.8	12.8
	w_{max}/R_w	39.6/0.16	27.1/0.12
H = 1.2	q_g	40.3	11.8
	w_{max}/R_w	31.75/0.20	23.6/0.13

From table 3 can be seen that the personal computor with an effect of the same size as the lamp gives a much greater air volume flow than the lamp. One reason for this is of course that the PC has a greater surface than the lamp, so the virtual origin is at a lower level. The air volume flow for the PC is almost equal to the the air volume flow for the cylinder. The velocities are however much higher for the PC and the plumes are narrower. The influence of the gradient is smaller both for the PC and the lamp than for the cylinder. One can also see that the plumes are not fully developed for any of the test subjects, the velocities are increasing up to a quite high level.

Conclusion

The mean temperature gradient in a room with displacement ventilation can be calculated with a simple formula when the ventilation air flow per m^2 floorarea is known.

The convection flows from different heat sources were measured at different temperature gradients in an office room. The measurements above a simulator of a person gave results in agreement with earlier presented measurements above persons for gradients around 0.6 °C/m. The influence of a temperature gradient around 1.5 °C/m was less, at lower levels in the room, in our measurements than in those earlier presented. At higher levels the convection flow disintegrated.

Different ventilation flow rates gave the same temperature gradient above the object but different convection flows. The deviation in the results between different laboratories was quite large.

The influence of the temperature gradient on the convection flows was less for heat sources giving higher initial velocities.

In this paper are presented some of the work done in this research project. Further measurements will be carried out to try to clarify the influence of different parameters on the convection flows.

Acknowledgement

The Swedish Council for Building Research supported this work

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SUMMARY

The convection flows from different heat sources in a room are the driving forces for the displacement ventilation. The volumes of these flows have to be known in order to see if the ventilation will behave as expected. Convection flows from different heat sources in a surrounding with an even temperature has earlier been investigated. However the convection flows in surroundings with temperature gradients are not as well known. In rooms with displacement ventilation there is always a temperature gradient. The temperature gradients are dependent on the geometry of the room, the internal loads and the ventilation flow. This paper presents a method to calculate the gradients in the room, which have been experimentally proved. It also presents measurements of the air volume flows in plumes above common heat sources in rooms with different temperature gradients.