BEHAVIOUR OF CONVECTIVE PLUMES WITH ACTIVE DISPLACEMENT AIR FLOW PATTERNS

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ABSTRACT

This study is a part of a research project named 'Convective Flows and Vertical Temperature Gradient within Active Displacement Air Distribution'. The project was started in 1996 in order to determine guidelines for air flow rate dimensioning of the system.

Aim of this study was to determine the characteristics of the thermal plumes of the heat sources used in the project. The characteristics were determined in order to apply the results to different kinds of plumes, which may occur with active displacement air distribution system. The focus was to determine the conditions where the plumes are able to penetrate the active displacement air flow pattern.

Two other papers named *Convective Flows and Vertical Temperature Gradient with the Active Displacement Air Distribution* and *Test Room and Measurement System for Active Displacement Air Distribution* regarding this study are also submitted to be presented at this conference.

KEYWORDS

Plumes, Full-scale experiments, Convective flows, Convective heat transfer, Air flow patterns

INTRODUCTION

This study is part of a research project named 'Convective Flows and Vertical Temperature Gradient within Active Displacement Air Distribution'. The project was started in 1996 in order to determine guidelines for air flow rate dimensioning of the system.

The aim of this study was to find the characteristics of the thermal plumes of the heat sources used in the project. This was studied by carrying out experiments both with and without active displacement air flow pattern. The characteristics were determined in order to apply the results to different kind of plumes, which may be occur with active displacement air distribution system. The focus was to determine the conditions where the plumes are able to penetrate the active displacement air flow pattern.

First the plumes were studied without supply air flow by varying the power and type of the heat source. The results were compared with thermal convection equations.

Then the plumes were measured with active displacement air distribution system in order to determine the conditions where the convective plumes can penetrate through the supply air flow patterns. The measurements were carried out by varying the power of the heat source and subtemperature of supply air using constant supply air flow rate. Also the effect of two different nozzle sectors was examined. A description of active displacement air distribution system has been presen.ed by Sandberg (1998).

METHODS

The behavior of the plumes was examined by measuring plume air flow rates and the heat flow rates at several heights above the heater. Flow rates through the zone boundary at the height of 4,0 m were the most important issues to examine.

Measurement methods and test arrangement

Measurements were carried out in a test room (3x10x6 m^3) by using a measurement robot to move the sensors. Velocity vectors were measured with Kaijo Denki ultrasonic anemometers and air temperatures with Craftemp thermistors. Velocity measurements were mainly done in the middle section of the test room. Additional measurements were also done with Dantec omnidirectional hot sphere anemometers in other sections. Measurement time was 120 s in each point.

Accuracy of Craftemp thermistors was examined before test. Probes were selected so that in normal room temperature the deviation

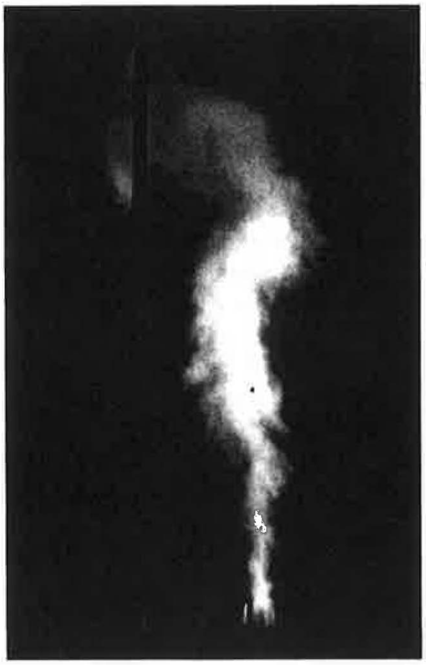


Figure 1. Smoke test with convective heat source (900 W)

of the measured temperatures was within 0,1 °C. Accuracy of the ultrasonic anemometer was ± 0,02 m/s.

Properties of the test room and tracer gas measurements have been described by Koskela (1998).

Velocity and temperature profiles of the convection plume were measured in two perpendicular planes above each heater. In conditions where the supply air flow patterns had no influence on the plumes (case 1) both measurements were carried out in the middle section of the room and the heaters were rotated 90° between the measurements. When supply air flow disturbed the plume (cases 2 and 3) both planes were measured simultaneously. Measurements on the perpendicular plane were in those cases done with hot sphere anemometers.

All the experiments were visualized by using smoke and a digital video camera.

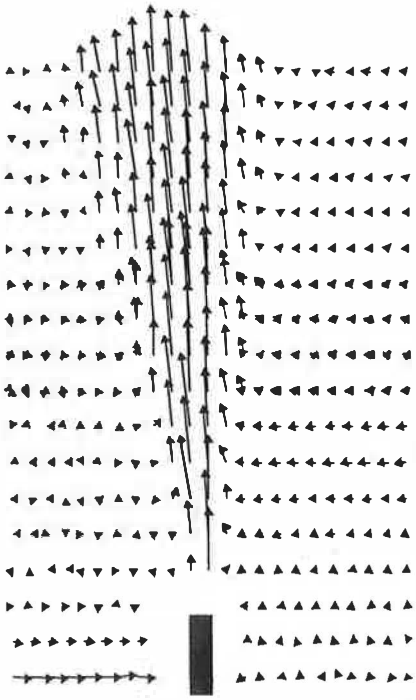


Figure 2. An example of the measured velocity vectors in the plume with the convector heat source (900 W)

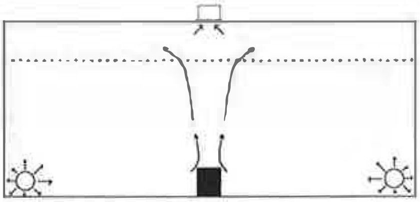
The heaters were mainly placed on the floor in the middle of the room. Radiators were also placed at the height of 1,00 m on top of a box.

Smoke test in Figure 1 and velocity vectors in Figure 2 represent the same measurement situation (Convective heater, 900 W).

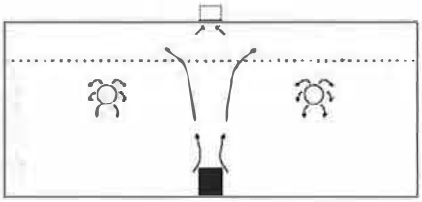
Velocities and air temperatures were examined above the heaters at several heights with 10 cm distance between measurement points

Heaters were tested in three different basic test arrangements (Figure 3):

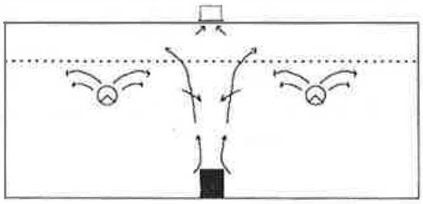
1. Undisturbed case: supply air was distributed to the lower part of the room far from the heater to minimize the effect of the supply air on the plume.
2. Supply air was distributed at the height of 3,0 m with 360° nozzle sector and low supply air flow rate (40 l/sm)
3. Supply air was distributed at the height of 3,0 m using 240° nozzle sector (27 l/sm)



Case 1. Supply on the floor



Case 2. Nozzle sector 360



Case 3. Nozzle sector 240

Figure 3. Description of the test cases

In each case exhaust was placed in the middle of the ceiling.

Three different kinds of heat sources were studied. The heating element in all three heaters was a 1150 x 560 mm² heating foil used in common electric convectors. The power of the heater was adjusted with a transformer.

1. **Cylinder** heat source was made of a ventilation duct (ø=315 mm). A heating element was mounted along a smaller duct (ø=200 mm) inside the source (figure 4). The cylinder was painted matt black.
2. **Convector** was made of stainless steel. Heating element was hanging on a steel rod inside the convector.
3. **Radiator** was made of a heating element placed under a matt black painted steel sheet. Heating element was insulated with a 30 mm mineral wool plate on the floor.

The structure of the heaters is shown in Figure 4.

Calculation of the plume air flow and heat flow rate

The calculation was done based on the hypothesis that Gaussian distributions can be used to approximate velocity distributions in both measured sections (Mundt 1996, Welling 1993). A least-square method was used to determine the plume widths (R_v), symmetry axis (x_0 and y_0) and centerline velocities (v_c). The velocity is given by

$$v(x,z) = v_{cx}(z)\,e^{-(x-x_0)^2/R_{vx}^2} \qquad (1)$$

Both distributions were fixed to real symmetry axes of the plume using formula

$$x' = x - x_0 \text{ and } y' = y - y_0 \qquad (2)$$

Relatively far form heaters the mean velocity distribution can be approximated with a single Gaussian distribution (r = distance from plume centre)

$$v(r,z) = v_c(z)\,e^{-r^2/R_v^2} \qquad (3)$$

Plume air flow rates were calculated using formula 4.

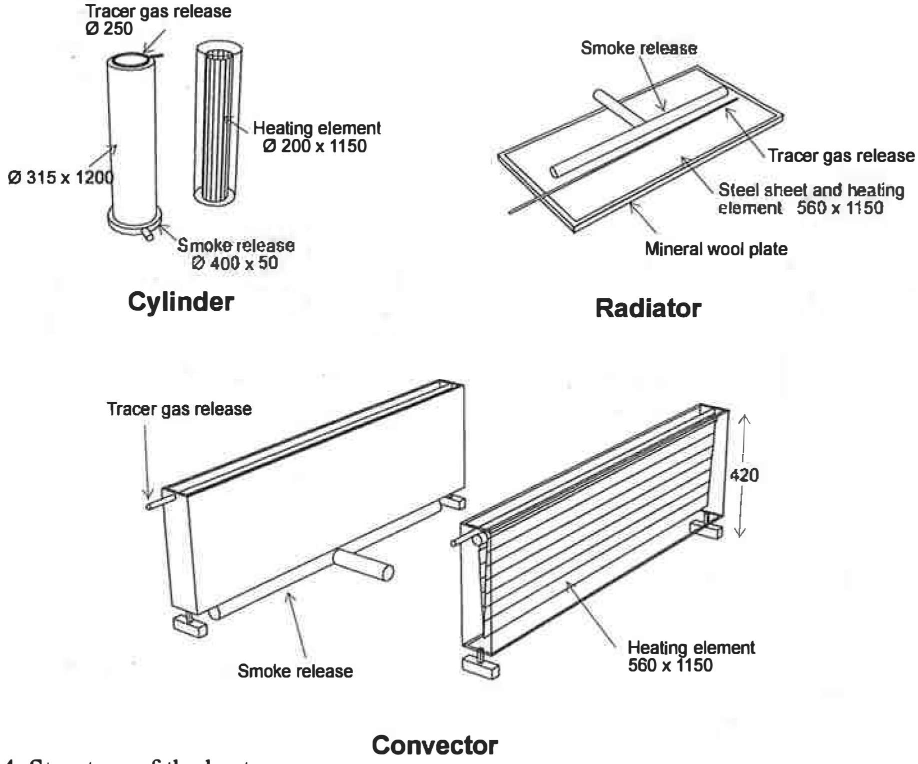


Figure 4. Structure of the heat sources

$$q(z)=2\pi\int_0^\infty v(z)r\,dr=\pi\,v_c(z)\,R_v^2 \qquad (4)$$

Measured temperature difference distributions in the plumes can be approximated similarly by Gaussian curves

$$\Delta T(r,z)=\Delta T_c(z)\,e^{-r^2/R_T^2} \qquad (5)$$

Enthalpy flux of the plume can be calculated by integrating velocity and temperature difference profiles

$$H'(z)=2\pi\rho c_p\int_0^\infty v(r,z)\,e^{-r^2/R_v^2}\,\Delta T(r,z)\,e^{-r^2/R_T^2}\,dr=$$

$$\pi\,\rho c_p v_c(z)\Delta T_c(z)\frac{R(z)_v^2\,R(z)_T^2}{R(z)_v^2+R(z)_T^2} \qquad (6)$$

Heat flow rate of the plume can then be estimated using the temperature of the lower zone T_{lz} as a reference temperature according to the formula 7.

$$Q(z)=\left(\frac{H'(z)}{q(z)\rho c_p}+T(z,r\to\infty)-T_{lz}\right)q(z)\rho c_p \qquad (7)$$

Plume air flow rates were also integrated numerally. Maximum values of the profiles were mounted into the middle of the plume. The air flow rate on a circular zone was calculated us a product of mean value of vertical velocity components and the area. The total air flow rate can be calculated from the following formula

$$q(z)=\sum_{i=1}^{n}q_i=\pi\sum_i\left(r_{i,\max}^2-r_{i,\min}^2\right)\overline{v_i} \qquad (8)$$

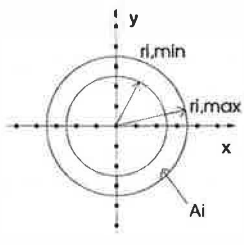


Figure 5. Principle of numerical integration of the plume flow rate

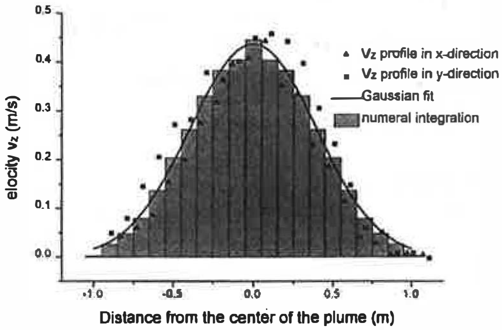


Figure 6 Measured velocity profiles with Gaussian fitting and numeral integration

An example of Gaussian and numeral fittings is presented in Figure 6.

Enthalpy flux was calculated from formula 9 and heat flow rate of the plume from formula 10.

$$H(z) = \rho c_p \sum_i \overline{v_i}\,\overline{T_i} \qquad (9)$$

$$Q(z) = \left(\frac{H(z)}{q(z)\rho c_p} - T_{lz}\right) q(z) \rho c_p \qquad (10)$$

Model for plume air flow rate

Formulas for plume flow rates without temperature stratification are found in the literature for some special heaters (Mundth (1996), VDI 3802 (1997)). Air flow rates q [l/s] are presented as a function of convective power P_c [W] and the distance from virtual origin $z+z_{virt}$ [m].

Point source

$$q = 5.5\, P_c^{1/3} (z + z_{virt})^{5/3} \qquad (11)$$

Line source

$$q = 14\, P_{cl}^{1/3} (z + z_{virt})^{5/3}\, l \qquad (12)$$

Horizontal surface

$$q = 0.5\, P_c^{1/3} (z + W)^{5/3} \qquad (13)$$

In this study the plume volume flow rates were fitted to formula 14.

$$q(z) = A\, P_c^{1/3} (z + z_{virt})^{5/3} \qquad (14)$$

Virtual origins for different heaters were determined from measured plume widths at several heights using least-square method. Coefficiences A were determinated from measured air flow rates and virtual origins with the same method.

Measurements

Measured cases are presented in Table 1. Convective power was calculated from the total power of the heater by subtracting the radiant power.

Table 1 Measured cases

Heater	Power (W)	Convective power (W)
Test arrangement 1		
Cylinder	450	200
Cylinder	900	400
Convector	900	750
Convector	450	375
Radiator	450	150
Radiator	900	300
Radiator *	900	300
Radiator *	450	150
Test arrangement 2		
Cylinder	450	200
Cylinder	900	400
Convector	900	750
Convector	450	375
Radiator (*	900	300
Radiator (*	450	150
Test arrangement 3		
Cylinder	900	400
Cylinder	450	200
Convector	450	375
Convector	900	750
Radiator (*	450	150
Radiator (*	900	300

*) at height of 1,00 m on top of a box

Subtemperature of supply air varied between 0,5 °C and 3,5 °C in different cases.

RESULTS

Measured plume heat and air flow rates at the height of 4 m are presented in Table 2. The values are based on Gaussian integration. Measured air flow rates are also compared with air flow rates calculated from formula 14.

Measured air flow rates in disturbed cases (cases 2 and 3) compared to the undisturbed case 1 are presented in Figure 7 and corresponding heat flow rates in Figure 8.

Table 2 Measured plume heat and air flow rates for undisturbed case

	Convective power (W)	Temperature gradient (°C / m)	Air flow rate (l/s)	Air flow rate vs. formula	Heat flow rate (W)	% of convective power
Cylinder 450 W	200	0,1	289	98	106	53
Cylinder 900 W	400	0,2	370	100	363	91
Convector 450 W	375	0,2	315	83	329	88
Convector 900 W	750	0,4	431	91	707	94
Radiator 450 W	150	0,2	329	78	157	105
Radiator 450 W, 1.0m	150	0,2	250	111	174	116
Radiator 900 W	300	0,3	552	104	155	52
Radiator 900 W, 1.0m	300	0,3	274	96	331	110

The results of numerical integration differed from Gaussian integration less than 10 %. For radiator, the difference was in some of the cases higher.

Calculated experimental coefficient A and virtual orgin z_{virt} for the heaters are presented in Table 3.

Table 3 Calculated experimental coefficiences A and virtual origins z_{virt}

Heater	A	z_{virt} (m)
Cylinder	5,2	1,2
Radiator	5,6	1,0
Radiator 1,0 m	4,4	1,0
Convector	1,5	4,5

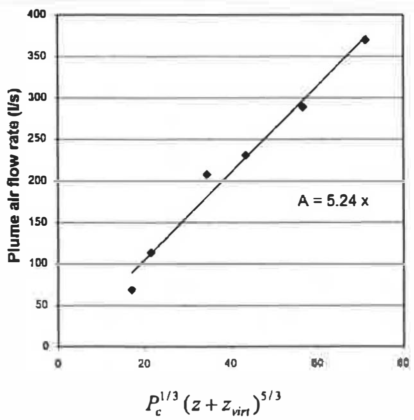


Figure 6. Determination of experimental coefficient A for the cylinder heat source

Determination of experimental coefficient A for cylinder is presented in Figure 6.

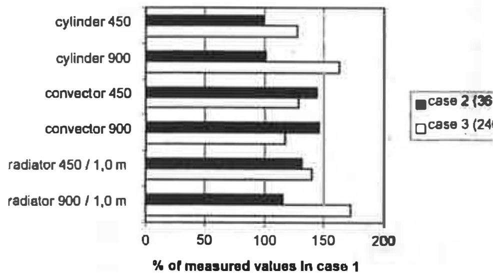


Figure 7. Measured air flow rates in cases 2 and 3 compared to case 1

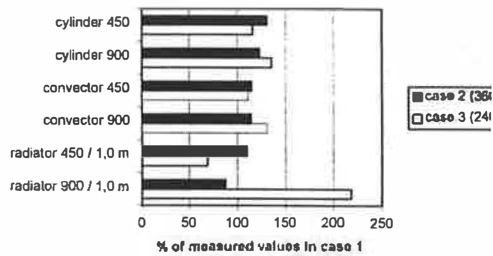


Figure 8. Measured heat flow rates in cases 3 and 3 compared to case 1

DISCUSSION

Undisturbed plumes

The measured air flow rates of the undisturbed plumes followed the fitted plume equations in most cases within 10 %. In two cases the difference was 20 %. The measured heat flow rates were in most cases within 15 % of the calculated convective power of the heaters. In two cases, however, the measured heat flow rate was only half of the convective power. The reason for this was not found. It can be due to the error in measurement of small temperature differences between the plume and ambient air.

Disturbed plumes

The measured air flow rates of the plumes were higher with supply air compared to the undisturbed cases. In case 2 the difference was 45 % for convector and 15 - 30 % for radiator, but for cylinder there was no increase. In case 3, where the flow pattern of supply air was stronger, the increase in plume flow rate was even higher, 20 - 70 %. This can be due to the increase in temperature difference caused by the introduction of subtempered supply air.

The measured heat flow rates were also in most of the cases 20 - 40 % higher in the cases with supply air compared to the undisturbed cases. The values were thus higher than the calculated convective power of the heaters. This can be explained by the difficulties in measurement of the reference temperature in situations with temperature gradients in the room.

For all the cases measured so far, the plumes penetrated the supply air flow pattern. The test program will be continued with cases of stronger supply air influence to the plume in order to find situations, where the penetration is decreased.

ACKNOWLEDGEMENTS

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REFERENCES

Koskela, H., Hautalampi, T., Sandberg, E. and Lindgren, M. (1998), Test Room and Measurement System for Active Displacement Air Distribution, submitted to Proceedings of Roomvent '98. Stockholm, Sweden

Mundt, E. (1996) The performance of displacement ventilation systems, Experimental and theoretical studies, Royal Institute of Technology, Stockholm

Sandberg, E., Koskela, H. and Hautalampi, T. (1998) Convective Flows and Vertical Temperature Gradient with the Active Displacement Air Distribution, submitted to Proceedings of Roomvent '98. Stockholm, Sweden

VDI 3802, Raumlufttechnische Anlagen für Fertigungsstätten (Air conditionin systems for factories) (1997), VDI, Düsseldorf

Welling, I. (1993) An investigation of the SVD-method for analyzing convection plumes, Lappeenranta Regional Institute of Occupational Health, Lappeenranta