INDUCING VENTS AND THEIR EFFECT ON AIR FLOW PATTERNS, THERMAL COMFORT AND AIR QUALITY

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ABSTRACT

It is possible to make high wall inducing vents with low air resistance, combined with natural ventilation or a mechanical exhaust. By means of these vents draughts may be prevented and high efficiencies in fresh air and contaminant removal may be realised, the latter being determined by the position of the outlet. Existing equations related to air flow patterns and Computational Fluid Dynamics (CFD) computer programs can be used, provided that the equations and the CFD program (Phoenics) are modified in order to get better agreement with measurements. On the basis of this research practical solutions may be found for a better indoor climate.

INDUCING VENTILATION

In The Netherlands the commonest way to control indoor climate is natural ventilation, usually combined with a mechanical exhaust. In order to save energy, much attention is currently being paid to the control of the mass flow by developing self-regulating vents. In addition to this aspect, draught and air quality also require more attention. Mass flows over 8 l/s (0 ^oC) may easily cause draughts (Heikkinen et al., 1993). The characteristics of ventilation systems should be:
1. economy
2. control of the mass flow of outdoor air
3. good thermal comfort
4. good air quality
5. noise insulation (traffic)
6. architectural integration.

In order to acquire more knowledge on better ventilation, we compared various inducing vents (installed just beneath the ceiling), two of which are discussed in this paper (fig. 1):

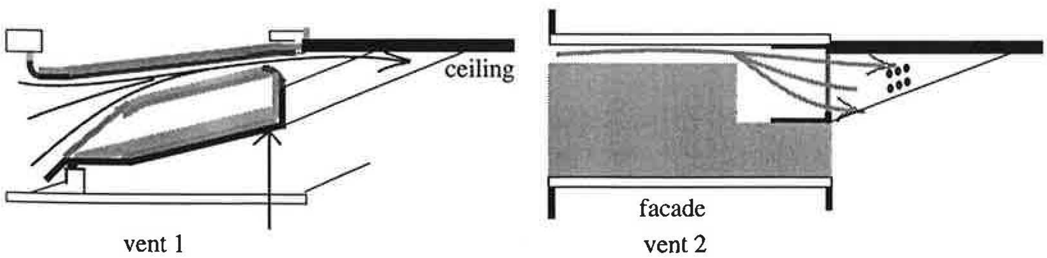

Fig. 1. Vent 1 is a specially designed inducing vent (10 x 1000 mm, or with a different height of the inlet). Vent 2 is a slot (15 x 690 mm), combined with a grate (area 48 x 690 mm, 3 mm holes).

Measuring methods

A flow finder was used to measure the mass flow of the air, which can be derived from the position of a a valve in the flow finder through which the measured air flows; the differences in pressure before and after the valve should be zero. Deviations in the resistance of the system are corrected by a fan, thus keeping inaccuracies in the measurement under 2 %. When the characteristic pressure difference (MKM, pressure difference gauge, 0 - 20 pascal) is known, the mass flow through a vent in the facade of a test chamber (l x w x h = 3.9 x 2.5 x 2.5 m) can be derived. Temperature differences have only minor effects on the mass flow. Air velocities were measured at 0 - 0.2 m beneath the ceiling (Schmidt SS 20.01 anemometer). Air velocity and turbulence intensity were measured at 1 m from the facade and at a height of 1.1 m above the floor (Thies 4.3027.10.000 anemometer). Average and standard deviation values of the measurements can be calculated with the LabView data acquisition program. The average room and outdoor temperature were 20 °C and 0 °C respectively. The Predicted Percentage Dissatisfied (PPD) due to draught can be calculated with these measurement values (Fanger, et al., 1988).

Statistical methods

With the following equations (Schramek, et al., 1993) it is possible to estimate draught problems, air velocity (v_x) and the mixed air temperature (t_x) at some distance (x_a) from the air inlet :

$$x_o = 2\, h_{eff}/m \quad (1)$$

$$v_x/v_o = \sqrt{(2h_{eff}/mx_a)} \quad (2)$$

$$\Delta t_x = \sqrt{(3x_o/4x)}\, .\, (t_i - t_x) \quad (3)$$

$$Ar = g\, h_{eff}\, (t_a - t_i) / (v_o^2\, T_i) \quad (4)$$

$$y = 0.4\, m\, Ar\, (x_a/h_{eff})^{2.5}\, h_{eff} \quad (5)$$

Ar = Archimedes number, g = gravitational acceleration (m/s^2), h_{eff} = real or corrected height of the air inlet (m), m = turbulence coefficient, t_a = average outdoor temperature (°C), average room temperature = t_i (°C) and T_i (K), v_o = average air velocity in the air inlet, x_o = initial length of the flow, before it has fully developed, y = distance of the jet centreline from the ceiling.
If the air flow passes along a horizontal plane, such as a ceiling, the actual distance from the ceiling will be smaller and can be calculated as:

$$y_{eff} = y/\sqrt{2} \quad (6)$$

The measurements have been compared both with these calculations and with computer simulations.

RESULTS

Measurements

Draught problems can be reduced or solved by inducing vents (table 1). The thermal comfort of vent 1 is slightly better than that of vent 2; vent 1 has a larger throw. In order to prevent draught problems the Archimedes number should be 0.0007 (30 l/s), and lower as the mass flow increases. Vent 1 (without radiator heating) produces a throw of more than 1 m at 30 l/s, and of more than 2 m at 45 l/s. A large throw at 20 l/s and at 10 mm height of the inlet, can be obtained with a length of vent 1 of 0.4 m instead of 1 m. The form of the vent and the size of the inlet determine the inducing qualities, such as the air velocities produced: with inlet sizes of 5 x 1000 or 7.5 x 1000 mm (20 l/s), U_{1m} will be 0.87 and 0.55 m/s respectively. Moreover, the cold air of vent 1 mixes very quickly - usually within a distance of 0.25 - 0.5 m - with the ambient room temperature, and at 1.1 m above the floor the air velocities are under

0.05 m/s. This is the result of considerable differences in the air velocities in the air stream and of a thin air stream in the air inlet, which bring about high turbulence in the air flow above the breathing zone. These measurements show that the temperature profile near the inlet follows that of the air velocity.

Table 1. Comparison of vent 1, h = 10 mm and vent 2, h = 15 mm (without heating)

	Mass flow (l/s)	ΔP (pascal)	PPD (%)	U_{1m} (m/s)	U_{2m} (m/s)	Ar
Vent 1	10	1.17	0	0.05		0.0069
Vent 2	10	1.26	0	0		0.0226
Vent 1	20	3.76	0	0.19		0.0017
Vent 2	20	3.79	0	0.09		0.0057
Vent 1	30	5.66	0	0.86		0.0007
Vent 2	30	7.51	10	0.22		0.0025
Vent 1	40	10			0.34	0.0004
Vent 2	40	12.17	8	0.3		0.0014
Vent 1	45	12.84			0.6	0.0003
Vent 2	45	15.26	13	0.27		0.0011

Radiator heating (40 - 70 °C) below vent 2 will increase its U_{1m} values (0.19 - 0.38 m/s), its PPD values remaining 0 %. Floor heating, however, will have an opposite effect.

Calculations with equations 1 - 6

These equations partly explain the favourable results of the measurements of the two inducing vents (PPD < 13 %, table 1). With increasing air velocity, the Archimedes number (with $1/v_o^2$) and y are smaller. If mass flows are lower than 45 l/s, these equations should be used with great care. Usually m = 0.15 - 0.25, but its value may be much larger (ca. 0.35 - 3) if the mass flow and h_{eff} are small: the height of the air stream may be far less than the geometric size of the inlet. Equations as well as the CFD program will generally estimate air velocities (v_x) too high (fig. 2) and mixed temperatures (t_x) too low.

Computer simulations

To calculate a jet flow passing along a wall, the CFD program (Phoenics) requires the use of the Prandtl One Equation Model (Nieuwstadt, 1989), combined with the Simple Low Reynolds Model for calculation of the effects of the heat sources. The value of the coefficients in these models should be low, about 0.05. Phoenics solves turbulence model flow equations in finite-volume method (Launder et al., 1974). The computational grid near the air inlet is very small, about 1 mm x 1 mm, and its first grid lines near the wall 0.5 mm. Because of the contraction of the air flow the size of the inlet is small: h_{eff} = 0.5 - 1 mm. Computer simulations of the air flow from vent 1 have been carried out, either with a radiator beneath the windows or with floor heating and the presence of a person at a distance of 1.5 m from the façade (defined as a "plate" 50 x 100 cm, 34 °C, with an "inlet", 1 olf = 5 mg/s contamination). The balance of the mass flow and of the heat flow should be approximately zero. The simulated mixed temperatures (10 °C at a distance of 0.3 m from the inlet) and air velocities (3 m/s at 0.15 m from the inlet) are comparable with the measurements (fig. 2). The simulated air velocities (with radiator heating)

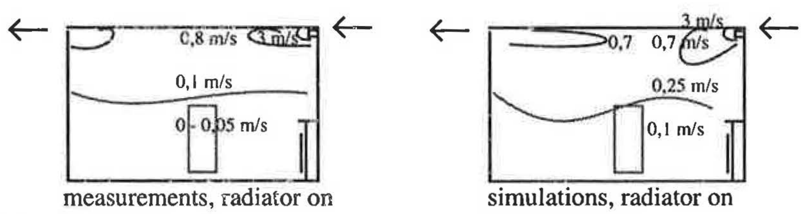

Fig. 2. Comparison of measurements and computer simulations of air velocities with vent 1 (20 l/s)

at a distance of 1 m from the inlet are 0.5 - 0.9 m/s, and 0.1 - 0.2 m/s in the breathing zone. These values are also comparable with the measurements, but the air velocities in the breathing zone are lower and there is more contraction of the air flow (fig.2). With different positions of the outlet (fig. 3) the simulations show considerable differences in contamination removal efficiency (ε_c):

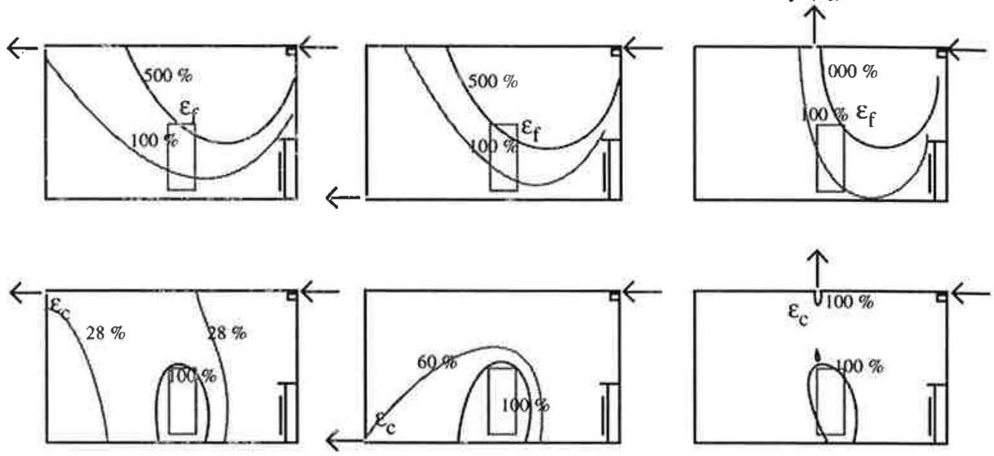

Fig. 3. Fresh air efficiency (ε_f) and contamination removal efficiency (ε_c) of vent 1, 20 l/s, with an outside temperature of 0 °C and a room temperature of 21 °C with radiator heating (300 W) and 3 different outlet positions.

A low outlet gives a high contaminant removal efficiency (ε_c), but a mass flow (Q) > 20 l/s may cause draughts. An outlet placed in the middle of the ceiling results in a high ε_c and a high ε_f, but Q > 20 l/s may cause short circuiting. In these cases self-regulating vents will be required. With floor heating (400 W) the ε_c value will be 7.5 % , 80 % and 100 % respectively.

Fresh air efficiency (ε_f) and contamination removal efficiency (ε_c) (Niu, 1994) have been calculated as follows :

$$\varepsilon_f = \frac{C_x}{C_i}\frac{V}{Q} \times 100\ \% \qquad (7)$$

$$\varepsilon_c = \frac{C_x}{C_i} \times 100\ \% \qquad (8)$$

C_x = concentration of fresh or contaminated air on the spot x, C_i = initial concentration in the mass flow Q (l/s) of fresh or polluted air, V = volume of the room (litres).

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