Air Distribution in Rooms, (ROOMVENT 2000)
Editor: H.B. Awbi

THE NEUTRAL HEIGHT IN A ROOM WITH DISPLACEMENT VENTILATION

H. J. Xing [1] and H.B. Awbi [2]

[1] Whitby Bird & Partners Engineers, 60 Newman Street, London W1P 4DT,UK
[2] Department of Construction Management & Engineering, The University of Reading, Reading, RG6 6AW, UK

ABSTRACT

This paper investigates the relationship between the neutral height for air distribution and the ventilation load in a room with displacement ventilation. An environmental chamber equipped with a displacement ventilation system has been used to carry out the neutral height measurements with the presence of a heated mannequin and other heat sources in the chamber. The total room load used was varied from 104 W to 502 W, i.e., corresponding to a ventilation load from 10 W/m^2 to 60 W/m^2. The prediction of the neutral height was based on plume theory. Comparison between the experimental data and the predictions showed good agreement. The predictions also showed that a human body could draw uncontaminated fresh air from the lower zone that improves the air quality in breathing zone.

KEYWORDS

Displacement ventilation, Neutral Height, Buoyant Flow, Indoor air quality

NOMENCLATURE

q_l	Volume flow rate in the plume (l/s)
P_c	Convective heat release rate from the source (W)
R_s	Heat source radius (m)
s	Temperature gradient in the room (K/m)
z	Height of source (m)
z_1	Height calculated using Eqn. 3 (m)
z_{den}	Plume rising height where the temperature difference is zero (m)
z_{virt}	Height of the virtual source (m)
z_0	Height from virtual origin to the heat source(m)
ΔT	Temperature difference between source and surroundings (K)
δ	Boundary layer thickness (m)

INTRODUCTION

Displacement ventilation (DV) has grown popular in recent years for comfort ventilation in rooms with low thermal loads such as in offices. The main principle of displacement ventilation is based on the air density differences which give the room air a tendency to move up toward the exhaust. Compared to the traditional mixing ventilation, displacement ventilation has the possibilities of creating both high temperature effectiveness and high ventilation effectiveness.

The neutral height in a room with DV is one of the important paramenters to evaluate when investigating the room air quality. The methods used to determine the neutral height both experimentally and theoretically have been described by Stymne et al (1991). Figure 1 summaries typical temperature and pollutant concerntration profiles in a room with displacement ventilation. It is shown that the neutral height corresponds to the height where the mean wall temperature is equal to the mean room air temperature. It is also equal to the mean contaminant concentration in the recirculation zone.

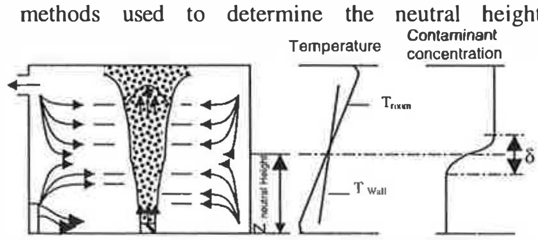

Figure 1: Room air flow pattern and profiles of temperature and concentration

Stymne et al (1991)) also described theoretically the neutral height position where the vertical component of the flow in the ambient is zero, so that the only vertical transport occurs within the plume. This is similar to the decriptions made by Nielsen (1993) and Mierzwinski (1992).

This paper presents the relationship between the neutral height and the room heat load obtained from chamber tests at the University of Reading. The experimental data and results are compared with the theortical predictions based on plume theory.

THEORETICAL PREDICTION OF NEUTRAL HEIGHT IN A ROOM

Theoretically, the neutral height is the height where the air flow balance exists, i.e. the total upward flow rate is equal to the supply flow rate. It is then necessary to know the theoretical equation for the plume flow rate above the heat source. Mundt (1996) proposed models for the calculation of the plume flow rate above heat sources. In her models, the effect of the ambient temperature stratification was taken into account.

For a point source, the calculation is made in four steps:

1. Vertical extended sources: calculate the boundary layer thickness (m)

$$\delta = 0.048 \cdot \left(z/\Delta T\right)^{1/4} \qquad (1)$$

For a horizontal source: the boundary layer thickness is 0.

2. Calculate the location of the virtual source (m)

$$z_{virt} = 4.18 \cdot (R_s + \delta) \qquad (2)$$

3. For different heights z above the source calculate z_l using:

$$z_l = 2.86 \cdot (z + z_{virt}) \cdot s^{3/8} \cdot P_c^{-1/4} \qquad (3)$$

4. For different z_l calculate m_l and the volume flow rate q_l

$$m_l = 0.004 + 0.039 \cdot z_l + 0.380 \cdot z_l^2 - 0.062 \cdot z_l^3 \qquad (4)$$

$$q_l = 2.38 \cdot P_c^{3/4} \cdot s^{-5/8} \cdot m_l \qquad (5)$$

The height where the temperature difference disappears (z_{den}), can be calculated from the Eqn. 6:

$$z_{den} = 0.74 P_c^{1/4} s^{-3/8} \quad (6)$$

where P_c is the convective part of the heat release rate which is equal to the total heat release rate multiplying by a coefficient k. k is 0.7 ~ 0.9 for pipes and channels, 0.4~ 0.6 for smaller components and 0.3 ~0.5 for larger machines and components Nielsen (1993). Mundt (1996) recommended k = 0.8~1.0 for point illumination, k = 0.5 for people and extended surfaces.

For plumes from heat sources located close to a wall or a corner, the plumes may be attached to the wall due to the Conanda-effect (Nielsen (1993)). The entrainment will be lower than the entrainment in a free plume and this will influence the location of the neutral height in the room.

Eqns. 7 and 8 give respectively the flow rates for a single plume against a wall and for N identical plumes located close to each other:

$$q_l = 0.0032 P_c^{1/3} (z + z_0)^{5/3} \quad (7) \qquad q_{lN} = N^{1/3} q_l \quad (8)$$

EXPERIMENTAL SET-UP

The test chamber used for this investigation is a 2.78m x 2.78m x 2.3m (L x W X H) office module with an 0.5m x 0.5m air inlet and an 0.2m x 0.2m outlet, situated in the University of Reading.

Three different types of displacement flow diffusers were used to supply the conditioned air to the chamber. These are a flat diffuser (DV1), a semi-circular diffuser (DV2) and floor swirl diffuser (DV3), see Figure 2. The supply air flow rate was set at 3 levels, to give 3.2 ac/hr, 5 ac/hr and 7 ac/hr.

a) flat wall unit b) semi circular unit c) swirl unit

Figure 2: The diffusers used in the tests

The heat sources include a mannequin of seated or standing postures with heat output of about 100 W, an office light of 36 W heat output, a computer simulator generating about 150 W and heating plates of variable heat outputs giving 95 and180W. Different distributions of these heat sources were used to simulate an office with different heat loads. The total heat load for the experiments were varied from 140 W to 502 W, which represent typical heat loads in an office of this size.

Combining different air supply conditions and heat source distributions, there was a total of 12 experimental conditions for the flat diffuser, 12 conditions for the semi-circular diffuser and 9 conditions for the floor swirl diffuser.

The temperature and velocity were measured at 25 key points in the room for each test condition by using four-wire Platinum Resistance Thermometer (accuracy = ±0.15 K). Contaminant concentration was also measured at 12 key points in the chamber for each condition using Bruel & Kjaer gas sampling system with SF_6 as tracer. The temperature, velocity and gas sampling data were logged onto a computer by software.

Table 1 lists the experimental tests and the output of heat sources for DV1, DV2 and DV3 respectively. Full details of the experimental set-up and the measurement techniques can be found in Hatton, Xing & Awbi (1999)

The total loads given in Table 1 are the sum of ventilation and conduction heat transfer. When calculating the neutral height using the Eqns. in section 2, only the ventilation load is considered.

TABLE 1
EXPERIMENTAL CONDITIONS FOR DV1, DV2 AND DV3

Configuration No.	Mannequin (W)			Computer simulator(W)			Plates (W)			Light (W)	Total Load (W)		
	DV1	DV2	DV3	DV1	DV2	DV3	DV1	DV2	DV3	DV1,2,3	DV1	DV2	DV3
1 (man seated)	104	129	126	0	0	0	0	0	0	36	140	165	162
2(man seated)	114	100	100	150	146	146	0	0	0	36	300	282	282
3(man seated)	111	98	103	150	146	155	95	95	86	36	392	375	380
4(man seated)	116	74	79	150	146	155	190	190	181	36	492	446	451
5(man seated)	116	108	n/a	150	146	n/a	190	190	n/a	36	492	480	n/a
6(man standing)	139	120	126	0	0	0	0	0	0	36	175	156	162
7(man standing)	124	100	112	150	146	160	0	0	0	36	310	282	308
8(man standing)	121	120	106	150	146	150	95	95	95	36	402	397	387
9(man standing)	126	74	90	150	146	156	190	190	188	36	502	446	470
10(man standing)	129	113	n/a	150	146	n/a	190	190	n/a	36	505	485	n/a
11(man standing)	114	99	n/a	150	146	n/a	0	0	n/a	36	300	281	n/a
12(man standing)	124	100	n/a	150	146	n/a	0	0	n/a	36	310	282	n/a

RESULTS

Using the method of Stymne et al (1991) for determing the neutral height in a room, the neutral height for each experimental condition was determined for the flat and semi-circular differs using both wall and room mean temperature data, and also from the distribution of the contaiminant concentration in the chamber.

TABLE 2
PREDICTED NEUTRAL HEIGHT FOR DV1, DV2 AND DV3

Experimental configuration number	Air change rate /hour	Total ventilation heat load (W/m^2)	Neutral height (m)		
			DV1	DV2	DV3
1	5	12.9	1.98	2.1	-
2	5	23.9	1.58	1.57	1.58
3	5	30.8	1.2	1.3	1.3
4	5	42.5	1.25	1.35	1.28
5	7	47.7	1.4	1.4	n/a
6	5	22.64	1.95	1.95	1.55
7	5	33.1	1.58	1.58	1.3
8	5	37.87	1.3	1.3	1.28
9	5	42.6	1.25	1.25	n/a
10	7	49.1	1.4	1.26	n/a
11	3.2	38.8	1.38	1.38	n/a
12	3.2	40.11	1.38	1.38	n/a

Applying the theory given in earlier, the flow rate for each thermal plume was calculated at different heights by assuming that all heat sources, except the heated plates, as point sources. A k value of 0.5 has been taken for each point source. The flow rate in the plume produced by the heated plate is obtained using Eqns. (7) and (8) with a value for k of 0.5. The flow rate from each source at different heights was added together. Then the neutral height for each configuration was defined to be the height at which the sum of the plume flow rates equals to the supply flow rate. The measured room temperature gradient between 0.1m and 1.1m was used for the calculation of plume flow rates. Table 2 gives the predicted neutral heights for DV1, DV2 and DV3.

DISCUSSION

To investigate the relationship between the ventilation load and the neutral height in the room, the experimental neutral height results and those predicted were plotted in Figures 3 and 4. Figure 3 shows the measured neutral height for DV1 and DV2 using the mean wall temperature method, whereas Figure 4 is based on the mean concentration.

In Figures 3 and 4, the solid line represents the predicted neutral height and the dots represent the experimental data. The results show that the neutral height decreases as the ventilation load of the room increases. However, when the room load is greater than about 45 W/m^2, the neutral height increases as the room load increases further. It is clear that the flow rate not only depends on the heat source strength, but also on the room temperature gradient. When the room load becomes high, the room temperature gradient is reduced due to the increase in temperature in the lower part of the room. This means an increase in ventilation rate is needed to maintain acceptable room conditions.

Figures 3 and 4 also show that the predicted data fit reasonably well with the experimental results. The measurements based on mean wall and mean room air temperatures produced better agreement with the prediction. This may be due to the difficulty of accurately measuring gas concentration. It appears that the predicted neutral heights are higher than the experimental data for most of the ventilation loads. The reasons for this are two-fold: one is the confinement of the ceiling on the rising thermal plume. When the thermal plume free rising height is larger than the ceiling height, this may generate a downward flow which displaces the upper layer towards the lower layer in the room. The other cause may be due to the unstable condition of the surroundings and the fluctuating weather conditions. Large differences were found for configurations with low inlet air change rates.

To investigate the air quality in the breathing zone, the neutral height above a seated and a standing mannequin was calculated using Eqn. (6) and compared with that predicted for the room. Figure 5 shows a comparison between the predicted room neutral height and the neutral height above the mannequin. For a seated mannequin, these two heights are similar in value. However, for a standing mannequin, it is shown that the neutral height above the mannequin is higher than that for the room. This is due to the entrainment of fresh air to the mannequin up to the breathing zone which is within the mixed zone of the room. This is consistent with the results obtained by Stymne et al (1991), in which it was shown that the occupants will draw uncontaminated air from the lower zone, and experience better air quality at the breathing level than that of the surrounding air even if the neutral height interface is below the head.

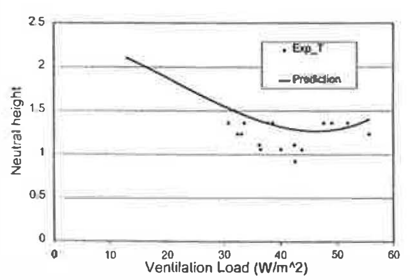

Figure 3: Neutral height in the room versus ventilation load (based on mean temperature)

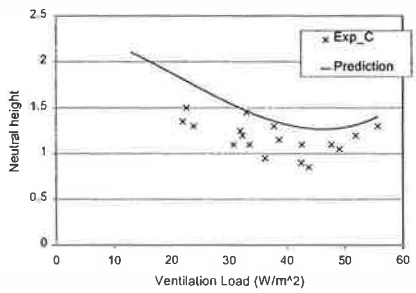

Figure 4: Neutral height in the room versus ventilation load (based on mean concentration)

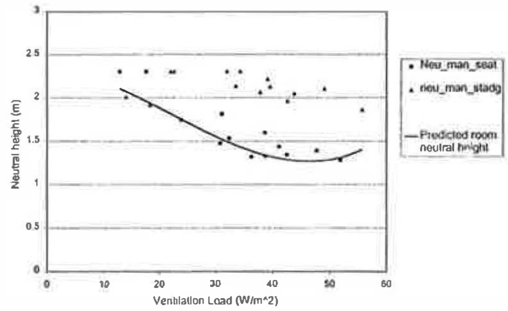

Figure 5: Comparison of the predicted room neutral height and the neutral height above the mannequin

CONCLUSIONS

This paper presented results from prediction and measurement of the neutral height in a room with displacement ventilation. Comparison between measurements based on mean room wall temperature agreed well with predictions. However, the agreement between measurements based on the mean concentration and prediction is less good. It is also shown that the human body can draw uncontaminated fresh air by the body plume that results in a better air quality in breathing zone.

Reference

Hatton, A. Xing, H. J. and Awbi, H. B. A (1999). Study of the Air Quality in Rooms Ventilated with Displacement Systems. *Internal Report*, Department of Construction Management & Engineering, University of Reading.

Mierzwinski, S. (1992). Testing and Modeling of Thermal Plumes in Rooms. *International Symposium on Room Air Convention and Ventilation Effectiveness*, University of Tokyo.

Mundt, E (1996). The Performance of Displacement Ventilation System. *Report for BFR project #920937-0.*

Nielsen, P. V. (1993). *Displacement Ventilation*, Aalborg University.

Nielsen, P. V. (1995). Vertical Temperature Distribution in a Room with Displacement Ventilation. *IEA Annex 26: Energy-Efficient Ventilation of Large Enclosures*, Rome.

Stymne, H. Sandberg, M. and Mattsson, M. (1991). Dispersion Pattern of Contaminants in a Displacement Ventilated Room-Implications for Demand Control. *12th AIVA Conference*, Ottawa, Canada.