and/or in combination, giving priority to the critical enclosure(s), and recalculate the required outdoor air quantity.

d) In variable air volume systems, determine multiple "worst-case" system configurations as described in "Variable air volume (VAV) systems" of this paper, and repeat (a) to (c) above for each configuration.

- Where possible, relief air to exhaust systems should always be drawn from relatively "dirty" enclosures with low Amenity Indices.
- Where possible, relief air transferred from one enclosure to another should always be from the "cleaner" enclosure (with higher Amenity Indices) to the "dirtier" enclosure (with lower Amenity Indices).
- Serious consideration should be given by HVAC designers to provision of supplementary tempered outdoor air systems in large buildings to provide flexibility for future changes of usage in parts of the building. Matching flexibility should also be considered in common exhaust systems.
- The development of approved test methods for determination of efficiencies of odour-removing air cleaners should be a high priority for manufacturers and testing authorities, and would significantly extend the options available for optimising ventilation system design and operation.

Acknowledgements

The author wishes to express his thanks to Vincent Aherne of Standards Australia, and the members of the ME/62 Committee for their assistance in providing information during the development of the draft Standard.

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Ventilation by displacement — its characteristics, design and other related developments

By Y. Li, PhD; A.E. Delsante, B.Sc.(Hons), PhD; and M. Sandberg*

Recent literature, mainly from Scandinavian countries, on displacement ventilation is reviewed in this paper. The paper first discusses the thermal-fluid flow and contaminant removal characteristics in displacement ventilated rooms and the flow behaviour of low-velocity air supply registers. This discussion outlines the main differences between displacement ventilation and mixing ventilation systems, which have implications for various design components in displacement ventilation.

Engineering models for calculating near-field length, airspeed in the farfield, air temperature near the floor and the ceiling, cooling load, vertical air temperature gradient, concentration in the upper pollutant zone, and the clean zone height are summarised. Preliminary work on integrating the four-node model into an Australian building thermal modelling code, CHEETAH, is presented. This work will eventually enable improved calculations of cooling loads due to heat conduction through walls and solar radiation through windows to be done for displacement ventilation systems in realistic buildings.

General guidelines for design displacement ventilation systems using these simple prediction tools are provided. Additionally, other related developments on near-body flows, effects of moving bodies, raised floor systems and chilled ceiling systems are briefly mentioned.

Introduction

Air distribution

The concept of ventilation has evolved to include not only the fresh air requirements, but also how the supplied air is distributed. A good air distribution system delivers sufficient fresh air to occupied regions and removes contaminants (including heat) from occupied regions as quickly as possible. There are four basic types of air distribution patterns, namely fully mixed flow, piston-like flow, short-circuiting and stagnant flow patterns.

A piston-like flow pattern is the most efficient way to deliver fresh air and remove contaminants. But it is always hard or expensive to maintain such an ideal flow pattern. In displacement ventilation, such a flow pattern is partly achieved by using the simplest natural convection principle, ie. warmer air rises. Thermal plumes over a person or a computer, and upward natural convection along a warmer wall surface, all tend to rise.

These upward flows transport the contaminants generated in the occupied regions or in the same source where the heat is generated. Such a flow behaviour is utilised in displacement ventilation to maintain a vertical upward displacement flow pattern. In a conventional mixing ventilation system, the room convection flows are not directly used, but rather are "destroyed" to some extent to achieve a flow pattern dominated by supply jets.

In many situations, displacement ventilation has been proved to be more efficient than mixing ventilation for simultaneously removing excess heat and achieving good air quality. It should be mentioned that displacement ventilation is primarily designed for cooling purposes. Its use alone is not efficient for heating.

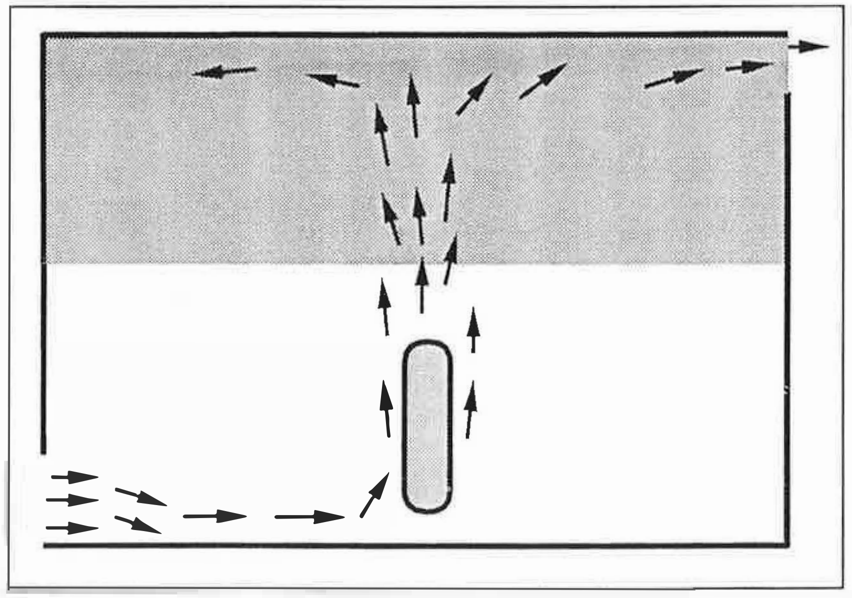
Natural convection flows

Very early studies on ventilation, eg. Shaw [20], have suggested that natural convective flows over the human body are the most important factor in room ventilation. Shaw asked: if natural convection around a body did not exist, what would happen to the air surrounding the head? He suggested an interesting comparison between a fish in water and a man in air. Water density cannot be substantially affected by fish body temperature as occurs to air by the human body. If a fish were to breathe water in and out, as humans breathe air in and out, a cloud of used water would be formed in front of the fish which would be breathed over and over again. Shaw interestingly speculated that this might be the reason that fish use gills to direct used water out behind the head. Fish cannot take advantage of the natural convective plume to expel the used water.

The author repeatedly suggested that "for the purpose of ventilation, convection is the primary condition of success though it is the cause of many failures". It has been realised now that the near-body natural convection currents (not only the plume over the human body) are very important to the effectiveness of displacement ventilation.

In a simple displacement ventilation system, cool air is supplied with a low velocity at floor level, which is then transported upward by natural convection and plumes. The used air is exhausted at the ceiling level (see Figure 1). Using smoke flow visualisation, one can find that there are generally two distinct zones — one lower clean zone containing fresh air and one upper polluted zone.

Figure 1. Basic flow principles in a room ventilated by displacement



A short history of displacement ventilation

The idea of displacement ventilation is not new. One can find examples in many old buildings with natural ventilation systems with openings in both roof and vertical walls. When the stack effect dominates ventilation, the outdoor air is sucked in from the lower openings, falls down to the floor level, spreads along the floor, and then displaces upward together with plumes, and finally leaves the building from its roof openings.

In natural displacement ventilation systems it is not always possible to control the air quality and supply air velocity for thermal comfort requirements, whereas mechanical displacement ventilation uses low velocity diffusers to achieve a better comfort control. The first widespread installation of displacement ventilation was in the earlier 1970s, due to a Swedish ventilation company, Bahco (now Stratos Ventilation), who designed such a system for the welding industry [22]. Since then, prediction and design tools have become available for the system, which will be summarised below. The system has now been used in both industrial buildings and offices.

Purpose of paper

The purpose of this paper is to introduce the thermal and flow characteristics in displacement ventilation, present simple prediction models for both flows and heat transfer, discuss some major design parameters and outline some other related developments. Where appropriate, some new research results will also be introduced.

Characteristics

The thermal and flow characteristics of displacement ventilation are discussed here by examining both the global flow patterns and the local flow elements.

Global flow pattern

The main global flow characteristics are the contaminant and thermal stratifications [4].

- Contaminant stratification — globally, there are two zones, one lower clean zone and one upper polluted zone. In the lower clean zone, the flow is dominated by horizontal spread of the supply fresh air and rising convection flows over heat sources and warmer wall surfaces. There is generally no strong transport of contaminants across the interface between the two zones except within rising convection currents. This interface will be called the clean zone height. This characteristic is shown in Figure 1 and is demonstrated by measurement in Figure 2.
- Thermal stratification — a vertical temperature gradient is generated outside the plumes with lighter air floating above heavier air. The ceiling is warmer than other surfaces. This gives rise to radiation transfer from the ceiling, mainly to the floor. As a result, this makes the floor warmer than the air layer adjacent to the floor and the ceiling cooler than the air layer below the ceiling (see Figure 3).

The above global flow characteristics mean that the flow in displacement ventilation cannot be modelled or analysed with a fully mixed approximation. This is one of the major difficulties encountered when designing displacement ventilation.

Local flow elements

There are at least three important flow elements, namely supply gravity currents, plumes and natural convection flows along vertical surfaces.

- Supply gravity currents — Sandberg and Mattson [19] showed the flow from the low velocity terminal is not a jet, but a gravity current. The entrainment of ambient air into the gravity currents is hindered by the thermal stratification. The flow rate in the current stays almost the same as the supply airflow rate. Unlike jets, gravity currents may spread across the entire floor area. This lateral spread is compared with a high-momentum jet spread in Figure 4. Lateral spreading of gravity currents helps

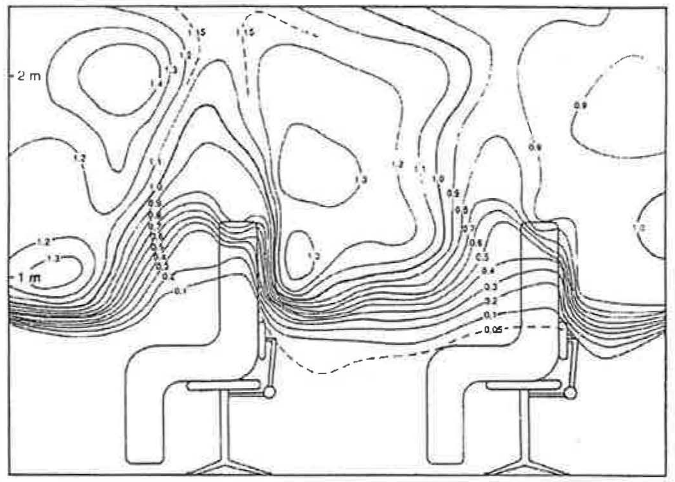


Figure 2. Measured concentration field in a room ventilated by displacement. Contaminant source was located at the ceiling level (from [23])

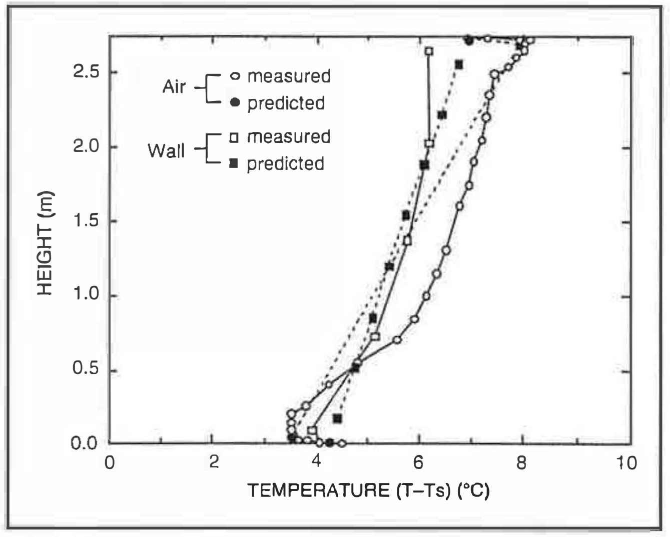
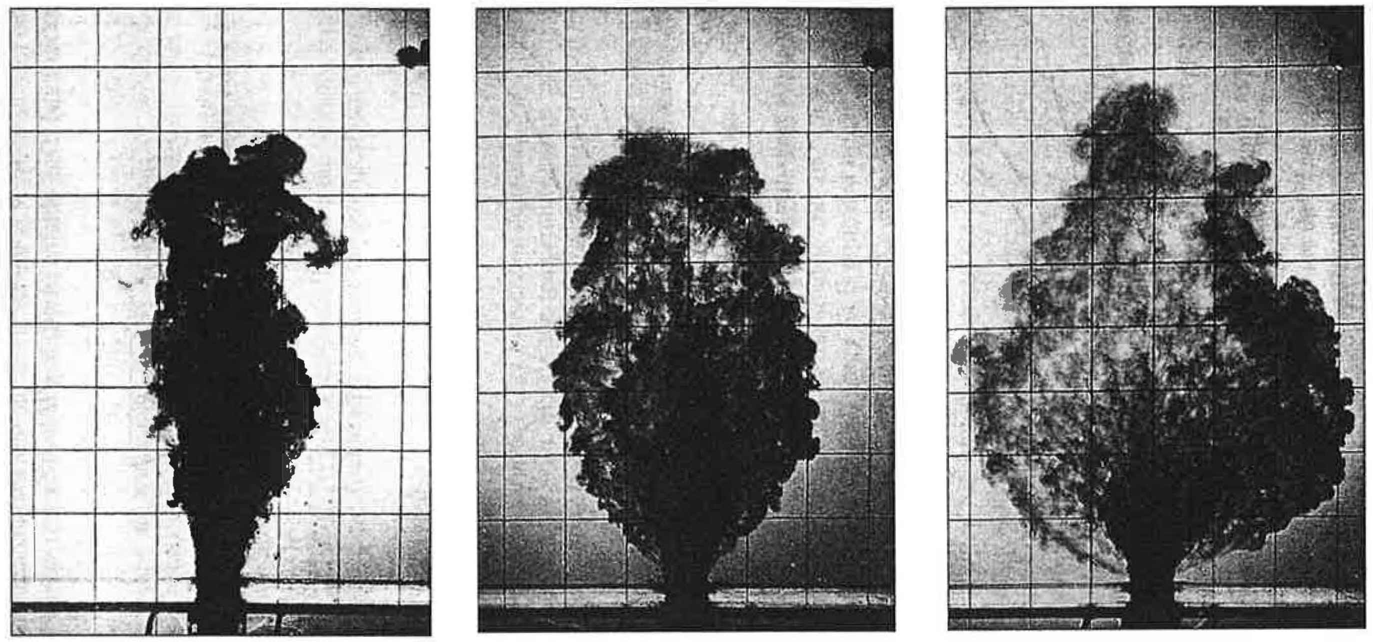


Figure 3. Measured temperature profile in a room ventilated by displacement (from [9])

displacement ventilation to deliver fresh air to a wider occupied region. However, the buoyancy force of a gravity current decreases gradually as it advances along the floor, being heated by the convective heat transter along the floor surface.

- Plumes [6] — plumes are generated over heat sources. By entrainment, flow rates in the plumes increase along their height. Due to mass conservation, at the clean zone height, the total airflow rate in the plumes and other rising convection currents equals the supply airflow rate. This in fact gives the simple two-zone model of displacement ventilation, which will be detailed later.
- Natural convection flows along vertical surfaces — in the lower part of vertical walls, the wall temperature is generally higher than the room air (see Figure 3), so upward natural convection occurs. On the other hand, in the upper part of vertical walls, downward natural convection generally develops. Generally, the velocity magnitudes in these wall surface flows are smaller than those in thermal plumes. However, these wall flows cover a much larger area and the total flow rates in these wall flows can be very significant [7]. For a constant temperature wall and a constant-temperature ambient air, the flow rates in the boundary layers can be analytically calculated. However, here we have both a wall surface temperature gradient and an air vertical temperature gradient. To our knowledge, there is no theoretical work being done to calculate flow rates in such natural convection boundary layers. For simplicity, the constant-temperature assumption is generally used.



Additionally, after the plumes penetrate the clean zone height, they may continue to rise until they hit the ceiling. The flow rates in the plumes also continue to increase. At the ceiling level, the total upward flow is generally greater than supply or return airflow rate.

The flow thus turns around the corners and flows downwards. These interactions generally cause the flow in the upper zone to be rather mixed.

Figure 4. Comparison of the spreading of jets and gravity currents: left - low Archimedes number (a jet); middle - moderate Archimedes number; and right - high Archimedes number (a gravity current) (from [17]).

Engineering prediction models

Engineering models aim at predicting main design parameters, including the airspeed in the far-field of a supply register, the clean zone height of the clean zone, the vertical air temperature gradient, air temperature in the occupied region and in the upper region, the concentration profile, and the cooling load. The models summarised here provide only simple tools for preliminary ventilation concept design. Fine tuning of the final design may require more detailed study such as computational fluid dynamics simulation (see for example [7]), which will not be discussed here.

Air speed in the occupied regions

Close to a low-velocity supply register, there is generally a region with uncomfortable and large vertical airflows. This region is defined as the near-field of a low-velocity supply register. Accordingly, the region beyond is called the far-field (see Figure 5). Two parameters are of design interest:

- the near-field length, X_D; and
- the velocity magnitude in the far-field, V_f

In order to illustrate the physics, the simplest case of a two-dimensional gravity current is considered here. Sandberg and Holmberg [17] showed that for a two-dimensional gravity current,

$$X_D = U_{in}\sqrt{\frac{H}{g'}} = \frac{H}{\sqrt{Ar}} \tag{1}$$

$$V_f = \left(\frac{B}{n}\right)^{1/3} \tag{2}$$

where U_{in} is the supply velocity, H is the height of the supply register, g' is the reduced gravity, Ar is the Archimedes number (defined as $g'H/U^2_{in}$), B is the buoyancy flux, and n is the unit length. For a three-dimensional gravity current, the unit length n in equation (2) can be replaced by the width of the room w.

When a room is fully insulated, the buoyancy flux can be calculated as

$$B = \frac{gE}{\rho c_p T} \tag{3}$$

where E is the heat source power, g is the gravity, ρ is the density, c_p is the heat capacity and T is the average air temperature. It can be seen here for displacement ventilation, both the near-field length and the velocity magnitude in the far-field are governed by the buoyancy force.

Near-floor and near-ceiling air temperatures

To illustrate the calculation of near-floor and near-ceiling air temperatures, let us assume a room which is perfectly tight (no infiltration) and perfectly insulated (only internal heat sources exist). The room has a volume of V (m^3). The amount of pollutant generated in the room is G (kg/s). The total heat source in the room is E (W). The supply airflow rate is q_v (m^3/s). The outdoor air has a pollutant concentration of c_s and a temperature of T_s. The air density is ρ and its specific heat capacity is c_p.

Assume:

- the flow is divided into a cold gravity current zone, a stratified region and a warm gravity current zone;
- the temperature in the stratified region is linear;
- the supply air is spread over the floor area without entrainment;
- the near-ceiling air temperature is equal to the extract air temperature; and
- the radiative equations are linearised due to moderate temperature differences.

An integral energy balance for the room as a whole gives

$$q_v \rho c_p (T_e - T_s) = E \tag{4}$$

Integral energy balances for the floor surface and the cold gravity current zone give

$$\alpha_r A(T_c - T_f) = \alpha_f A(T_f - T_f^a) \tag{5}$$

$$\alpha_r A(T_c - T_f) = \alpha_c A(T_c^a - T_c) \tag{6}$$

$$q_v \rho c_p (T_c - T_f) = \alpha_f A(T_f - T_f^a) \tag{7}$$

where with linearisation $\alpha_r = 4T^3_0\sigma$, and T_0 is calculated approximately as the mean of the assumed floor and ceiling temperature.

A is the floor or ceiling surface area, and α_f and α_c are convective heat transfer coefficients at floor and ceiling surfaces respectively.

From equations (5) and (7)

$$\lambda = \frac{T^a_f - T_s}{T_e - T_s} = \left[\frac{q_v \rho c_p}{A}\left(\frac{1}{\alpha_f} + \frac{1}{\alpha_r} + \frac{1}{\alpha_c}\right) + 1\right]^{-1} \tag{8}$$

The mean vertical temperature gradient can then be calculated to be

$$s = (1 - \lambda)\frac{E}{\rho c_p q_v H} \tag{9}$$

where H is the room height. The temperatures of T_e, T^a_f and T_f can be calculated as

$$T_e = \frac{E}{\rho c_p q_v} + T_s \tag{10}$$

$$T_c^a = T_e \tag{11}$$

$$T_f^a = \lambda (T_e - T_s) + T_s \tag{12}$$

$$T_f = \frac{\alpha c_p q_v (T_f^a - T_s)}{\alpha_c A} + T_f^a \tag{13}$$

$$T_c = \frac{\alpha_c T_c^a + \alpha_r T_f}{\alpha_c + \alpha_r} \tag{14}$$

This is the four-node model (see Figure 6) developed by Li *et al.* [8] and Mundt [12]. The predicted results are shown in Figure 3.

Clean zone height

If the total airflow rate in the vertical wall boundary layers is q_w and the total flow rate in the thermal plumes is q_p, the clean zone height is determined by (see [18])

$$q_p + q_w = q_v \tag{15}$$

For a wall surface with a temperature difference of $\Delta T = T_w - T_{\infty}$, [13]

$$q_w = 0.00275 \Delta T^{0.4} h^{1.2} w \tag{16}$$

where h is the height and w is the width of the wall.

For a convective heat source with a heat power of E (W), [13]

$$q_p = 0.005 E^{1/3} (h + R)^{5/3} \tag{17}$$

where R is the diameter of the heat source.

If there are multiple heat sources and multiple wall surfaces, the total flow rates should be considered. Mundt [13] gives a summary of airflow rates over different heat sources.

Concentration field

In the upper polluted zone, we assume that the extract pollutant concentration, C_e, is the same as the room concentration, C_{au}. Thus,

$$C_{au} = C_e = C_s + \frac{G}{q_v} \tag{18}$$

where G is the pollutant generation rate and q_v is the total ventilation flow rate. C_s is the concentration in the supply air. In the lower clean zone, the concentration C_{al} is the same as that supplied, C_s.

$$C_{al} = C_s \tag{19}$$

Cooling load prediction

In the models discussed above, the cooling load is assumed to be known. In realistic buildings, the cooling loads include not only the internal sensible heat sources, but also heat transmission through building envelopes. Many conventional cooling load programs do not consider the non-uniformity of air temperature fields, thus they are not applicable to displacement ventilation without further improvement. One possible improvement in these programs is to integrate a simple four-node model such as that described above.

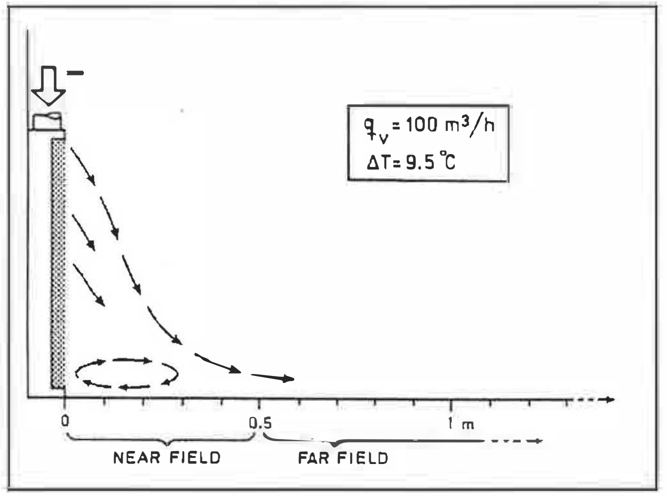


Figure 5. Near-field and far-field in displacement ventilation (from [2]). q_v is the supply airflow rate and ΔT is the air temperature difference between supply and exhaust.

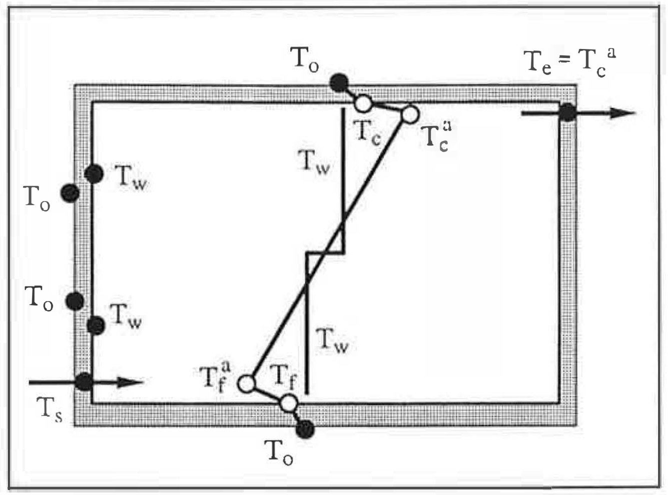


Figure 6. Illustration of four-node model of displacement ventilation (see [9])

A research version of the CHEETAH program [1] has recently been developed which does this. CHEETAH calculates hourly temperatures in up to ten zones of a building, using real weather data. While zones are usually associated with volumes (eg. rooms), they can also be associated with surfaces. Thus, the floor and ceiling surfaces, the air near the floor and the air near the ceiling are designated as zones, and heat is transferred between them via convective and radiative exchanges.

The four-node model can be easily extended in CHEETAH to an n-node model, the only limitation being the number of available zones. To illustrate the implementation, a 5-node model of a simple room was created, where the fifth node is the wall surface. The room size is 10m x 10m x 3m and there are no windows. The floor and ceiling consist of 50mm of external insulation and 20mm of concrete while the walls consist of 50mm of insulation only. The wall construction differs simply to illustrate the great flexibility obtained by implementing the n-node model into an existing building thermal simulation program. The heat source in the room is 2500 W, and no infiltration is assumed. All outdoor surfaces are driven simply by a sinusoidal temperature. Air at 19°C is supplied at floor level at a constant flow rate of 0.25 m^3/s.

Figure 7 shows the node temperatures in this room. Because of radiative exchanges between the floor and ceiling, the floor air is cooler than the floor surface, and the ceiling surface is cooler than the ceiling air. The temperature difference between the surfaces is approximately 5 K, which gives an estimate of the stratification in the room. Note the phase differences between wall surface temperature and floor or ceiling temperatures. These are due to the differences in thermal capacity. With a knowledge of these temperatures, the cooling load can be calculated.

Some design aspects

In designing a displacement ventilation system, care should be taken to ensure that the following factors are considered properly: freedom from draughts, comfort limits, the vertical temperature gradient, convective flows, cooling capacity, ducting, cooling load calculation, layout of air supply registers, control [15], and sound levels in a room.

Generally, design guidelines require that the supply air temperature should be around 19°C and the supply air velocity should be less than 0.2 m/s for comfort ventilation. These may vary for different applications.

Indoor air quality and thermal comfort

Two major requirements in displacement ventilation design are:

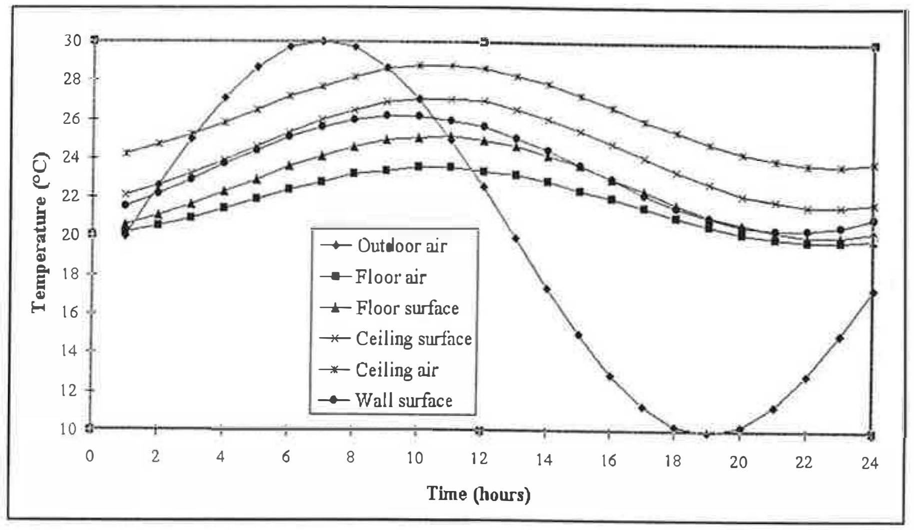


Figure 7. Time-dependent air and surface temperatures in a room with displacement ventilation calculated by CHEETAH

- the clean zone height should be above the occupied region — for indoor air quality considerations; and
- the vertical air temperature gradient should be within a comfortable range, say 3°C—for thermal comfort considerations.

From the four-node model, it can be seen that the higher the heat power of the heat sources in the room, the higher the vertical air temperature gradient. Vertical air temperature gradient can be lowered by supplying a higher airflow rate. However, there is generally an upper limit to the total supply airflow rate, specified by different ventilation codes. Therefore, there is generally a maximum cooling capacity, eg. around 25 W/m^2 in offices, due to the thermal comfort requirement on vertical temperature gradients (see [16, 24]). Other studies show that this upper limit may be increased.

A recent experimental study by Wyon and Sandberg [25] showed that it is the operative temperature and not the vertical temperature gradient which affects the thermal comfort of the whole body and the local thermal discomfort. In their experiments, test persons were exposed to a range of vertical temperature gradients, namely 0, 2 and 4 K/m. The results showed that the vertical temperature gradient did not affect thermal comfort. Therefore, if the feet are too cold or the head is too warm it is because of local temperature, not the temperature gradient. At this stage, it is reasonable to suggest that the eventual influence of temperature gradients is under debate.

Minimum supply airflow rates can be determined from both indoor air quality and thermal comfort considerations. From an indoor air quality point of view, the height of the clean zone should be above the occupied region, say at head height. The two-zone model can be used to determine a minimum supply airflow rate. From the thermal comfort point of view, the four-node model can be used to determine another minimum supply airflow rate. To satisfy both indoor air quality and thermal comfort, the final ventilation flow rate should be the largest of the calculated flow rates.

The two-zone model requires the input of wall surface temperature which is the output of the four-node model, which requires the input of the airflow rate. There is a possibility that these two models can be integrated.

Selection of supply registers

Once the supply airflow rate is chosen, and the supply air parameters such as air temperature and velocity are determined, the supply registers can be chosen. As the supply airstream is not a jet any longer, the existing knowledge on mixing ventilation jets cannot be used directly for displacement ventilation design. The equations provided above may be used to estimate the near-field length and a good design should keep the near-field length as short as possible. There is generally information available from manufacturers on different diffusers, which can be used. As each room has different layout and space constraints, more advanced prediction tools such as CFD may be used for selection of supply registers.

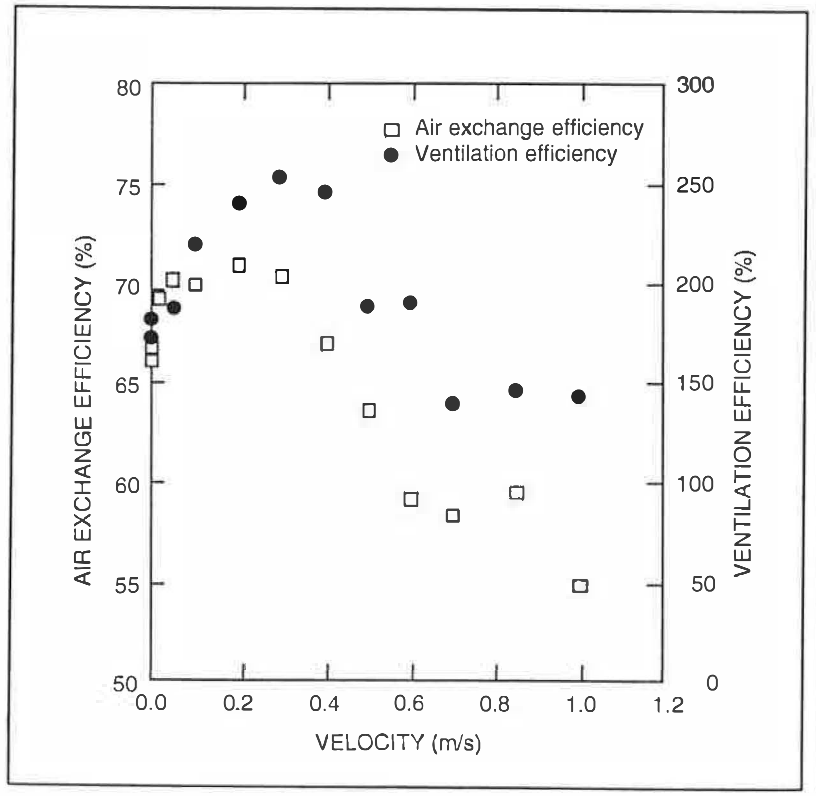
Other related development

It is easy to speculate that moving bodies such as a walking person can have a strong influence on displacement ventilation performance. A recent study using a "walking" dummy showed that a very slow movement increases the air exchange and ventilation efficiency. However, at higher velocities, the efficiency decreased and gradually approached those of a mixing ventilation system (see Figure 8).

In Figure 8, the ventilation efficiency is defined as the quotient of the outlet concentration and the room average concentration, while the air exchange efficiency is defined as the quotient of the nominal time constant and two times the mean age of air in the room. Thus, the ventilation efficiency is a measure of how quickly a contaminant is removed from a room, and air exchange efficiency is a measure of how effectively the air in a room is replaced by fresh air from the ventilation system.

However, the study used a dummy walking in a regular and monotonous way and in reality, human walking is quite random. The effects of random walking have not been studied.

Figure 8. The effects of moving bodies on ventilation efficiency (from [II])



Near-body natural convection flows

The near-body flow pattern is perhaps one of the most important fundamental flows in building ventilation. The performance of a ventilation system depends very much on this flow pattern and the quality of air in these airstreams. The interaction of the natural convection flows and the surrounding flows in the room plays an important role in ventilation performance.

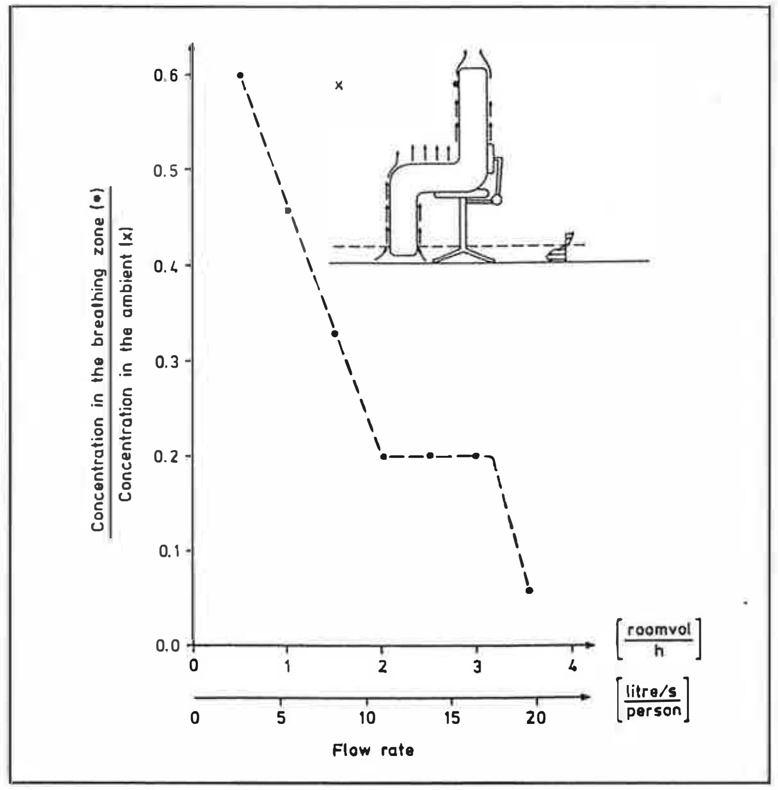


Figure 9. The concentration in the breathing zone relative to the ambient concentration (from [2])

The near-body flows were discussed by Etheridge and Sandberg [2] (Figure 9). It was found that the expiratory flow from the nose and mouth is generally taken away upwards by the airstream moving past the face. The air inhaled is generally taken from the boundary layer flow around the body. Consequently the inhaled air is brought from below to the mouth and nose and the quality of the air inhaled is determined by the quality of the air below the mouth. This implies that the air quality in the breathing zone in ventilation by displacement can be better than that of the ambient air at the same height, which means even if a person's nose or mouth is located above the clean zone height, the inhaled air quality can still be better than that in the polluted zone. This may allow some freedom in designing a clean zone height in some situations.

Raised floor systems

It has been reported that many buildings with raised floor systems have been developed in Europe and Japan [10, 21]. Generally, in raised floor systems, floor-mounted diffusers are used. Some of these diffusers give a vertical circular free jet with swirl, which has been studied by Nielsen *et al* [14]. A simple model for calculating the vertical air temperature gradient was developed by Ito and Nakahara [5].

Chilled ceiling systems

Chilled ceiling systems can be used to partly remove the cooling load in a room in addition to displacement ventilation. The benefits on thermal comfort are obvious as it reduces the supply airflow rate and thus the near-field velocity, and it also reduces the vertical air temperature gradient. A simple implementation of the system is to run cold water through the ceiling. When the ceiling is smooth, heat exchange is mainly carried out by thermal radiation. When the ceiling is ribbed, convection dominates the heat transfer. It should be noted that chilled ceiling systems can neither supply fresh air nor remove latent heat.

It has been reported that the cooling capacity of a chilled ceiling system can be as high as 130-150 W/m^2 for a convective system and 60-90 W/m^2 for a radiative system [3].

Conclusions

Displacement uses the basic principle that warmer air rises to achieve a partly piston-like flow pattern in a room. In many situations, it has been proved to be more efficient than mixing ventilation for simultaneously removing excess heat and achieving good air quality. Its basic thermal and flow characteristics are very different from that of mixing ventilation. Simple engineering models are summarised for predicting near-field length, airspeed in the far-field, air temperature in the occupied regions, the vertical air temperature gradient, concentration in the polluted region, the clean zone height and the cooling loads. Two major design parameters from both indoor air quality and thermal comfort points of view are the clean zone height and the vertical air temperature gradient.

Some design aspects have been discussed. Other related developments covered briefly here include the effects of moving bodies, near-body natural convection flows, raised floor systems and chilled ceiling systems.

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* *Drs Yuguo Li and Angelo Delsante are Research Scientists with CSIRO Division of Building, Construction and Engineering, Highett, Victoria. Dr Mats Sandberg is Professor of Fluid Mechanics and head of the Indoor Environment and Ventilation Laboratory in the Royal Institute of Technology, Cavle, Sweden. This paper was presented at the 1997 AIRAH Conference, Hobart, Tasmania.*

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