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# Studies on the 'Ventilation Effectiveness' and Modification of the Ventilation System in the Waiting Hall of a Railway Station

**Key Words**
Ventilation effectiveness
Macroscopic numbers
System design
Field measurement

**Abstract**
The concept of ventilation effectiveness for mechanical ventilation of an airspace is reviewed and associated parameters for describing the performance of ventilation systems are described. The idea is applied to a study of the thermal environment in the waiting hall of a railway station. Two ventilation schemes: one with a ventilation system only and the other with an air-conditioning system were considered. The proposed air-conditioning system would provide air at a temperature only 5 °C below the ambient but with a higher air circulation rate. This would reduce the amount of energy used for environmental protection compared to more conventional systems providing full air-conditioning. It is found that using such a scheme it is possible to provide a better thermal environment using less energy than with a full air-conditioning system.

## Introduction

Population density in the north-eastern part of the New Territories of Hong Kong has increased significantly in the past decades and travelling by trains is the main means of transportation for rapid transit. Everyday, thousands of people pass through the railway stations and, inevitably, spend some time in the waiting hall. A proper ventilation system [1] is necessary and has to be designed to provide a good indoor environment. In waiting halls with only a mechanical ventilation system, complaints of discomfort due to air stuffiness and a hot environment are frequently reported. Therefore, proposals were put forward to install air-conditioning systems in the waiting halls of railway stations. As the space volume for such a waiting hall is large, a partial air-conditioning system was proposed to help cut down the energy consumption. A study was carried out to assess the resulting environment with different ventilation schemes.

Two ventilations schemes: one with a ventilation system only and the second with partial air-conditioning were evaluated by carrying out detailed investigations in a waiting hall. In the second case, cool air with a temperature 5 °C lower than ambient would be discharged by the proposed air-conditioning system. A higher air speed than usual would be used to give a sense of air movement to the occupants as such a solution is more favoured in a subtropical area like Hong Kong. This would give a balance between the high energy consumption needed to operate a normal air-conditioning system and the provision of an acceptable thermally comfortable environment [1, 2].

This paper reports on investigational studies starting from reviewing the concept of 'ventilation effectiveness' in a large space with mechanical ventilation. Field mea-

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surements and numerical simulations were carried out to evaluate the effectiveness of the two proposed ventilation schemes for the waiting hall.

## Brief Review on Ventilation Effectiveness

Ventilation systems [1] are expected to achieve three goals: maintenance of thermally acceptable environment in the occupied zone, provision of outdoor air and dilution of internal contaminants.

Many flow parameters have been proposed in the literature to provide a quantitative description of the environmental characteristics of a ventilated space. The primary flow parameters are the mean air speed (or velocity), temperature, relative humidity and contaminant concentration. Well-known secondary parameters include the local age of air, purging air flow rate [3], scales of ventilation efficiency, local ventilation rate and different versions of air change efficiency and ventilation efficiency. Usually, they are calculated from the value of the measured speeds at different positions and so depend only on the air flow pattern. However, some parameters depend also on the contaminant level. The ventilation effectiveness is one of the examples and this is a key factor in determining the health and comfort conditions in the ventilated space. As there is no common agreement on how to describe the ventilation effectiveness, different terms have been used. Some of them can be evaluated by experimental or numerical methods.

As reviewed by Sandberg [2], the performance of a general ventilation system would be determined firstly by the distribution of ventilation air in the room and secondly on the distribution of air contaminants. The age distribution of the air, the net flow rate of ventilation air, and the mean age of all air present in the room are key parameters for the first part. The local flow rate of contaminant and the age distribution of contaminants are important for the second part. Mixing of the fresh air and the contaminated air is important.

The age of air was proposed [1, 4] to give the time elapsed since a given sample of air entered a space. Air change efficiency and ventilation efficiency were then derived to evaluate ventilation effectiveness [4, 5]. The air change efficiency is a measure of how effectively the air at every location in a room is replaced by outdoor air. It compares the average residence time for the air molecules in the room at ideal piston flow ventilation to the actual average residence time in that room with the specified ventilation system. Alternatively, the ventilation efficiency measures how quickly a contaminant is removed from the room. The contaminant concentration in the exhaust air is related to the average concentration in the room through this parameter. Different forms of air change and ventilation efficiencies were found in the literature, but a systematic analysis method was proposed by Kato and Murakami [6]. Instead of making use of a single parameter, several scales of ventilation efficiency (SVE) were suggested to describe the spatial distribution of age, residual lifetime and residence time of air. The scales of ventilation efficiency were extended [7] using a contribution ratio to describe the flow field induced by a multiplicity of supply air outlets. The air change effectiveness of an entire building was studied by Fisk and Faulkar [8]. Here, air change effectiveness parameters that indicated the extent of short circuiting, mixing and displacement air flow in the building were defined. Other forms of ventilation efficiency [1] have been derived based on the use of the tracer gas technique to evaluate ventilation effectiveness.

As an alternative to the 'age-of-air' approach, ventilation effectiveness has also been defined based on a 'two-zone' model [9, 10]. In this model, the space under consideration is divided horizontally into two perfectly mixed zones. Calculation of the ventilation effectiveness can be made from an analysis of the interzonal air flow and the steady-state condition in the respective zone. A similar evaluation method has been adopted in Appendix F of the ASHRAE Standard 62-1989 [11].

The parameters defined to describe ventilation effectiveness as noted above can be grouped under three main categories. Group 1 parameters, commonly known as ventilation efficiencies, are direct measures of contaminant removal by the ventilated air. These parameters depend on the contaminant concentration field and the velocity field. Therefore, the parameters will be affected by the location of a contaminant source and the physical characteristics of that contaminant. Group 2 parameters are commonly known as the air change efficiencies. They are measures of the extent of short circuiting or displacement air flow between the air supply and exhaust and they are independent of the contaminant diffusion field. Group 3 parameters are those correlating the ventilation rate for a given space with that of the entire building enclosure. The effectiveness of a building with several interdependent flow fields (multi-zone air flow) will be described.

In designing a ventilation system, engineers found that the air change efficiency is easier to use than the ventilation efficiency. This is because the contaminant diffusion

**Table 1.** Review of the parameters describing ventilation effectiveness

| Parameter | References | Unit | Expression | Nomenclature |
|---|---|---|---|---|
| Air change per hour (ACH) | e.g. CIBSE [12] 1986 | $h^{-1}$ | $ACH = \frac{Q}{V_{RM}}$ | $Q$ = volume flow rate in $m^3\,h^{-1}$<br>$V_{RM}$ = volume of room in $m^3$ |
| Ventilation rate (VR) | e.g. CIBSE [12] 1986 | $m\,s^{-1}$ | $VR = \frac{Q}{A_f}$ | $A_f$ = room floor area in $m^2$ |
| Specific flow (n) | | $s^{-1}$ | $n = \frac{Q}{V_{RM}}$ | $Q$ = volume flow rate in $m^3\,s^{-1}$ |
| Nominal time constant $\tau_n$ | Sandberg et al. [4] 1983 | s | $\tau_n = \frac{V_{RM}}{Q}$ | – |
| Air change time ($t_r$) | Sandberg et al. [4] 1983 | s | $t_r = 2T_m$ | $T_m$ = room mean age of air (residence time) |
| Room mean age of air ($T_m$) | Sandberg et al. [4] 1983 | | | |
| (a) Concentration decay | | s | $T_m = \frac{\int_0^\infty t\cdot c(t)dt}{\int_0^\infty c(t)dt}$ | t = time<br>c(t) = concentration at time t |
| (b) Concentration growth | | s | $T_m = \frac{\int_0^\infty t(1-\frac{c(t)}{c(\infty)}dt}{\int_0^\infty (1-\frac{c(t)}{c(\infty)}}dt$ | – |
| (c) Pulsed injection | | s | $T_m = \frac{1}{2}\cdot\frac{\int_0^\infty t^2\cdot c(t)dt}{\int_0^\infty t\cdot c(t)dt}$ | – |
| Local purging flow rate ($U_p$) | Sandberg et al. [4] 1983 | $m\,s^{-1}$ | $U_p = Q\frac{c_c}{c_p}$ | $c_c$ = concentration at exit<br>$c_p$ = concentration at point p |
| Air change efficiency ($E_a$) | Sandberg et al. [4] 1983 | – | $E_a = \frac{\tau_n}{t_r} = \frac{\tau_n}{2T_m}$ | – |
| Local air change efficiency ($E_p$) | Sandberg et al. [4] 1983 | – | $E_p = \frac{\tau_n}{T_p}$ | $T_p$ = mean age of air at point p |
| Ventilation efficiency ($E_v$) | Sandberg et al. [4] 1983 | – | $E_v = \frac{c_c(\infty) - c_s(\infty)}{\langle c_p(\infty)\rangle - c_s(\infty)}$ | – |
| Local ventilation index ($E_{vp}$) | Sandberg et al. [4] 1983 | – | $E_{vp} = \frac{c_e(\infty) - c_s(\infty)}{c_p(\infty) - c_s(\infty)}$ | – |
| Scale of ventilation efficiency (SVE) | Kato and Murakami [6] 1988 | | | |
| (a) SVE1 | | – | $SVE1(x_s) = \frac{c_o(x_s)}{c_s\int_{v(x)}dx}$<br>where $c_s = q/Q$<br>$c_o(x_s) = \int_{v(x)}c_x(x_s,x)dx$ | $c_s(x_s,x)$ = contaminant concentration at x with contaminant generation at $x_s$<br>q = generation rate of contaminant<br>$c_s$ = perfect mixing concentration |
| (b) SVE2 | | – | $SVE2(x_s)^2 = \int_{v(s)}\frac{(x - x_G(x_s))^2\cdot c_x(x_s,x)dx}{c_o(x_s)}$<br>where $x_G(x_s) = \int_{v(x)}\frac{x\cdot c_x(x_s\cdot x)dx}{c_o(x_s)}$ | – |
| (c) SVE3 | | | $SVE3 = \frac{c'_x(x)}{c_s}$ | $c_x$ = contaminant concentration in case of uniform contaminant generation |

**Table 1** (continued)

| Parameter | References | Unit | Expression | Nomenclature |
|---|---|---|---|---|
| Ventilation effectiveness (Ev) (zone model) | ASHRAE Standard 62–1989 [13] | – | $Ev = \frac{1-s}{1-rs}$ | s = fraction of the supply air delivered to a ventilated space that bypasses the occupied portion of that space. r = fraction of return air that is recirculated. |
| Air exchange effectiveness | Fisk et al. [14] 1989 | | | $T_N$ = age of air exhaused to the outside |
| $AEE_{global}$ | | – | $AEE_{global} = \frac{T_N}{T_m}$ | |
| $AEE_{BL}$ | | – | $AEE_{BL} = \frac{T_N}{T_{BL}}$ | $AEE_{BL}$ = AEE at breathing level $T_{BL}$ = age of air at breathing level |
| Normalised local age of air (NLA) | Fisk et al. [14] 1989 | – | $NLA = \frac{T_N}{T_{BL}}$ | – |
| Air diffusion effectiveness (ADE) | Fisk et al. [14] 1989 | – | $ADE = \frac{T_{RG}}{T_{BL}}$ | $T_{RG}$ = age of a return grille located near the $T_{BL}$ measurement location |
| Ventilation efficiency in the occupied zone ($\varepsilon_{oc}$) | Nielsen [15] 1992 | – | $\varepsilon_{oc} = \frac{c_R}{c_{oc}}$ | $c_R$ = concentration in the return opening $c_{oc}$ = mean concentration in the occupied zone |
| Mean ventilation efficiency ($\bar{\varepsilon}_T$) | Nielsen [15] 1992 | – | $\bar{\varepsilon}_T = \frac{c_R}{\bar{c}}$ | $\bar{c}$ = mean concentration in the whole room |
| Temperature efficiency ($\varepsilon_t$) | Nielsen [15] 1992 | – | $\varepsilon_t = \frac{\theta_R - \theta_o}{\theta_{oc} - \theta_o}$ | $\theta_R$ = temperature in return opening $\theta_o$ = temperature in supply opening $\theta_{oc}$ = mean temperature in the occupied zone |
| Relative contaminant removal effectiveness ($\mu$) | Haghighat et al. [16] 1992 | – | $\mu = \frac{T_m}{T_{mc}}$ | $T_m$ = average age of air $T_{mc}$ = average age of contaminant |

field is difficult to determine at the design stage. To further simplify the analysis, they looked for a design parameter which is a single number such as the air exchange rate that would describe the handling capacity of the ventilation system at any specified location. The relevant parameters (such as the purging flow rate and SVE2 [6]) derived so far are all group 2 parameters and can be calculated only when the contaminant field is known. It is highly desirable to get design parameters for specifying ventilation effectiveness which can be calculated only from the velocity field. It seems that the local ventilation rate is a probable candidate although there are many other mathematical expressions derived [12–16] to describe ventilation effectiveness as shown in table 1.

## Local Ventilation Rate

The local ventilation rate can be defined as the specific ventilation flow rate at a given location of the air flow field. The local ventilation rate has been expressed in terms of the specific air flow rate for the entire flow field divided by the local age of air which is normalised for the entire flow field [14]. The local air speed has not been considered for this study and there is no direct relationship assumed between the local ventilation rate and the local mean air velocity. However, it is well recognised by engineers that the mean air velocity is directly related to the local ventilation rate, especially at locations near the supply air where the local age of air is low. In fact, local mean air velocity should have a dominating influence on the local ventilation rate when the age of air is approaching zero. The local ventilation rate $v_{sp}$ at a point P is defined as:

$$v_{sp} = \begin{cases} \dfrac{v_s}{T_n} & \\ v_p & \text{if } \dfrac{v_s}{T_n} > v_p \end{cases} \tag{1}$$

where $T_n$ is the normalised local mean age of air at the point P, $v_s$ is the specific flow rate for the entire flow field given by the ratio of the supply air flow rate to the volume of room, $v_p$ is the specific flow rate at point P given in terms of the area $A_p$ and volume $Vol_p$ of the control volume:

$$v_p = \frac{A_p\sqrt{u^2 + v^2 + w^2}}{Vol_p} \tag{2}$$

The local age of air can be calculated based on the numerical method proposed by Matsumoto and Kato [17].

The local ventilation rate is commonly viewed by engineers as the specific rate of supply of ventilated air to the specified location of the flow field. This is similar to the macroscopic air exchange rate used in conventional ventilation design in that perfect mixing is assumed. An analogy can also be made between the local ventilation rate and local air exchange rate. However, by definition, the local air exchange rate is an ambiguous and sometimes misleading parameter [4]. Therefore, the local ventilation rate can be considered as a supplement to the host of flow parameters defined to described ventilation effectiveness [2, 4]. The use of the local ventilation rate in the study of flow field in HVAC applications has been discussed in detail elsewhere [18].

## Age of Air

A simple method for solving the age directly from its transport equation was given by Matsumoto [19]. A transport equation was derived by assuming that air particles are delivered only by convection. Suppose the age of a fluid element at a point is given by T(x,y,z,t). When the fluid element is moved to another point at time $\Delta t$ later, and if the motion is dominated by convection with air velocity components u, v, w along the x, y, z directions, the equation for T(x,y,z) is:

$$T(x + u\Delta t, y + v\Delta t, z + w\Delta t, t + \Delta t) = T(x,y,z,t) + \Delta t \tag{3}$$

Equation 3 can be expanded as:

$$T(x + u\Delta t, y + v\Delta t, z + w\Delta t, t + dt) = T(x,y,z,t) + \left(\frac{\partial T}{\partial x}\right)u\Delta t + \left(\frac{\partial T}{\partial y}\right)v\Delta t + \left(\frac{\partial T}{\partial z}\right)w\Delta t + \left(\frac{\partial T}{\partial t}\right)\Delta t \tag{4}$$

Equating the right hand side of equations 3 and 4:

$$u\left(\frac{\partial T}{\partial x}\right) + v\left(\frac{\partial T}{\partial y}\right) + w\left(\frac{\partial T}{\partial z}\right) + \left(\frac{\partial T}{\partial t}\right) = 1$$

For a very short time interval, the age of air does not change and so the transport equation for the age of the fluid element is:

$$u\left(\frac{\partial T}{\partial x}\right) + v\left(\frac{\partial T}{\partial y}\right) + w\left(\frac{\partial T}{\partial z}\right) = 1 \tag{5}$$

This is a type of convection equation composed of the convection term and the source term ( = 1). The equation can be solved under the known flow field and the boundary conditions of age T. The age can be given as zero at the inlet, and $\partial T/\partial n = 0$ is assumed on the other boundaries. A solution procedure to calculate the age distribution by this method has been illustrated [19].

The diffusion concept was included by considering the transport equation for the residence time introduced by Sandberg [9]. An equation was derived [20] from the conservation law in terms of the diffusivity D:

$$u\left(\frac{\partial T}{\partial x}\right) + v\left(\frac{\partial T}{\partial y}\right) + w\left(\frac{\partial T}{\partial z}\right) = \frac{\partial}{\partial x}\left(D\frac{\partial T}{\partial x}\right) + \frac{\partial}{\partial y}\left(D\frac{\partial T}{\partial y}\right) + \frac{\partial}{\partial z}\left(D\frac{\partial T}{\partial z}\right) + 1 \tag{6}$$

In this equation, the tracer is transferred by both convection and diffusion. It can be considerd as an extension of equation 5. But strictly speaking, the two equations are different as the age T given by equation 5 is the age of fluid elements with zero diffusivity and the age T given by equation 6 is the residence time of tracer with diffusivity D. Since the diffusivity of fresh air is nearly zero, equation 5 can be used to calculate the age of air entering through the supply duct.

A more complicated equation has been derived from equation 5 by Kato and Murakami [6]:

$$\langle u\rangle\frac{\partial\langle T\rangle}{\partial x} + \langle v\rangle\frac{\partial\langle T\rangle}{\partial y} + \langle w\rangle\frac{\partial\langle T\rangle}{\partial z} = \frac{\partial}{\partial x}(-\langle T'u'\rangle) + \frac{\partial}{\partial y}(-\langle T'v'\rangle) + \frac{\partial}{\partial z}(-\langle T'w'\rangle) \tag{7}$$

where T′ and u′, v′, w′ are the variations of age T and velocity components u, v, w. The symbol $\langle x\rangle$ shows the average ensemble operation of variable x. The equation takes account of turbulent diffusion and the terms $\langle T'u'\rangle$, $\langle T'v'\rangle$ and $\langle T'w'\rangle$ can be approximated by the gradient transport hypothesis

$$\frac{v_t}{\sigma_s}\frac{\partial\langle T\rangle}{\partial x}, \frac{v_t}{\sigma_s}\frac{\partial\langle T\rangle}{\partial y}, \frac{v_t}{\sigma_s}\frac{\partial\langle T\rangle}{\partial z}$$

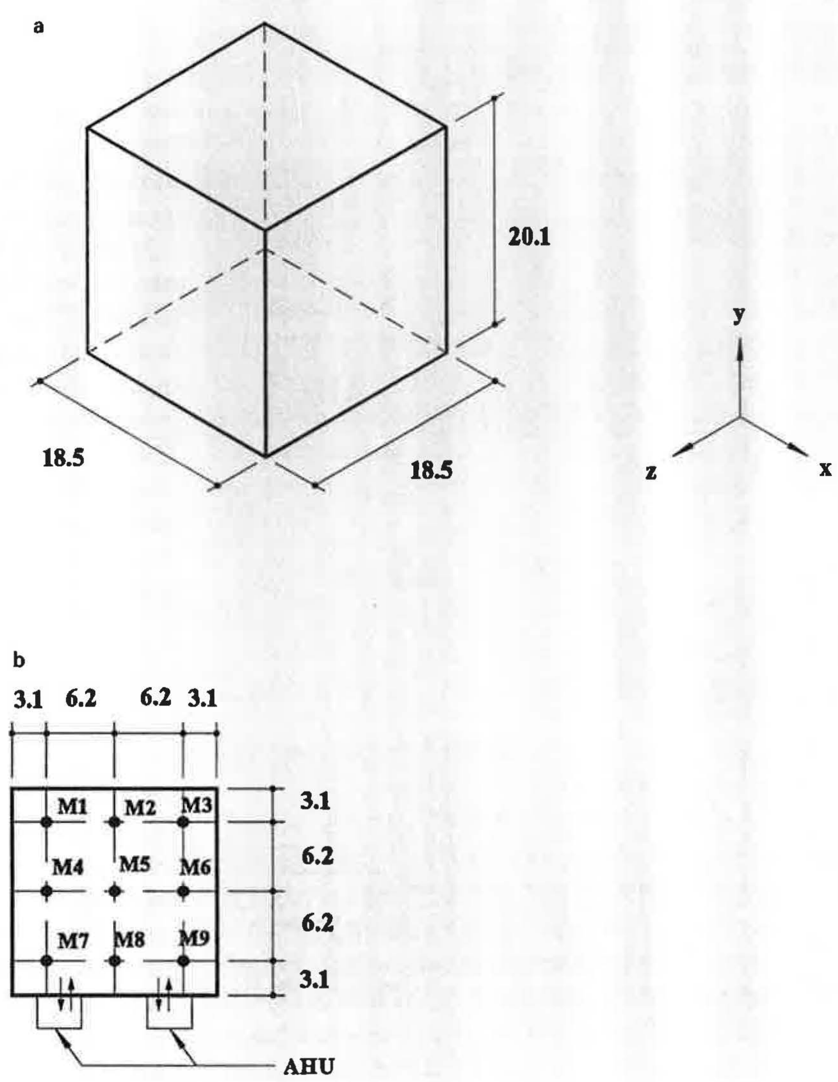


**Fig. 1.** Waiting hall of a railway station. All measuring points at 1.2 m above floor level. All dimensions in metres. AHU = Air-handling units. **a** Geometry. **b** Floor plan for measurement.

If it is modelled appropriately, this equation can give a better prediction of the age than the others.

However, equation 5 is considered to be good enough to predict the age of air for most purposes and it is relatively simpler to compute the equation. This is the method used in the present study.

### Ventilation Effectiveness of a Waiting Hall

A study was conducted to assess the ventilation efficiency in the waiting hall of a railway station employing two alternative ventilation systems. The dimensions of the hall are shown in figure 1. A mechanical ventilation system was installed with the air handling unit shown in figures 1b and 2a to provide a healthy and odour free environment. However, ever since the hall was first opened for use, there have been numerous complaints in hot summer seasons about a lack of air movement and poor air quality. The mechanical ventilation system was subsequently replaced with a partial air-conditioning system in an effort to improve the thermal conditions.

Many partial air-conditioning systems have been successfully installed in tropical countries to provide marginal cooling to occupants at a relatively low energy cost (as compared with a conventional air-conditioning system). With such a system, the occupied zone of the air-conditioned space is controlled at a temperature 3–6 °C below

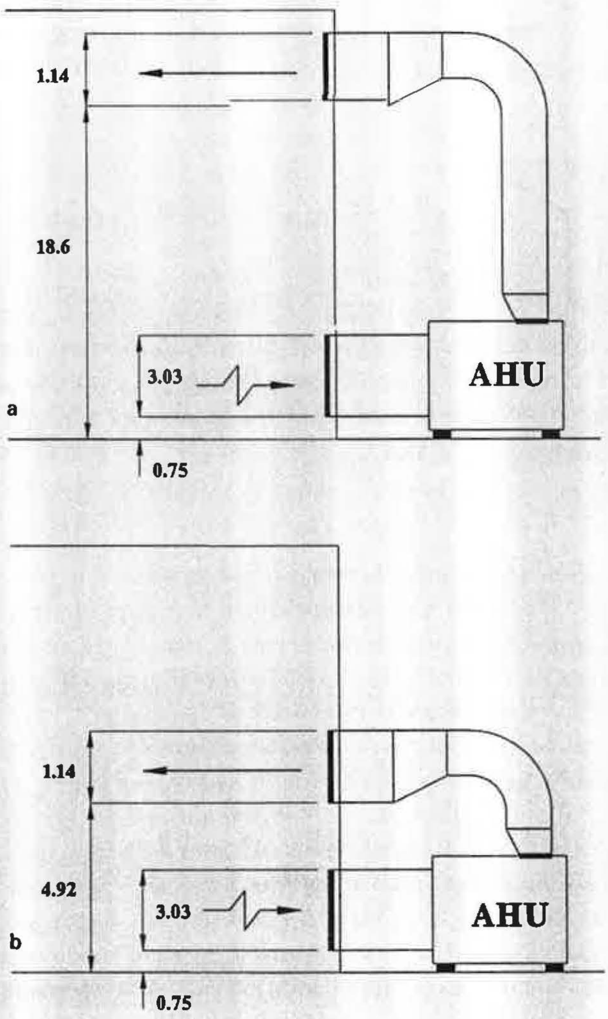


**Fig. 2.** Locations of air intake and exhaust. All dimensions in metres. Not to scale. AHU = Air-handling units. **a** VS1: mechanical ventilation. **b** VS2: air-conditioning system.

the outside air temperature in summer. To compensate for the possible discomfort due to elevated room temperature (which is usually of the order of 28–30 °C), the air circulation rate in the occupied zone is maintained at a high value to introduce a cool draughty feeling. The room sensible and latent load is much smaller than that of a conventional full air-conditioning system because of the smaller temperature differential between the indoors and outdoors. The saving in chiller power more than offsets the slight increase in fan power needed to create the draughty feeling. The size of the chiller plant needed will also be much smaller.

The proposed partial air-conditioning system for the waiting hall is shown in figure 2b. The cool air discharge is set at a lower level than that of a mechanical ventilation system to maximise the air circulation rate in the occupied zone. In the study, the original mechanical ventilation scheme is labelled as VS1 and the partial air-conditioning scheme is labelled as VS2.

One of the prime objectives of the study was to evaluate the ventilation effectiveness of the two ventilation schemes. Two approaches were adopted: field measurement and numerical analysis. From field measurements, macroscopic parameters describing ventilation effectiveness can be determined and analysed against the more enhanced parameters calculated from numerical simulation. The results obtained from the two approaches should complement each other and be able to provide a better picture of the effectiveness of the ventilation regime.

## Field Measurements

Measurements were made in the occupied zones 1.2 m above floor level at locations shown in figure 1b. Grid lines were marked on the floor for positioning the measurement points labelled as M1–M9. A standard tripod was used to support the transducer. Measurements were made 1.2 m above the floor level during a 5-min interval. Only two technicians were engaged in each test to avoid causing disturbance to the flow field. They would stay at least 900 mm away from the test rig. Because of the small temperature difference between the skin of people and the surroundings, the buoyancy effect due to the occupant is assumed not to affect the experimental results considerably.

The air speeds were measured using a Brüel & Kjær Indoor Climate Analyzer 1213 which was calibrated in an accredited laboratory.

The age of air was measured by the tracer gas concentration decay method using a Brüel & Kjær multi-gas monitor type 1302. Sulphur hexafluoride ($SF_6$) was used as the tracer gas. The age of the air $T_p$ at the point P was calculated from the expression:

$$T_p = (t_{stop} - t_{start})\, C_{av}/C_{start} \tag{8}$$

where $t_{start}$ is the time when tracer injection stopped at the beginning of a tracer gas decay; $t_{stop}$ is the time of the final tracer gas measurement at P during the tracer gas decay; $C_{av}$ is the time average tracer gas concentration at P between time $t_{start}$ and $t_{stop}$ and $C_{start}$ is the tracer gas con-

**Table 2.** Summary of the physical configuration of the test sites

| Ventilation scheme | VS1: with mechanical ventilation only | VS2: with air-conditioning system |
|---|---|---|
| Approximate number of occupants during normal working hours | 150 | 150 |
| Design air changes per hour | 48.6 | 48.6 (higher than the normal value for air-conditioned spaces) |
| Nominal time constant, s | 74.1 | 74.1 |
| Parameters measured | | |
| Mean air speed, m $s^{-1}$ | | |
| Mean | 0.63 | 0.69 |
| SD | 0.2 | 0.38 |
| Percentage dissatisfied, % | | |
| Mean | 24 | 29 |
| SD | 6 | 11 |
| Normalised age of air | | |
| Mean | 1 | 0.75 |
| SD | 0.09 | 0.18 |
| Local ventilation rate, $h^{-1}$ | | |
| Mean | 52.7 | 68.2 |
| SD | 6.29 | 14.58 |

centration at P at time $t_{start}$. A more detailed discussion of the experimental procedure has been reported [18].

The percentage dissatisfied [21] due to draught (or elevated air speed [20] for Hong Kong) was also obtained.

The mean and standard deviation of the measured mean air speed, mean age of air and local ventilation rate at occupied zone for the test site are summarised in table 2. The mean age of air was normalised against the overall mean for the entire space. Values of the mean velocities over all the site exceeded the limits recommended by the Chartered Institution of Building Services Engineers (CIBSE) [10] as a higher air speed is required to achieve the free evaporative cooling effect on the skin. The local age of air distribution was related to the configuration of the ventilation system and the value decreased from 1 to 0.76 when the supply air outlets were closed to the occupied zone.

The following points were noted when comparing ventilation parameters for the two regimes:

(1) There was a 9.5% increase in the mean air speed in the occupied zone with ventilation scheme VS2. A considerable increase in standard deviation of the measurements was observed, suggesting uneven air distribution within the occupied zone. The increase in mean air speed and standard deviation are expected because air diffusers were installed at a low level to create a draughty feeling in the occupied zone.

(2) The percentage dissatisfied increased from 24% to 29% with the ventilation scheme VS2. Contrary to the prediction that most people feel discomfort with a cool but draughty environment, it was suggested [22] that local people would prefer 'draughty' surroundings at a higher room temperature.

(3) Both normalised age of air and local ventilation rate were improved with VS2 because the air diffusers were closer to the occupied zone than with the ventilation scheme VS1.

From the above study of macroscopic parameters, ventilation scheme VS2 proved better than VS1 with respect to its effectiveness. This, however, is because of the thermal environment of the room and the fact that the system is used for cooling. As a heating system, it would probably give a completely different result since the air draught would then be considered as an unwanted form of cooling. As a state of uneven ventilation within the occupied space contributes to the acceptance of a ventilation scheme, a more enhanced spatial analysis of ventilation parameters using computer simulation is required for a better understanding of the system.

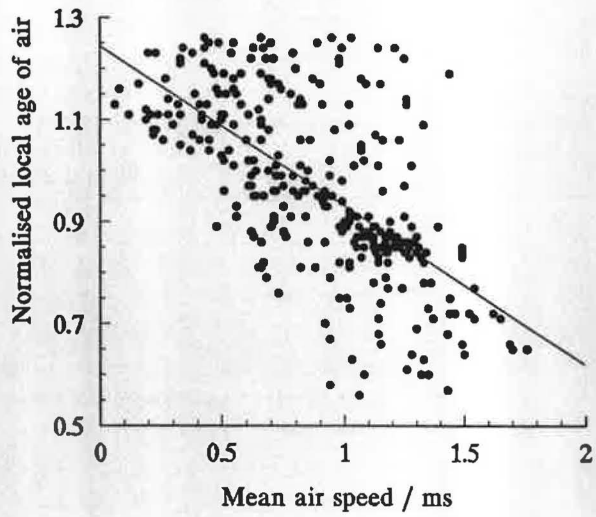


**Fig. 3.** Predicted correlation between mean age of air with mean air speed for ventilation scheme VS1.

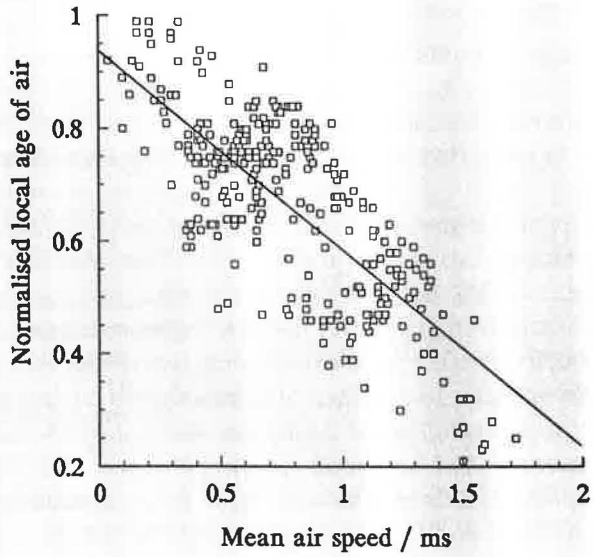


**Fig. 4.** Predicted correlation between mean age of air with mean air speed for ventilation scheme VS2.

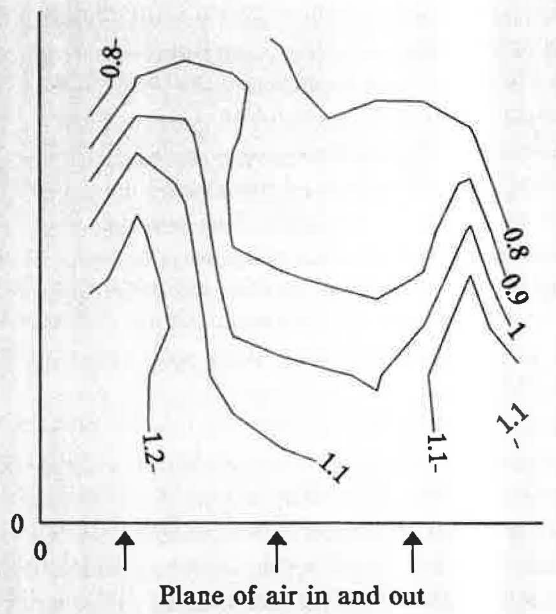


**Fig. 5.** Mean age of air contour at the occupied zone for ventilation scheme VS1.

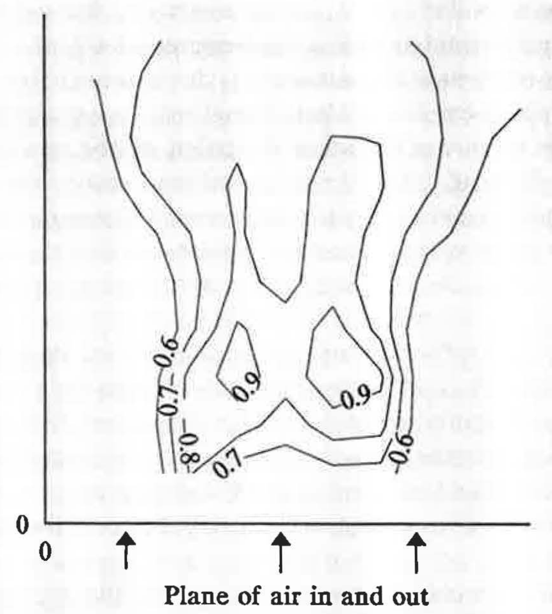


**Fig. 6.** Mean age of air contour at the occupied zone for ventilation scheme VS2.

[redacted]

## Numerical Experiments

The air flow pattern and temperature in the waiting halls are simulated by computational fluid dynamics (CFD) [23]. This is a self developed software and described in detail by Chow and Fung [24]. The assumptions made in the simulation are:

(1) The mean time-averaged values of the air flow variables, including the air velocity components and temperature, are predicted.

(2) The two-equation k-ε turbulence model is used to describe the turbulence effect and model the higher order fluctuation term.

(3) The set of partial differential equations describing conservation of momentum and enthalpy is converted to a quasi-linear form by expressing the flow variables at a point in terms of its neighbouring variables using the power law scheme.

(4) A Cartesian co-ordinate system (x,y,z) is used with the y-axis designated as the vertical direction.

(5) The SIMPLER [25] is used for solving the velocity-pressure-linked equations (in linear form).

After predicting the flow field, the local age of air can be predicted. Correlation relationships can be derived between the local age of air with the local mean air speed. The mean air speed $u_m$ and the mean age of air $T_m$ are plotted in figures 3–4. It can be seen that the mean age of air increased as the mean air speed decreased. The following linear correlation relationship can be derived from the results with a correlation coefficient of 0.647 for ventilation scheme VS1:

$$T_m = 1.244 - 0.313\ u_m. \tag{8}$$

Similarly, for ventilation scheme VS2, the following line was calculated from the data with a correlation coefficient of 0.761:

$$T_m = 0.939 - 0.352\ u_m. \tag{9}$$

Comparing the two ventilation schemes, a better correlation was found for scheme VS2 with an air-conditioning system although using a higher value of mean air speed may not necessarily improve the age of air. Contour diagrams of the mean age of air for the hall on a typical horizontal plane cutting across the occupied zone for VS1, with ventilation system only, and for VS2, with the air-conditioning system, are shown in figures 5 and 6, respectively.

The figures indicate that, with VS2, there is a general improvement in the mean age of air in the vicinity of air diffusers – a direct benefit of locating air diffusers near the occupied zone. A knowledge of the spatial distribution of such ventilation parameters as the age of air is important to help identify areas which may have an air quality problem and so aid the subsequent detailed design of the air distribution system.

## Conclusion

A review of the different parameters used to describe ventilation effectiveness has been conducted. At present, there is no well-accepted and straightforward method to determine the effectiveness of a given ventilation regime. The choice of a particular analytical approach will very much depend on the nature of the application and the type of analysis to be adopted. In this study of the ventilation effectiveness of the waiting hall in a local railway station, field measurements and numerical simulation were used to determine the ventilation parameters needed to describe the thermal environment of the occupied space. The mean air speed, percentage dissatisfied with air draught, age of air and local ventilation rate in the occupied zone were evaluated for two alternative ventilation design schemes. It was concluded that a ventilation scheme (VS2) with partial air-conditioning offers a better environment than one without (VS1). The study illustrates how the effectiveness of a given ventilation regime can be analysed in a systematic manner using experimental or computational techniques.

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