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# Measurements and Computations of Ventilation Efficiency and Temperature Efficiency in a Ventilated Room

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ABSTRACT

*In order to improve the indoor air quality in a room and to save energy, the air movement and contamination distributions in the room with ventilation have been studied experimentally and numerically. The experiment is carried out in a full-scale climate room with different air supply systems, heat gains from the venetian blinds and ventilation rates. The measurements concern room airflow patterns and air temperature, velocity and contamination concentration fields, etc. The airflow computer program PHOENICS and the cooling load program ACCURACY have been applied for the numerical simulations. PHOENICS solves the conservation equations of air mass, momentum, energy, concentration, kinetic energy and dissipation rate of kinetic energy. ACCURACY, which considers the influence of room air temperature distributions, is employed for the determination of cooling load, wall surface temperatures and convective heat transfer on room enclosure surfaces. These are the boundary conditions required by PHOENICS.*

*The agreements between the computations and the measurements are good. The ventilation efficiency and temperature efficiency which are used for evaluation of indoor air quality and energy consumption are reported for each case. Additional application of these computations to annual energy analysis is also discussed.*

**Key words: Indoor air quality, air distribution, energy conservation, ventilation, measurement, computation.**

## INTRODUCTION

Since the energy crisis of the 1970s, the insulation of buildings has been improved to reduce heat loss in winter, heat gain in summer and infiltration. As a consequence, the air-conditioning load in the buildings is smaller, and therefore, the ventilation required has been reduced. However, the reduction of the ventilation rate causes an increase of the concentration of indoor air contaminants. This is one of the reasons that the problem of indoor air quality in office buildings has received much attention from ventilation engineers over the past few years. Many contributions concerning indoor air quality have been presented [1 - 3], and one of the practical control methods presently used is by space air distribution [1].

Due to the decrease of mechanical ventilation, the airflow in office buildings has changed from a forced into a mixed convection. Buoyancy becomes one of the dominant factors of indoor airflow. In most cases, there is a temperature gradient in a room. If the outlets of a room are at different heights, the air extracted is of different temperatures. Because air-conditioning load and inlet air temperature are related, a high location of the outlets requires a smaller amount of supply air in the cooling period. If the inlet air is supplied directly into the occupied zone, the average room air temperature can be higher and this will reduce the cooling load. Therefore, the inlet and outlet locations of a room have a large influence on building energy consumption.

Our aim is to study airflow, contaminant concentration distributions and temperature

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fields in a room in order to find reliable methods both to control the pollutant dispersion and to use energy more efficiently. This research is carried out by experimental and numerical methods. Three different kinds of air terminal devices which can be located at different positions under different kinds of heat gain from the venetian blinds and under various ventilation rates are used for the investigation.

The second aim of this research is to assess the agreement between measurements and computations with an airflow program and a cooling load program.

## EXPERIMENTAL SETUP

The experiments were carried out in a full-scale climate room which is 5.6 m long, 3.0 m wide and 3.2 m high. In this room, four kinds of ventilation systems were used as shown in Fig. 1.

— *System 1:* an inlet air diffuser 68 cm high and 50 cm wide was located on the floor near the rear wall. Two outlets were in the rear wall near the ceiling. A table 175 cm long, 145 cm wide and 85 cm high was placed near the window.

— *System 2:* the air diffuser was installed on the floor near the window and there were two outlets in the rear wall near the ceiling. In this system, the table cannot be placed near the window otherwise the inlet air is confined under the table.

— *System 3:* two vertical jets were installed on the floor near the window. The inlet size for both of them was 100 cm long and 1 cm wide. Two outlets were in the rear wall near the ceiling. The table was located near the window.

— *System 4:* two horizontal jets in the upper part of the rear wall were used. The inlet size of each was 25 cm long and 2 cm high. Two outlets were in the rear wall near the floor.

The venetian blinds near the window can be electrically heated uniformly in order to simulate solar radiation. The floor surface temperature of the space above the room was controlled to be the same as that of the room and the ceiling temperature of the space below to be the same as that of the room. The total heat exchange through the side walls and the rear wall was controlled to be zero. The air temperature outside the window was set at 23 ℃. The sensor for the control of the room air temperature was placed in the middle of the occupied zone ($x$ = 2.8 m, $y$ = 1.5 m and $z$ = 0.9 m). The set point of the sensor was 23 ℃. In order to simulate a contaminant, a constant helium source combined with 25 W of heat was introduced at the point $x$ = 3.7 m, $y$ = 0.7 m and $z$ = 1.1 m. The constant flow of helium, 0.5% of the total ventilation rate, is supplied for a period of one hour.

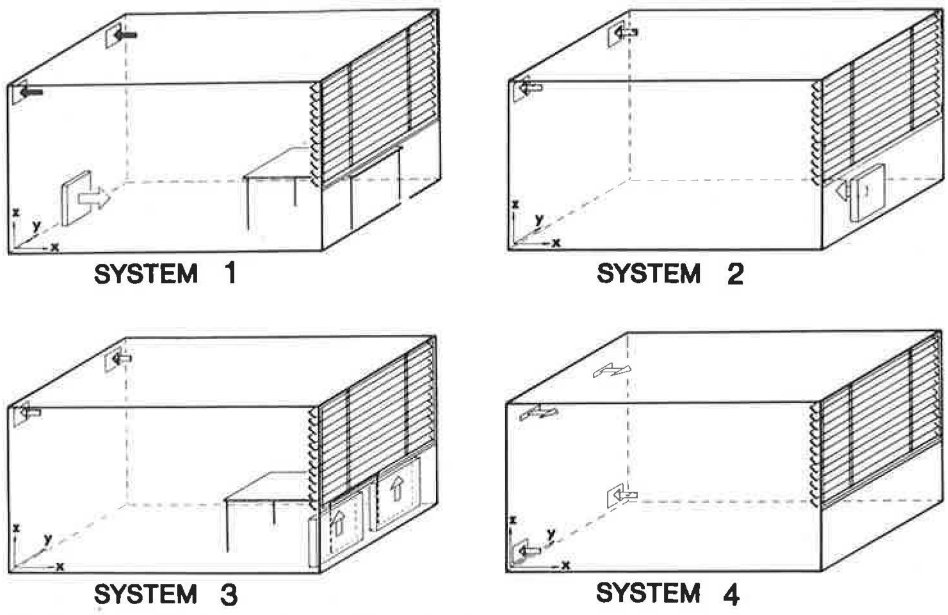


Fig. 1. The ventilation systems of the climate room.

The measurements concern airflow patterns, air temperature, velocity and helium concentration fields, enclosure surface temperatures, inlet mass flow, and inlet and outlet air temperatures. The airflow patterns are observed by using smoke. The velocities are measured through hot-wire anemometers and the temperatures by thermocouples, all of them connected to a data logger. The anemometers and thermocouples are positioned in the middle section of the room ($y = 1.5$ m) as shown in Figs. 2(A) and 2(B). The helium concentration is measured by a portable gas monitor and its sampling tubes are placed in the positions $y = 0.7$ m as shown in Fig. 2(C).

## THEORETICAL BASIS

*Brief description of the computer program PHOENICS*

The computer program PHOENICS is an airflow program developed by Rosten and Spalding [4], etc. It is used for the calculations of the distributions of room air velocity, temperature and helium concentration. The computation method involves the solution, in finite-domain form, of three-dimensional equations for the conservation of mass, momentum, energy, concentration, turbulence energy and dissipation rate of kinetic energy, with wall function expressions for solid boundary conditions. The governing equations can be expressed in a single form:

$$\underbrace{\mathrm{div}(\rho \vec{V}\phi - \Gamma_\phi \,\mathrm{grad}\,\phi)}_{\text{convection + diffusion}}$$

$$= S_{\mathrm{Buoyancy}} + S_\phi$$

$$= \text{buoyancy source} + \text{other sources}$$

where $\phi$ stands for any one of the following: 1, $u$, $v$, $w$, $k$, $\epsilon$, $H$ and $C$. When $\phi = 1$, the equation changes into the continuity equation. A complete description of the theoretical basis can be found in refs. 4 - 6. The wall function used here has been modified by the authors in order to obtain better agreement on heat exchange through enclosure surfaces between computations and measurements.

*Brief description of the computer program ACCURACY*

In order to determine the boundary conditions for the airflow program PHOENICS, such as wall surface temperatures, inlet air temperatures, etc., the cooling load program ACCURACY is used [7]. ACCURACY is the acronym of A Cooling-load Code Using Room Air Currents (the final Y is added for euphony). It is based on the room energy balance equation method and can consider the influence of room air temperature distribution. It

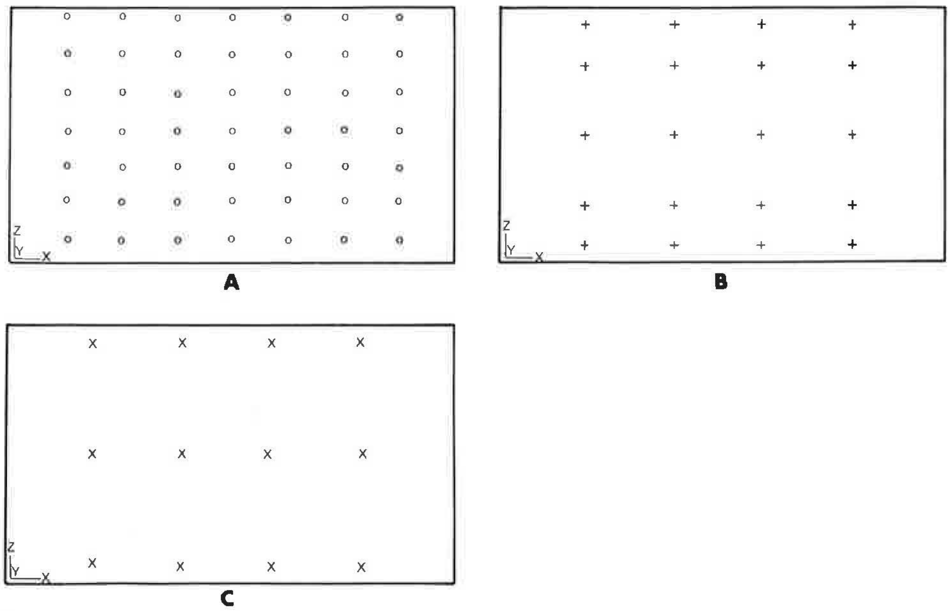


Fig. 2. Measuring positions. (A) Anemometers in the section $y = 1.5$ m; (B) thermocouples in the section $y = 1.5$ m; (C) concentration sampling points in the section $y = 0.7$ m.

can also be applied for the determination of transient behaviour of concentration from the airflow patterns.

The main difference between the program ACCURACY and other cooling-load programs based on the room energy balance equations is that the former considers transient temperature differences between the air near the inside surfaces of an enclosure and the middle point of the room as shown in Fig. 3(A). These temperature differences can be obtained from the temperature distributions which are updated hourly from an airflow computer program, such as PHOENICS. However, it is too expensive to get time-dependent airflow and temperature distributions of a room from the airflow computer program based on the $k$–$\epsilon$ turbulence model. This is due to the properties of the fluid and turbulence-governing equations as well as the numerical method which requires a large amount of grid numbers to give a good prediction. If a few typical airflow patterns of the room are precalculated and stored in a disk, hourly room air temperature distributions can be determined from the corresponding airflow pattern as shown in Fig. 3(B). This is possible because when the boundary conditions vary in a certain range, the airflow in a room changes very little and temperature fields have a large change. The latter method of calculations is much cheaper, and therefore, it is employed in ACCURACY for the determination of air-conditioning load and room air temperature distributions. A more detailed description of the computer program ACCURACY is given in ref. 7.

ACCURACY calculates the radiative heat exchange between the room enclosure surfaces, the transient heat conduction into those surfaces and determines the enclosure surface temperatures and air-conditioning loads. The convective heat exchange between the venetian blinds and the room air can be calculated in the cooling load program ACCURACY and is inserted in the airflow program PHOENICS as a thermal source near the window.

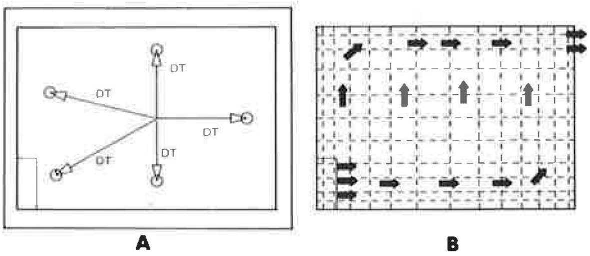


Fig. 3. Program ACCURACY uses airflow patterns. (A) Temperature differences are considered in the program; (B) the temperature differences are calculated from airflow patterns.

The transient behaviour of concentration can also be determined in ACCURACY from the airflow field precalculated, because direct determination from PHOENICS is too expensive.

## RESULTS

In this Section, the experimental and computational results of room airflow, temperature and contamination distributions will be reported in three groups:

(1) different ventilating systems;

(2) different ventilation rates;

(3) different heat gains from the venetian blinds.

In order to evaluate objectively the effects of ventilation and its influence on indoor air quality and energy saving, two technical terms, ventilation efficiency ($\eta_V$) and temperature efficiency ($\eta_T$), are introduced. They are defined as:

$$\eta_V = \frac{C_{out} - C_{in}}{C_{occu} - C_{in}}$$

and

$$\eta_T = \frac{T_{out} - T_{in}}{T_{occu} - T_{in}}$$

where $C_{occu}$ is the air contaminant concentration in an occupied zone, and $T_{occu}$ is the air temperature in an occupied zone. In this paper, $C_{occu}$ is the air contaminant concentration at the point $x = 2.2$ m, $y = 0.7$ m and $z = 1.6$ m and $T_{occu}$ is the air temperature at the point $x = 2.2$ m, $y = 1.5$ m and $z = 1.6$ m because the measurements presented are carried out only at a few points. It is not possible to obtain a precise average value of the zone of occupation from the measurements.

When room air is perfectly mixed, $\eta_V = 1$ and $\eta_T = 1$. For well-organized room air velocity distributions, $\eta_V$ and $\eta_T$ have values higher than one in cooling situations.

*Different ventilating systems*

In this group, the mechanical ventilation rate was controlled at five air changes per hour (ach) (0.075 $m^3/s$) for all the four systems. The heat introduced on the venetian blinds was 600 W for systems 1 and 2, and 950 W for systems 3 and 4. The higher the heat on the venetian blinds, the lower the inlet air temperature required. Therefore, the 600 W heat was chosen for systems 1 and 2 to prevent the inlet air from causing draughts in the occupied zone because it was supplied directly into the lower part of the room through the air diffuser. However, for systems 3 and 4, where the inlet jets were used to supply air outside the occupied zone, the induction by the jets increased the inlet air temperature and made it possible to use a lower inlet air temperature. This is the reason that the heat on the venetian blinds was controlled at 950 W for these two systems.

The airflow, temperature and helium concentration distributions acquired from the computations and measurements of system 1 are shown in Fig. 4. The temperature efficiency and the ventilation efficiency are given in Table 1. The agreement between the computations and the measurements is good although there are some discrepancies in the temperature and concentration fields. The difference between the computations and the measurements in most cases occurs in the place where the variation in gradient is large. This means that a slight difference in the measuring location can cause a significant error. From Table 1, it can be calculated that

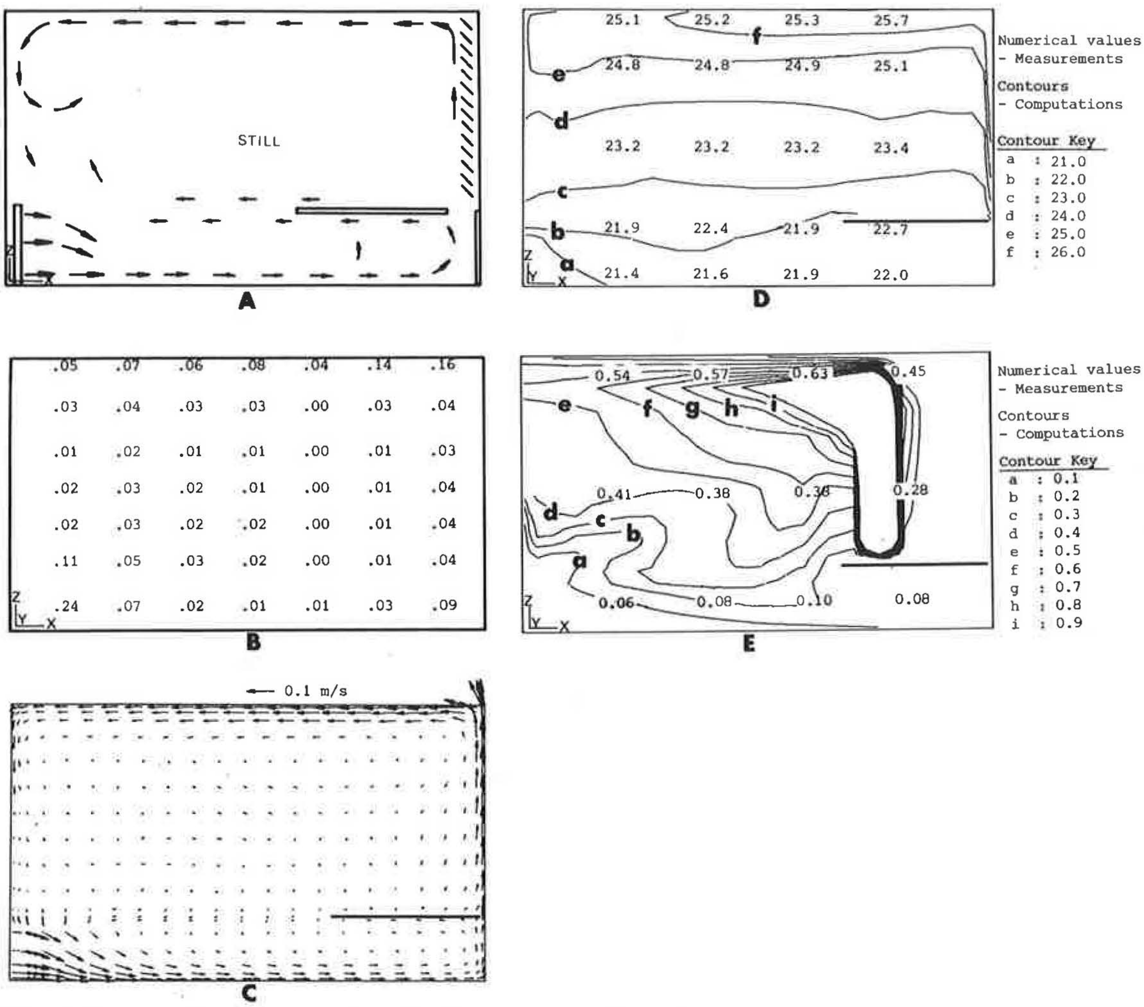


Fig. 4. Computed and measured results for system 1. (A) Airflow pattern observed by using smoke in the section $y = 1.5$ m; (B) measured air velocities (m/s) in the section $y = 1.5$ m; (C) computed air velocity distribution in the section $y = 1.5$ m; (D) computed and measured temperature field in the section $y = 1.5$ m (°C); (E) computed and measured concentration field in the section $y = 0.7$ m (%).

TABLE 1

Measured and computed ventilation and temperature efficiencies with different ventilation systems*

| Systems | Heat on venetian blinds (W) | $T_{in}$ (°C) | $T_{out}$ (°C) | $T_{occu}$ (°C) | $\eta_T$ | Temp. grad. (°C/m) | $C_{in}$ (%) | $C_{out}$ (%) | $C_{occu}$ (%) | $\eta_V$ |
|---|---|---|---|---|---|---|---|---|---|---|
| 1 (meas.) | 600 | 19.7 | 25.0 | 23.2 | 1.51 | 1.20 | 0.0 | 0.51 | 0.38 | 1.34 |
| 1 (comp.) | 600 | 19.1 | 25.0 | 23.4 | 1.37 | 1.53 | 0.0 | 0.50 | 0.41 | 1.22 |
| 2 (meas.) | 600 | 19.3 | 25.0 | 23.2 | 1.46 | 1.53 | 0.0 | 0.50 | 0.40 | 1.25 |
| 2 (comp.) | 600 | 18.7 | 24.8 | 23.2 | 1.36 | 1.33 | 0.0 | 0.51 | 0.40 | 1.02 |
| 3 (meas.) | 950 | 15.2 | 24.9 | 22.9 | 1.26 | 1.41 | 0.0 | 0.52 | 0.11 | 4.73 |
| 3 (comp.) | 950 | 14.9 | 24.9 | 22.9 | 1.25 | 1.32 | 0.0 | 0.52 | 0.13 | 4.00 |
| 4 (meas.) | 950 | 12.5 | 22.8 | 22.9 | 0.99 | −0.07 | 0.0 | 0.50 | 0.78 | 0.64 |
| 4 (comp.) | 950 | 12.6 | 22.9 | 22.9 | 1.00 | 0.04 | — | — | — | — |

*Ventilation rate is 5 ach.

the heat extracted from the room was smaller than the heat supplied on the venetian blinds because a part of the heat was transferred outside of the window. Figure 4 indicates that the inlet discharges the fresh air horizontally across the floor and that the air velocity near the ceiling is large due to the buoyancy effect from the hot venetian blinds. In the remaining part of the room, there is a large stagnant zone and the ventilation efficiency in this case is high (refer to Table 1). The reason for this is that the buoyancy effect of the 25 W heat source causes the helium to rise immediately upwards to the ceiling. As a result, the helium concentration near the ceiling is very high and it is removed directly through the outlets. This phenomenon is explained by the computed velocity and temperature fields in the section $y = 0.7$ m which is shown in Fig. 5. The concentration distributions in the sections $y = 0.7$ m (where the helium is introduced), $y = 1.5$ m and $y = 2.3$ m are shown in Fig. 6. The required inlet air temperature of simulation shown in Table 1 is lower than that of the measurements. The possible reason for this is that the measurements are not carried out under a real steady situation or it could be due to the discrepancies caused by the computation. Because the ceiling and the floor of the room are of heavy concrete of which the heat capacity is very large, a very long time is needed to reach steady conditions. If it is in a dynamic period, the heat transferred into the ceiling is larger than that obtained from the floor, and therefore, the measured inlet air temperature is lower.

As can be seen from the results above, the airflow in the room is very complex and a hot-wire probe can be used only for measurements near the inlets, where the velocity is relative high. The measurement of air velocity below 0.05 m/s with the hot-wire anemometers is very difficult because of the calibration of the probes and the impact of the free convection on the heat transfer from the probes.

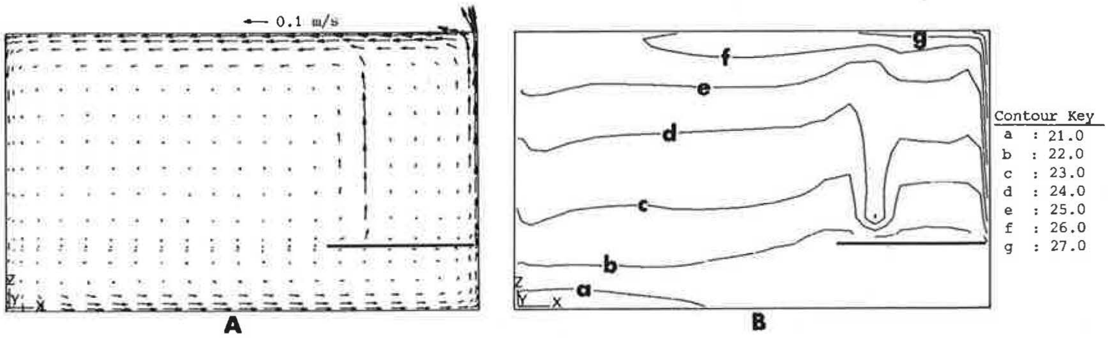


Fig. 5. Computed results of system 1 in the section $y = 0.7$ m (A) Air velocity distribution (m/s); (B) temperature field (°C).

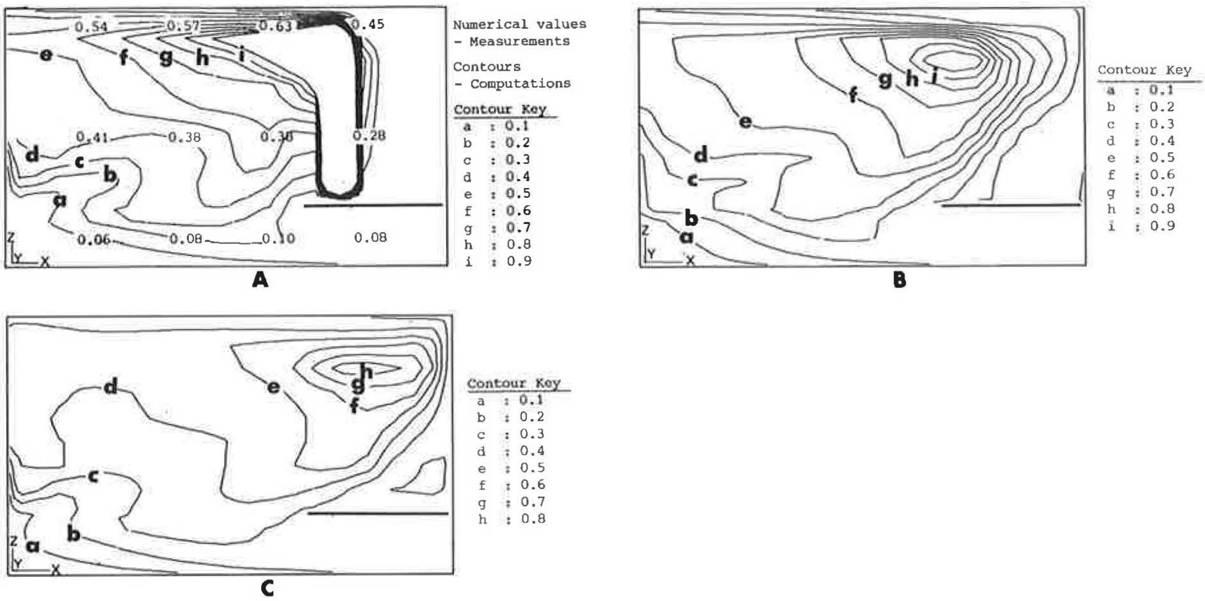


Fig. 6. Computed and measured concentration distributions in different sections for system 1. (A) Computation and measurement in the section $y$ = 0.7 m (%); (B) computation in the section $y$ = 1.5 m (%); (C) computation in the section $y$ = 2.3 m (%).

In system 2, the numerical and experimental results which are presented in Fig. 7 show two large eddies in the room. One is formed by the air discharge from the inlet air diffuser and the other from the buoyancy produced by the hot window. The measured results indicate that both the hot air near the ceiling and the cold air near the floor can reach to the rear wall so that the two eddies are larger than those of computations. The temperature efficiency, temperature gradient and ventilation efficiency of this system are almost the same as those of system 1. Although the fresh air keeps the lower part of the room very clean, the concentration at standing level ($z$ = 1.6 m) is nearly the same as that in the upper part of the room. For this reason, system 1 is better for acquiring good air quality.

The measured and simulated field results of system 3 are shown in Fig. 8. The air velocity presented in the Figure is that in the section $y$ = 0.7 m in which the helium is introduced. However, the temperature and concentration fields shown in the Figure are those in the middle section of the room where the measured results are available. The induction by the inlet air increases the inlet air temperature and will not cause a draught after it falls back into the occupied zone of the room. The ventilation efficiency is very high as shown in Table 1 and in Fig. 8, although such a system is not very commonly used for cooling. However, the airflow pattern of this system is very sensitive to the inlet air temperature, inlet dimensions and ventilation rate. If the ventilation rate, for instance, is 7 ach, the ventilation efficiency can drop to 0.79 because the inlet air brings the contaminant directly into the occupied zone. If the opening of the inlet is too large or, in other words, the inlet air velocity is too low, the cold air will drop down directly to the occupied zone and has no induction effect. This will cause a draught.

System 4 is a very common one. Due to its outlets on the lower part of the room, the system is very bad both for saving energy and obtaining good indoor air quality in summer. This can be explained from the ventilation efficiency and temperature efficiency shown in Table 1. The room air mixture is very good, therefore the air temperature gradient is very small (see Table 1 and Fig. 9). The computed concentration distribution is not available. The calculated air velocity fields and temperature distributions in the section of inlet located and in the middle section are presented in Fig. 9. The simulated room air temperatures in the section $y$ = 1.5 mostly are larger than 22.0 ℃ and smaller than 23.0 ℃ so that no contour for 22.0 ℃ can be drawn in the Figure. They are in good agreement with the measured ones.

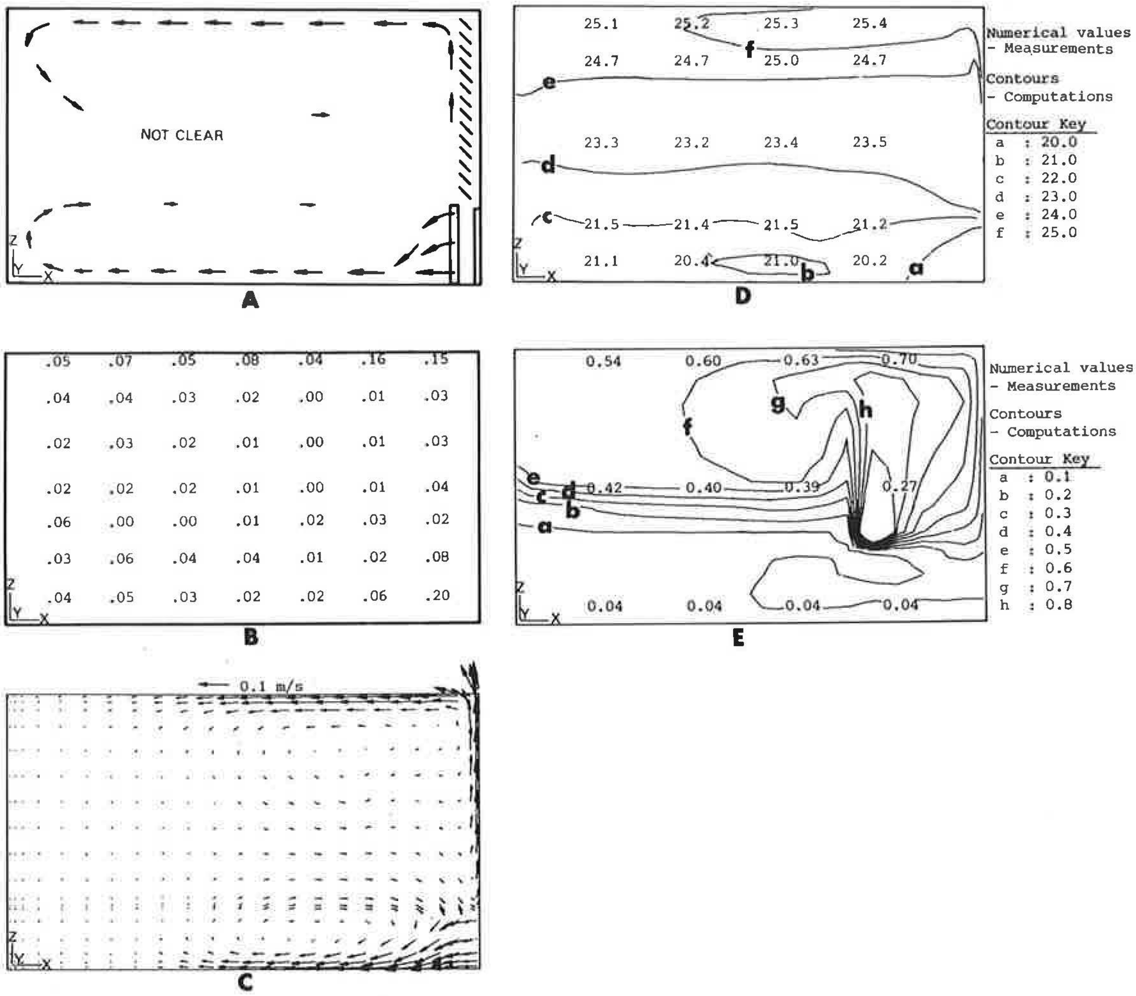


Fig. 7. Computed and measured results for system 2. (A) Airflow pattern observed by using smoke in the section $y = 1.5$ m; (B) measured air velocities (m/s) in the section $y = 1.5$ m; (C) computed air velocity distribution in the section $y = 1.5$ m; (D) computed and measured temperature field in the section $y = 1.5$ m (°C); (E) computed and measured concentration field in the section $y = 0.7$ m (%).

Among these four systems, system 1 seems the best one both for saving energy and obtaining good indoor air quality in cooling situations. For system 2, a table placed in the middle near the window is not suitable even with a distance of 1.0 m to the window. When the table is placed in this position, the air temperature under the table is less than 20 °C. This is not permitted. Although system 3 presents the best ventilation effect, it is too sensitive to the inlet air temperature, ventilation rate and inlet geometry. This will cause difficulties in control. The airflow pattern of the system is more complicated and is not easily observed by using smoke. System 4 is not good either for energy saving or better air quality in summer. The situations discussed here are concerned with a dominant heat gain from the window. If inner heat sources in a room are dominant, the results can be different. More research on this subject will be done in the future.

*Different ventilation rates*

System 1 was used to study the influence of ventilation rate on the ventilation efficiency and temperature efficiency. The heat introduced on the venetian blinds for this group was fixed at 600 W. The study concerned the airflow, temperature fields and helium concentration distributions when mechanical ventilation rates were controlled at 3 ach, 5 ach and 7 ach.

The computed results presented in Table 2 implied that the larger the ventilation rate, the higher the temperature efficiency and the

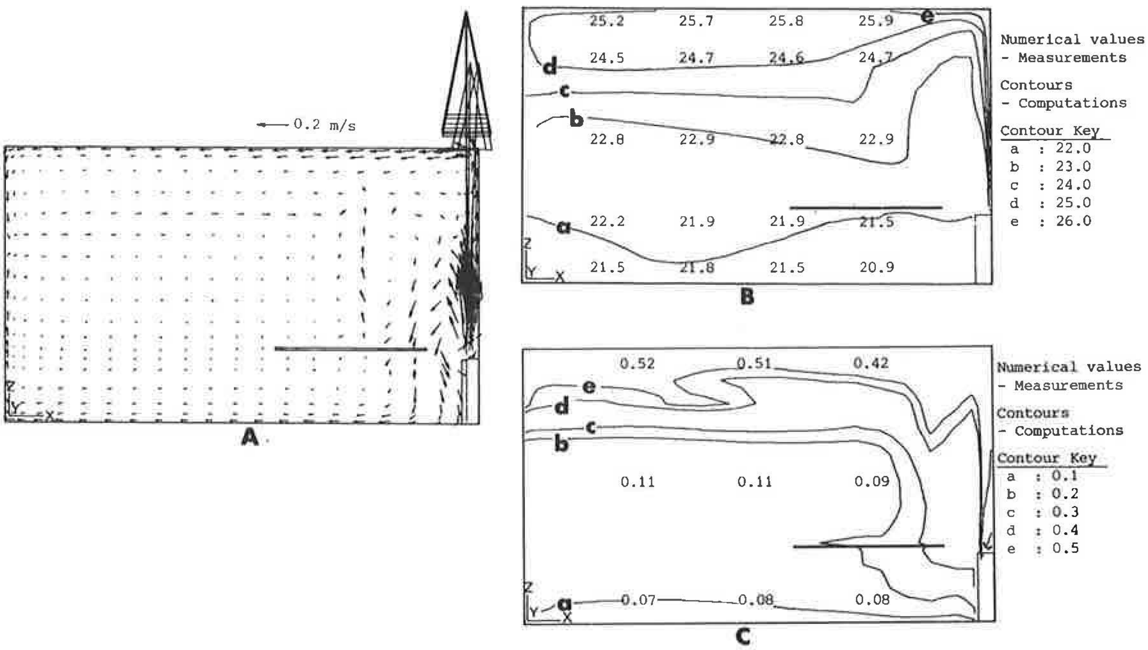


Fig. 8. Computed and measured results for system 3. (A) Computed air velocity distribution in the section $y$ = 0.7 m; (B) computed and measured temperature field in the section $y$ = 1.5 m (°C); (C) computed and measured concentration field in the section $y$ = 1.5 m (%).

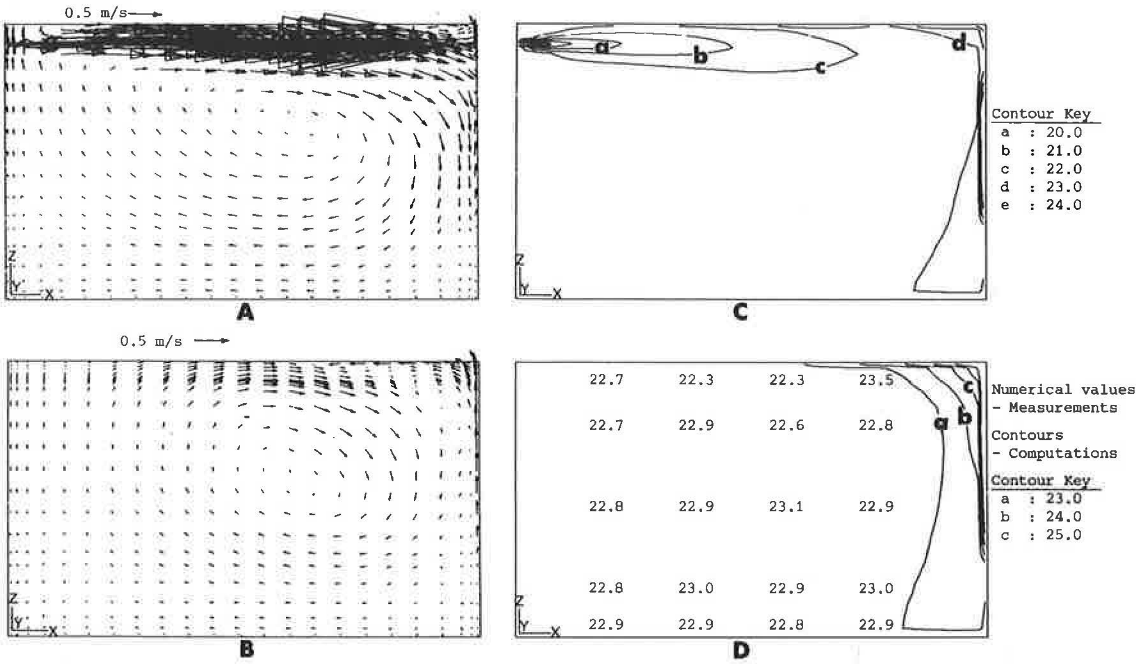


Fig. 9. Computed and measured results for system 4. (A) Computed air velocity distribution in the section $y$ = 0.75 m (through the inlet); (B) computed air velocity distribution in the section $y$ = 1.5 m; (C) computed temperature field in the section $y$ = 0.75 m (°C); (D) computed and measured temperature field in the section $y$ = 1.5 m (°C).

ventilation efficiency and the smaller the temperature gradient. When room ventilation rate increases, the required inlet air temperature increases in cooling situations. According to the definition of temperature efficiency, both $T_{\text{out}} - T_{\text{in}}$ and $T_{\text{occu}} - T_{\text{in}}$ become smaller. However, the change rate of $T_{\text{occu}} - T_{\text{in}}$ is much larger, therefore $\eta_T$ is higher. But this

TABLE 2

Measured and computed ventilation and temperature efficiencies for system 1 with different ventilation rates*

| Ventilation rates (ach) | $T_{in}$ (°C) | $T_{out}$ (°C) | $T_{occu}$ (°C) | $\eta_T$ | Temp. grad. (°C/m) | $C_{in}$ (%) | $C_{out}$ (%) | $C_{occu}$ (%) | $\eta_V$ |
|---|---|---|---|---|---|---|---|---|---|
| 3 (meas.) | 16.0 | 25.6 | 23.8 | 1.23 | 1.68 | 0.0 | 0.50 | 0.39 | 1.28 |
| 3 (comp.) | 16.0 | 25.8 | 23.5 | 1.31 | 1.67 | 0.0 | 0.49 | 0.44 | 1.11 |
| 5 (meas.) | 19.7 | 25.0 | 23.2 | 1.51 | 1.20 | 0.0 | 0.51 | 0.38 | 1.34 |
| 5 (comp.) | 19.1 | 25.0 | 23.4 | 1.37 | 1.53 | 0.0 | 0.50 | 0.41 | 1.22 |
| 7 (meas.) | 20.8 | 24.8 | 22.9 | 1.90 | 1.23 | 0.0 | 0.50 | 0.32 | 1.56 |
| 7 (comp.) | 20.7 | 24.9 | 23.4 | 1.56 | 1.27 | 0.0 | 0.48 | 0.28 | 1.71 |

*Heat on the venetian blinds is 600 W.

does not mean that it saves energy, since the energy consumption of a building also depends on the ventilation rate.

As shown in Fig. 10, the buoyancy influence in the upper part of the room is the same under these three ventilation rates because the heat gain from the venetian blinds is the same. However, the mechanical ventilation controls air distribution in the lower part of the room. The higher the ventilation rate, the more fresh air is injected into the occupied zone. Therefore, the concentration of contaminant in the zone of occupation is lower so that $\eta_V$ is higher. This can be seen from the concentration fields and the numerical air velocity distributions illustrated in Fig. 10.

Although measured results of velocity distributions are not presented in Fig. 10, they are the same as the computed ones.

The transient behaviour of the helium concentration of this group is presented in Fig. 11. The smaller the ventilation rate, the more difficult it is to remove the helium which was used as contaminant.

*Different heat gains from the venetian blinds*

In this group, the mechanical ventilation rate was fixed at 7 ach for system 1. The ventilation rate was higher than that used in practice because the major concern was to study the influence of heat gain. If a too small amount of heat was used, it would be very difficult to overcome the influence from other disturbances. In addition, the larger the amount of heat used, the higher the ventilation rate required to maintain comfort in the zone of occupation. The study concerned the air velocity, temperature and concentration of helium distributions when heating amounts of 300 W, 600 W and 950 W were applied to the ventilation blinds.

The computed and measured air temperature, concentration fields, and calculated airflow patterns are shown in Fig. 12. The measured velocities present the same results. The temperature efficiency and the ventilation efficiency are presented in Table 3. If the heat gain from the venetian blinds is larger, the required inlet air temperature is lower and the air movement in the upper part of the room is stronger. The stronger air movement forms a larger eddy. The inlet air, which is of lower temperature, stays in a thin layer on the floor and induces a certain amount of air from the room (see Fig. 12). Due to these two reasons, the mixture in the room is better. As seen in the results, the concentration of contaminant in the occupied zone increases so that $\eta_V$ is smaller.

According to the results presented in Table 3, the influence of the heat gain from the ventilation blinds on temperature efficiency is small.

## DISCUSSIONS AND FURTHER REMARKS

The computations using the airflow computer program PHOENICS are very expensive. In the situations discussed above, the CPU time on an IBM 3083 JX1 computer is about 80 minutes for one case if the calculation is started with uniform initial field values and a total grid number of $18 \times 18 \times 23$. When the computation uses the field results as initial values from a similar case, about 20 minutes CPU time is required. Nevertheless, it is still too expensive.

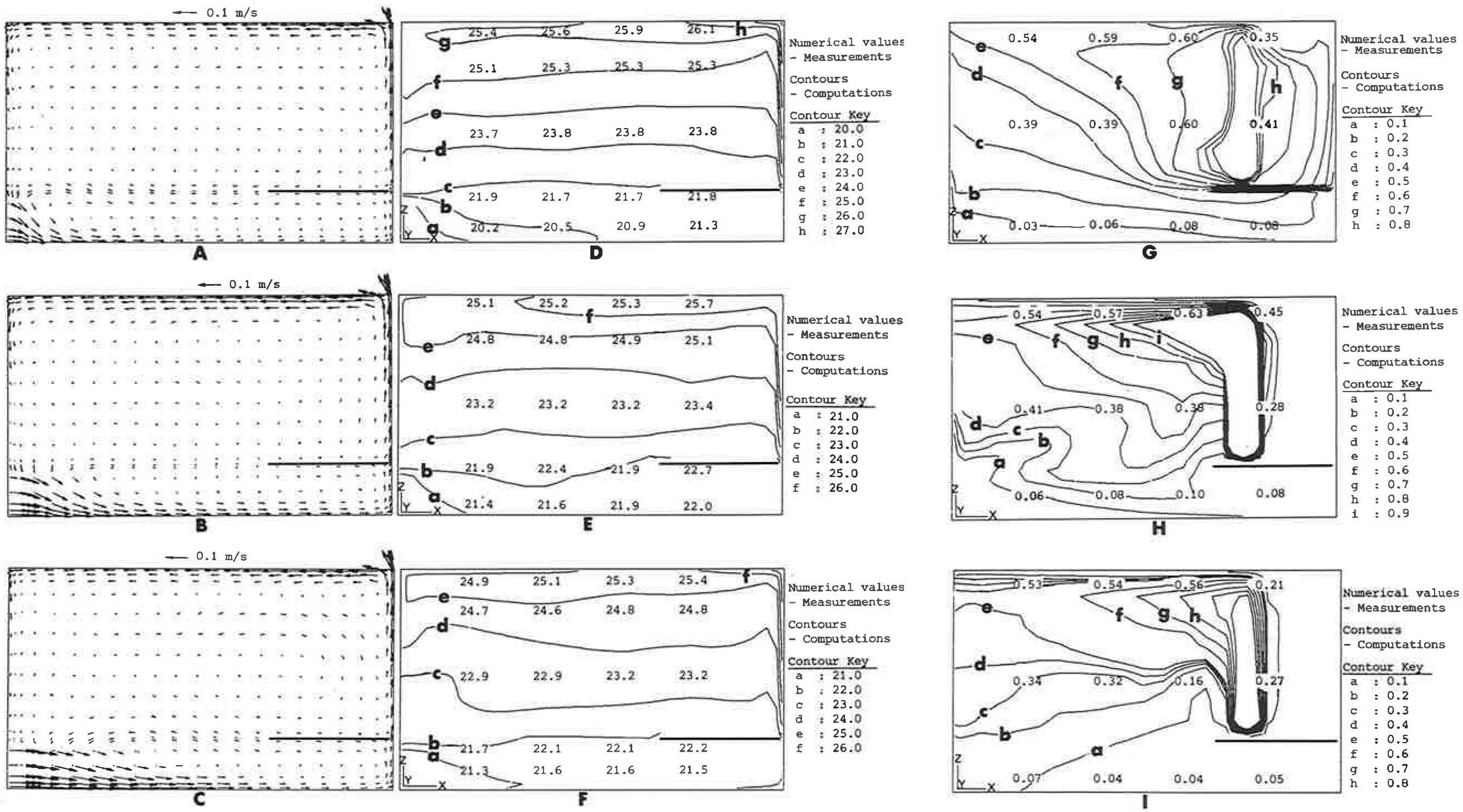


Fig. 10. Computed and measured results for system 1 with different ventilation rates and with 600 W gain from the venetian blinds. For (A), (B), and (C), the computed air velocity distributions in the section $y = 1.5$ m with ventilation rates 3, 5, and 7 ach, respectively; for (D), (E) and (F), the computed and measured temperature fields in the section $y = 1.5$ m with ventilation rates 3, 5, and 7 ach, respectively (°C); and for (G), (H) and (I), the computed and measured concentration fields in the section $y = 0.7$ m with ventilation rates 3, 5, and 7 ach, respectively (%).

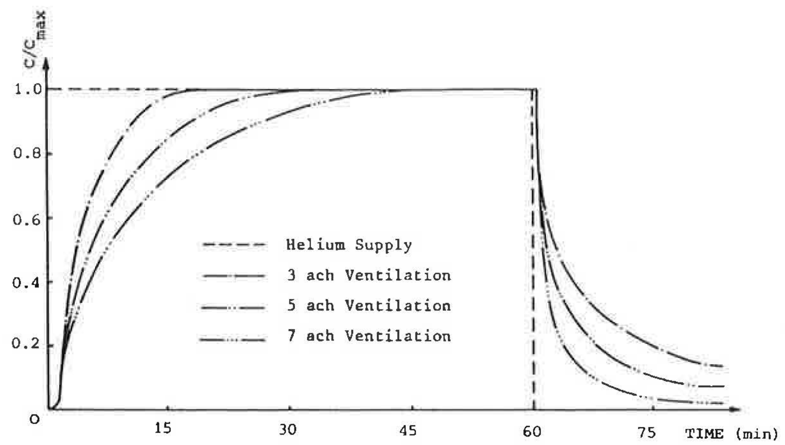


Fig. 11. Transient behaviour of the room helium concentration in the point ($x$ = 2.2 m, $y$ = 1.5 m and $z$ = 3.0 m) for system 1 with different ventilation rate.

TABLE 3

Measured and computed ventilation and temperature efficiencies for system 1 with different heat gains from the venetian blinds*

| Heat on the venetian blinds (W) | $T_{in}$ (°C) | $T_{out}$ (°C) | $T_{occu}$ (°C) | $\eta_T$ | Temp. grad. (°C/m) | $C_{in}$ (%) | $C_{out}$ (%) | $C_{occu}$ (%) | $\eta_V$ |
|---|---|---|---|---|---|---|---|---|---|
| 300 (meas.) | 21.3 | 24.1 | 23.6 | 1.22 | 0.68 | — | — | — | — |
| 300 (comp.) | 21.7 | 24.0 | 22.7 | 2.30 | 0.67 | 0.0 | 0.53 | 0.30 | 1.76 |
| 600 (meas.) | 20.8 | 24.8 | 22.9 | 1.90 | 1.23 | 0.0 | 0.50 | 0.32 | 1.56 |
| 600 (comp.) | 20.7 | 24.9 | 23.4 | 1.56 | 1.27 | 0.0 | 0.48 | 0.28 | 1.71 |
| 950 (meas.) | 19.6 | 27.1 | 23.8 | 1.79 | 1.72 | 0.0 | 0.50 | 0.38 | 1.32 |
| 950 (comp.) | 19.4 | 26.9 | 23.7 | 1.74 | 1.67 | 0.0 | 0.49 | 0.37 | 1.32 |

*Ventilation rate is 7 ach.

As it can be seen from the above results, the required inlet air temperatures, between two different systems but with the same heat gain and ventilation rate such as systems 1 and 4, are so large that this has to be considered in the energy analysis of the building. Normal cooling load programs which assume room air temperature to be uniform are not suitable for the energy analysis because they will present the same results such as inlet and outlet air temperature, air-conditioning load for all kinds of systems. Using the cooling load program ACCURACY, Chen and van der Kooi [7] have demonstrated the differences among a number of approximated methods used for considering the influence of room air temperature distributions on air-conditioning load and inlet air temperature required. These methods are:

(1) using fixed room air temperature gradients;

(2) assuming the room air temperature gradients are proportional to the cooling load;

(3) obtaining the room air temperature distributions from the room airflow patterns which are determined from several specific cases.

The third method gave the best results on the calculations of air-conditioning load and inlet air temperature required, as well as providing detailed hourly information on room airflow distributions, air temperature and contaminant concentration fields. The airflow patterns used were precalculated from the airflow program PHOENICS. The costs to determine the field results from ACCURACY are much cheaper than those calculated directly from PHOENICS especially when a transient problem is involved. In an hourly simulation for a one-month period, only four minutes CPU time was needed for ACCURACY. The cost of the computations for

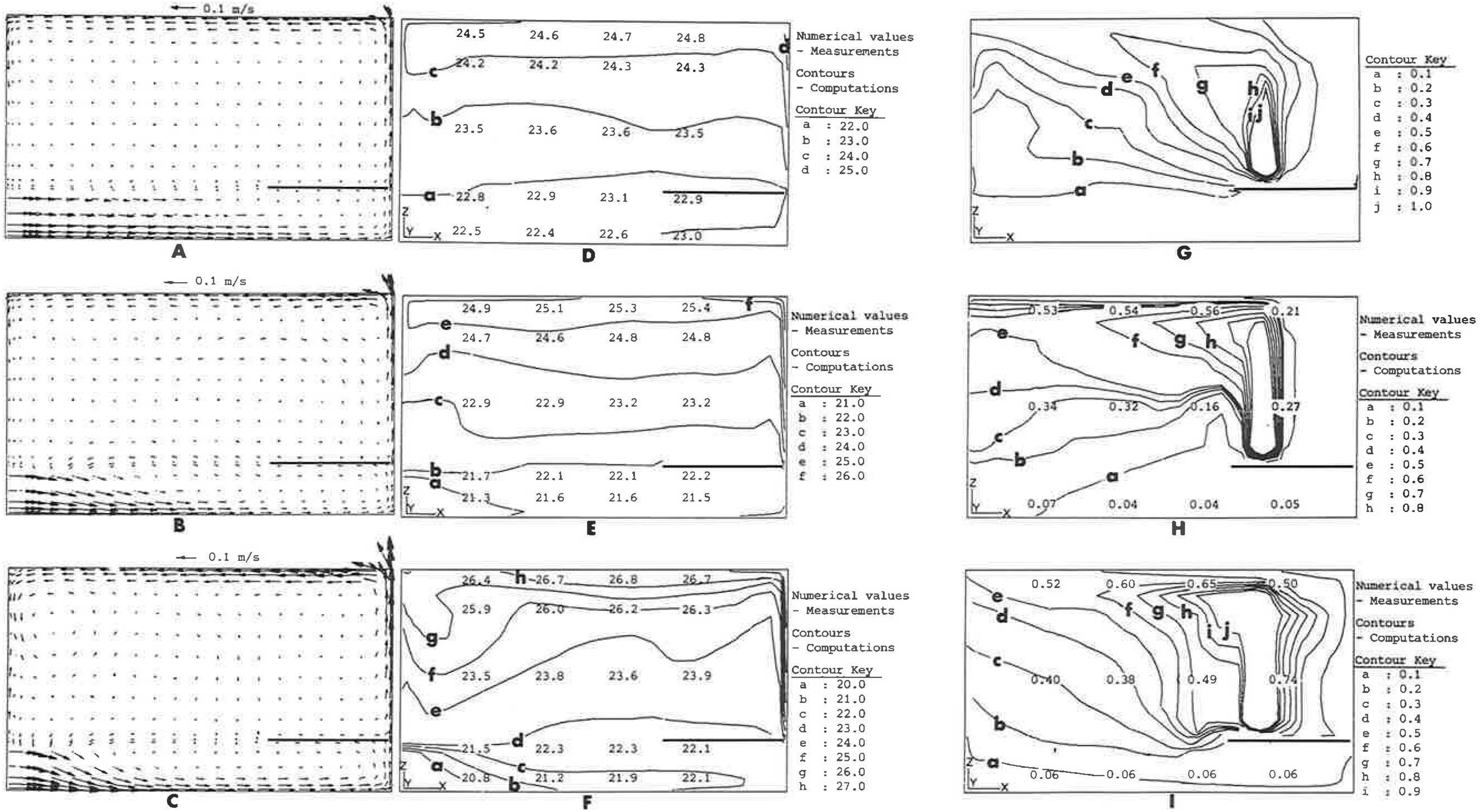


Fig. 12. Computed and measured results for system 1 with different heat gain from the venetian blinds and with ventilation rate 7 ach. For (A), (B), and (C), the computed air velocity distributions in the section $y = 1.5$ m with heat gain 300, 600, and 950 W, respectively; for (D), (E) and (F), the computed and measured temperature fields in the section $y = 1.5$ m with heat gain 300, 600, and 950 W, respectively (°C); and for (G), (H) and (I), the computed and measured concentration fields in the section $y = 0.7$ m with heat gain 300, 600, and 950 W, respectively (%).

obtaining the precalculated airflow patterns was excluded. Of course, the method used in ACCURACY was an approximated one, but nevertheless, it is precise enough for engineering purposes. However, ACCURACY is costly for the applications of annual hourly cooling load computations.

From the airflow computations for system 1 as discussed above, it is sufficient to use the airflow patterns which are calculated based on ventilation rates of 3, 5, 7 ach and heat gains of 300, 600, 950 W to approximate indoor air velocity distributions of the system for all cooling situations. The air temperature gradient of the room of the system ($\Delta T$) can be written as the function of the air-conditioning load ($Q$) and the ventilation rate (Vent), etc.

$$\Delta T = f(Q, \mathrm{Vent})$$

In an additional subroutine of the program ACCURACY, this function can be determined from the precalculated airflow patterns and control strategy. Different air-conditioning systems, such as fixed inlet air temperature system, fixed inlet airflow system, variable air volume system and their combinations, etc., can be easily handled. The result is as precise as that calculated by the third method. The cost is the same as that of the second method (2.5 times more expensive than the normal uniform temperature method because iteration is required in some steps). This cost is about 20 times cheaper than the third methods. Though, this method will not present hourly air temperature distribution and contaminant concentration field.

## CONCLUSIONS

The following paragraphs provide a summary of the more important conclusions which may be extracted from the present work:

(1) Most of the computations presented in this paper are in good agreement with the measured results. A few discrepancies were found.

(2) The temperature and ventilation efficiencies of systems 1 and 2 are very high. They seem to be the best ones both for saving energy and obtaining good air quality in cooling situations. However, system 2 is not as good as system 1. System 3 presents higher ventilation efficiency but is not easy to control. System 4 is a representative of commonly used systems where room air is perfectly mixed. Its temperature and ventilation efficiencies are very low.

(3) According to the results of system 1, the higher the ventilation rate, the larger the temperature efficiency and the ventilation efficiency and the smaller the temperature gradient. In addition, the energy consumption also depends on the ventilation rate. Obviously, it is easier to remove the contaminant in the case with a high ventilation rate.

(4) The larger the amount of heat introduced on the venetian blinds for system 1, the lower the ventilation efficiency and the larger the temperature gradient. Its influence on temperature efficiency is small.

(5) If room heat gain and ventilation rate are the same, the required inlet air temperature of the four systems is different. In cooling situations, the supplied inlet air temperature of systems 1 and 2 can be higher than that of system 4. This has to be considered in building energy analysis. The computer program ACCURACY can be employed for this purpose. A new calculating method is suggested.

It should be pointed out that the major results and conclusions of this paper are tentative. For example, ventilation efficiencies might be highly sensitive to the location of pollutant sources and the rate of heat generation at the sources or to physical characteristics of the room and its furnishing. Further research on the sensitivities seems to be necessary.

## LIST OF SYMBOLS

| | |
|---|---|
| $C$ | concentration ($m^3$ helium/$m^3$ air) (%) |
| $C_{in}$ | inlet air contaminant concentration ($m^3$ helium/$m^3$ air) (%) |
| $C_{occu}$ | air contaminant concentration in occupied zone; in this paper it is the air contaminant concentration at the point $x = 2.2$ m, $y = 0.7$ m and $z = 1.6$ m ($m^3$ helium/$m^3$ air) (%) |
| $C_{out}$ | outlet air contaminant concentration ($m^3$ helium/$m^3$ air) (%) |
| $H$ | specific enthalpy (J/kg) |
| $k$ | kinetic energy of turbulence (J/kg) |
| $Q$ | air-conditioning load (W) |

| | |
|---|---|
| $S_\phi$ | source term of general fluid property (—) |
| $T$ | temperature (℃) |
| $T_{in}$ | inlet air temperature (℃) |
| $T_{occu}$ | air temperature in occupied zone; in this paper it is the air temperature at the point $x = 2.2$ m, $y = 1.5$ m and $z = 1.6$ m (℃) |
| $T_{out}$ | outlet air temperature (℃) |
| $u$ | velocity component in $X$-direction (m/s) |
| $V$ | flow velocity (m/s) |
| Vent | ventilation rate of a room (ach) |
| $v$ | velocity component in $Y$-direction (m/s) |
| $w$ | velocity component in $Z$-direction (m/s) |
| $\Gamma$ | diffusive coefficient (N s/m$^2$) |
| $\epsilon$ | dissipation rate of turbulence energy (J/kg s) |
| $\eta_T$ | temperature efficiency (—) |
| $\eta_V$ | ventilation efficiency (—) |
| $\rho$ | air density (kg/m$^3$) |

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