Submitted to "BUILDING & ENVIRONMENT" #4156

INTERPOLATION THEORY AND INFLUENCE OF BOUNDARY CONDITIONS ON ROOM AIR DIFFUSION

QINGYAN CHEN*
ALFRED MOSER*
PETER SUTER*

This paper analyzes the errors caused by the interpolation from existing cases for assessing indoor air flow, air quality and thermal comfort in an office. A sensitivity study is then provided to determine the influence of several boundary conditions on indoor air diffusion. The research is conducted numerically by using a low-Reynolds-number k-ε model. It can be concluded that the interpolation errors caused by the variations of solar radiation, window size, heat source location due to lighting, and the surface temperatures of interior walls are small and can be quantitatively determined. But it is difficult to estimate the errors introduced by the variations of furniture location and size.

1. INTRODUCTION

CORRECT AIR diffusion, as well as the proper quantity of conditioned air, is essential for good air quality and comfortable conditions in forced ventilation systems. Room air movement is affected by the building's geometry, diffuser configuration and location, air velocity and direction at the diffuser, ventilation rate, exhaust location, internal obstructions, and thermal sources by occupancy and/or equipment. Detailed investigation on these parameters is therefore necessary.

During the last two decades, a large number of experimental and numerical results have been achieved for the prediction of room air motion. However, most of the studies were aimed at specific cases. It is difficult to gain sufficient information from publications for a specific design case. This is because the probability is so small to have two absolutely identical cases, and it is not easy to estimate the error caused by the difference between the specific design case and a case from publications. A new experiment or numerical computation is often required for a new design. A full-scale experiment is prohibitively expensive and it is impossible to develop an undistorted similitude model for room air motion when there is internal heat production within the room, because the Reynolds number (ratio of inertial force to viscous force) and the Archimedes number (ratio of thermal buoyancy force to inertial force) - both important dimensionless terms in determining room air distribution - lead to contradictory scaling factors. The development of computer facility and turbulence modelling enable us to investigate the complex flow phenomena in a room. However the complexities of turbulence theory and numerical techniques require a skillful engineer. It may take three to six months for an engineer who is familiar with the general concepts of computer modelling of fluid flow processes to get familiar with a well-developed computer code so that he can apply the code for his flow problem with confidence.

Hence, it is necessary to develop a concept for a design tool that allows the design engineer to assess air flow pattern, comfort, and indoor air quality without performing a full-scale experiment or running a complex flow field simulation code. Therefore, an air flow database has been constructed in the International Energy

* Energy Systems Laboratory, Swiss Federal Institute of Technology, ETH-Zentrum, CH-8092 Zurich, Switzerland.

Agency Annex 20 - "Air flow patterns within buildings". Since the database contains a number of pre-calculated cases that are prepared entirely without any knowledge of specific design case by the user of the database, there are certainly differences between a pre-calculated case in the database and a specific design case, even if the the specific design case is within the domain of the database. It is necessary to estimate the error caused by the differences and to examine what kind of approximations can be introduced to interpolate the database. This also applies to the situation when the designer has found a similar case from publications. The aim of the present paper is to interpolate between known cases, to derive as much information as possible from the available cases, and to study the influence of boundary conditions on room air movement.

2. INTERPOLATION THEORY

The errors caused by interpolation at different levels of approximation will be discussed in this section. First, it should be explained what is meant by "interpolation" within a knowledge base. It is the art of deducing useful information for a specific design from databases or the cases available in publications. It is noted that the databases or the cases available from publications are not exactly the same as the specific design case, but the specific design case is usually similar to at least one of the cases available in the databases or publications. The intention to achieve accurate interpolation seems ambitious. Therefore, different levels of approximation are proposed here with different chances for success. Even if perfect interpolation cannot be realized, it should be possible to reach some level of generalization valid for a certain sub-domain of cases.

Different classes of interpolation could be envisaged. The term "design case" here refers to the geometry and flow conditions the designer intends to investigate; it represents the "point" where the cases in the databases or publications should be interpolated. Later we use the expression "existing case" referring to one of the configurations, geometry and flow parameters etc., which has been available in the databases or publications. Here are some categories of situations and remarks on the associated interpolation error:

1) The design case coincides exactly with one of the existing cases. This class refers not to a proper interpolation but corresponds to the conventional procedure of making a numerical prediction of the flow for a design case.

ERROR: The correctness of the numerical result depends on the following:

(a) The quality of the input data. This includes the specifications of boundary conditions. For example, an air supply diffuser is often of complex shape and geometry. Different levels of approximation for the diffuser are usually introduced in numerical simulation.

(b) The quality of the physical model. In the numerical techniques, assumptions are often required in the conservation equations of motion in order to make them solvable. For example, the details of turbulent flow are difficult to calculate and engineers are mainly interested in the mean values. Therefore, one turns to so-called turbulence models by which it is possible to compute the mean values. The application of the turbulence model often result a certain degree of error.

(c) The truncation error of the finite difference representation. The magnitude of this error varies with the mesh size used.

(d) Possible errors resulting from incomplete convergence. It is a common practice to use the sum of the absolute residuals of the variables solved in each cell as a criteria to monitor the convergence. If the sum is small enough, the results are regarded as converged ones. Actually, the sum of the absolute residuals is an error.

2) The design case is geometrically similar with one of the existing cases and the numerical values of important non-dimensional input parameters are the same. Geometry similarity here includes not only the similarity of room dimensions but also the size, shape, and location of air supply diffusers, radiators, windows, and other sources of energy, momentum, and/or mass. Here the rules of dimensional analysis and modelling theory apply and allow translation of calculated results to a wide range of physical realizations, as long as they have the same geometrical proportions.

ERROR: Here, the error sources (a) through (d), above, also apply, of course. In addition, the following should be considered:

(e) Consistency of less important non-dimensional input parameters. Sometimes, certain less important non-dimensional input parameters cannot be matched. For instance, the Prandtl number, the Mach number, or Boussinesq approximation ($\Delta\rho/\rho$) in which room air density is taken as constant and considers buoyancy influence on air movement in the momentum equation in room air movement. Most of these parameters might have only a negligible influence, such as the Mach number and Boussinesq approximation. But in many other cases, unmatched values such as turbulent Prandtl number would cause a significant error [1].

3) The design case is geometrically similar with one of the existing cases but the numerical values of some important non-dimensional input parameters are slightly different. The non-dimensional output variables are accepted without change and are applied to the design case as they are.

ERROR: Effects (a) through (e) apply.

(f) The error resulting from second-order effects. This approximation is almost like a linear extrapolation. For instance, the radiator in a design case is of the same dimension as that in an existing case but the surface roughness is different. The difference in surface roughness results in a different heat transfer coefficient. This implies that the linear influence of ΔT on heat transfer of the radiator is correctly taken into account, but the heat transfer coefficient is not correct. A sensitivity study on the roughness is required to estimate the error.

However, the second-order effects may be eliminated if the general dependence of (non-dimensional) output parameters on input parameters is approximately known from the relations such as:

$$Nu = c\, Re^m\, Pr^n$$

The constant c and exponents m and n are taken from an empirical relation that is applicable to the particular flow conditions or from numerical sensitivity studies.

Extrapolation is strongly discouraged if the non-dimensional input parameter is one of dominant parameters in the design case, and is very different from the existing case. As mentioned above, in mixed-convection case, both the Reynolds number (ratio of inertial force to viscous force) and the Archimedes number (ratio of thermal buoyancy force to viscous force) are dominant non-dimensional terms in determining room air distribution. Scaling method cannot be used because the two terms lead to contradictory scaling factors or one of the two parameters cannot be matched.

4) Design case and existing case are not absolutely geometrically similar, and the numerical values of important non-dimensional input parameters are the same or slightly different. For instance, the room size is similar but with a slight difference in radiator location, furniture location, heat source location, window size, and/or air diffuser size and location.

ERROR: Effects (a) through (f) apply.

(g) The error from non-similar geometries. It is necessary for different types of rooms and air-conditioning or heating systems. Error caused by these parameters can be determined and can be corrected via corresponding sensitivity study.

5) Design and existing cases are neither geometrically similar nor have the same non-dimensional input parameters. Here, an attempt could be made to numerically interpolate between a number of neighbouring cases, in terms of dimensionless variables. These neighbouring cases should resemble the design case in terms of dimensionless variables and geometric proportions. The reference cases is scaled to convenient physical dimensions (e.g., same heat load in Watt and same room area in m^2 as the design case). These physical reference cases are then presented to the designer. It is up to him now, with his experience and judgement, to use the available information and translate it to his particular design. This interpolation would certainly not be advisable if there is a drastic change of air flow patterns or flow regime between the available cases. The procedure is completed by translating the non-dimensional results into physical quantities.

The regional models [2-4] are one of the examples. It is based on the idea that room air flow can be divided into flow regions, and within which a simpler approach may be used to scale experiments and models. Understanding regional flow characteristics in a room will help to develop methods for correcting distortions in similitude models so the results from a reduced scale model study can be properly extrapolated to the prototype of the model.

ERROR: Effects (a) through (g) apply.

(h) The feasibility and reliability of this method. When the data are scaled to the physical dimensions of the application, not all characteristic constants and non-dimensional parameters can match the design case. The remaining differences of independent variables will cause a certain deviation of dependent variables. The designer will have several options to approximate an optimum match. One option would be to calculate the dimensionless parameters of the design case for comparison with the corresponding values of the nearest existing case, and to set the unknown non-dimensional dependent variables equal to those of the existing case. An error remains, because numerical results are applied to a situation that is not described by the same values of characteristic parameters. The error may be judged according to the differences between these parameters.

Certainly, one condition for a match of two cases is that both are characterized by the same set of input parameters and a homology of shape. It would not make sense to try to match a room under forced ventilation with one under purely free convection. Regarding shape, cases with an air supply opening at floor cannot be compared with rooms where the opening is at ceiling level.

From the five different levels of approximation, we may conclude that levels one and two have been discussed extensively in almost all the case studies published. Level five is a case-dependent problem, and it is not easy to get a general conclusion. Therefore, we will mainly discuss levels three and four in the following sections. A more general discussion on the interpolation theory has been detailed in [5].

3. CLASSIFICATION OF INPUT PARAMETERS

Since indoor air flow patterns are related to inlet size, shape, and location, outlet location, air supply parameters such as air temperature, velocity, and turbulence intensity, room geometry, furniture, wall temperatures, and internal heat sources and their locations, etc., interpolation is always necessary to gain information for a design case that is different from an existing case.

If the geometry of the design case is not close to or the same as the existing case, scaling rules should be used. As the scaling factors are generally contradictory in a room with mixed convection, the useful information can be obtained from this kind of interpolation only when flows in different local regions within a room may be dominated by different mechanisms, such as inertial force, thermal buoyancy force, viscous force, or turbulent mixing. Hence, there are limitations.

If a database is established, there should not be any problems with scaling factors because the geometry of a design case is more or less the same as at least one of the existing cases. For some special cases, such as the airflow pattern in a theatre, it will be more economical and reliable to do computation or experiments with real boundary conditions instead of producing thousands of "theatres" to hand in a database. In other words, a database has a domain and should contain only the cases which are within common interests such as for office and hotels. Aimed at this expertise, a database has been set up for assessing indoor air diffusion, air quality, and thermal comfort in offices in the International Energy Agency Annex 20 - "Air flow patterns within buildings". The domain can be divided into several sub-domains by the parameters significantly affecting air flow patterns such as summer cooling, winter heating, and ventilation system (all rooms with similar geometry). In the database, two ventilation systems, displacement ventilation and well-mixed ventilation, are selected under summer cooling conditions. For simplicity, the displacement ventilation system is used for demonstration in this paper.

Within each sub-domain, all the parameters can be divided into two classes. The parameters of the first class have a significant influence on indoor air flow patterns. In other words, the first class parameters not only are the elements of important non-dimensional input parameters, but also can vary in a wide range. For example, space load, which is connected to Archimedes number, is one of those. Those parameters should be studied in detail within the range. However, the database will only be able to store a limited number of the cases with several different space loads (Archimedes numbers). The second class parameters have minor influence on air flow pattern if they vary within a limited range. The surface temperature of an interior wall is of this kind.

In the present study, a displacement ventilation system for office buildings is selected as a sub-domain for demonstration. It is unnecessary to apply similitude theory, since actual physical dimensions can be employed which are straightforward in practical applications. The sub-domain is defined as rectangular rooms with one air supply device and one window (exterior wall). The first-class parameters include:

1. room length and width;
2. room height;
3. inlet and outlet location; and
4. space load.

For ordinary small offices, the length and width of the room should be under 8 m and the height 3.5 m. For a displacement ventilation system, the inlet is always located in a wall at the floor level while the outlet is at the ceiling level. The ventilation rate is related to space load and, therefore, is not regarded as a first class parameter. Space load of a room with displacement ventilation should not exceed 40 W/m^2 floor area without using a ceiling cooling panel.

The second class parameters for the displacement ventilation systems are:

1. inlet size and shape;
2. inlet air temperature, mass inflow, and turbulence intensity;
3. space load due to solar radiation through window;
4. lighting;
5. window size;
6. the locations of internal heat sources;
7. furniture size and location; and
8. the surface temperatures of interior walls, etc.

Since air velocity for displacement ventilation must be very small (for example 0.25 m/s), inlet size becomes less important. Due to thermal comfort reasons, supply air temperature should not be too low. This means that air temperature and mass inflow can only change in a very small range. With a low air velocity, the influence of turbulence intensity of the supply air is less important on air flow patterns and thermal comfort. In modern buildings, the application of window shading devices eliminates a major part of solar radiation so that the influence of solar radiation from the window is small. Internal heat sources in office buildings are known and will be studied as a first class parameter. However, the location of the internal sources and furniture can be quite different from office to office. In fact, it is unnecessary to have a very precise investigation on the locations of the internal sources and furniture. A designer would be satisfied if he can obtain a sense on the degree of the variation.

With the information from a database for the first class parameters and for the second class parameters, a designer should be able to assess airflow pattern, thermal comfort, and indoor air quality without doing a costly experiment or a complicated flow field simulation.

4. INFLUENCE OF THE SECOND CLASS PARAMETERS ON AIR DIFFUSION

In a previous paper [6], the influence of air supply parameters such as inlet size, shape, air temperature supplied, inlet air velocity, and turbulence intensity of supplied air on indoor air diffusion has been studied in detail. This paper will only concentrate on the following boundary conditions (geometry and input parameters):

1. variation of space load due to solar radiation through window;
2. variation of window size and heat source location due to lighting;
3. variation of the surface temperatures of interior walls; and
4. variation of furniture location and size.

The advantages and disadvantages of numerical and experimental prediction approaches have been discussed in [6]. The numerical technique appears more appropriate to study the influence of boundary conditions on the field distributions of air velocity, temperature, and contaminant concentration in a room, hence, it has been employed in the present research. According to the flow characteristics in a room and the accuracy of modelling results, the low-Reynolds-number k-ε model is chosen [7].

4.1. *Case setup*

The sensitivity studies of the influence of boundary conditions on indoor air diffusion are conducted for an office with a displacement ventilation system as shown in Figure 1. The office size is 4.5 m in length, 4.5 m in width and 2.5 m in height with some furniture, two computers, and two occupants. For simplicity, no aerodynamic blockage is considered for the computers. We name the case shown in Figure 1b as a standard case. For the standard case, the contaminant from the occupant A is assumed to be 0.01 ml/s, simulating a smoking person. The surface temperatures of interior walls are 22.3°C. To simulate a summer cooling situation, a convective heat gain of 150 W is assumed from the window due to solar radiation. The window size is 3.1 m wide and 1.1 m high. The heat sources from occupant is 80 W and computer 120 W. The inlet, a flat, non-spreading diffuser, is assumed to be a simple slot, i.e. the effective area equal to the gross area. Its size is 0.6 m in width and 0.6 m in height. The turbulence intensity of the supply air is 40%. The supply airflow rate is 70 l/s or 5 ach with a supply temperature of 19.0°C.

4.2. *Variation of space load due to solar radiation through window*

In this group, all the geometry and input parameters are the same as those in the standard case except the convective heat gain from the window. Figure 2 shows the results with different heat gains side-by-side. The top row, 2a, 2b, and 2c, illustrates the computed field distributions of air velocity in section y = 2.25 m (mid-width section), the central row air temperature in section y = 2.25 m, and the bottom row smoke concentration in section y = 1.7 m (section via the smoke source). Sub-figures 2a, 2d, and 2g are for the case without convective heat gain from the window, 2b, 2e, and 2h for the standard case i.e. with a convective heat gain of 150 W from the window, and 2c, 2f, and 2i for the case with 300 W convective heat gain from the window.

From sub-figures 2a, 2b, and 2c, we can see that the airflow patterns in the lower part of the room look very similar although the heat gain from the window is different. The temperature and concentration distributions also confirm that the air distributions are irrelevant to the heat gain from the window in the lower part of the room. The convective heat gain from the window results in a upward buoyancy force that has a significant impact on the air diffusion near the window and ceiling areas. This implies that the regional model [2-4] could be a useful tool for the study of indoor air diffusion.

The temperature difference between the head level (1.1 m above the floor) and the ceiling increases with the heat gain from the window. The relationship between the temperature difference and the space load (the sum of all kinds of convective heat gain) may be approximated by a linear one. However, the linear interpolation may only be accepted with a small variation of the convective heat gain from the window. Chen and van der Kooi [8] pointed out that the relationship is no longer linear if the variation of the space load is large.

The interpolation for the concentration distribution looks much more complicated. The smoke concentration distribution with 0 W and 150 W convective heat gain from the window are more or less the same as indicated in sub-figures 2g and 2h. The concentration in the upper part of the room is higher in the case with 300 W heat gain (sub-figure 2i). This is because the flow due to the buoyancy from the window is stronger which forms a downward re-circulation in y-z plane and brings the contaminated air to the upper part of the room. If the concentration distributions are regarded as invariant to the convective heat gain from the window, it will lead to a less than 20% of interpolation error (between sub-figures 2h and 2i). Should a linear interpolation be introduced for the concentration difference between the head level and the ceiling, the interpolation error is also less than 20% (between sub-figures 2g and 2h or 2h and 2i). Perhaps a non-linear interpolation would be necessary for the concentration for more precise applications.

4.3. *Variation of window size and heat source location due to lighting*

This group is divided into two parts. The first one concerns the variation of window size, and the variation of heat source location due to lighting comprises the second part. In studying the variation of window size, all the geometry and input parameters, including the total convective heat gain from the window, remain unchanged as in the standard case. The window size for the standard case is 3.1 m in width and 1.1 m in height. A smaller window, which is 1.3 m wide and 1.1 m high, is used for comparison.

Sub-figures 3a, 3d, and 3g are the computed field distribution of air velocity, temperature, and concentration in different sections of the office with a smaller window. The results for the standard case are also illustrated in sub-figures 3b, 3e, and 3h for comparison. Again, we have found that, though the distributions of air velocity, temperature, and smoke concentration in the upper part near the ceiling are slightly different between the two cases, they are nearly the same in the zone of occupation. It may be concluded that the influence of window size on indoor air diffusion is negligible as long as the convective heat gain from the window is the same.

With respect to the heat source location due to lighting, the following case is taken for comparison with the standard case. We simulate a case without convective heat gain from the window but with 150 W convective heat gain from the lighting on the ceiling. The 150 W convective heat gain is uniformly distributed in an area of 2.4 m long and 0.4 m wide on the ceiling above the two occupants. Normally, no additional lighting is required if there is an exterior window providing sufficient light. A zero heat gain from the window implies no window or an ideally insulated window. Thus, the standard case with 150 W heat gain from the window and the case with 150 W heat gain from the lighting on the ceiling simulate two different heat source locations due to lighting. For a situation with heat gains from both window and the lighting, it may be regarded as an intermediate one between the two cases studied.

The numerical results for the case with 150 W heat gain from the lighting in the ceiling are given in sub-figures 3c, 3f, and 3i. The airflow in lower part of the room is controlled by the air supply from the low diffuser, and the heat source location due to lighting has little influence on the flow distribution. The temperature gradient near the ceiling for the case with ceiling lighting is much larger than the standard case with window lighting. But the distributions of velocity, temperature and smoke concentration in the occupied zone do not change very much. The temperature and smoke concentration in the standard case are a bit lower. This is because the flow generated from the buoyancy in the window results in higher velocities near the ceiling level, and removes the heat and contaminant more efficiently. The maximum temperature difference in the occupied zone between the two cases is less than 0.5°C and the maximum concentration difference is less than 10% of the averaged concentration.

4.4. *Variation of the surface temperatures of interior walls*

Since the variation of the temperatures of interior wall surfaces are small in most cases, the walls are often treated as isothermal ones. This treatment is acceptable for a wall to an adjacent room with the same thermal conditions. However, certain discrepancies may be expected if the neighbour room is a corridor or basement with a different air temperature. Hence, it is necessary to estimate the error introduced if the surface temperatures of interior walls in a design case are different from those in an existing case.

Three different surface temperatures of interior walls, 20.3°C, 22.3°C (the standard case), and 24.3°C are studied in this group, while the other boundary conditions are the same as the standard case. Note that the surface temperatures of interior walls refer to the temperatures of the floor, ceiling, and all the wall surfaces except the window.

The computed distributions of air velocity, temperature and smoke concentration in different sections of the room are shown in Figure 4. Sub-figures 4a, 4d, and 4g correspond to the case with a 20.3°C interior wall temperature, 4b, 4e, and 4h to the standard case with a 22.3°C interior wall temperature, and 4c, 4f, 4i to the case with a 24.3°C interior wall temperature. The numerical computations show that the corresponding space loads for the three cases are 359 W, 473 W, and 588 W respectively. The space load is defined as the energy difference between the supply and exhaust air. The vertical temperature differences between the points 0.1 m from the ceiling and the floor, in the centre of the room, are 3.8°C, 5.0°C, and 6.2°C respectively, as shown in sub-figures 4d, 4e, and 4f. Very similar to the results discussed in section 4.2, the relationship between the temperature difference between the air near the ceiling and the floor and the space load is a linear function if the temperature variation is not too large. However, the air flow distribution in the lower part of the room is affected by the interior wall temperatures. This is due to the heat exchange with the vertical walls and floor. In the case with a 20.3°C wall temperature, the flow near the vertical walls is downward. But for the case with a 24.3°C wall temperature, the flow is upward near the lower part of the vertical walls and downward near the upper part. This results in a higher vertical temperature gradient at the mid-height of the room.

The interior wall temperatures were varied by 4.0°C among the three cases, a very wide range. In many practical situations, a variation of 1.0°C in one of the walls is possible. In these circumstances, the vertical temperature difference is less than 0.2°C, and is, therefore, negligible. With different surface temperatures of interior walls, the mean air temperature of the room is different. However, the mean air temperature can be easily determined from the space load.

The influence of the interior wall temperatures on the smoke concentration distributions are more complicated. The concentration in the case with 20.3°C wall temperature is more uniform since the downward airflow along the walls brings the contaminated air to the lower part of the room. If more detailed information is not available for interpolating from an existing case, the smoke concentration distributions may be assumed to remain unchanged. This results in an error of about 20% of the average concentration at 4°C variation of interior wall temperatures.

4.5. *Variation of furniture location and size*

With the same room size, the furniture location and size, and the internal heat sources from equipment can be very different. It is a common practice to assume that each occupant requires about 10 m^2 area. In the standard case, the room area is 20.25 m^2, and therefore, two occupants are assumed. Each occupant has a bookshelf and a table with a filing cabinet. In order to obtain a general sense of the variation of furniture location, a furniture distribution near the rear wall, as shown in sub-figure 1a, is used in comparison with the standard case. The two occupants and computer heat sources are moved together with the tables but the remaining thermal boundary conditions remain unchanged. In many offices, there is more furniture than in the standard case. In order to study the influence of furniture size on indoor air diffusion, a new case has been setup in which the tables in the standard case (sub-figure 1b) are replaced by larger ones (sub-figure 1c).

The computational field distributions of air velocity, temperature, and smoke concentration for the three cases are shown in Figure 5. Sub-figures 5a, 5d, and 5g are for the case with furniture distributed at the near rear wall (distribution 1), 5b, 5e, and 5h for the standard case (distribution 2), and 5c, 5f, and 5i for the case with larger tables (distribution 3).

There is no doubt that furniture distribution has a significant influence on indoor air distribution by comparing sub-figures 5a, 5d, and 5g with sub-figures 5b, 5e, and 5h. However, there are something in common. For example, the fresh cold air still goes along the floor level and before being heated up by the internal heat source and the heat from window. This results in temperature stratification in the room air and the similar temperature distributions. The temperature differences between the air near the ceiling and near the floor are not very large. The smoke distribution could be quite different because the smoke source location has been changed.

From sub-figures 5b, 5e, 5h, 5c, 5f, and 5i, we can see that the influence of the furniture size on air diffusion is very small in this particular case. This is so because the tables happened to be located mainly in the stagnant zone. The results can be very different if another filing cabinet is placed at the left side of the tables.

The above results indicate that it is difficult to get a general interpolation error due to the variation of furniture location and size. The designer should use his best knowledge to interpolate the results by level five interpolation. When a database is prepared, the furniture distribution should be as reasonable as possible to reduce the interpolation error. In addition, the influence of internal heat sources on air diffusion is very important but not discussed here, since they are first class parameters.

It should be pointed out that the study of the influence of boundary conditions on indoor air diffusion is not necessarily restricted to single room configuration (one zone). Jiang [9] has studied the air diffusion in a ventilated two-zone enclosure with a connecting open door. The study concerns the influence of door location and air supply and exhaust locations on air flow pattern, indoor air quality, and thermal comfort.

5. CONCLUSIONS

In this paper, five levels of interpolation have been outlined to interpolate the results of the cases from a database or from publications for a specific design case in order to assess air flow pattern, indoor air quality, and thermal comfort. The error caused by different levels of interpolation has been analyzed. It is necessary to provide a sensitivity study for estimating the error caused by level three and level four interpolations. These imply that some important boundary conditions may vary in a very limited range, and the influence of the variation of the boundary conditions on indoor air flow, air quality, and thermal comfort should be estimated.

An office with a displacement ventilation system is used as a prototype to study the field distributions of the air velocity, temperature, and smoke concentration with the variations of the following boundary conditions:

- space load due to solar radiation through window;
- window size and heat source location due to lighting;
- the surface temperatures of interior walls; and
- furniture location and size.

The interpolation errors caused by the variations of space load due to solar radiation through the window, window size, heat source location due to lighting, and the surface temperatures of interior walls are small, and can be quantitatively determined. But it is difficult to estimate the errors introduced by the variations of furniture location and size.

REFERENCES

1. G. P. Hammond, Turbulent Prandtl number within a near-wall flow. *AIAA Journal*, **23** (11), 1168-1169 (1985).
2. M. B. Timmons, Use of physical models to predict the fluid motion in slot-ventilated livestock structures. *ASAE Transactions*, **27** (2), 502-507 (1984).
3. J. H. Ferziger, Application of zonal modeling to ventilation simulation, in *Building Systems: Room Air and Air Contaminant Distribution* (Ed., L. L.

Christianson). American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc., Atlanta (1989).
4. J. S. Zhang, L. L. Christianson, and G. L. Riskowski, Regional airflow characteristics in mechanically ventilated room under non-isothermal conditions. *ASHRAE Transactions*, **95** (2), (1990).
5. A. Moser, Interpolation in data base of computed flow field. Presented at the 4th Expert Meeting of IEA Annex 20, Lommel, Belgium, 1989.
6. Q. Chen, P. Suter, and A. Moser, Influence of air supply parameters on indoor air diffusion. Submitted to *Building and Environment*, (1990).
7. Q. Chen, A. Moser, and A. Huber, Prediction of natural convection in a room by a low-Reynolds-number k-ε model. *ASHRAE Transactions*, **96** (1), (1990).
8. Q. Chen and J. van der Kooi, ACCURACY - a computer program for combined problems of energy analysis, indoor airflow, and air quality. *ASHRAE Transactions*, **94** (2), 196-214 (1988).
9. Z. Jiang, *Intra and Inter-Zone Heat and Mass Transfer in Multi-Zone Buildings*, Ph.D. Thesis, Concordia University, Canada (1990).

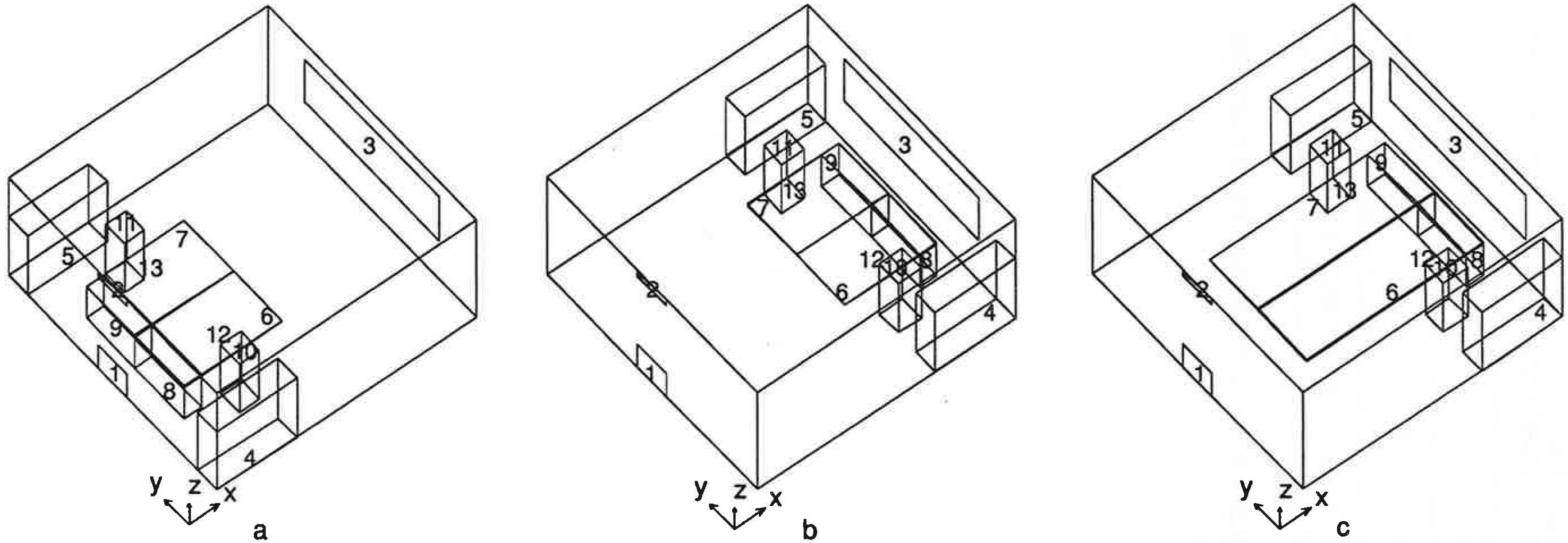

Figure 1. Sketch of the office. (a) Furniture distribution 1; (b) furniture distribution 2; (c) furniture distribution 3.
1 - inlet, 2 - outlet, 3 - window, 4 - bookshelf A, 5 - bookself B, 6 - table A, 7 - table B, 8 - filing cabinet A, 9 - filing cabinet B, 10 - occupant A (smoking person), 11 - occupant B (non-smoking person), 12 - computer A, 13 - computer B (no aerodynamic blockages for the computers)

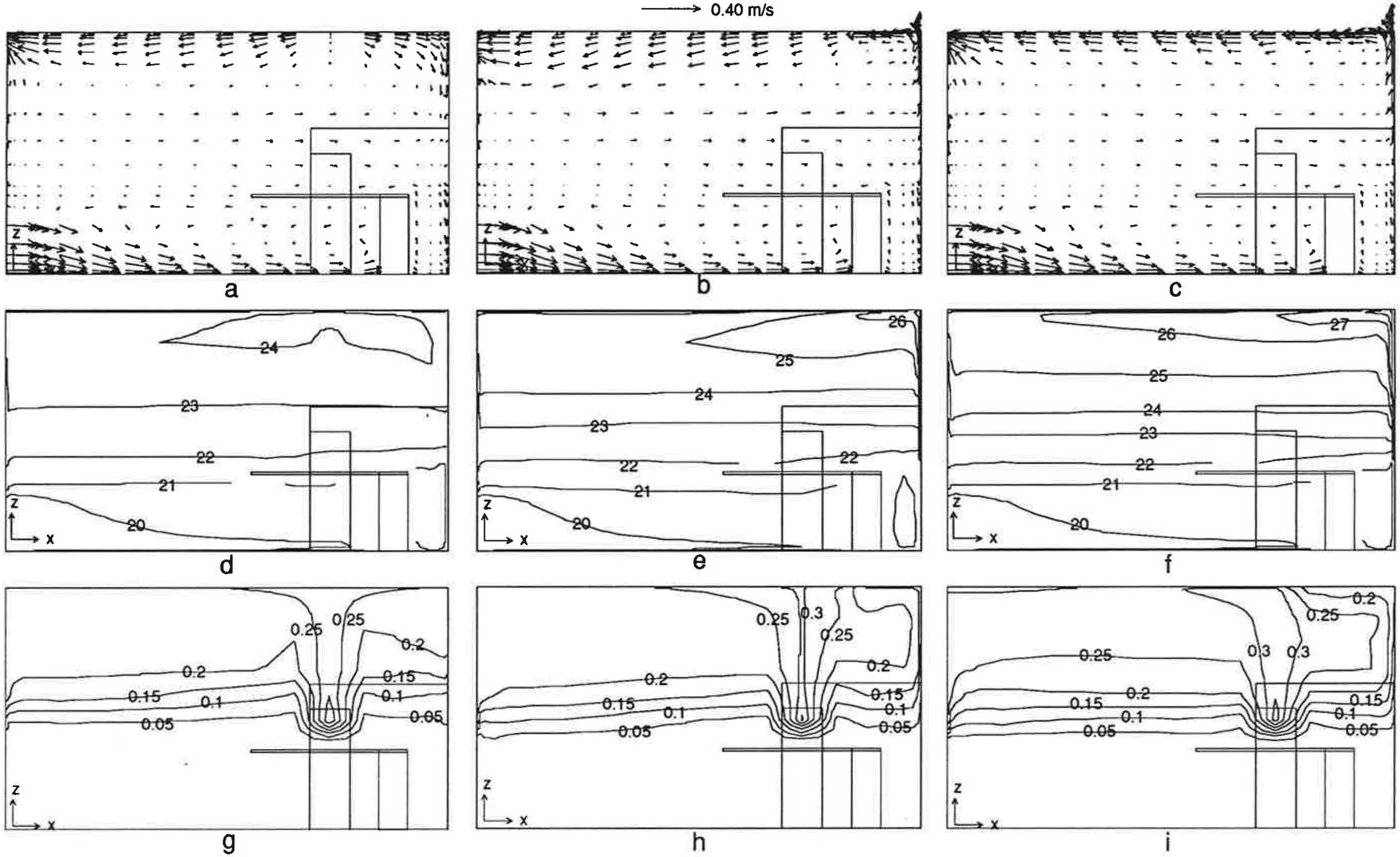

Figure 2. Computed field distributions in the office with different convective heat gain from the window. For (a) (b) (c) velocity in section y = 2.25 m with 0 W, 150 W, and 300 W heat gain respectively; for (d) (e) (f), temperature in section y = 2.25 m with 0 W, 150 W, and 300 W heat gain respectively [℃]; for (g) (h) (i), smoke concentration in section y = 1.7 m with 0 W, 150 W, and 300 W heat gain respectively [ppm].

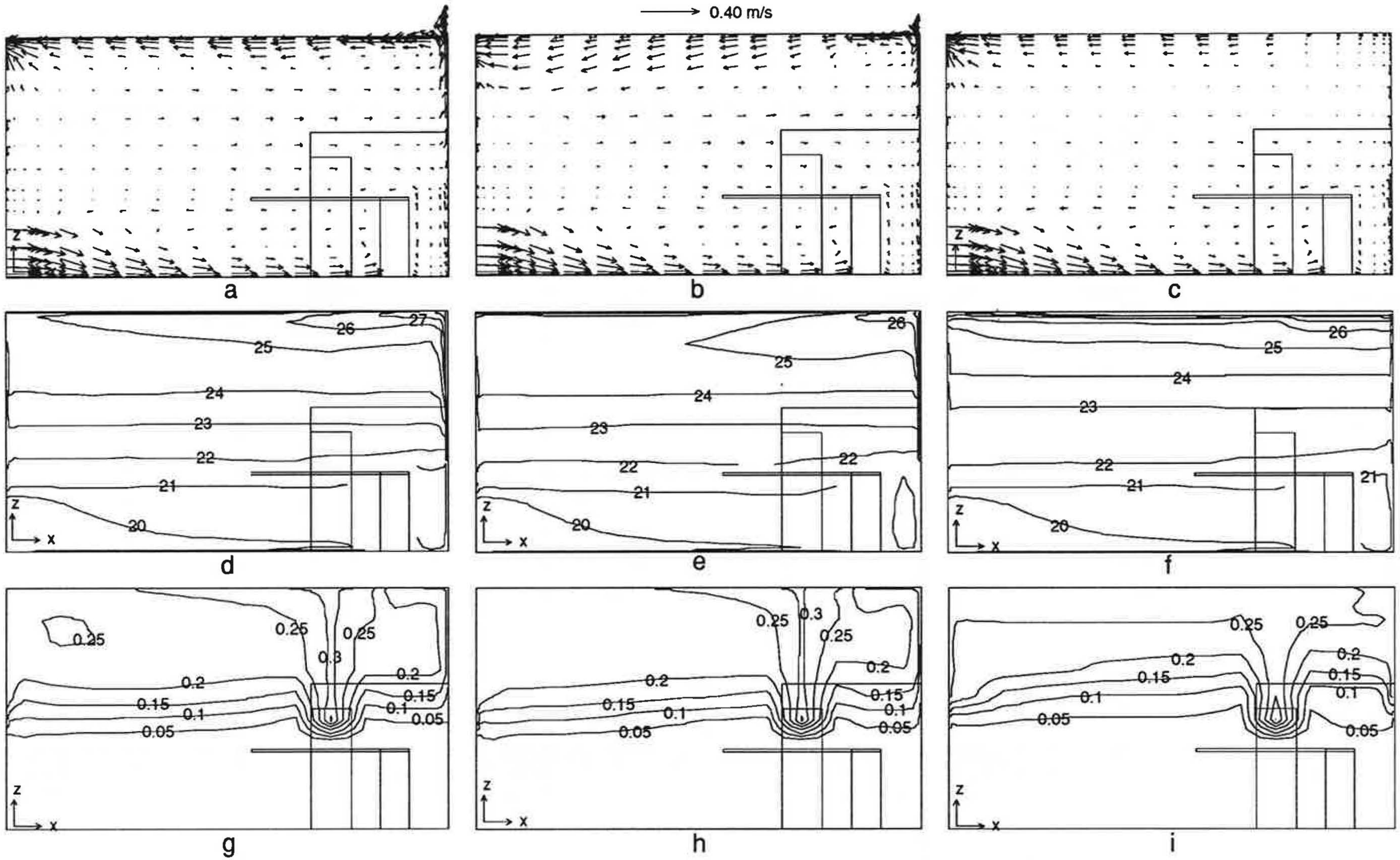

Figure 3. Computed field distributions in the office with different window sizes and heat source locations due to lighting .* For (a) (b) (c) velocity in section y = 2.25 m; for (d) (e) (f), temperature in section y = 2.25 m [°C]; for (g) (h) (i), smoke concentration in section y = 1.7 m [ppm].

* Sub-figures (a) (d) (g) is for the case with a window size of W = 1.3 m and H = 1.1 m and a convective heat of 150 W from the window.
Sub-figures (b) (e) (h) is for the case with a window size of W = 3.1 m and H = 1.1 m and a convective heat of 150 W from the window.
Sub-figures (c) (f) (i) is for the case without heat gain from the window but with a convective heat of 150 W on the ceiling above the occupants.

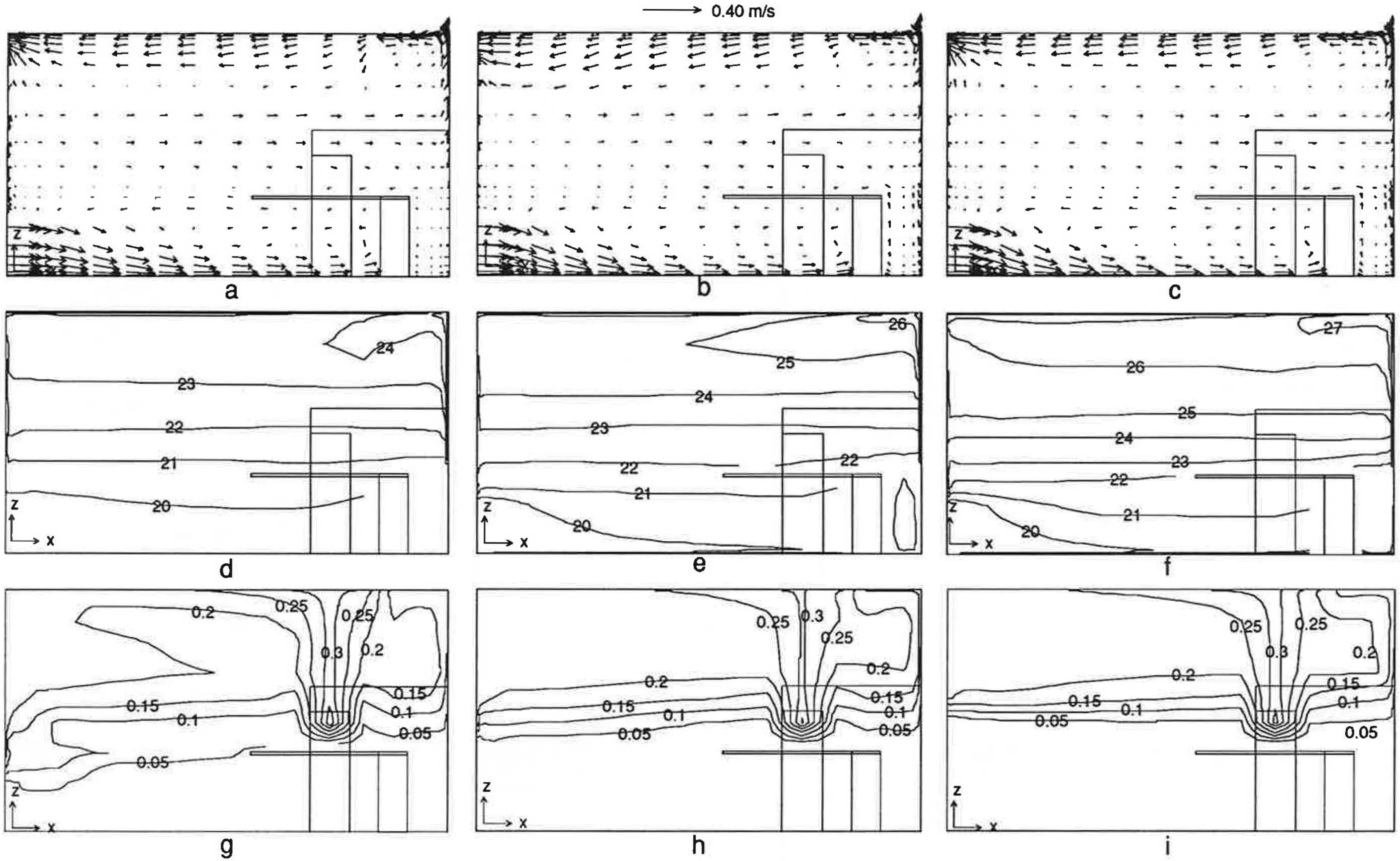

Figure 4. Computed field distributions in the office with different temperatures of interior wall surfaces. For (a) (b) (c) velocity in section y = 2.25 m with 20.3, 22.3, and 24.3°C wall temperatures respectively; for (d) (e) (f), temperature in section y = 2.25 m with 20.3, 22.3, and 24.3 °C wall temperatures respectively [°C]; for (g) (h) (i), smoke concentration in section y = 1.7 m with 20.3, 22.3, and 24.3 °C wall temperatures respectively [ppm].

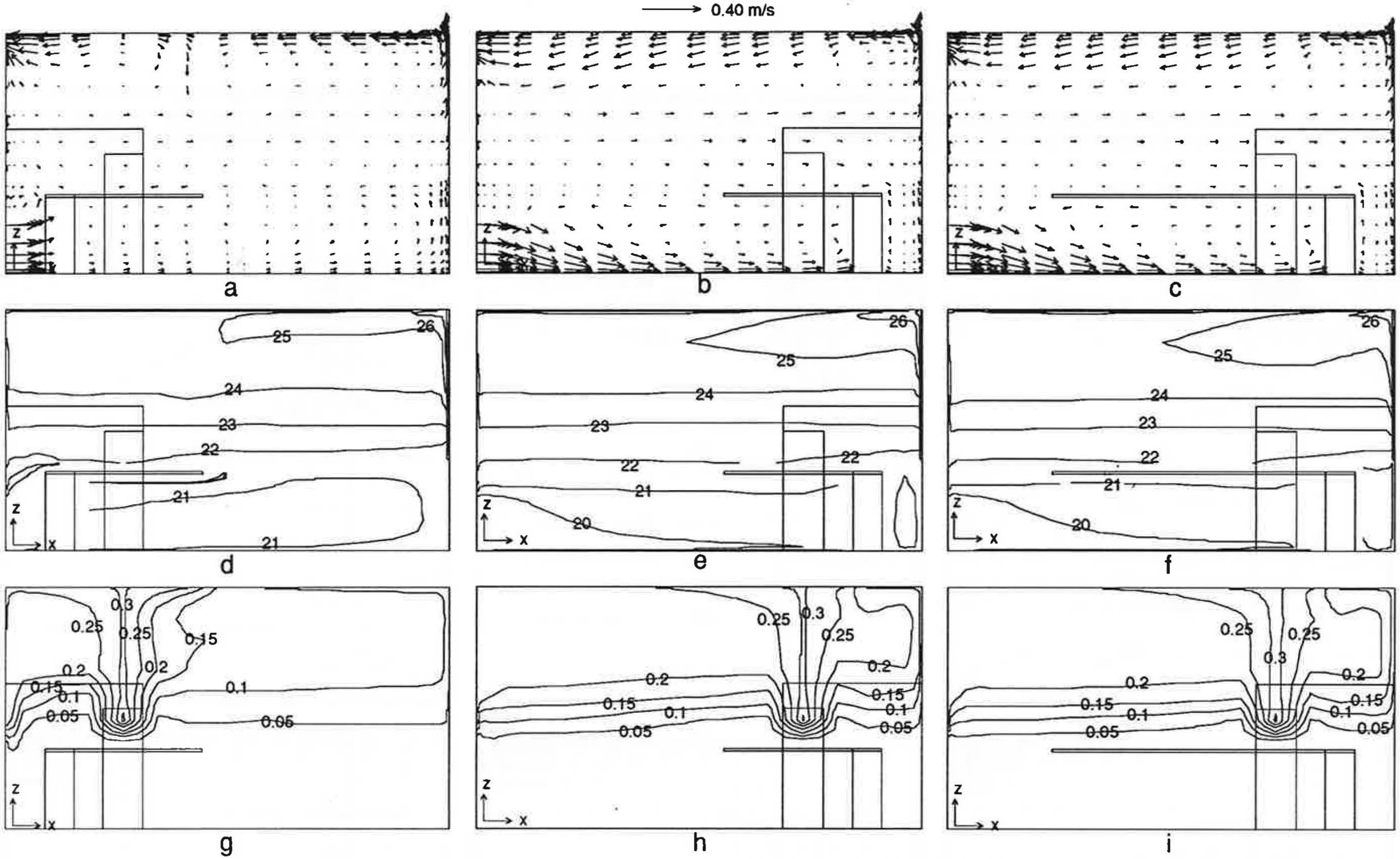

Figure 5. Computed field distributions in the office with different furniture distributions . For (a) (b) (c) velocity in section y = 2.25 m with furniture distributions 1, 2, and 3 respectively; for (d) (e) (f), temperature in section y = 2.25 m with furniture distributions 1, 2, and 3 respectively [℃]; for (g) (h) (i), smoke concentration in section y = 1.7 m with furniture distributions 1, 2, and 3 respectively [ppm].