ACCURATE NUMERICAL SIMULATION OF AIR FLOWS IN VENTILATED MULTI-ROOMS

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ABSTRACT

A numerical study of turbulent air flow in ventilated multi-room configurations, where both of buoyancy- and radiation-effects are of importance, is described in this paper. Our computer code solves, in finite difference form, the transient-state conservation equations for mass, momentum and thermal energy. The two equation k-e model with buoyancy terms (Boussinesq approximation) is employed for modelling of turbulence. In addition, a two-band model based on Gebhart's method has been implemented enabling the study of the effects of long wave radiation and solar radiation through windows.

Our purpose here is to present a fast multigrid solver with local grid refinements. The much faster convergence rate of that solver compared with more standard single grid schemes (such as SIMPLE) allows the usage of a large number of computational elements. This together with a rational distribution of the node points is expected to achieve an improved spatial resolution and numerically accurate prediction. The model equations and the numerical methods are briefly presented. The effect of wall insulation and radiation on indoor flow and temperature fields is studied. The natural convection in partially divided rectangular enclosures is siimulated. A local grid refinement near the supply outlet and the exhaust inlet in a 3D forced convection room is demonstrated. A good agreement is shown between the predicted results and the previously reported experimental data.

INTRODUCTION

It is now well established that indoor air flow is one of the most important parameters influencing thermal comfort, indoor air quality and energy consumption. How to assess this aspect in ventilation and air conditioning design depends on two questions: 1. What kind of air flow or pattern should be in the occupied space? 2. Can the indoor air flow, especially in the occupied space be accurately predicted? This paper addresses the second question with respect to accurate numerical simulation. Consideration of accurate numerical predictions of air flows includes two issues; solving the correct modelling equations with correct boundary con-ditions and solving the discrete equations to an adequate degree of accuracy.

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Air flow within a building is mainly caused by mechanical ventilation system (i.e. forced convection), by a temperature difference between the warm and cold zones (i.e. natural convection), or by a combination of them (i.e. mixed convec-tion). This air flow is influenced by the room geometry, infiltration and exfiltration, position and geometry of outlets and inlets, supply air temperature and velocity, distribution and emission of heat sources, size of the windows, the amount of insulation and the outside weather conditions, etc. Over the past twenty years, more and more attention has been focussed on the numerical simulation in ventilated rooms. Several reviews on the subject exist, (e.g. Nielsen, 1989). The literature survey shows the range of indoor air flow simulations has been enlarged to include turbulent, three dimensional, and buoyancy-effected flows (Chen, 1988; Davidson, 1989). It is not an easy task to cover all the above mentioned influencing factors through a modelling system of partial diffe-rential equations in a simulation code. For example, very few computational studies con-sidering the wall radiation and insulation in numerical simulation of indoor air flow, have been reported in the literature. As a fundamental configuration, natural convection and surface radiation interaction in a 2D enclosure has been analysed numerically for laminar cases (Kim et al. 1984) and for turbulent cases (Fusegi et al. 1989). Recently, lower levels of stratification in air full-scale and gas (R114) small scale experi-ments have been observed when compared to a small-scale water experiments (Olson et al. 1990). This phenomenon suggests a further study on whether the absence of radiation in the water experiments may contribute to the different flow patterns and thermal stratification. In addition to the above mentioned studies on air flow in a single room, a number of additional studies have been undertaken on airflow in multi-room buildings. Natural convection in a 2D partitioned enclosure has been numerically studied for the laminar case by Chang et al. (1982), for the turbulent case by Acharga et al. (1990). With respect to the first issue of the accurate numerical simulations, effects of wall radiation and insulation and interior partition will be investigated and discussed here.

Finally, numerical algorithms produce only an approximate solution to the governing partial differential equation system. Errors arise from the various components of numerical methods: grid generation (including coordinate transformations), discretization and iterative procedure. For any consistent numerical approximation, the error defined as the difference between the numerical and exact solution, is reduced as the grid is refined. Therefore, grid refinement is a natural mean of improving accuracy. The much faster convergence rate of the multi-grid solver compared with more standard single grid schemes (such as SIMPLE) allows the usage of a large number of computational elements. This together with a rational distribution of the node points is expected to achieve an improved spatial resolution and more accurate predictions.

The next two sections of the paper provide a brief description of the turbulent flow equations, the two-band radiation model for surface radiation and wall insulation effect, and the multi-grid solver with local grid refinements. The application studies are reported in the following section.

INDOOR AIRFLOW MODELLING

It is the physical characteristics of indoor air flow that gives a basis to choose the proper model. First, air flows are almost always turbulent in ventilated rooms. It is known that the development of turbulence closure model has not advanced to the stage where an universal model exists and can be applied generally. The k-ε two equation model used here has also been employed rather frequently in previous studies in the field of ventilation and air-conditioning. Secondly, buoyancy is expected to have a significant effect. Reduction of ventilation rate, together with heat transfer and heat generation within the building, can generate non-uniform distributions of indoor air temperatures in many cases. In displacement ventilated rooms, the flow is in general driven by buoyancy force. Simple eddy-diffusivity model for turbulent heat flux and the Boussinesq approximation is used in our code. Lastly, there is more energy exchange by radiation at room temperature than commonly realised, in particular, for heating systems with heated surfaces, e.g. a heated ceiling, a radiator, and for the displacement ventilation system with a vertical temperature gradient, and for the rooms with sunlight through windows. A two-band radiation model (Li & Fuchs, 1991b) is employed here.

Model equations

The indoor airflow code VentAir which is under development by the authors solves a general system of equations of the form:

$$\frac{\partial(\rho\phi)}{\partial\tau}+\frac{\partial}{\partial x_i}(\rho u_i\phi - \Gamma_\phi\frac{\partial\phi}{\partial x_i}) = S_\phi \qquad (1)$$

The dependent variable ϕ take the forms of u, v, w, t, k, ε, and 1 (for the continuity equation). The corresponding coefficients Γ_ϕ and sources S_ϕ can be found in (Li et al. 1991a). The wall functions in (Rodi, 1980) have been employed.

Two-band radiation model

The indoor wall surface temperature can be obtained from the energy balance equation at the surface. The basic assumptions of the model are as follows: 1. Air is transparent. 2. All energy is emitted and reflected diffusely. 3. All surfaces are grey. If we divide the wall boundary into area elements, for each surface elements i, we have:

$$q_r = q^l_i + q^s_i \qquad (2)$$

where

q^l_i =net rate of long-wave radiation heat transfer for a surface elements i
q^s_i =net rate of short- wave radiation for a surface element i
q_r =total radiant heat transfer rate at i.

The two-band model allows to separate the effects of long- and short-wave radiations. This results in very little computational effort. The model also allows the surfaces to have different radiation properties in each band.

NUMERICAL METHOD

SIMPLE procedure

Two versions of codes exist, designated

here as VentAirI and VentAirII. Both isothermal- and nonisothermal-flows can be predicted by VentAirI. The system of equations of type(1) are solved by the SIMPLE procedure (Patankar, 1980) in VentAirI. In this procedure, the staggered grid is employed to obtain a consistent connection between the pressure and the velocities. The hybrid upwind/central differencing scheme is used to discretize the advection terms. The continuity equation is rewritten into an equation for the pressure correction. The resulting algebraic equation are solved by use of TDMA(Tri Dia-gonal Matrix Algorithm) line by line method. It was recognized that the stability furnished by the hybrid scheme was achieved at the expense of accuracy. Numerical diffusion can be significant compared to the physical diffusion. Many other schemes for the advection terms have been developed. All these scheme require, however, an adequate spatial resolution. Most calculation carried out in this paper were performed on progressively finer grids to reduce the numerical diffusion and test the relative accuracy of the grid-dependent result. For the applications in next section, the VentAirI has been used if not stated otherwise.

Multi-grid Procedure with Local Grid Refinements

The main difference between the two versions of the codes is that the mutli-grid procedure with local grid refinements has been incorporated into VentAirII. The code can only be used to predict isothermal flows presently. The convergence rate of VentAirI is in general, strongly dependent on the grid fineness as well as the Reynolds and/or Rayleigh numbers. One major numerical disadvantage of VentAirI is its slow convergence rate. This has resulted in the need to develop new procedure for which very fine grids do not result in unacceptable CPU times. The multi-grid method is an iterative procedure which ideally exhibits grid-independent convergence rate. A number of multi-grid algorithms for the numerical solution of the Navier-Stokes equations may be found in the literature, (e.g. Fuchs et al. (1984), Brandt et al. (1979)). For high-Reynolds-number flows, which is often the case in ventilated rooms, the convective part of the equations is the most difficult to address. The details of the multi-grid procedure used in VentAirII can be found in Bai et al.(1991).

The mutli-grid procedure has been extended naturally to handle locally refined grids. To resolve large gradients in the flow field, the grid is refined locally so that the local spatial scales are resolved. The advantage of this scheme is that it is possible to resolve large gradients in the flow field without influencing the convergence rate of the multi-grid scheme. A detailed description of the original local grid refinement procedure can be found in Fuchs (1986).

APPLICATIONS

Influence of wall conduction, radiation on indoor air temperature field

As a fundamental configuration, natural convective heat transfer in rectangular enclosures has received more theoretical, experimental, and computational attention. There are numerous studies dealing with it, but most of them do not account for wall heat conduction and radiation (Kim (1984)). A test for turbulent natural convection-surface radiation interactions has been carried out with VentAirI by Li et al. (1991b). The calculated result shows a good agreement with the numerical results by Fusegi et al. (1989). The effect of wall conduction on the turbulent natural convection-surface radia-tion interactions will be further studied here. The geometry of the problem is kept as same as Li et al. (1991b). For a square enclosure, the side walls are maintained at two different constant temperatures. The floor and ceiling are thermally insulated or heat conducting. The surface of the entire enclosure is black for radiation. The Prandtl number of the

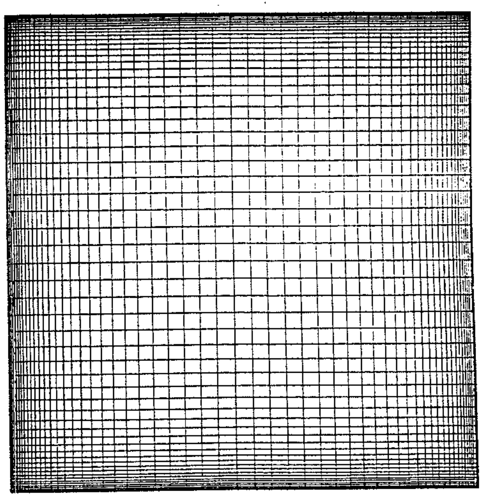


Fig. 1 The mesh used for calculation of conduction, natural convection and radiation interactions in a vertical differentially heated square enclosure.

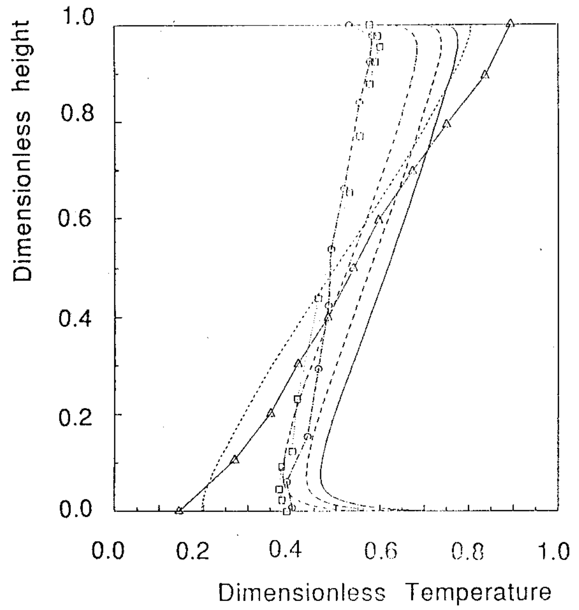


Fig. 2 The core vertical temperature profiles at mid-plane of the enclosure. solid line: natural convection and surface radiation; dashed line: pure natural convection; long dash: natural convection and surface radiation with less conduction heat loss; dotted line: natural convection and surface radiation with more conduction heat loss; solid line with square mark: R114 experiment; solid line with circle mark: full- scale air experiment; solid line with triangle mark: water experiment.

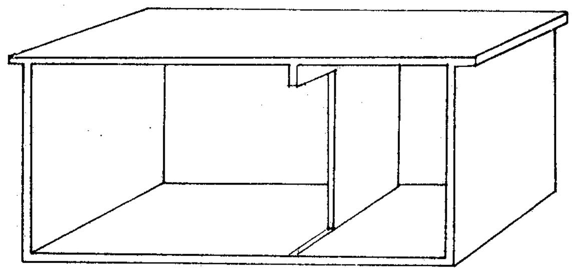


Fig. 3 Configuration of a two-cell room.

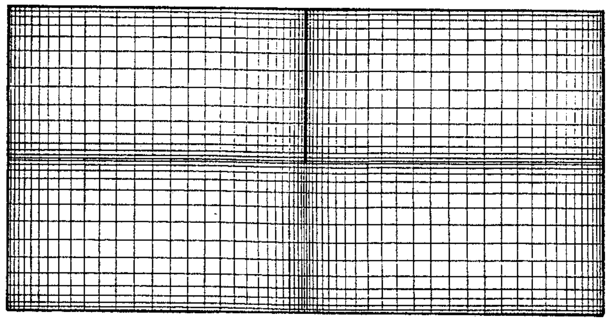


Fig. 4 The mesh used for calculation of natural convection in partially divided rectangular enclosures. A 60*30 mesh is shown for clarity.

fluid is held fixed at 0.686. The following results are for Grashof number $1.46*10^9$ and non-uniform grid 58*58 (see Fig.1). The calculated core vertical temperature profiles at mid-plane in the enclosure are shown in Fig.2. As a comparison, the measured core vertical temperature at the mid-plane has also been depicted in the figure. The data shown for comparison are results from the gas(R114) scale experiment (Olson et al. 1990, Ra=$2.2*10^{10}$, Al=0.35), full scale air experiment (Olson et al. 1990, Ra=$2.6*10^{10}$, Al=0.32), water scale experiment (Nansteel et al. 1985, Ra=$.8*10^{10}$, Al=0.5). The water experiments show much larger temperature stratification than the two gas experiments. The water case and the two gas cases correspond to the pure convection and the natural convection-radiation interactions in the calculations, respectively. In the water experiment, the radiation heat transfer was absent, sine water is opaque. It is found that the addition of conduction results in a better agreement between the calculation and the experiment (Fig.2). This is so since in the experiment the ceiling and floors have not been perfectly insulated (Olson et al. 1990). The results indicate that the thermal boundary conditions (wall conduction and radiation exchange) on room wall boundaries influence the indoor air flow and indoor air temperature distribution. The frequently employed adiabatic or prescribed wall temperature boundary conditions may not be always suitable.

Natural convection in partially divided rectangular enclosures

Simulation of natural convection in an enclosure with a partial vertical divider can be considered as a first step towards the multi-room air flow simulation, (Fig.3). Enclosures with a single divider projecting vertically upwards or vertically downwards, and enclosures with two dividers, one projecting upwards and the other downwards, have both been experimentally studied, (Acharya et al. (1990)). The air

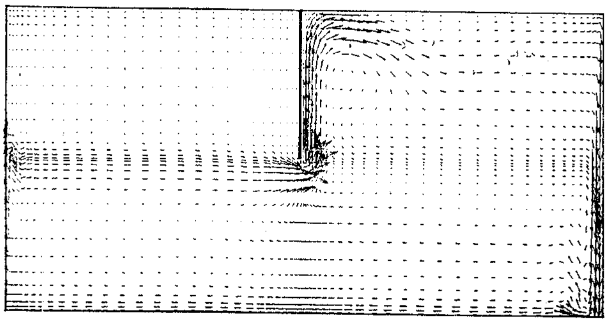
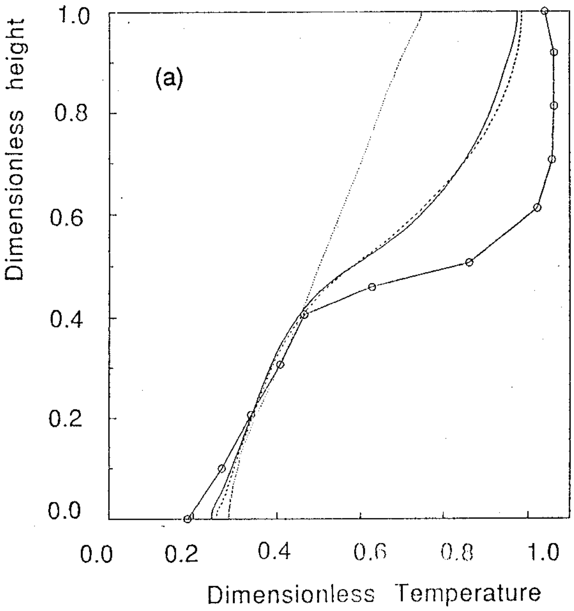
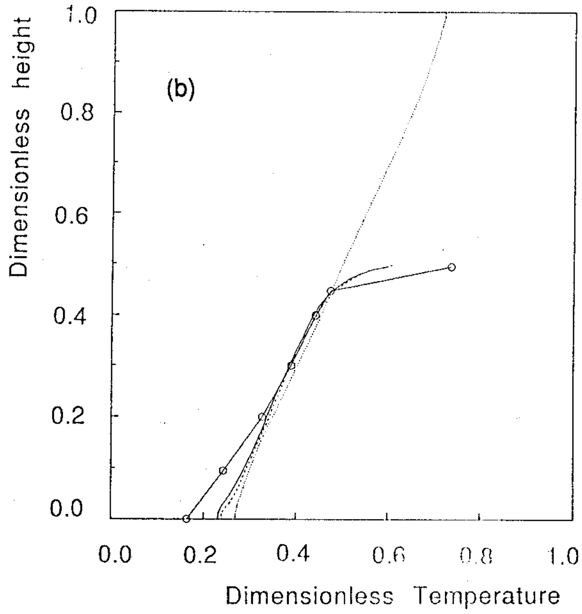


Fig. 5 The calculated flow field in the partially divided rectangular enclosure.



flow in multi-cell enclosure is affected by many parameters, such as the dimensions of the room, the size and the location of the partition opening, and the conditions at the walls, the ceiling and the floor. It is expected that numerical simulations can be used to analyse the variation of each of these parameters. A numerical simulation in 2D partially divided, water-filled rectangular enclosures for which experimental measurements are available (Nanstell et al. (1981)), has been performed here, .The height of the enclosure is 15.2 cm, and the width is 30.5 cm. The heated (right) and cooled (left) vertical walls were hold at temperatures of 63.0^{o}C and 29.6^{o}C, respectively. The partition is adiabatic with aperture ratio of .5. The Prantdl number is 3.8. The Rayleigh number based on the width of the enclosure is $4.21*10^{10}$. The local water temperatures within the enclosure were measured



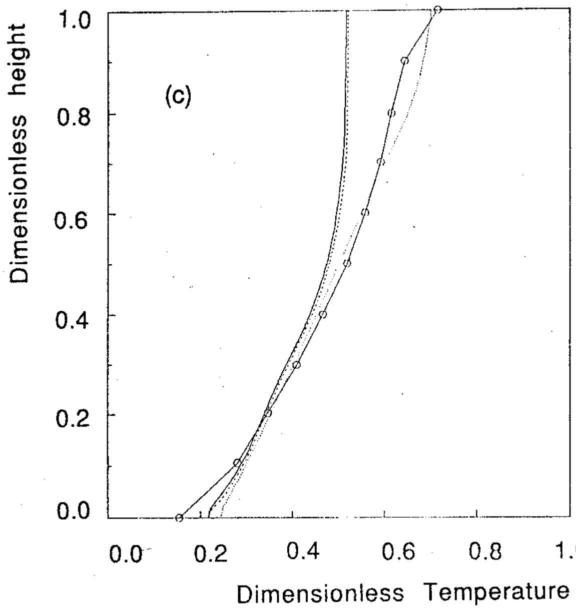


Fig. 6 The calculated and measured vertical temperature profiles at the position x/L = .25(a),x/L = .5(b),x/L = .75(c). solid line: 120*60 mesh, with partition; dashed line: 200*100 mesh, with partition; dotted line: 120*60 mesh, without partition; solid line with circle: water experiment.

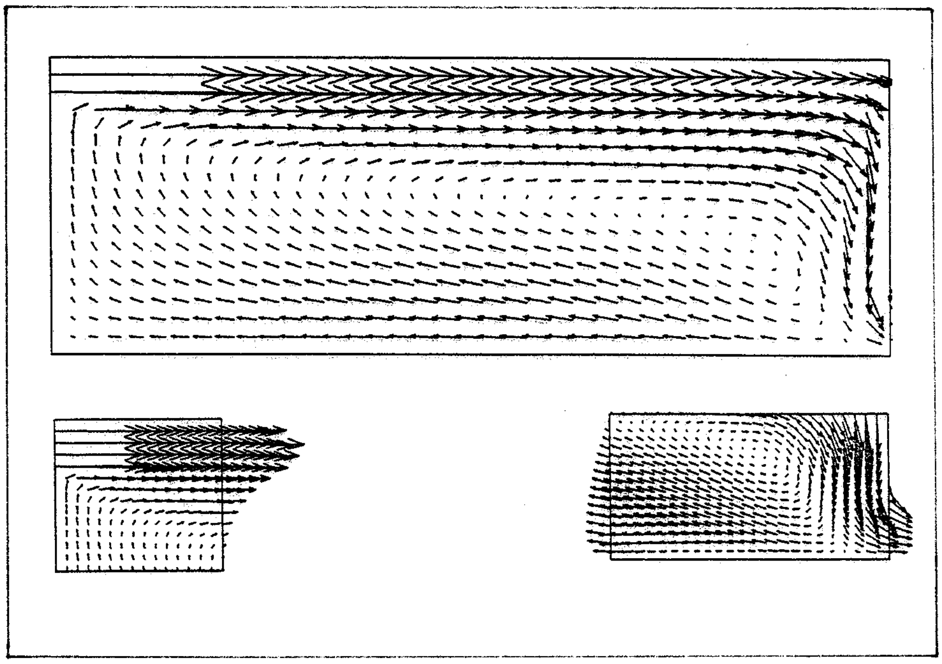


Fig. 7 The flow field on the symmetry plane. Local refined grids are imposed near the inlet and the outlet.

with thermoscouple probe assemblies, (Nansteel et al. (1981)). For the calculations, we used a 120*60 and 200*100 non-uniform grid. The mesh near the walls and the partitions are refined. The distribution of the mesh is shown in Fig. 4, where only a 60*30 mesh is shown for clarity.

The calculated flow pattern is very similar to the one observed in the experiment (Fig.5). The calculated and measured vertical temperature profiles at the positions x/L=.25, .5, and .75 are plotted in Figs. 6. It can be seen that the temperature profiles near the bottom half of the enclosure are almost identical to those without partition. This is due to the fact that the flow pattern is almost unaffected by the presence of the partition there. However, in the upper half of the enclosure, the floor partition caused a temperature difference across the partition, due to the warm ceiling jet being blocked by the partition. The agreement between the calculated profiles and the measured profiles is in general good. The measured temperature in the upper left part exceeds the average hot wall temperature.

Local grid refinement near the inlet and outlet in a 3D room geometry

The great improvement of convergence rate by the multi-grid procedure has been addressed in numerous papers over the past years (e.g. Fuchs et al.(1984)). The convergence rate of **VentAirI** is very slow. For example, to achieve a converged solution for the surface radiation and natural convection interaction problem with grid 58*58 (i.e. 5 order reduction of the residuals), about 3000 iterations are needed. The same 3D problem with Reynolds number of 5000 as in Li et al. (1991) is chosen to perform local grid refine-ments with a multi-grid procedure. The small inlet and outlet openings are resolved by locally refined grids placed at these locations. Fig. 7 shows the flow field in the forced ventilated room. Local refined grids are imposed at the inlet and the outlet boundaries. The code **VentAirII** with multi-grid procedure exhibits a much faster convergence than the corresponding single-grid methods. The non-isothermal version of **VentAirII** is under development.

CONCLUSIONS

A numerical code for simulating turbulent air flows in ventilated single- and multi-rooms has been developed. The buoyancy- and radiation-effects are included by Boussinesq approximation and the two-band radiation model, respectively. The code solves, in finite difference form, the transient-state conservation equation for mass, momentum and thermal energy. A multi-grid solver with local grid refinements has been implemented. It is much faster than standard single grid methods (such as SIMPLE procedure). The faster solver together with a rational distribution of the node points is expected to improve numerical efficiency and accuracy.

The effect of wall insulation and radiation on indoor air flow and temperature fields has been studied. The results indicate that the thermal boundary conditions (wall conduction and radiation exchange) on room wall boundaries influence the indoor air flow and indoor air temperature distribution. The natural convection in partially divided rectangular enclosure and the local grid refinement near the inlet and outlet in a 3D forced convection room have also been studied. The predictions show good agreement with the measured results in the literature.

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