Simplified Method for Indoor Airflow Simulation

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ABSTRACT

At present, numerical simulation of room airflows is mainly conducted by either the Computational-Fluid-Dynamics (CFD) method or various zonal/network models. The CFD approach needs a large capacity of computer and a skillful expert. The results obtained with zonal/network models have great uncertainties.

This paper proposes a new simplified method to simulate three-dimensional distributions of air velocity, temperature, and contan-finant concentrations **in** rooms. The method assumes turbulent viscosity to be a function of length-scale and local mean velocity. The new model has been used to predict natural convection, forced convection, mixed convection, and displacement ventilation in a room. The results agree reasonably with experimental data and the CF1) computations. The simplified method uses much less computer memory and the computing speed is at least 10 times faster, compared with the CFD method. The grid number can often be reduced so that the computing time needed for a three-dimensional case can be a few minutes in a PC.

INTRODUCTION

Proper design of indoor environment requires detailed information of indoor air distribution, such as airflow pattern, velocity, temperature, and contarruinant concentrations. The information can be obtained by experimental measurements and computational simulations. Experimental measurements are reliable but need large laboreffort and time. Therefore, the experimental approach is not feasible as a general design tool. Two approaches of computational simulations are available for the study of indoor air distribution. The first approach is the computational-fluld-dynamics (C FD) methods and the second is simplified flow simulation methods.

Computational-Fluid-Dynamics Methods

The CFD methods solve the Navier-Stokes equations for flows. For laminar flows the computed results are accurate and reliable. However, it is difficult to predict turbulent

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flows. Very fine numerical resolution is required to capture all the details of the indoor turbulent flow. This type of simulation is direct numerical simulation. The direct numerical simulation for a practical flow needs a huge computer system that **is** not available.

Indoor airflow simulations use turbulence models to compute the mean values. This can be done with the capacity and speed of present computers. Eddy-viscosity models are the most popular turbulence models. The CFD program with eddy-viscosity models solves air velocities, temperature, contaminant concentration, and turbulent quantities in a space. The space is divided into 10,000 to one million cells to achieve a reasonable accuracy for a threedimensional flow problem. $30 - 21$ **Castled**

In addition, the CFD program user should have good knowledge of fluid dynamics, numerical $\%$ technique, and indoor air distribution. However, a large computer and a skillful ill not \mathcal{A} 冠5 元 記 \sim λ

 $\sigma_{\rm T}$ $f(x, 4, 4, 2, ...)/f$ 1. **CAPPOS** $\mathbb{R}^{\mathbb{Z}_2}$. user wir-uarantee success. Chen:(1997) reported many failures in using the CFD method in a group with more than ten-year experience. Obviously, most HVAC designers and architects do not have the computer capacity and the CFD knowledge. Therefore, in predicting indoor air distribution and designing a comfortable indoor environment, application of the CFD method is limited. (线) 可说 $\mathcal{L}_{\mathbb{R}}$. $\lambda_{\rm B} = 1.5$ $E_{\rm I}$ $\mathcal{C}_{\mathcal{P}}$. C_L 2.11 .

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 $\label{eq:3.1} \begin{array}{cccccccccc} \mathbb{P}^1_{\mathbb{C}} & \mathbb{P}^1_{\$ 9 대 백이 도행 왜 그리 $-.35$ The second approach does not use a turbulence model. The approach uses a much coarse cell system. In most cases, the total cell number for a space is less than 10,000.

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 $J_{\mathcal{F}}$, where $J_{\mathcal{F}}$

A very simple rnethod (Lebrun and Ngendakumana 1987) is to fix airflow patterns and use empirical flow laws for different flow components, such as 'ets, plumes, etc. In many cases, the airflow patterns are difficult to impose even by an experienced fluid dynamics engineer. method has limited applications.

Another popular method is the network model (Walton 1989). The model determines flow within a space by Bernoulli's equation. The method works reasonably for parabolic flows and is ruseful to analyze combined problems of HVAC systems, infiltration, and multi-room airflow simultaneously: However, the uncertainty is large if the method is applied for a room presented by several different cells or sub-volumes. with the psychological state that them a thin

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The method proposed by Wurtz and Nataf (1994) is to calculate indoorgair pressure using a degraded equation for the momentum. The airflow between two zones is determined by the pressure differential. Because of the poor representation of the momentum, the method does not work for pressure and buoyancy driven flows, i.e. flows set up by temperature differences in the air.

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A recent zonal model developed by Inard et al. (1996) calculates flow rate for zones with small momentum through pressure distribution. Although the results are consistent with experimental data, the model may not be applied for high momentum

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flows. In addition, the method uses a discharge coefficient that must be determined through experiment. (2) ism , aon

When a room is subdivided by a partition wall or a large opening, all the above models use a discharge coefficient to calculate flow due to pressure or temperature difference. This will further reduce the reliability of the methods since a general form for the discharge coefficient

3.3 has not been established. Calculations for new geometries require an CFD run or experiment Prento determine the discharge coefficient. $0 - 2$

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 $1.5 - 10$ **Sustification of Need**

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 $10^{5} - 26 - 50$ 18 $\mathcal{L}^{(1)} = \mathcal{L}^{(1)} = \mathcal{L}_{\mathcal{A}} \mathcal{L}^{(1)} = \mathcal{L}_{\mathcal{A}} \mathcal{L}^{(1)}$ $T = 5t^2 - 9$ Many HVAC design engineers and architects have limited knowledge of fluid flow and do not have the access to a large computer. It is important to develop a simplified model to simulate indoor airflow in a personal computer. The flow program should then be coupled with an energy analysis program to simulate simultaneously airflow, thermal comfort, and energy consumption of HVAC systems. The program will also allow the temperature of interior walls to be predicted. The program would serve as a tool to accurately provide design information and ~ 1 to properly size HVAC systems and assure comfort conditions exist at all important locations M within the space. Mr Fig. 1977, Branch and Contract the Re- \sim 10 $-$ 10 $+$ 11 2^{n} , 2^{n} , 3^{n} , 3^{n} , 4^{n} , 5^{n} , 1^{n} , 1^{n} , 1^{n} \mathbb{C} α \mathbb{T}

The goal of the present investigation is to develop a program which will provide design information to establish acceptable comfort conditions through the interior space. Precise rigor and exact predictions will be relaxed to allow the program to be easily used by 11VAC engineers with a minimum of training and modest desktop personal computers. The following **Section describes a new simplified method.** \mathbb{R}^n and \mathbb{R}^n and \mathbb{R}^n are \mathbb{R}^n

 $\sim 10^{-1}$.

 $\label{eq:12} \alpha_{\rm eff} = \frac{N_{\rm eff}}{N_{\rm H}} = 1.001$

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EXAMPLIFIED METHOD AND STUDENTS age was the $\sim 10^{-11}$ ನ್ಯಾಲಿಕ್ ಮಾಡಿ a. 29% *** n. 1 1 50% → 1. 12 % and the state of the state of 调控 百宝 $11 - 100$ **Soverning Flow Equations** $\mathbb{E} \left[\begin{array}{ccc} 0 & \cdots & 0 \\ 0 & \cdots & 0 \end{array} \right]$ ~ 10 \sim $^{-1}$

Most indoor airflows are turbulent. Often airflow calculations use the Buossinesq approximation. The approximation takes air density as constant in the momentum terms and considers the buoyancy influence on air movement by the difference between the local air weight and the pressure gradient. With an eddy-viscosity model, the indoor airflows can be the described by the following time-averaged Navier-Stokes equations for the conservation of mass, momentum, energy, and species concentrations: \mathbb{R}^n and \mathbb{R}^n and \mathbb{R}^n are \mathbb{R}^n

Mass continuity: $\label{eq:3.1} \mathcal{F}^{\text{eff}}_{\text{M}}\mathcal{F}_{\text{max}} = \mathcal{F}^{\text{eff}}_{\text{M}}\mathcal{F}^{\text{eff}}_{\text{M}}$ and the state of the state $\mathcal{L}_{\mathbf{a},\mathbf{b}} = \left\{ \begin{array}{ll} 0 & \text{if } \mathbf{b} \\ \text{if } \mathbf{b} \end{array} \right.$ $\label{eq:Ric} \mathbf{B}(\mathbf{e}) = -\mathbf{W} = -\mathbf{w} - \mathbf{v} = -\mathbf{z} - \mathbf{b} = -\mathbf{v} = -\mathbf{b}^{\mathrm{R}} = -\mathbf{b}^{\mathrm{R}}.$ 작은 오른의 아이들은 아이들이 없었다. τ . $\frac{\partial V}{\partial x_i} = 0$, $\frac{\partial$ $36 - 8Vi$ by $\sqrt{15}$ 5

where $Vi =$ mean velocity component in xi-direction α and α xight = coordinate (for 1= 1, 2, 3, xi corresponds to three perpendicular axes).

Momentum:

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where $p =$ air density

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 $p = pressure$ pt.ff = effective viscosity

- T₁, temperature in a reference point
- T temperature
- gravity acceleration \overline{g}

 Vj = velocity component in xj-direction

 $p =$ thermal expansion coefficient of air

The last term on the right side of the equation is the buoyancy term.

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The turbulent influences are lumped into the effective viscosity is the sum of the turbulent viscosity and larninar viscosity, @I:

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 $PICff = pt t + R$ $104 - 220 - 15$ GAT . (3) $-1.9.02$ **Controlled State**

The Prandtl-Kolmogorov assumption, the turbulent viscosity expresses as the product of turbulence ki energy, k, and turbulent macroscale, 1, that is a proper length scale for turbulence interactions:

"PLt Cvpk 1/21 $\lim_{n \to \infty} \frac{1}{n} \lim_{n \to \infty} \frac{1}{n$ a stadio k

 $766 = 35^{20} \pm 1^{14}$ where C, 0.5478, an empirical constant. Depending on how to solve the unknown parameters k an eddy-viscosity models have different forms. The simplest model is probably the Prandtl's mixing-le model (Prandtl 1926) and complicated ones use multi-equations for turbulence transport. The standar E model (Launder and Spalding 1974) is the most widely used two-equation model.

From the results of our previous investigations, Tu is about 10% for many room airflows. We h found a single algebraic function to express the turbulent viscosity as a function of local mean velocity $\frac{1}{2}$ and a length scale, 1: $\frac{1}{2}$, $\frac{1}{2}$ モウム ヨイリード

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This equation is a universal one without adjustable constants between different flow conditions.

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The pound of a universe part of the second storage in the constant diff 2000 in (6.1) Momentum:

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\frac{\partial \rho V_i}{\partial t} + \frac{\partial \rho V_i V_j}{\partial x_j} = -\frac{\partial \rho}{\partial x_i} + \frac{\partial}{\partial x_j} \left[\mu_{eff} \left(\frac{\partial V_i}{\partial x_j} + \frac{\partial V_j}{\partial x_i} \right) \right] + \rho \beta (T_o - T) g_i \tag{2}
$$

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where $\rho = \text{air density}$

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 V_i = velocity component in x_i -direction

 $p = pressure$

 μ_{eff} = effective viscosity

 β = thermal expansion coefficient of air

 T_o = temperature in a reference point

 $T = temperature$

 $g =$ gravity acceleration

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The last term on the right side of the equation is the buoyancy term.

The turbulent influences are lumped into the effective viscosity is the sum of the turbulent viscosity, μ_t , and laminar viscosity, μ :

 $2\, \mathrm{g}^{-1/2}$, where g

 $\label{eq:G} \mathcal{G}_{k}(\mathcal{S}) = \mathcal{G}(\mathcal{G}) \quad \text{and} \quad \mathcal{G}_{k}(\mathcal{S}) = \mathcal{G}_{k}(\mathcal{S})$ $\mu_{\rm eff} = \mu_{\rm t} + \mu^{-1} \qquad \qquad \qquad \ldots \qquad \qquad \qquad \gamma_{\rm eff} = \gamma_{\rm eff} \qquad \qquad \gamma_{\rm eff} = \gamma$ (3)

The Prandtl-Kolmogorov assumption, the turbulent viscosity expresses as the product of turbulence kinetic energy, k, and turbulent macroscale, l, that is a proper length scale for turbulence interactions:

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$$

where $C_v = 0.5478$, an empirical constant. Depending on how to solve the unknown parameters k and l, eddy-viscosity models have different forms. The simplest model is probably the Prandtl's mixing-length model (Prandtl 1926) and complicated ones use multi-equations for turbulence transport. The standard k- ε model (Launder and Spalding 1974) is the most widely used two-equation model.

 $\mathcal{P}_1 = \sum_{i=1}^n \mathbb{P}^1_{i} \mathcal{P}_i \mathcal{P}_i = \mathbb{P}_1 \mathcal{P}_1 \mathcal{P}_2 \mathcal{P}_2 \mathcal{P}_3 \mathcal{P}_4 \mathcal{P}_5$ The contract of state infinite and an accountable that a last

> From the results of our previous investigations, Tu is about 10% for many roon airflows. We have found a single algebraic function to express the turbulent viscosity as a function of local mean velocity, V, and a length scale, l:

$$
\mu_{\rm t} = 0.03874 \, \rho \, \text{V} \, \text{l} \tag{5}
$$

This equation is a universal one without adjustable constants between different flov conditions.

Energy:

To determine the temperature distribution and the buoyancy term in Equation (2), the conservation of energy must be solved.

> apT apVjT a aT "T,eff at *axiaxi* axi $+ q / Cp$ (6)

where FT,ff effective turbulent diffusion coefficient for T q = thermal source CP = specific heat

In our work we have estimated the effective diffusive coefficient for temperature in Equation FT, ff, by:

 Reff (7) FT'ff - Pr,ff

where the effective Prandd number, Pr,ff, is 0.9.

Species concentrations:

For determination of pollutant or water vapor concentration distribution the conservation of mass must be combined with the equation of transfer of the species.

> apc Ovic a ac 17c,eff - +SC
at *axi* axi axi (8)

where C = species concentration

Fc,ff =effective turbulent diffusion coefficient for C Sc= source term of C

Similar method to the energy equation is used to determine the effective diffusive coefficient species concentration in Equation (8), 1-c,,ff:

F 1-1 Cff (9)

C'eff - Seeff

where effective Schmidt number, Sc,ff, is 1.0.

Equations (1) to (9) form the new simplified model.

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tarian realizanta enc. s.f., s.d (c) of $t^2 \to \infty \times \mu \mathbb{B}$

 $\overline{\mathcal{M}}$ Energy:

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the conservation of energy must be solved. $\mathbf{1}^{\circ}$ $\frac{\partial \rho T}{\partial t} + \frac{\partial \rho V_j T}{\partial x_j} = \frac{\partial}{\partial x_j} \left(\Gamma_{T,\text{eff}} \frac{\partial T}{\partial x_j} \right) + q/C_p$ (6)

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To determine the temperature distribution and the buoyancy³ term in Equation (2)

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where $\Gamma_{T,eff}$ = effective turbulent diffusion coefficient for T $q =$ thermal source C_p = specific heat \mathbb{R} $\mathbb{$

In our work we have estimated the effective diffusive coefficient for temperature in M. D. RH 消息説像 Equation (6), $\Gamma_{T,eff}$, by:

$$
\Gamma_{\text{T,eff}} = \frac{\mu_{\text{eff}}}{\mathbf{P}_{\text{eff}}} \tag{7}
$$

where the effective Prandtl number, Pr_{eff} , is 0.9.

Species concentrations:

For determination of pollutant or water vapor concentration distribution the conservation of mass must be combined with the equation of transfer of the species.

$$
\frac{\partial \rho C}{\partial t} + \frac{\partial \rho V_j C}{\partial x_j} = \frac{\partial}{\partial x_j} \left(\Gamma_{C,eff} \frac{\partial C}{\partial x_j} \right) + S_C
$$
\n(8)

where $C =$ species concentration

 $\Gamma_{C,eff}$ = effective turbulent diffusion coefficient for C

 S_C = source term of C_{dR} and is a reserve entered enter the term of C_{dR}

Similar method to the energy equation is used to determine the effective diffusive ant adj and the theory coefficient for species concentration in Equation (8), Teenmanoon ix entry

bottrdary conditions of the surface."

$$
\Gamma_{\text{C,eff}} = \frac{\mu_{\text{eff}}}{\text{Sc}_{\text{eff}}} \tag{9}
$$

where effective Schmidt number, Sc_{eff} is 1.0.

Equations (1) to (9) form the new simplified model.

Boundary Conditions

Boundary conditions are necessary for the mathematical solution of the govern' ing flow equations. There are three types of boundaries of practical importance: free boundary, symme surface, and conventional boundary.

Single graphs Free boundary

The boundary surface may be adjacent to an inviscid stream. Examples are air supply outle and return inlet. For a supply outlet. the boundary conditions are:

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where subscripts "supply" are the parameter values at the supply outlet.

Pressure is normally given for a return inlet and zero gradients normal to the surface are assumed for other parameters:

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axi 10^{15} ~ 3.8 $\sigma = -\frac{1}{K}$ \mathcal{U} 哀討し $2\sqrt{2} \times \sqrt{2}$ - 5 $A = \mathbb{R}^{13}$ e. S $\Omega = 210$. The α $\,6\,$ ाउँ $\label{eq:3.1} \mathbb{E}[\frac{\partial V^2}{\partial t} - \frac{1}{2} \frac{d\mathbf{g}}{d\mathbf{g}} \mathbf{v} - \mathbf{w} \mathbf{v}^T] = - \frac{2\mathbf{g} \mathbf{f}^T \mathbf{v}}{2 \mathbf{g}^T} \mathbf{v} \mathbf{v} = \mathbf{v} \mathbf{v}^T \mathbf{v}^T$ $\mu^{(2k+1)}$ \sim χ J^{11} $\frac{1}{\pi}$

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Boundary Conditions

Boundary conditions are necessary for the mathematical solution of the governing flow equations. There are three types of boundaries of practical importance: free boundary, symmetry surface, and conventional boundary.

• Free boundary

The boundary surface may be adjacent to an inviscid stream. Examples are air supply outlet and return inlet. For a supply outlet, the boundary conditions are:

$$
V_i = V_{\text{supply}}
$$

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$$
T = T_{\text{supply}}
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$$
C = C_{\text{supply}}
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\n(10)

where subscripts "supply" are the parameter values at the supply outlet.

Pressure is normally given for a return inlet and zero gradients normal to the surface are assumed for other parameters:

$$
p = p_{return}
$$

\n
$$
\frac{\partial V_i}{\partial x_i} = 0
$$

\n
$$
\frac{\partial T}{\partial x_i} = 0
$$

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$$
\frac{\partial C}{\partial x_i} = 0
$$
\n(11)

where p_{return} is the pressure at a return inlet.

Symmetry surface

If the x_i coordinate is normal to the symmetry surface, the following equations describe the boundary conditions of the surface:

$$
\frac{\partial V_i}{\partial x_i} = 0
$$

$$
\frac{\partial T}{\partial x_i} = 0
$$

$$
\frac{\partial C}{\partial x_i} = 0
$$

(12)

Conventional boundary

This type of boundary surfaces includes wall, ceiling, and floor Surfaces and the surfaces of furniture, appliance, and occupants. If xi coordinate is parallel to the surface, the boundary conditions are:

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 $T = @$ lelT ∂X

 $q = h(Tw - T)$ (13)

S C C source

where $T =$ shear stress

h = convective heat transfer coefficient

Csource = species concentration source

The convective heat transfer coefficient is determined from the follows](, equation which

is similar to the Reynolds analogy:

Pt eff C p

Preff AX j (14)

where Axj is the distance between the surface and the first arid close to the surface.
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APPLICATION EXAMPLES

This section demonstrates the new simplified method by applying it to predict indoor airflows of.

Natural convection Forced convection

Mixed convection

0 Displacement ventilation

Natural, forced, and mixed convoction roprosont tho basic olomonts of room airflows. For simplicity, two-dimenslonal case selected to demonstrate the new simplified model. The displacement ventilation case used is three-dimensional with complicated boundary conditions. The displacement ventilation case is a test of the overall performance of the new model.

Natural Convection

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 $\label{eq:2.1} \begin{array}{ccc} \mathbf{A}^{\mathrm{H}} & \mathbf{A}^{\mathrm{H}} & \mathbf{A}^{\mathrm{H}} \\ \mathbf{A}^{\mathrm{H}} & \mathbf{A}^{\mathrm{H}} & \mathbf{A}^{\mathrm{H}} \end{array}$

For natural convection, the experimental data of 01son and Glicksman (1991) as shown in Fig. 1 will be used. $\label{eq:4} \mathcal{L} = \frac{1}{2} \sum_{i=1}^n \frac{1}{2} \sum_{i=1}$

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 $\label{eq:2.1} \mathcal{E}_{\mathcal{A}} = \mathcal{E}_{\mathcal{A}} + \mathcal{E}_{\mathcal{A}} + \mathcal{E}_{\mathcal{A}} + \mathcal{E}_{\mathcal{A}}$

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 $\label{eq:R1} \begin{array}{c} \left\langle x \right\rangle \stackrel{A_1}{\longrightarrow} \left\langle x \right\rangle \stackrel{A_1}{\longrightarrow} \left\langle x \right\rangle \stackrel{A_2}{\longrightarrow} \left\langle x \right\rangle \stackrel{A_3}{\longrightarrow} \left\langle x \right\rangle \stackrel{A_2}{\longrightarrow} \left\langle x \right\rangle \stackrel{A_3}{\longrightarrow} \left\langle x \right\rangle \stackrel{A_4}{\longrightarrow} \left\langle x \right\rangle \stackrel{A_5}{\longrightarrow} \left\langle x \right\rangle \stackrel{A_6}{\longrightarrow} \left\langle x \right\rangle \stackrel{A_6}{\longrightarrow} \left\langle x$

 $2\times 10^{-5} E^2 \cdot 495 \mathcal{C} \approx 0.5 \cdot 10^{-25} \qquad \mathrm{W}$

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Conventional boundary

This type of boundary surfaces includes wall, ceiling, and floor surfaces and the surfaces of furniture, appliance, and occupants. If x_i coordinate is parallel to the surface, the boundary conditions are:

$$
\tau = \mu_{eff} \frac{\partial V_i}{\partial x_j}
$$

q = h(T_w - T)
S_C = C_{source} (13)

where τ = shear stress

 $h =$ convective heat transfer coefficient

 C_{source} = species concentration source

The convective heat transfer coefficient is determined from the following equation which is similar to the Reynolds analogy:

$$
h = \frac{\mu_{eff}}{Pr_{eff}} \frac{C_p}{\Delta x_i}
$$
 (14)

where Δx_i is the distance between the surface and the first grid close to the surface.

APPLICATION EXAMPLES

This section demonstrates the new simplified method by applying it to predict indoor airflows of:

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- Natural convection
- Forced convection
- Mixed convection
- Displacement ventilation

Natural, forced, and mixed convection represent the basic elements of room airflows. For simplicity, two-dimensional cases are selected to demonstrate the new simplified model. The displacement ventilation case used is three-dimensional with more complicated boundary conditions. The displacement ventilation case is a test of the overall performance of the new model.

Natural Convedion

For natural convection, the experimental data of Olson and Glicksman (1991) as shown in Fig. l will be used.

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 $\mathbf{p}^{(n)} \in \mathbb{R}^{n \times n} \times \mathbb{R}^{n \times n}$

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layer thickness are not correct for the simplified model due to the large cell size used. However, we found only the Lam-Bremhorst model could predict the reversed flow when we tested quite a few eddy-viscosity models.

Fig. 3 presents the dimensionless temperature profiles in the vertical center line. The simplified model predicts the temperature profile better than the CFD model in this particular case.

Forced Convection

The forced convection case uses the experimental data from Restivo (1979) shown in Fig. 4. The Reynolds number is 5000 based on bulk supply velocity and the height of air supply outlet. The air supply outlet $h = 0.056$ H, and exhaust inlet $h' = 0.16$ H.

Fig. 5 compares the airfow patterns by the simplified model and the CFD method with standard k- ϵ model (Launder and Spalding 1974). The computed velocity profiles are compared in Fig. 6 with experimental data in two vertical sections $x/H = 1$ and $x/H = 2$ respectively and two horizontal sections, $\sqrt{H} = 0.972$ (through the air supply outlet) and $y/H = 0.028$ (through the air exhaust inlet). The results of the simplified model show a jet decay that is too strong. Hence, the primary flow near the ceiling and the return flow near the floor are smaller than the data. In this case, the k - ε model predicts a satisfactory result. Nevertheless, the simplified model could predict the second recirculation on the upper right corner, though the recirculation is too large. However, the k- ϵ model fails to predict the recirculation.

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Fig. 2. Comparison of the airflow patterns for natural convection: (a) simplified method, (b) CFD method, (c) smoke visualization.

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Fig. 5. Comparison of the airflow patternsfor theforced convection: (a) simplified
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Fig. 3. Comparison of temperature profile in vertical line at the middle of the room with natural convection.

Fig. 5. Comparison of the airflow patterns for the forced convection: (a) simplified method, (b) CFD method..

Mixed Convection and a set of the set of the

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The mixed convection case uses the experimental data from Schwenke (1975). The cas similar to the forced convection but the room len-th is 4.7 H and the height of the air su outlet h = 0.025 H. The right wall is heated but the ceiling and floor are adiabatic. Schwe conducted a series measurements with different Archimedes numbers, Ar, ranging from 0. **to 0.02.** *¹ ·*

Fig. 6. Comparison of velocity profiles in differentsections of the room with forced convection: (a) atvW = 1, (b) at x1H = 2, (c) (it y1H = 0. 9 72, and (d) at y1H = 0. 028. 2.35 外表 100

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Mixed Convection

The mixed convection case uses the experimental data from Schwenke (1975). The case is similar to the forced convection but the room length is 4.7 H and the height of the air supply outlet $h = 0.025$ H. The right wall is heated but the ceiling and floor are Schwenke conducted a series measurements with different Archimedes adiabatic. numbers, Ar, ranging from 0.001 to 0.02.

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Fig. 6. Comparison of velocity profiles in different sections of the room with forced convection: (a) at $x/H = 1$, (b) at $x/H = 2$, (c) at $y/H = 0.972$, and (d) at $y/H = 0.028$.

Fig. 7 compares the computed airflow pattern by the simplified method with that by the CFD method with the standard k-F- model. The two results are similar. The airflow pattern is very sensitive to the Ar. The computed and measured penetration depths, xe, versus different Ar numbers are compared in Fig. 8. The x, is the horizontal distance of air movement along the ceiling before it falls to the floor. The simplified model works better in high Ar but the CFD model better in low Ar.

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Fig. 7. Comparison of the aifflow patternsfor the mixed convection: (a) simplified method, (b) CFD method...

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Fig. 8. Comparison of the penetration length versus Archimedes numberjor the room with mixed $\{T_i\}$ convection. ÷. 抗常 \pm iv

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Fig. 7 compares the computed airflow pattern by the simplified method with that by the CFD method with the standard k- ε model. The two results are similar. The airflow pattern is very sensitive to the Ar. The computed and measured penetration depths, x_e, versus different Ar numbers are compared in Fig. 8. The x_e is the horizontal distance of air movement along the ceiling before it falls to the floor. The simplified model works better in high Ar but the CFD model better in low Ar.

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Fig. 7. Comparison of the airflow patterns for the mixed convection: (a) simplified $\mathcal{R}^{\mathbb{Z}}$ method, (b) CFD method... $\omega^{\frac{2}{2}}$: y.,

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online or au on Eig. 8. Comparison of the penetration length versus Archimedes number for the room with mixed convection.

Displacement Ventilation

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Fig. 9 shows the application of the simplified method and the CFO method with the standard k-F, model for the prediction of room airflow with a displacement ventilation system. The room dimension is 5.6 m long, 3.0 m wide, and 3.2 m high. A convective heat source, of 530 W on the window was used to simulate a summer cooling condition. The supply airflow rate was five air-change per hour. The corresponding supply air temperature was 19 OC. A box placed near the table was heated by a 25 W lamp to simulate a person sitting next to the table. The heat strength is considerably lower than that generated from an occupant. However, a helium source was also introduced in the box as a tracer gas to simulate contaminant from the occupant, such as \cos or tobacco smoke. The helium flow rate was 0.5% of the air supply rate. Since helium is much lighter than the air and the helium source was relatively strong in the room, the combined buoyant effect from the thermal source (heat from the lamp) and the mass source (helium) was as strong as that generated from an occupant.

> The computations were carried out with different grid numbers with the in a simplified the state of the problem

method: 31 x 28 x 26 (the same as the CFD method), 16 x 14 x 12, 10 x 10 x 10, and \therefore 6x

⁷·x 6., ··A grid number of 16 *x* 14 *x* 12 is minimum in order to represent the room geometry, such as the inlet, outlets, window, and table. Fig. 9 shows similar airflow patterns and the distributions of air temperature and helium concentration computed by the simplified method with 16 x 14 x 12 grids and the CFD method with 31 x 28 x 26 grids. Fig. 10 further compares the computed results with experimental data. The velocity and temperature profiles are at the center of the room and helium concentration profile at a line near to the center of the room. The agreement between the computed and measured results is reasonably good. The results are nearly identical between the simplified and CFD methods if the grid number is the same. It is possible to use a minimum grid number of $6 \times 7 \times 6$ with which the table in the room cannot be represented. The accuracy of the results is relaxed but it does predict the main features of displacement ventilation, such as temperature gradient, non-uniform distribution of contaminant concentration, and higher risk of draft near the inlets at the floor level. The minimal grid number seems less than that used in zonal models. Therefore, the simplified method has a great potential to be used in an hour-by-hour energy simulation program to take into account the impact of non-uniform temperature distribution on energy consumption.

Note in all of the cases, the simplified model, Equation (5), is exactly the same. No adjustable constants were used in the computations. The simplified model is universal for room. airflow simulation: -2

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 $DISCUSSION$ Table 1 shows the total grid number used in the four cases by the simplified model

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and the CFD models. It also shows the memory needed and CPU time used. The \sim

convergence residuals are the same between the simplified and CFD computations. The residuals, R, are defined as:

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 $71 - a$ **ALCOHOL:** $\label{eq:4} \begin{array}{ll} \alpha & \alpha & \beta \\ \alpha & \beta & \gamma \end{array} \quad \mbox{and} \quad \begin{array}{ll} \alpha_{\rm{Q}} & \gamma_{\rm{P}} \end{array}$ -1 \sim 100 μ $\label{eq:1.1} \mathbb{E} \left[\begin{array}{ccc} \mathbb{E} & \mathbb{E} & \mathbb{E} & \mathbb{E} \\ \mathbb{E} & \mathbb{E} & \mathbb{E} & \mathbb{E} \end{array} \right]$ -1 $\{\zeta \rightarrow \zeta\}$ -1.47 11.713777 $\mathcal{L} = \{ \sigma_1, \sigma_2, \ldots, \sigma_{n-1} \}$ de comme and \mathcal{F} \rightarrow - \mathcal{H} - \mathcal{X} - \mathcal{V} \mathcal{L}^{L} in \mathbf{A} . The set of \mathbf{A} $\label{eq:2.1} g_k(\theta)=\theta-\gamma-\gamma-1, \quad \text{and} \quad \quad k=1,\ldots, \quad \forall k\in\mathbb{N}.$ - 50 2 31 31 $\mathbb{R}^n \times \mathbb{R}^n$. In , we can $\sqrt{Y} = -2\pi \sqrt{1-\frac{1}{2}}$ $\label{eq:4} \mathcal{L}(\mathbf{z}) = \mathcal{L}(\mathbf{z}) = \mathcal{L}(\mathbf{z}) = \mathcal{L}(\mathbf{z}) = \mathcal{L}(\mathbf{z}) = \mathcal{L}(\mathbf{z})$ $\label{eq:V} V = \frac{1}{2} \left(\begin{array}{ccc} 1 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0$ 1927 16:12 $\frac{1}{3}$ x $\mathbf{u}^{\mathcal{A}}\colon\mathbb{C}^{\mathbb{Z}/2}\longrightarrow\mathbb{C}^{\mathbb{Z}/2}\longrightarrow\mathbb{C}^{\mathbb{Z}/2}\longrightarrow\mathbb{C}^{\mathbb{Z}/2}\longrightarrow\mathbb{C}^{\mathbb{Z}/2}\longrightarrow\mathbb{C}^{\mathbb{Z}/2}\longrightarrow\mathbb{C}^{\mathbb{Z}/2}\longrightarrow\mathbb{C}^{\mathbb{Z}/2}\longrightarrow\mathbb{C}^{\mathbb{Z}/2}\longrightarrow\mathbb{C}^{\mathbb{Z}/2}\longrightarrow\mathbb{C}^{\mathbb{Z}/2}\longrightarrow\mathbb{C}^{\mathbb{Z}/2}\longrightarrow\mathbb{C}^{\mathbb{Z}/$ **PERMIT AT ALC:** $M_{\rm H}$, $S_{\rm eff}$ = $=$

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Displacement Ventilation

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DISCUSSION

Table 1 shows the total grid number used in the four cases by the simplified model and the CFD models. It also shows the memory needed and CPU time used. The convergence residuals are the same between the simplified and CFD computations. The residuals, R, are defined as:

 $f\circ f = \left(\frac{a}{r}\right)$ and $\overline{\langle \overline{\chi} \rangle}$ $p \in \mathcal{D}(\mathcal{G}_t)_{k \in \mathbb{Z}}$

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Fig. 9. Comparison of the airflow patterns and distribution of air temperature ('C) and helium concentration (%): (a), (b), and (c) simplified method and (d), (e), and (f) CFD $method$.

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Fig. 1 0. Comparison of the profiles ofair velocity, air temperature, and helium concentration in a vertical line ofthe room.

 $\label{eq:2.1} \begin{split} \mu(\alpha) &= \kappa e^{-\frac{2\pi i}{\hbar^2}} \frac{\kappa}{\kappa} \\ &= -\frac{\kappa}{\hbar^2} \frac{\sqrt{\kappa}e^{\kappa}}{\kappa} = 0 \end{split}$

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Fig. 10. Comparison of the profiles of air velocity, air temperature, and helium concentration in a vertical line of the room.

Table 1. Comparison of computizg performance (@f the simplified and CFD models

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Y Y, Y, 1 residuals in a cell

 R_1 = $k=1$

reference value (15)

where $NX = total$ cell number in x direction $NY = total$ cell number in y direction $NZ = total$ cell number direction

The reference value is the total air supply rate for mass continuity and heat from a heated/cooled wal energy. The present investigation uses R < 0.001 for mass continuity and R< 0.0 1 for energy.

The computations were conducted in a 486 personal computer. The simplified model uses much memory than the CFO model. The simplified method is at least 10 times faster than the CFO method. results show that most room airflow simulation can be done with a personal computer and the compu time for each case is in the order W

of a few seconds for a two-dimensional problem and a few minutes for a three-dimensional case.

1. CONCLUSIONS

This paper proposes a new simplified method for the prediction of room airflow pattern and the distributions of air temperature and contaminant concentrations. The model is derived from the Navier-Stokes equations. Using the concept of eddy-viscosity, turbulent viscosity is approximated by a length scale and mean velocity. The main difference between the simplified method and the conventional CFD approach with a k-F-

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Table 1. Comparison of computing performance of the simplified and CFD models

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The reference value is the total air supply rate for mass continuity and heat from a heated/cooled wall for energy. The present investigation uses $R₁ < 0.001$ for mass continuity and $R < 0.01$ for energy $\frac{1}{3}$ bink, $\frac{1}{2}$ is the solution of the M

The computations were conducted in a 486 personal computer. The simplified Suit T main model uses much less memory than the CFD model. The simplified method is at least 10 times faster than the CFD method. The results show that most room airflow simulation 此种价 1 明 9 can be done with a personal computer and the computing time for each case is in the order 3.95 of a few seconds for a two-dimensional problem and a few minutes for a three-dimensional case.

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Prande, L. 1926. "Ubher die ausgebisse Turbuienz," Proc. 2nd br. - Chroness for Apur.

This paper proposes a hew simplified method for the prediction of room airflow pattern and the distributions of air temperature and contaminant concentrations. The to vera vint. . "model is derived from the Navier-Stokes equations." Using the concept of eddy-viscosity, turbulent viscosity is approximated by a length scale and mean velocity. The main difference between the simplified method and the conventional CFD approach with a k- ϵ

model is that the former does not solve transport equations for turbulent quantities. The simplified method use a new zero-equation model.

The study demonstrates the capability of the simplified model by applying it to predict the airflow with natural convection, forced convection, mixed convection, and \blacksquare displacement ventilation in rooms. The predicted results are compared with experimental data and the results of CFD simulations. The simplified method can predict reasonably good indoor airflow patterns and the distributions of air temperature and contaminant concentrations.

Since the simplified model does not solve transport equations for turbulence, the computer memory needed is much smaller, and the convergence speed is 10 times faster than that with a CFO model. With the simplified model, simulation of a three-dimensional, steady-state flow in a room can be made in a personal computer. In addition, the user does not need the knowledge of turbulence modeling.

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