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Evaluation of Displacement and Mixing Modes of Mechanical Ventilation Enclosure

تقييم نماذج التهوية بالإزاحة والخلط في حيز

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ملخص البحث

يقدم هذا البحث دراسة عددية ومقارنة بين أداء حالات التهوية بالإزاحة والتهوية بالخلط تحت شروط حديه مختلفة. تظهر نتائج الدراسة على هيئة الراحة الحرارية وجودة الهواء الداخلي. تم دراسة أداء أربعة حالات بالتهوية بالإزاحة وحالتين بالتهوية بــالخلط. تم دراسة تأثير كل من عدد رينولدز وعدد شميدت على أداء بالتهوية عن طريق الإزاحة والخلط. أظهرت النتائج في حالة التهويــة بالخلط بأن اقل أداء ظهر في الحالة التي يكون الدخول والخروج من اعلي (السادسة) بينما أفضل أداء في الحالة التي يكون الــدخول من اعلي والخروج من أسفل على الجانبين (الخامسة) عن طريق التهوية بالإزاحة الحالة التي يكون الدخول من اسفل على الجــانبين والخروج من اعلي (الرابعة). أظهرت الدراسة أيضا أن كل من معدل انتقال الكتلة والطاقة وفاعلية إزالة الحرارة والتلوث وكفاءة نظافة الهواء تتأثر بتغير كل من عدد رينولدز وعدد شميدت.

Abstract

A numerical simulation using a computational fluid dynamics model is used to investigate and compare the performance of displacement and mixing ventilation modes under different boundary conditions. The companion paper reports the results in terms of the thermal comfort and the indoor air quality. Four modes (cases 1, 2, 3 and 4) of displacement ventilations and two modes (cases 5 and 6) of mixing ventilations are studied. The influence of Reynolds and Schmidt numbers on the performance of displacement and mixing ventilations are presented. The results show that the lower performance in case 6 for mixing ventilation while the higher performance in case 5 for mixing ventilation and case 4 for displacement ventilation. The heat and mass transfer rates, the heat and contaminant removal effectiveness and the air clearing efficiency are influenced by the change of the Reynolds and Schmidt numbers.

Keywords: CFD; Mixed convection; Displacement and mixing ventilation; heat/Contaminant sources

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1. Introduction

Prediction of indoor environment is very important for analysis of energy consumption and indoor air quality. It is well known that there exist temperature variations in different zones of a room. The temperature of the ''occupied zone'' where the occupants stay is usually different from that of the ''discharge zone'' of the conditioning air. In room air distribution, there are usually two methods of supplying the air: either mixing ventilation or displacement ventilation. In mixing ventilation, air is normally supplied at high level over the ceiling which is then deflected down into the occupied zone by the opposite walls thus causing a mixing of the air jet with room air. In displacement ventilation, the air is supplied at low level, usually over the floor, and then rises up due to buoyancy before it is extracted at high level. In displacement ventilation, both natural and forced convection must be taken into account. The interaction between the two convective flows is important to achieve the effectiveness of ventilation. Visualizing the processes may provide a new means to understand the philosophy.

The air movement inside the room also affects the performance of displacement ventilation system. Melikov et al. [1] studied the air flow characteristics in the occupied zone. Flow visualization indicated the convective flow generated above the heat sources and the development of the stratified layer. Xing et. al.[2], Park and Holland [3], Wyon and Sandberg [4], Gan [5 and 6], Lin et al. [7], Deng et al. [8], Chen et al. [9], and Nielsen [10] presented a study of the air distribution or air quality or thermal comfort in the breathing zone in a room or enclosure with displacement ventilation. Lian [11] investigated the effect of the type of outlet, distance between the occupant and outlet, velocity and temperature of supply air, and the type of outlet on displacement ventilation. The results show that the main influence on the thermal comfort was the distance between the occupant and the supply. Shaun and Andrew [12] investigate the steady state natural ventilation of a room heated at the base and consisting of two vents at different levels. These findings suggest how the appropriate ventilation strategy for a room can be varied depending on the exterior temperature, with mixing ventilation more suitable for winter conditions and displacement ventilation for warmer external temperatures.

Awbi [13] compared the effectiveness of mixing and displacement ventilation in terms of heat and contaminant removal. The CFD simulations and the measurements suggest that displacement ventilation was more energy efficient than a mixing system. Deng and Tang [14] investigated two-dimensional, steady and laminar for the displacement and mixed ventilation models. The results and comparisons showed that the displacement ventilation guarantees a high indoor air quality (IAQ) and was therefore a desired air-conditioning system. Linden et al. [15] studied the natural ventilation inside a room by using the ''emptying filling boxes method'' to investigate the plume spreading in a density stratified flowing fluid (brine). They compared the flow pattern of mixing ventilation and displacement ventilation under various loading conditions. They also proposed a formula to estimate the height of the displacement zone.

The goal of the present work was to study in detail and compare the performance of displacement and mixing ventilation modes. Four modes (cases 1, 2, 3 and 4) of displacement ventilations and two modes (cases 5 and 6) of mixing ventilations are studied. The influence of Reynolds and Schmidt numbers on the performance of displacement and mixing ventilations models are presented. The companion paper will report the results in terms of the thermal comfort and the indoor air quality. The thermal comfort parameters are expressed by temperature distribution and ventilation effectiveness for heat removal (ε_h). The indoor air quality is determined by investigating the contaminant distribution, ventilation effectiveness for the contaminant removal (ε_c), and the air cleaning efficiency (ξ).

2. Physical description of model

The physical model of displacement and mixing ventilation models under consideration here is schematically shown in Fig. (1). It is a two-dimensional ventilated square section room, with sides of length L. A heat source (H.S.) of size L_{hs} and high temperature t_h, located at the center of the floor wall, and two contaminant sources (C. S.) of size L_{cs} and high concentration c_h each located at the center of the right and left walls. The fresh cold air (u_{in}, t_{in}, c_{in}) is supplied from the inlet and the polluted hot air exhausted from the outlet at different locating walls as shown in the figure. The inlet and outlet are of the same size, h. Other parts of the room are all considered adiabatic. The different locations of the air inlets and outlets for displacement ventilation models are shown in Fig. (1a, 1b, 1c, 1d.) and for mixing ventilation models in Fig. (1e, and 1f).

3. Governing equations

The analysis is based on the two-dimensional steady continuity, momentum, concentration and energy equations in dimensionless form. The flow is steady, incompressible, and laminar. The air and the contaminant gases are perfectly mixed. All the thermo-physical of the fluid properties are constant, except that of the density in the buoyancy term of the momentum equation following the Boussinesq approximation.

Considering the above assumptions, the governing equations are non-dimensionalized using scales L, u_{in}, $\Delta t=(t_h-t_{in})$, and $\Delta c=(c_h-c_{in})$ for length, velocity, temperature, and concentration respectively. Accordingly the dimensionless variables are (X,Y)=(x,y)/L, (U,V)=(u,v)/u_{in}, $P=p/\rho u_{in}^2$, $\theta=(t-t_{in})/\Delta t$, $C=(c-c_{in})/\Delta c$ and dimensionless parameters are defined as $Pr=\upsilon/\alpha$, $Gr=g\beta_t \Delta t L^3/\upsilon^2$, $Re=u_{in}L/\upsilon$, $Sc=\upsilon/D$, $Br=\beta_c \Delta c/\beta_t \Delta t$

The governing equations for airflow in a ventilated room include continuity, momentum, energy and concentration, can be written in the general form as the following:

$$\frac{\partial}{\partial X}(\beta\Phi U)+\frac{\partial}{\partial Y}(\beta\Phi V)=\frac{\partial}{\partial X}(\Gamma\frac{\partial\Phi}{\partial X})+\frac{\partial}{\partial Y}(\Gamma\frac{\partial\Phi}{\partial Y})+(S_u+S_p\Phi) \qquad (1)$$

The Φ, β, Γ, S_u and S_p' used in Eq. (1) are given in Table 1. The boundary conditions for U, V, θ and C are given as follows:

Inlet: U=1, V= 0, θ=0 and C=0

Outlet: $\partial U/\partial n=0$ and $\partial\theta/\partial n=0$, $\partial C/\partial n=0$

Walls: U= V= 0, θ=1 for the heat source and $\partial\theta/\partial n=0$ elsewhere, C=1 for contaminant source and $\partial C/\partial n=0$ elsewhere.

The local Nusselt number (Nu) and Sherwood number have been evaluated

$Nu(X) = -(\partial\theta/\partial Y)|_{Y=0}$ and

$Sh(Y) = (\partial C/\partial X)|_{X=0} + (\partial C/\partial X)|_{X=1}$

The average Nusselt and Sherwood numbers describe the heat and mass transfer rates on the surfaces of heat and contaminant sources are expressed as follows

$$Nu_m = \frac{1}{\varepsilon_{hs}}\left[\int_{(1-\varepsilon_{hs})/2}^{(1+\varepsilon_{hs})/2} -(\partial\theta/\partial Y)|.dX\right]_{Y=0} \quad (2)$$

$$Sh_m = \frac{1}{\varepsilon_{cs}}\left[\int_{(1-\varepsilon_{cs})/2}^{(1+\varepsilon_{cs})/2} -(\partial C/\partial X).dY\right]_{X=0} + \frac{1}{\varepsilon_{cs}}\left[\int_{(1-\varepsilon_{cs})/2}^{(1+\varepsilon_{cs})/2} -(\partial C/\partial X).dY\right]_{X=1} \quad (3)$$

Since ε_{hs} and ε_{cs} is set equal to 0.5 for the study

The ventilation effectiveness for heat removal (ε_h) reflects the ability of the ventilation system to remove heat and it is defined by:

$$\varepsilon_h = \frac{\theta_{out} - \theta_{in}}{\theta_m - \theta_{in}} \quad (4)$$

The Ventilation effectiveness for contaminant removal (ε_c) is a measure of how effective the ventilation system is in removing internally produced contamination. It is defined by:

$$\varepsilon_c = \frac{C_{out} - C_{in}}{C_m - C_{in}} \quad (5)$$

where θ_{in} and C_{in} are the dimensionless temperature and contaminant concentration of inlet air, (typically zero), respectively; θ_{out} and C_{out} are the dimensionless temperature and contaminant concentration of outlet air, respectively; θ_m and C_m are the mean values for dimensionless temperature and contaminant concentration of air in the occupied zone, respectively.

ASHRAE [16] recommends using the air cleaning efficiency, ξ, to evaluate such configurations, where ξ is calculated from:

$$\xi = 1 - C_{out} \quad (6)$$

4. Numerical procedure

The present two-dimensional problem has been treated numerically using the control volume finite difference method described by Patankar [17]. The computational domain in two dimensions has been discretised using control volumes of uniform size. The discretisation equations have been obtained by applying conservation laws over a finite size control volume surrounding the grid node and integrating over the control volume, Versteeg and Malalasekera [18]. Convection–diffusion terms have been treated by the upwind scheme and diffusion and source terms have been treated by the central differencing scheme. Resulting algebraic equations have been solved simultaneously using the Gauss-elimination procedure until convergence has been achieved. The convergence of results were assumed to be reached when the maximum relative changes of all velocity, temperature, stream function, concentration, and vorticity values between the consecutive iterations were less than 10^{-6}. The above mentioned numerical method has been implemented in a self-written FORTRAN computer code that has been developed to simulate systems of coupled non-linear partial differential equations in two dimensions. In order to obtain a numerical solution that was not affected by the grid size, different grid sizes: 40×40, 65×65, 81×81, 97×97 and 104×104 for the same problem were tested. Considering the accuracy requirement, the grid size used in this study was 97×97.

5. Model validation

To validate the mathematical and numerical model, the displacement mode for square enclosure as shown in Fig. 1a is simulated and compared with measured data from M. Cehlin et al. [19] and also compared with result from the commercial finite volumes code Fluent 6.0.

The experimental data obtained by thermal images taken by infrared thermography while numerical result obtained by the commercial finite volumes code Fluent 6.0 as shown in Fig. 2. Convection studied for Re =500, Gr = 10^6, Pr = Sc =0.7, Br = 1 ε_r =1, ε_{hs} = ε_{cs}=0.5, and ε_{in} =ε_{out} =1/8. The comparison of simulated isothermal line of present work with measured data by M. Cehlin et al. [20] and from code Fluent 6.0 is shown in Fig. 2, which indicates a fairly good agreement. Thus the mathematical model is validated and may be adopted to perform further numerical study. Also, the mathematical and numerical model is validated by El-Agouz [20].

6. Results and discussions

In the present study, the aspect ratios of the ventilated room, the heat source, the contaminant source, the inlet air and outlet air of what are kept constant as ε_r =1, ε_{hs} = ε_{cs}= 0.5, ε_{in} = ε_{out} = 1/8. The buoyancy ratio, Prandtl number and Grashof number are held fixed at Br = 1, Pr = 0.7 and Gr = 10^6. The effect of the Reynolds and Schmidt numbers on the performance of the different models for displacement and mixing ventilation is studied. During the study, the Schmidt number varied from 0.2 to 2.4 and Reynolds number varied from 50 to 1500.

Figure 3 shows the evolution of isothermal contours with Re of different cases at Sc =0.7. With the increase of the Reynolds number for different cases, the isothermal line decreases in occupied zone for all cases. The figure shows that the lower temperature is found in case 5 and the higher in Case 6 for the mixing ventilation. On the other hand, for the displacement ventilation the change between cases is small. These observations indicate that the decrease of the temperature level the accumulated energy is saved move.

Figure 4 shows the evolution of contaminant contours with Sc of different cases at Re =500. With the increase of the Schmidt number, the contaminant contours decreases in occupied zone for all cases. The figure shows that the lower contaminant concentration is found in case 5 and the higher in Case 6 for the mixing ventilation. On the other hand, for the displacement ventilation the change between cases is small. These observations indicate that the decrease of the contaminant contours level the indoor environmental conditions is improved.

The variation of average Nusselt number is plotted in Fig. 5 against Reynolds and Schmidt numbers for different cases. It shows the non-linear variation between Reynolds and average Nusselt number for given Schmidt number. At Re =1500, the average Nusselt number was about 21.92, 20.97, 21.46, 23.27, 24.69, and 5.95 for the cases 1 to 6 respectively. Mixing ventilation (case 6) is associated with higher average Nusselt number than displacement ventilation while, mixing ventilation (case 5) is associated with lower average Nusselt number than displacement ventilation. The figure shows that the best Nu_m is found in case 5 for the mixing ventilation and case 4 for the displacement ventilation.

The average Sherwood number, Sh_m, for different cases as function of Reynolds and Schmidt numbers is displayed in Fig. 6. The average Sherwood number increases with increasing the Re and Sc due to the forced convection fully dominates the airflow structure. At Re =1500, the average Sherwood number was about 21.35, 19.32, 17.74, 18.75, 19.69, and 14.53 for the cases 1 to 6 respectively. It is observed that, the mass transfer rate is highly influenced by the different cases of ventilation.

Fig. 7 gives the comparison of Mean temperature of different cases at different Reynolds and Schmidt numbers. θ_m in the occupied zone decreases with increasing Re due to the enhanced ventilation while, it is constant with increasing Sc. At Re =1500, the Mean temperature was about 0.069, 0.098, 0.085, 0.026, 0.007, and 0.163 for the cases 1 to 6 respectively. Mixing ventilation (case 6) is associated with higher mean temperature than displacement ventilation while, mixing ventilation (case 5) is associated with lower mean temperature than displacement ventilation. The figure shows that the best θ_m is found in case 5 for the mixing ventilation and case 4 for the displacement ventilation, Case 6 has the highest mean temperature for the studied ranges of Re and Sc.

Fig. 8 gives the comparison of Mean contaminant concentration of different cases at different Reynolds and Schmidt numbers. C_m in the occupied zone decreases with increasing Re and Sc due to the enhanced ventilation. Mixing ventilation (case 6) is associated with higher mean contaminant concentration about 77% than other cases and the other cases have little difference between them. The figure shows that Case 6 has the highest mean contaminant concentration for all values of Re and Sc.

The variation of ventilation effectiveness for heat removal (ε_h) is plotted in Fig. 9 as a function of Reynolds and Schmidt numbers for different cases. The figure clearly shows that ε_h decreases with increasing Reynolds and Schmidt numbers for all cases except in case 5 increases. At Re =1500, the ventilation effectiveness for heat removal was about 0.159, 0.195, 0.496, 1.217, 2.85, and 0.057 for the cases 1 to 6 respectively.The figure shows that the best ε_h is found in case 5 for the mixing ventilation and case 4 for the displacement one.

The variation of ventilation effectiveness for contaminant removal is plotted in Fig. 10 as a function of Reynolds and Schmidt numbers for different cases. The figure clearly shows that the values of the ventilation effectiveness for contaminant removal curves are little affected by Reynolds and Schmidt numbers for all cases. At Re =1500, the ventilation effectiveness for contaminant removal was about 2.87, 1.53, 4.968, 5.83, 4.51, and 1.12 for the cases 1 to 6 respectively.The figure shows that the best ε_c is found in case 4 for improved the air quality and the lower in case 6 for all values of Re and Sc.

Fig. 11 gives the comparison of air cleaning efficiency of different cases at different Reynolds and Schmidt numbers. The figure shows that as Re and Sc are increased, the air cleaning efficiency is increased due to the enhanced ventilation. In addition, the best ε_c is found in case 2 and the lower in case 4 for Re and Sc.

7 Conclusions

In this paper, numerical simulation is carried out to study the effects of different Reynolds and Schmidt numbers on performance of different displacement and mixing ventilation models. The enclosure heated at the bottom wall and two contaminant sources at right and left walls. The conclusions may be drawn as follows:

- The lower performance is found in case 6 for mixing ventilation while the higher performance in case 5 for mixing ventilation and case 4 for displacement ventilation.
- The heat and mass transfer rates, the heat and contaminant removal effectiveness and the air clearing efficiency influenced by the change of the Reynolds and Schmidt numbers.
- Mixing ventilation (case 6) is associated with higher mean contaminant concentration about 77% than other cases.
- Results for different cases show that the nearer the outlet is from the heat and contaminant sources, the more effective it is to vent heat and contaminant generated by the sources for displacement and mixing ventilation

Nomenclature

Br buoyancy ratio
c, C dimension and dimensionless concentrations
D mass diffusivity
g gravity acceleration
Gr Grashof number
h sizes of inlet or outlet
k thermal conductivity (W/m. K)
L length or length of the room
Nu_m average Nusselt number
Pr Prandtl number
Re Reynolds number
Sc Schmidt number
Sh_m average Sherwood number
t temperature
U, V dimensionless velocities in X and Y directions
u, v velocity components in x and y directions
x, y Cartesian coordinates
X, Y dimensionless Cartesian coordinates

Greek symbols

β thermal expansion coefficient
ρ density
ε_h ventilation effectiveness for the heat
ε_c ventilation effectiveness for contaminant
α thermal diffusivity
ν kinematic viscosity
ϕ general variable
ω dimensionless vorticity
θ dimensionless temperature
Ψ' dimensionless stream function
ε aspect ratio
ξ air cleaning efficiency

Subscripts

c concentration
cs contaminant source
h high
hs heat source
i local
in inlet
m mean
out outlet
r room
t thermal

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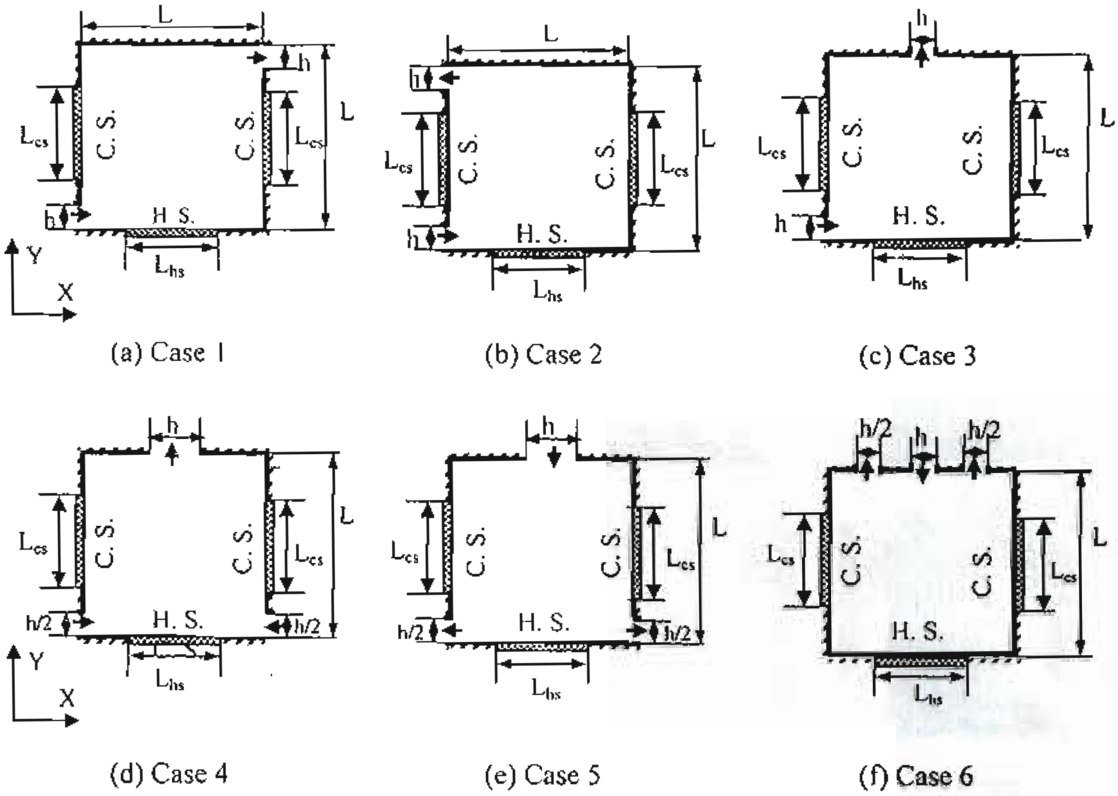
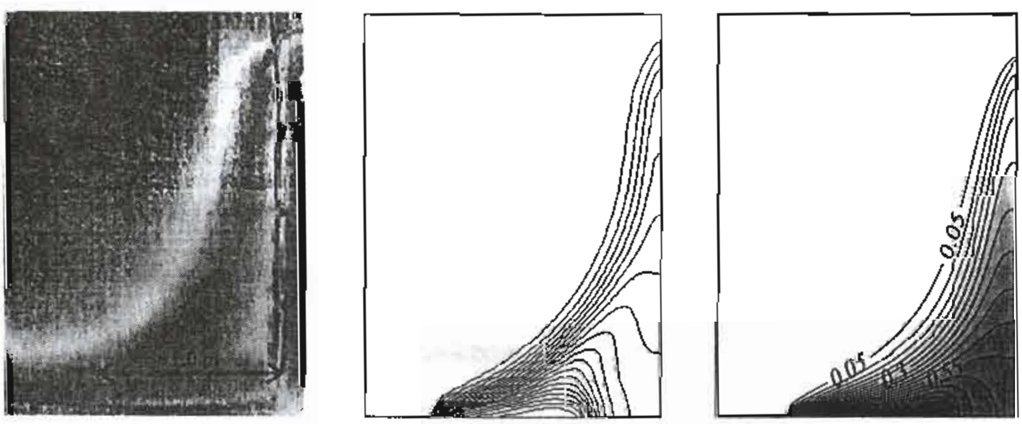


Fig. 1 Schematic diagram of physical configuration of considered displacement and mixing models



M. Cehlin et al. [19] Fluent 6.2 Present work

Fig. 2. Results comparison between the present work, M. Cehlin et al. [19] and Fluent 6.2 for isothermal lines

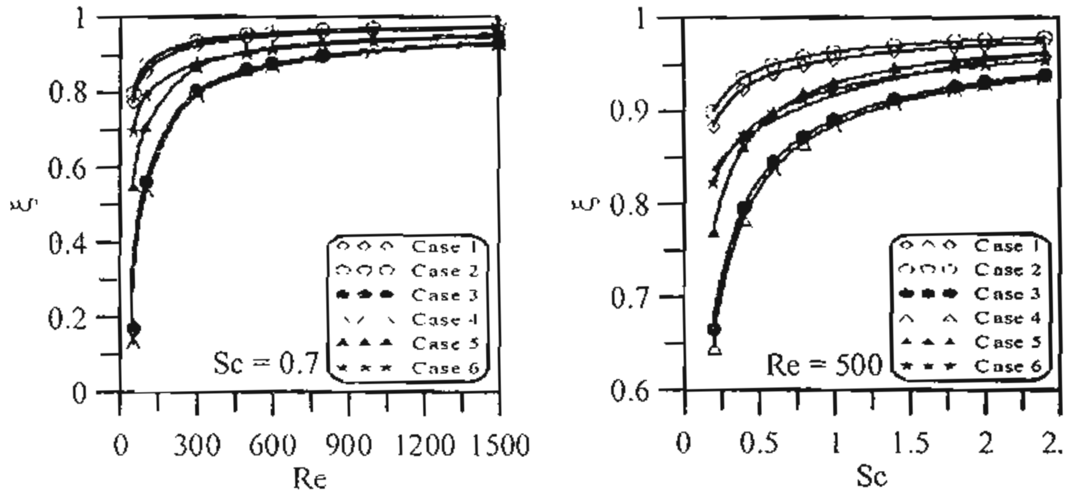


Fig. 11 Air cleaning efficiency versus Reynolds and Schmidt numbers for different cases

Table 1 The Φ, β, Γ, S_u and S_p used in Eq. (1).

Φ	β	Γ	S_u	S_p
U	0	0	$\partial\psi / \partial Y$	1
V	0	0	$\partial\psi / \partial X$	-1
ω	1	1/Re	$Gr / Re^2 [\partial\theta / \partial X + Br\ \partial C / \partial X]$	0
ψ	0	1	ω_P	0
θ	1	1/Re Pr	0	0
C	1	1/Re Sc	0	0