MACROSCOPIC AND MICROSCOPIC ANALYSIS OF ZONAL MODELS

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ABSTRACT

In this paper, microscopic and macroscopic flow analyses have been employed to identify and analyze the causes of discrepancies of power law zonal models. The analyses show that the use of constant flow coefficient (K_f) is one of the reasons for the discrepancy. Computational Fluid Dynamics (CFD) was employed to estimate K_f for isothermal condition. The variations of K_f as function of room height were investigated for two types and locations of diffusers. Five levels of Reynolds numbers (*Re*) were used. The flow coefficients for horizontal (K_{fx}) and vertical (K_{fy}) flow were estimated separately. The result has shown that K_{fx} and K_{fy} vary differently and can be affected by variability of the flow field, which can vary depending on the choice of inlet device and relative location of inlet. Implementation of the zonal model which employs a variable flow coefficient has shown a significant improvement compared to the existing zonal model.

INTRODUCTION

Zonal models are intermediate airflow models between the extremes of single/multi-zone and CFD. For single/multi-zone model a zone may represent a building or a section of a building such as a room. However, in the zonal approach, a room is divided into small number of macroscopic control volumes, which are usually larger than the one used for CFD application. The advantage gained is that the resulting systems of algebraic equations are smaller and by far easier to solve than the difference approximations (finite difference, finite volume) to the partial differential equations used in the CFD approach. Hence, zonal models can provide information on airflow and temperature distribution in a room faster than CFD and more accurate and detailed than single/multi-zone models.

Heat and mass transport between the zones are related using the macroscopic mass and energy conservation principles:

$$\sum_{j=1}^{n} m_j + S_M = 0 \qquad (1)$$

$$\sum_{j=1}^{n} q_j + S_q = 0 \qquad (2)$$

The detail of the relation used for the evaluation of airflow between zones is most important for the reliability and generality of the zonal models. This is due to the fact that the calculation of the zone to zone airflow leads to the description of the flow field. To that end, a number of approaches have been used since the inception of zonal models. These approaches can be classified as non-pressurized and pressurized zonal models (Teshome and Haghighat, 2004). The non-pressurized zonal models are the first generation zonal models which employ empirical and/or analytical relations to express the mass flow in jet, plumes and other zones in the room (Howarth, 1980; Inard, 1988). Whereas The pressurized zonal models, first proposed by Bouia (1993), employ a macroscopic equivalent of the momentum equation to relate the mass flow between neighboring zones with pressure difference between the zones. These models are commonly referred to as the power law zonal models:

$$m_j = K_f A_j \rho \sqrt{\frac{2\Delta P}{\rho}} \qquad (3)$$

These methods have more general applications than the non-pressurized zonal models but they have limitations in predicting specific flows (jets, plumes, etc) as they cannot correctly describe the momentum for such zones (Inard et al. 1996). In this regard, the power law zonal models have only been employed for zones outside of jets and plume region; and specific flow equations can be integrated to calculate the mass flow across the boundary of zones within jet and plume region (Wurtz, 1995, Haghighat et al. 2001). The zonal models in use to day are therefore the hybrid of non-pressurized and pressurized airflow models. Such hybrid models can provide more accurate information than either non-pressurized or pressurized zonal models. Furthermore, such models have been integrated with moisture transfer models; thermal comfort models; radiation models; contaminant source and sink models; air infiltration models such as COMIS; and personal exposure models. Teshome and Haghighat (2004) have presented detail review of zonal models.

Zonal models have been found to predict the main stream reasonably well for natural ventilation (Wurtz et al. 1999). For the case of forced convection, however it has been found that the hybrid zonal models have pronounced discrepancies in predicting the formation of recirculation (Haghighat et al. 2001; Mora et al. 2003). These authors have shown such discrepancies by comparison with CFD and validation with experimental data. The empirical jet models are well-developed and their integration shows a significant improvement of the prediction capability of the hybrid zonal model in the jet zones (Mora et al. 2003). It can therefore be stated that any discrepancy pertaining to the prediction of the hybrid zonal model should come from the power law equation (Equation 3). To address this problem, Axley (2001) proposed an alternative approach, surface-drag flow model, which was developed by considering the transfer of shear stress near wall surfaces using one-dimensional momentum balance. The model can be written in its simplified and rearranged form as:

$$m_j = \left[\frac{1}{2a^{3/2}\kappa} \sqrt{\frac{A_j(2n-1)}{2\Delta l_j}} \right] A_j \rho \sqrt{\frac{2\Delta P}{\rho}} \quad (4)$$

where, a = 1/7 (for turbulent boundary layer velocity profile) and n is the zone number starting from the wall (1,2,3,…). Axley (2001) has not directly suggested the use of variable flow coefficient K_f. But comparing the power law relation (Equation 3) and the simplified form of the surfaces-drag flow relation (Equation 4) shows that the latter is the form of power law relation with variable K_f, the first term in the right side of Equation 4. The flow coefficient, K_f in the power law model is an empirical coefficient which has been given a value of 0.83 (Wurtz et al. 1999; Haghighat et al. 2001). This can be due to the fact that the zonal model for indoor airflow and the orifice equation for flow through cracks and small openings employ the same equation (Equation 3). However, it cannot be physically plausible to use Equation 3, which was originally developed for flow through orifice, without modification for two entirely different applications. In that respect, we believe that the attempt made by Axley (2001) to improve the zonal model is a step in the right direction. Nevertheless, comparison of the flow field predicted by the two models, Equations 3 and 4 shows that the latter has made no significant improvement compared to the power law model (Mora et al. 2003). This can be due to the fact that the surface-drag flow model was developed by considering the transfer of shear stress for well-developed turbulent flows near walls and the modifications made have not completely avoided the inherent problem in the power law zonal models, predicting the formation of recirculation loop in the room.

The main objective of this paper is therefore to identify and analyze the most important causes of the discrepancy of the zonal models and propose improvement methodologies relevant to their application in the prediction of isothermal forced convection.

ANALYSIS OF THE LIMITATIONS OF THE ZONAL MODEL

Single/multi-zone and zonal models are macroscopic approaches for mechanistic modeling of indoor airflow and contaminant distribution. They are one-dimensional models, which assume a hydrostatic variation of air pressure within a zone, and use the power law equation with constant flow coefficient to evaluate zone to zone airflow. Except for some details, the way the conservation principles are applied and their solution method makes these two macroscopic models fundamentally similar. This shows that the power law relation has been used to describe flow in two entirely different scenarios. Indeed original developers of the zonal model were aware of this problem from the outset but few investigations were made to adopt an improved version of the power law model for indoor airflow application. Therefore, the roots of the discrepancies of the zonal model can be traced back to the assumptions used in deriving and applying the power law equation. It is from this background that we attempt to provide macroscopic (for one-dimensional nature and hydrostatic flow field) and microscopic (for flow coefficient) analyses of zonal models to explain the causes of the discrepancies and propose possible improvement methodology.

One-dimensional airflow model

The flow crossing a boundary of the zone is simply considered as the result of the pressure difference between the two neighboring zones in the flow direction. The information (convection and diffusion) from the upstream and downstream, lower and upper, and left and right is not taken into account. This has resulted in the uncoupling of the vertical and horizontal airflow equations. Hence, zonal models can be considered as one-dimensional airflow models applied for each zone in the horizontal and vertical directions and are still expected to give a two or three-dimensional information of the flow like the Navier-Stokes equations. Indeed, this problem can be alleviate by considering a number of possible combination of the horizontal and vertical velocities with pressure drop in the respective direction to get a form of reduced Navier-stokes equation. One typical example is the Euler equation employed by Griffith and Chen (2003).

Hydrostatic flow field

The hydrostatic flow field assumption is the physical basis for the macroscopic airflow models, i.e. zonal and single/multi-zone. It corresponds to no-flow condition which can be the product of the assumptions behind the orifice equation (Sandberg, 2004) and/or the definition of the discharge coefficient (Etheridge, 2004). In the case of flow through cracks and orifices, the values of the pressure difference, ΔP is obtained from pressures measured at positions away from the opening so that the influence of the flow is negligible. The orifice equation is therefore a reasonable model when the kinetic energy of the air flowing through small openings and cracks is completely dissipated in the static room air (Kato, 2004).

Flow coefficient

The flow coefficient K_f is an empirical coefficient, which has been considered to lump the effects of viscous losses and local contractions of the streamlines for flow through cracks and openings. In such instances, the coefficient can be estimated experimentally using the ratio of the actual mass flow rate to the theoretical one. Nevertheless, when there is no solid boundary, like in rooms, it could be difficult to get accurate experimental values of K_f (Inard et al. 1996). This can be one of the possible reasons for the use of constant K_f. As stated earlier, Axley (2001) used a variable K_f (Equation 4) though it has made no significant improvement in the prediction compared to the existing zonal models (Equation 3). This is not due to the physical implausibility of the use of variable K_f. It is rather due to the methodology used to get it. A simplified algebraic expression for K_f like Equation 4 was obtained in order to maintain the simplicity and improve the accuracy of zonal model. The problem with the expression is that it is not consistent with the flow pattern expected in indoor airflow since it was derived by assuming a well-developed turbulent flow near the wall. Moreover, the modifications made, when the equations were applied in the region away from the wall , are not good enough. In that respect, any adjustment made on K_f should be made to reflect the anticipated airflow distribution.

It is due to such understanding that we set out to demonstrate the physical basis for the need to use a variable and consistent K_f using the microscopic momentum (Navier-Stokes) equations. Moreover, the momentum equations are helpful to show the parameters represented by K_f. The finite volume discretization of the momentum equation for staggered grid structure described in (Patankar, 1980; Versteeg and Malalasekera, 1995) will be employed. For two-dimensional flow in the x and y directions, the equations can be written as:

$$a_e u_e = \sum a_{nu} u_{nb} + (P_P - P_E) A_e \qquad (5)$$

$$a_n v_n = \sum a_{nv} v_{nb} + (P_P - P_N) A_n \qquad (6)$$

Simplifying and rearranging Equation 3 gives K_f for any direction j:

$$K_{fj} = \frac{u_j}{\sqrt{\dfrac{2\Delta P_j}{\rho_j}}} \qquad (7)$$

Comparing Equations 5 and 6 with Equation 7, it can be seen that K_f includes local convection and diffusion in the coefficients a_e, a_n, $a_{nu,}$ a_{nv}, which vary through out the flow field. This consolidates the fact that the use of constant K_f throughout the flow field can be one of the possible reasons for the discrepancy of the power law zonal model. The use of variable K_f can also avoid the other causes of discrepancy of zonal model - one-dimensional nature and the hydrostatic assumption- as variable K_f takes into account the effect of convection and diffusion both in the direction of the main flow and perpendicular to it. Furthermore, expressions for K_f that are physically consistent with the pattern expected in indoor airflow can be obtained using Equations 5 through 7. This is very important as experimental values for only velocity and temperature distribution are commonly reported in literature due to the difficulty of measuring pressure difference between two positions in a room. It is because of this reason that we propose to use pressure and velocity data from CFD simulation for the estimation of K_f.

METHODOLOGY

CFD has been used to predict the airflow, temperature and contaminant distribution in a room (Haghighat et al.1992; Chen and Xu 1998; Topp et al. 2001). CFD simulation of airflow for isothermal rooms have been validated for number of cases and the patterns of the flow field are well-known. The similarity of patterns can be exploited to obtain general relations for flow coefficient, which can be employed for similar configuration.

The foregoing discussion revealed that making K_f variable may alleviate some of the discrepancies of the zonal model. This can be made possible by using data from CFD simulation so that K_f will have a physically consistent variation in the flow field. Hence, in this section two-dimensional CFD simulation of isothermal room will be used to generate pressure and velocity data for the estimation of K_f. The dimensions of the room employed for the CFD simulations are L/H = 2.16, and heights of inlet and exhaust are 0.004H and 0.0027H, respectively. Nielsen (1998) has used this geometry for two-dimensional slot for Re=2210 (based on the inlet height). Here, the same geometry will be used to investigate the effect of the diffuser type and its location with respect to *Re* on K_f. To that end, Two cases were considered as shown in Figure 1. Slot inlet on the left wall (Case I), and grille and ceiling diffusers on the ceiling (Case II). For each type of inlet configuration a simulation was performed for five levels of *Re*, 1000, 2210, 3500, 5000, 10000. For the CFD simulation, the inlet flow and turbulence conditions for k-ε turbulence model were estimated using equations given by Nielsen (1990). Furthermore, to model the inlet diffusers, the main region specification method (Huo et al. 2000), which adopted the well-developed jet formulas (ASHRAE 2001), was selected for the CFD simulations conducted using AirPak CFD software (AirPak. 2002). For the estimation of K_f using Equation 7, the flow field data from CFD simulation was not directly used. The room was first divided into smaller number of macroscopic zones commonly used in zonal models application. The pressure and velocity data which correspond to the centers and faces of these zones, respectively were then selected for the estimation.

RESULTS AND DISCUSSIONS

Estimation of K_f

The variation of K_{fx} and K_{fy} as a function of dimensionless height of the room (y/H) for case I and II are shown in Figures 2 through 4. Since the height of the room is commonly used as a reference length to express the other dimensions, we used the dimensionless height of the room to show the variation of K_{fx} and K_{fy} .The values of K_{fx} and K_{fy} shown are therefore the average values for all zones at that height. The results indicate the patterns of the variability of K_{fx} and K_{fy} depending on the type and location of the diffuser. For the range of *Re* used in this study, the variations of K_{fx} and K_{fy} show similar trend for the respective type and location of diffuser. The variation of K_{fx} and K_{fy} for wall and ceiling diffusers are shown in Figures 2 and 3. As can be seen in Figure 2, K_{fx} for the grille diffuser first decreases up to the mid-height of the room and then starts to increase from the mid height up. This variation of the flow coefficient can be due to the formation of recirculation, which makes the average horizontal velocity to increase away from the center of the recirculation. This results in higher pressure drop around the center and smaller away from it. Therefore, in accordance with Equation 7, the average flow coefficient in the power law model has to decrease towards the center and increase away from it. Unlike K_{fx}, K_{fy} increases towards mid-height of the room. This is due to the decrease in the average vertical velocity as we go away from the center of the recirculation. Hence, based on the zonal model, Equation 7, the value of K_{fy} needs to be higher in the central region of the recirculation. For the circular diffuser (Figure 3), the minimum and the maximum values for K_{fx} and K_{fy}, respectively, are shifted upward compared to grille diffuser (Figure 2). The direction of the inlet velocity for the former is horizontal, which results in a different flow field than the latter and shifts in the centers of the recirculation regions. However, the variation of K_{fx} and K_{fy} for the slot inlet (Figure 4) seems somewhat similar to that of grille inlet. Both have their minimum and maximum values for K_{fx} and K_{fy} at or very close to the mid-height of the room.

Comparison and validation

The use of a variable K_f in Equation 3 requires relations so that the results from one configuration can be reproduced to another similar configurations. It may be difficult to come up with a single equation to describe the variation of K_f in the flow field. However, the average variation of K_f for the horizontal and vertical flow depicted in Figures 2 through 4 can be useful to deduce basic relations since the commonly used inlet conditions fall within the range of *Re* values considered. The variation of K_{fx} and K_{fy} for slot and grille diffuser can then be approximated by

$$K_{fj} = B\left(\frac{y}{H}\right)^2 + C\left(\frac{y}{H}\right) + D \qquad (8)$$

where, the parameters *B*, *C*, and *D* vary depending on the choice of inlet. When Equation 8 is inserted in Equation 3 the improved zonal model can be obtained. This can still be solved by applying Newton-Raphson global convergence technique, which solves the non-linear system of equations resulting from the application of mass balance for each zone. Integration of jet model is also required for better representation of the flow in the jet region.

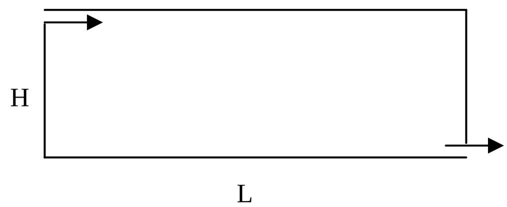
For comparison with CFD and validation, simulation has been conducted for a full-scale isothermal room with dimensions: H = 3m and L/H =3.0, height of inlet and outlet 0.056H and 0.16H, respectively. Re = 4773, which falls in the range of Re used for the estimation of K_f. Nielsen (1990) and Topp (1999) have used the same geometry to simulate indoor airflow and contaminant transport. The flow fields predicted by CFD and zonal models are depicted in Figures 5 through 7. The flow field predicted by CFD in Figure 5 shows a large recirculation centered in the second half of the room. The result agrees well with previous CFD simulations for the same geometry (Chen and Xu, 1998; Topp, 1999). Comparing the prediction by the improved zonal model (Figure 7) with that of CFD (Figure 5) and the existing zonal model (Figure 6) shows a substantial improvement in the prediction of the recirculation region. As shown in Figure 7(a), the magnitudes of the velocity below the center of the recirculation increases away from it like that of the CFD. Furthermore, validation of the CFD and zonal model prediction (Figure 8) shows that the improved zonal model has followed the trend very well and reasonably predicted the magnitude of the horizontal velocity, *u*. However, the existing zonal model could not even show the tendency to follow the trend. The magnitudes of the velocities are very close to zero except at the jet region, which is due to the integration of the jet model. Refining the zones in the recirculation region as shown Figure 7(b), has shown the direction of the velocities below the jet region and above the center of recirculation. The velocities are reversed unlike the prediction of CFD. Such discrepancy, like the existing zonal model, in the predicted flow field can be due to the difficulty of completely ensuring the momentum conservation using a variable flow coefficient in this region. Capturing the gradient of the shear in this region requires the flow to be two-dimensional since the shear stress varies in the normal direction to the main flow.

For non-isothermal forced ventilation (mixed convection), the airflow pattern is sensitive to the Archimedes number (*Ar*). Chen and Xu (1998) have performed CFD simulation for mixed convection. The penetration length of the jet for slot inlet is inversely related to *Ar*. For higher *Ar*, only one recirculation is expected to be formed. The penetration length is very small and the jet drops down near the inlet. The flow pattern is similar to Figure 5 but the velocity vectors in the reverse direction. For lower *Ar*, two recirculation regions are expected to be formed. The length of the first region is determined by the jet penetration. Similar methodology pursued in the previous section, for the estimation of K_f, can still be applied. But for the non-isothermal case *Ar* is used to generalize the result instead of *Re*. Moreover, the minimum *Ar* required for the formation of two recircualtion regions need to be estimated so as to obtain parametric relations of K_f (like Equation 8) for each recirculation region.

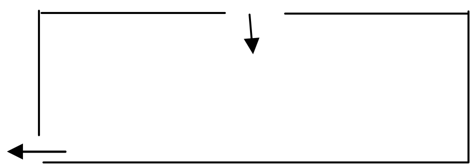
Finally, incorporating the variable K_f in the zonal models may not solve all the discrepancies pertaining to the existing zonal models. It can not also make the zonal models to be a complete substitute of CFD as far as accuracy of the models is taken into account. However, the improved zonal models have not only provided low computational cost and reasonable accuracy but also they have avoided, to certain extent, the physical deficiencies of the existing zonal models. Due to these facts, the improved zonal models can be worthwhile alternatives to CFD. Moreover, the growing trend for coupling airflow models with energy simulation demands a reduced computational cost which is still unattainable when CFD is used for the coupling. In this regard, the improvement of zonal models is one of the steps for their possible coupling with energy simulation models.

CONCLUSION

The one-dimensional nature, hydrostatic flow field assumption and the use of constant flow coefficient were found to be the most important reasons for the discrepancy of zonal model. The microscopic momentum equation was used to demonstrate the importance of using variable flow coefficient as well as to develop a methodology to estimate it. The estimations have shown that K_{fx} and K_{fy} vary differently and they can be affected by variability of the flow field, which can vary significantly depending on the choice of inlet device, and relative location of inlet. The implementation of zonal model with variable flow coefficient shows that a significant improvement in predicting the flow pattern as well as the magnitude of the velocities compared to the existing zonal model. The use of variable flow coefficient can improve the prediction capability of zonal models without affecting their relative simplicity compared to CFD.



(a)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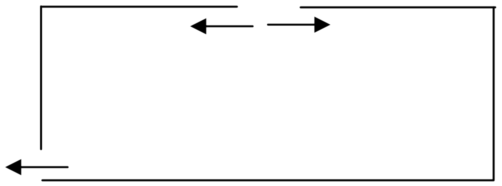
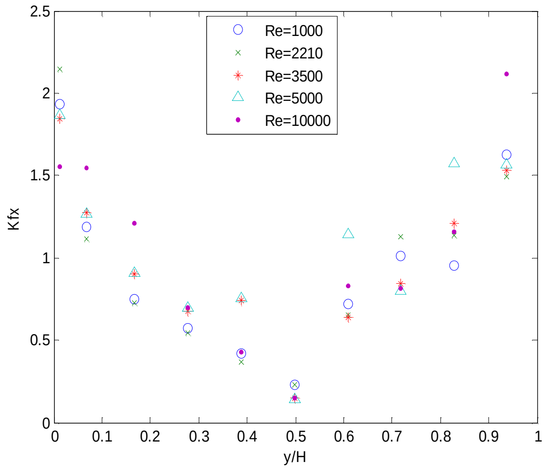
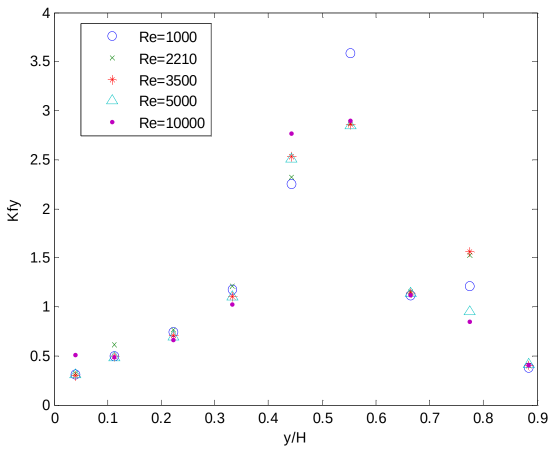


(c)

Figure 1 The cases considered for the estimation of K_{fy} Case I (a) and Case II (b and c)

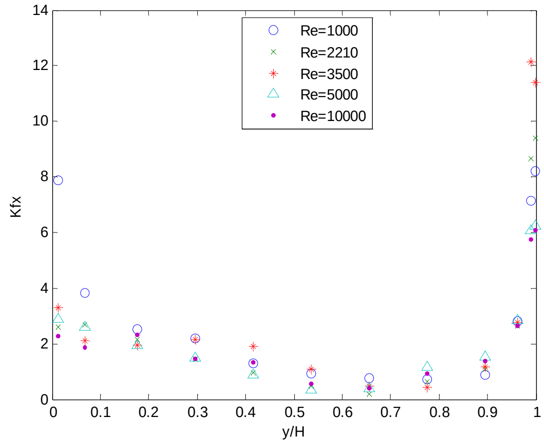


(a)

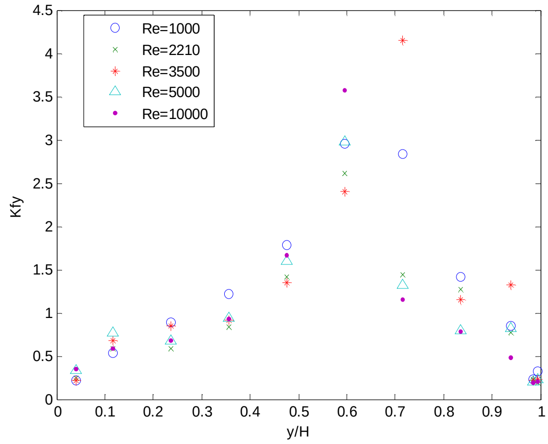


(b)

Figure 2 Average K_f for grille diffuser a) K_{fx} b)K_{fy}

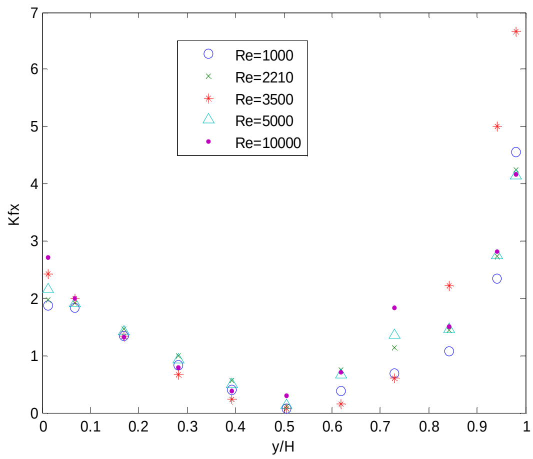


(a)

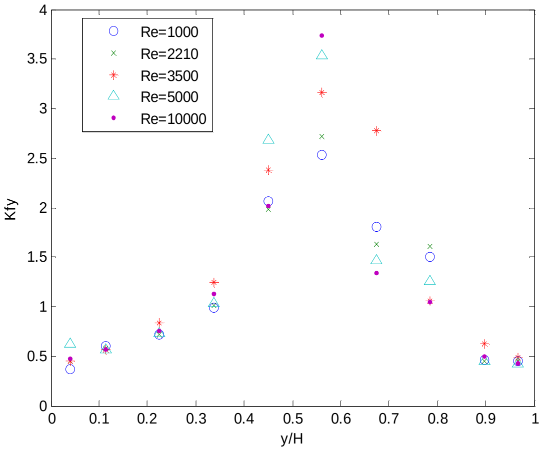


(b)

Figure 3 Average K_f for circular diffuser (a) K_{fx} (b) K_{fy}



(a)



(b)

Figure 4 Average K_f for slot diffuser (a) K_{fx} (b) K_{fy}

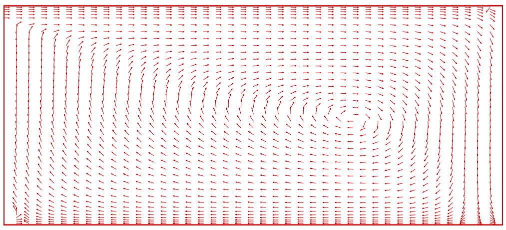


Figure 5 Flow field predicted by CFD (40x40 grids)

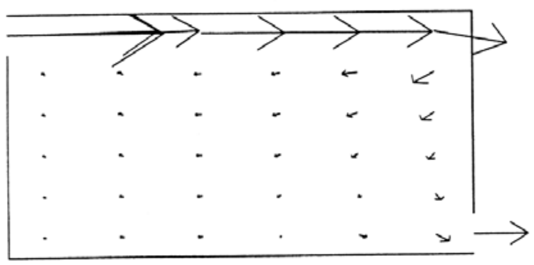
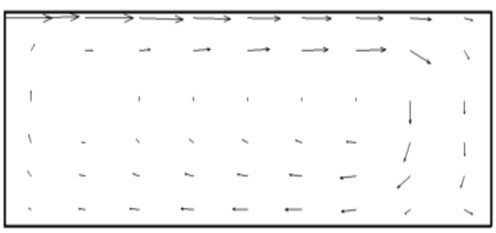
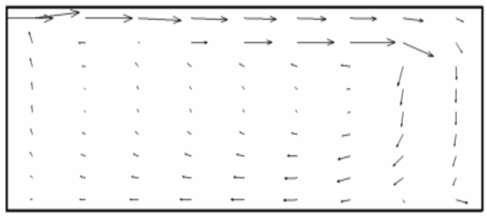


Figure 6 Flow field predicted by existing power law zonal model (6x6 grids).



(a)



(b)

Figure 7 Flow field predicted by improved zonal model (a) 9x6 grids and (b) 9x9 grids

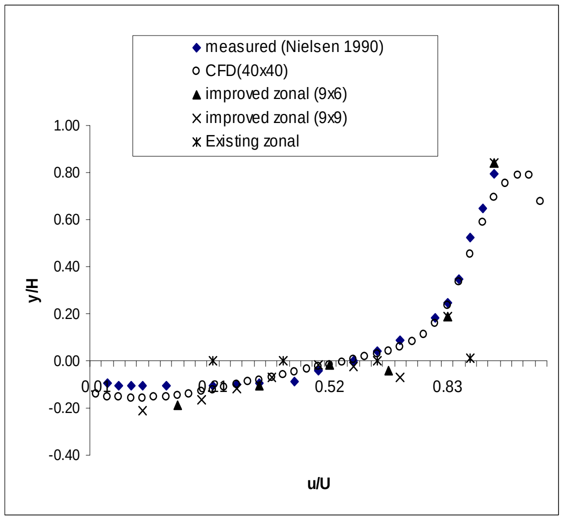


Figure 8 Validation for slot diffuser

NOMENCLATURE

a	1/7 (power law profile), coefficient
A	the area, m^2
Ar	Archimedes number
B,C,D	coefficients
g	gravitational acceleration, $m.s^{-2}$
h	height, m
K_f	flow coefficient, -
K_{fj}	flow coefficient in the j direction, -
K_{fx}	low coefficient in the x direction, -
K_{fy}	flow coefficient in the y direction, -
l	width of the face of zone,m
L	length of room, m
m	mass flow rate , $kg.s^{-1}$
P	pressure in the zone, Pa
q	the rate of energy flow, watts
S	source term, $kg.m^{-2}s^{-2}$
t	time, s
u	velocity in x direction, $m.s^{-1}$
v	velocity in y direction, $m.s^{-1}$

x horizontal distance, m

y verical distance, m

Subscripts and superscripts

e east cell face
E east cell center
$_j$ zone/face
$_M$ mass
N north cell center
n north cell face, number of zones
nb neighboring cell
nu north u-neighboring cells
nv north v-neighboring cells
P cell center
$_q$ energy

Symbols

ρ density of zone air, kg.m^{-3}
κ von Karman constant = 0.4
Δ difference
Σ sum

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