CFD MODELLING FOR SWIRL DIFFUSER AND ITS IMPLICATIONS ON AIR CHANGE EFFECTIVENESS ASSESSMENT TO GREEN STAR'S IEQ-2

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ABSTRACT

Swirl diffusers can create better air mixing to enhance indoor air quality and help achieve compliance with Green Star IEQ-2 through Air Change Effectiveness (ACE) measure but the lack of modelling guidelines gives rise to the use of various modelling approaches with different results. The ACE calculation depends strongly on the flow characteristics produced by the diffuser outlet that vary considerably between different modelling set ups. Proper calibration and correct definition of performance related parameters are important to effect the radially diffusing flow pattern along the ceiling (Coanda effect). This study demonstrates the common approaches, identifies the critical design parameters, analyses and discusses the differing outcomes in terms of flow pattern, air distribution and ACE. Comparisons of the simulated results from the proposed modelling method with experimental data are also carried out.

INTRODUCTION

Air Change Effectiveness and Indoor Air Quality

Commercial office spaces are progressively more reliant to new designs of HVAC equipment to achieve better indoor conditions measured against current industry standards such as Green star and NABERS. One of the most telling indoor air quality parameters when it comes to air distribution system and fresh air delivery is Air Change Effectiveness (ACE). ACE is commonly used as a measure for effective delivery of outside air by a ventilation system to the occupied space in a building.

According to ANSI/ASHRAE 129-1997, the ACE of a building is measured by comparing the age of air in the occupied space at the occupant breathing level (1 meter from floor) to the age of air of the space if the indoor air were perfectly mixed. The age of air in this context is the average time for air to travel from an inlet to a particular point in an indoor space. ACE is strongly governed by the air flow patterns within a room. Factors affecting it include ventilation system design and operation, outside air and supply air flow rates, mixing coefficient, air distribution system and configuration that determine the type of flow, locations of supply inlets and return outlets, air conditioning operation mode etc.

An effective outside air delivery with complete mixing gives an ACE value of 1. On a broad category, entrainment flow air distribution systems that create a flow pattern by entrainment of room air into a jet typically has ACE less than 1. Displacement flow systems that sweep the air through the room from one end to the other could have ACE as high as 1.2. Table 6-2 of ASHRAE 62.1-2007 provides reference information on ACE for different air distribution configurations to achieve an acceptable indoor air quality. Typical office air distribution systems with ceiling supply of cool air should be designed to achieve ACE of 1.0. Floor supply of cool air and ceiling return typical of displacement ventilation with low velocity providing unidirectional flow and thermal stratification could achieve 1.2 whereas systems with make up supply drawn in on the opposite side of the room from the exhaust/return may get only 0.8 and make up supply drawn in near to the exhaust and/return location could get as low as 0.5.

Green Star requirements as outlined in IEQ-2 Office V3 awards two points for compliance when the ACE value at breathing height measured at 1m from the floor is 0.95 or higher for at least 95% of the net lettable area (NLA) (Green Star Technical Manual, 2008). Whilst the Green Star guidelines outline the passing level of ACE for office space, it does not provide specific information with regards to the different air distribution systems, components and configurations, proportion of recirculated air and the different operating modes of the air conditioning system and at times, the target 0.95 for 95% NLA coverage may be unrealistically high for certain types of air distribution systems and configurations.

In general, CFD technique has been used to model buildings and spaces to calculate the mean age of air for ACE determination particularly in regards to assessing compliance to IEQ-2 of Green Star. While this practice is acceptable, there exists no guidelines for CFD modellers to follow. Although numerical solutions employed in CFD may well be capable of doing the task, proper modelling techniques, correct set up of modelling parameters and accurate representation of the computer virtual objects are important to get a meaningful result. More specifically, a number of other factors are sometimes overlooked leading to under achievement or in some cases overestimation of the ACE values. In the most stringent cases, VAV systems that utilise low turn down ratios associated with low flow rates achieve very poor ACE values exacerbated by short circuiting of supply air to the return outlets, particularly when re-circulated air age is factored in the supply air. The re-circulated air from the return air stream channelled back into the supply air stream has been found to corrode the freshness (age of air) of the supply air and affects the ACE calculation significantly. This subject matter is quite in-depth and is a topic of discussion in another study.

Some of the key areas demanding more attention in CFD modelling of air distribution systems for ACE calculation are as follows:

- Different air distribution systems or configurations have different target ACE. Displacement systems have a tendency to get better ACE than entrainment systems. Realistic expectations have to prevail early in the design stage to assess if IEQ-2 points could be achieved or not.

- Diffuser CFD modelling is quite complex. Correct swirl diffuser modelling is critical to establishing the right flow pattern in the room. Accurate diffuser representation in CFD models and diffuser performance characteristics such as vane angle, swirl, effective area, specified flow rate with regard to throw and exit velocity must be defined correctly.

- Short circuiting of supply air in entrainment flow systems. This is typical in an office ventilation set up where supply inlets and return outlets are both located on the ceiling. There are some strategies that can be used to minimise the short circuiting of fresh air thereby minimising penalty on ACE.

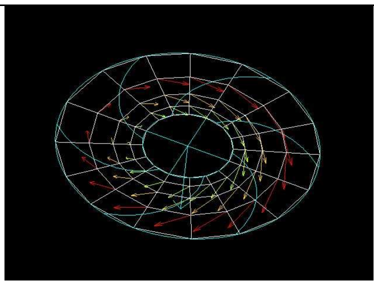
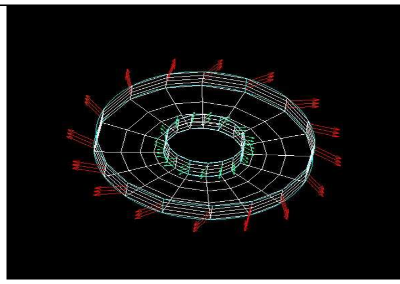
CFD modelling practices for swirl diffusers

Swirl diffusers are designed to provide effective indoor air diffusion through specially designed swirl deflection blades to produce a highly turbulent radial air flow pattern that will induce better mixing of room air. This also results in fast temperature equalisation to give stable room conditions with minimum temperature gradients. The excellent qualities of air distribution from high performing diffusers enable designers to aim for a high value of ACE. Swirl diffusers have recently become very popular because they generate radially high induction swirl air flow by drawing room air up into the supply air pattern to induce superior air mixing. Better mixing means better ACE.

Generally, diffusers come with a set of performance data derived from experimental results. For swirl diffusers, the most important parameters other than flow rate are the throw distances at specific terminal velocities (V_t normally measured at 0.75, 0.5 and 0.25 m/s) and the effective area. In order to reflect these performance data accurately, it is imperative to understand how to model the swirl diffuser correctly. This involves proper representation of its physical characteristics, correct definition of its performance data as well as other derivative parameters to ensure attainment of its performance characteristics as described in its specifications.

CFD modelling, as with other types of modelling, involves streamlining and simplifications to predict outcomes within reasonable accuracy for the given time, cost and effort constraints. The incorrect or inadequate representation or calibration of diffuser in a CFD model can lead to meaningless results. On the other hand, a full blown detailed representation of the diffuser is too costly and resource intensive. A swirl diffuser could sometimes be modelled as an axial fan with swirl for a specific purpose with acceptable results. However, in the assessment of ACE the characteristic flow pattern with radial air diffusion swirl flow is not to be overlooked or else errors in ACE calculation will occur. Diffuser performance is critical in the overall ventilation design effectiveness. Allison and North (2011) stress that importance but attribute diffuser’s uncertain performance particularly at low air flow rates of minimum turn down ratios to the lack of performance data at these flow rates which eventually may lead to incorrect diffuser calibration. They suggested using a global approach with generic diffuser performances as a compromise for all the different modelling methods developed out there bypassing the minor details in diffuser geometry to set a level playing field for ACE comparison. Proper diffuser performance data give a set of throw distances versus velocities at various flow rates down to 25% turn down ratio (Holyoake, 2006) and these should be entered correctly in a suitable format such that when a particular flow rate is called for, a correct operating point can be identified by interpolation of the entered data. This view, however, is not shared by Allison and North (2011) who argue that the swirl diffuser performance not be categorised in terms of throws and terminal velocities. The ACE was said to depend strongly on the type of flow the diffuser produced and yet the horizontal across the ceiling, vertical downward and somewhere in between types of flow resulting from their steady state diffuser modelling were calibrated using visual inspection of transient diffuser experiment in smoke tests and used to determine the ACE. In the CFD context, a swirl diffuser should be modelled as 3D circular fan usually with an internal hub (Einberg et al., 2004) as shown in Table 1. In Airpak/Fluent CFD code (ANSYS Airpak/ANSYS Fluent, 2010), the velocity decay constant as the main parameter in the momentum method calculation has to be adjusted until the right effective area is obtained. Effective area is one of the most important

Table 1 Different modelling parameters from the two different approaches for swirl diffuser.

Modelling Difference	Generic axial fan	Vortex (swirl) diffuser
Geometry	2D	3D
Velocity vector	Normal with swirl (tangential) component	Radial with swirl (tangential) component
Throw-terminal velocity data	not required	important performance parameters
Effective area	fan surface area	fixed value for adjustment by flow rate, throw and terminal velocity through velocity decay constant
Velocity decay constant adjustment	not required	required
Geometry, mesh, velocity boundary in CFD model	 Figure 1a. Generic fan CFD model	 Figure 1b. Swirl diffuser CFD model

parameters in diffuser performance and design and it has to be provided as part of diffuser performance data.

METHODOLOGY

ACE definitions and calculation formulas (ASHRAE Fundamentals, 1997, Federspiel, 1999)

Air Change Effectiveness (ACE): $\varepsilon_I = \dfrac{\tau}{\theta_{age}}$ (1)

Nominal ACE: $\varepsilon_{I,N} = \dfrac{\tau_N}{\theta_{age,N}}$ (2)

Time constant; τ = 1/ACH (3)

where:
θ_{age} = age of air in seconds, determined experimentally or numerically
ACH = Air Change rate per Hour, dimensionless
Subscript N denotes 'Nominal' which represents the entire building, zone or space

The ***Air Change Effectiveness at Breathing Height*** is therefore the zone Nominal ACE measured at breathing height where average local age of air at breathing height (taken as 1m from floor level) is used to divide the room time constant as per Equation (2). It is the ACE referred to in ASHRAE 62.1-2007 and Green Star IEQ-2.

The different modelling approaches and the resulting flow characteristics

In a scenario where the diffuser is represented with a fan, the parameters to be defined are fluid temperature, flow direction that can be specified as vectors or angles from the normal, flow rate, swirl which has two options i.e. magnitude or RPM and the turbulent parameters if the two-equation or RNG model has been selected to model turbulence. Specific to swirl that controls the flow direction in relation to the direction of blade revolution, it can be defined as a swirl magnitude that takes the ratios of fan's radial coordinate to the outer radius of the fan and the tangential to axial velocities or more easily as a factor to the fan's RPM with a certain assumption on how much of the tangential velocity is transferred to the fluid.

The vortex diffuser representation for swirl diffusers takes a more involved procedure to include performance data in the form of a set of terminal velocities at specified throw distances. Room supply conditions that contain flow rate, temperature, species, turbulence and swirl angle are to be specified as well. The swirl angle determines how much off the normal direction to the supply inlet surface the flow direction takes. The last step is to specify the modelling method of the diffuser from two options i.e. momentum or box method with the former more suited for vortex diffusers.

For proper representation of diffuser performance, the simulation requires that the airflow from the diffuser enters the room with momentum corresponding to the initial jet velocity instead of the typical velocity calculated from the volumetric flow rate and the geometric cross-sectional area that the diffuser occupies. A momentum source that accounts for the initial jet velocity is added to the diffuser to reproduce diffuser performance in the simulation in a much similar way with the approach used for linear jets. The implementation for ceiling diffusers (circular, square, and vortex) however, maintains the radial or lateral jet characteristic of the diffuser by modelling the circumferential distance of the diffuser and extruding it in the direction normal to the ceiling until the modelled flow area and the computed effective area are the same.

The experimental data on swirl diffuser provide duct size, flow rate with corresponding static pressure and throw distances for terminal velocity at three standard values typically 0.75, 0.5 and 0.25 m/s. Additionally, data on disc thickness, swirl angle and effective area can be sought to give a full spectrum of diffuser performance.

RESULTS AND DISCUSSIONS

Flow patterns and ACE

The generic axial fan typically produces a downwash flow as shown in Fig. 2a and is significantly different to the flow pattern typified by a vortex diffuser with 3D geometry created through the diffuser macro (see Fig. 2b).

With adequate amount of swirl at a certain flow rate the lift in the flow will be dominant to produce a radial flow as shown in Figs. 3a and 4a but is still not representative of the true radial diffusion flow from a vortex (swirl) diffuser (see Figs. 3b and 4b).

When it comes to age of air as the main determinant for ACE, the generic fan's downwash flow delivers the fresh air right to the measurement plane at 1m and hence the green patch of young age of air in Fig. 5a whereas the vortex diffuser's radial flow induces mixing that causes the fresh air to mix with the room air to result in a more homogenous age of air profile (see Fig. 5b). The implications of this work in favour of the room to get a higher value of ACE when the diffusers are represented with the generic axial fan. Though the right approach, the lower ACE value from the vortex diffuser is posing a challenge to achieve the target IEQ-2.

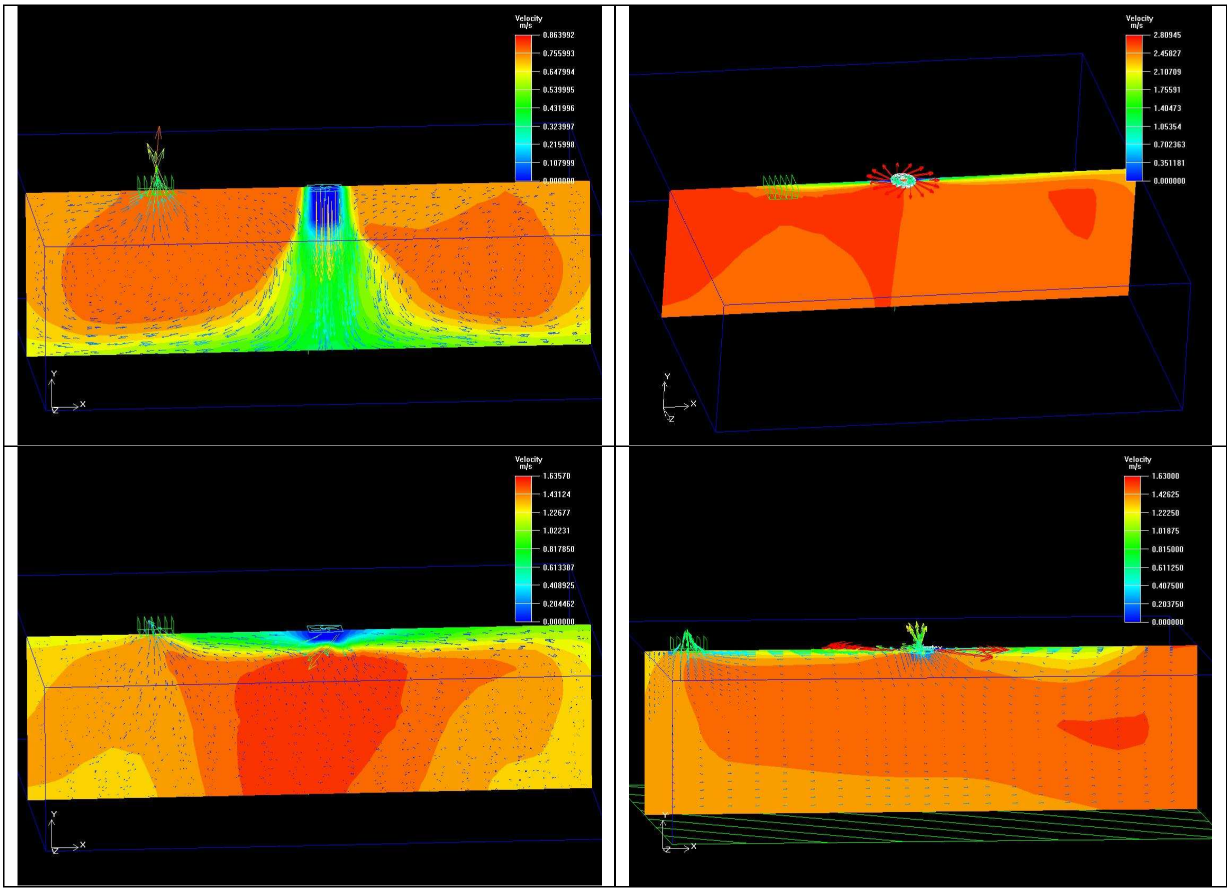


Figure 2a, 2b; 3a and 3b (top, left to right; bottom, left to right) Comparison of standard fan (left) and swirl diffuser (right) on flow pattern and air distribution.

Turn down ratio

The diffuser set up macro in Airpak was utilised to create the swirl diffuser geometry with 600 mm square nominal face, 300 mm duct size and to set up an input data file containing performance data at different flow rates associated with the throw distances, the turbulence properties and the prescribed effective area. The actual flow rate defined could then be interpolated from the performance data to give the throw distance, discharge velocity and Ar number. The effective area of the diffuser supplied by the manufacturer (Holyoake) was 0.0925 m^2 and for a flow rate of 250 L/s the throw distance was 3.1 m at terminal velocity of 0.25 m/s.

Figures 6a and 6b show that the flow patterns generated from a generic fan with swirl and a swirl diffuser are inherently different. Both fans have the

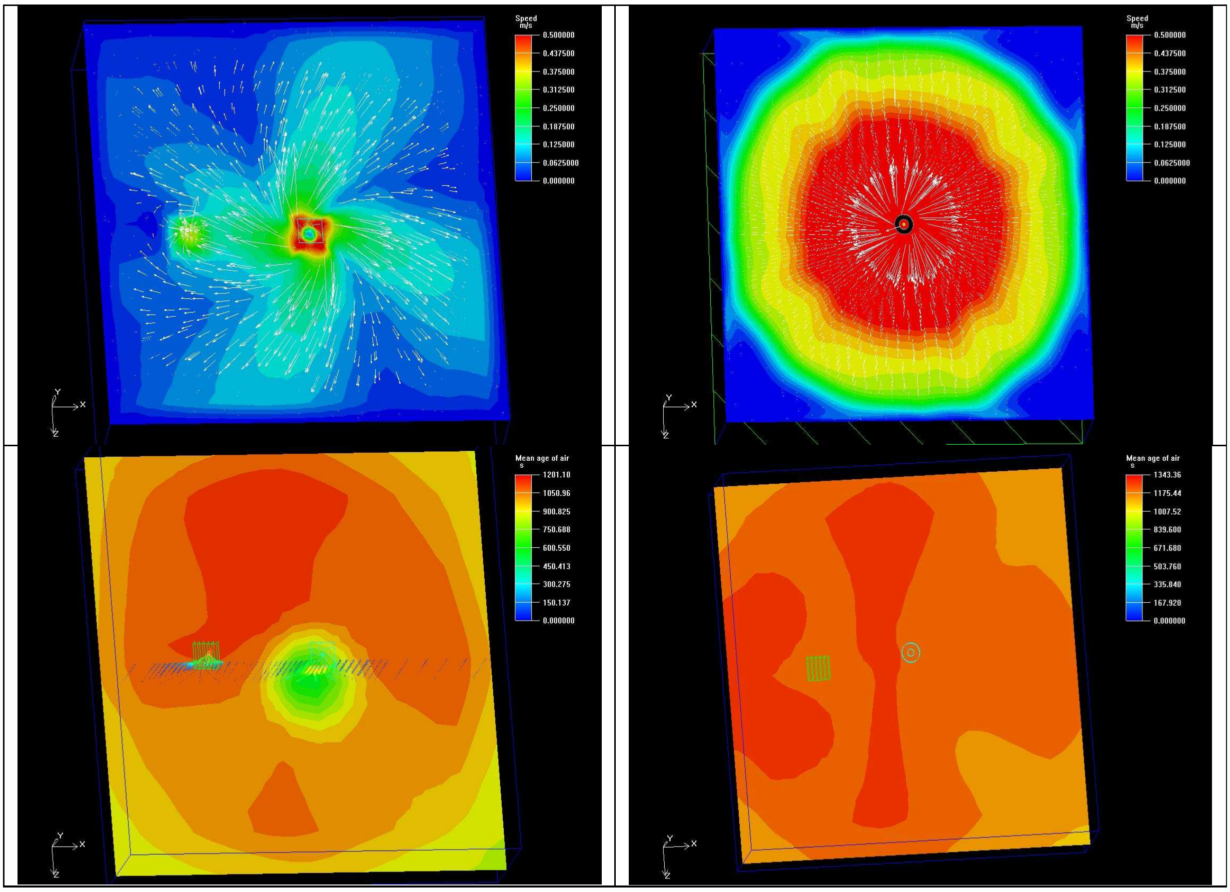


Figure 4a, 4b; 5a and 5b (top, left to right; bottom, left to right) Comparison of standard fan (left) and swirl diffuser (right) on flow pattern and age of air.

same flow rate and swirl but the disc thickness of the swirl diffuser coupled with the swirl angle propagates an uninhibited radial diffusion to create a horizontal coander effect along the ceiling.

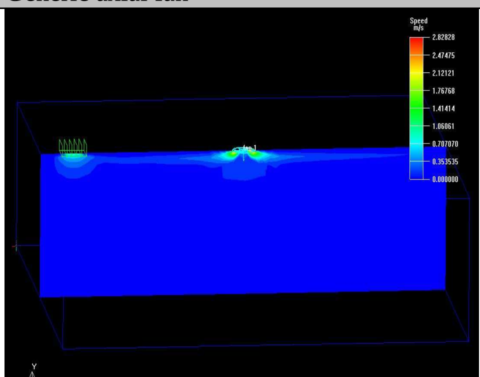
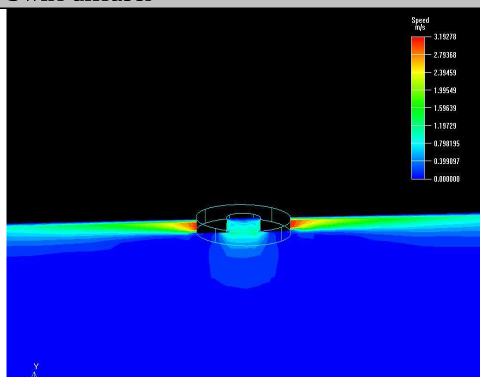
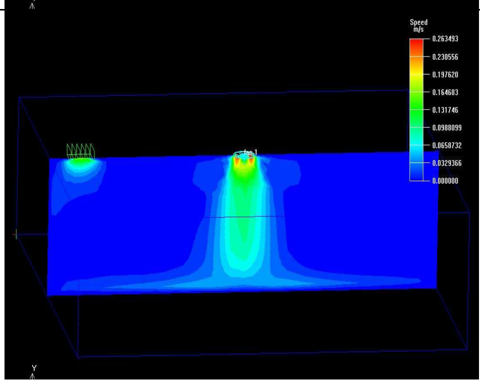
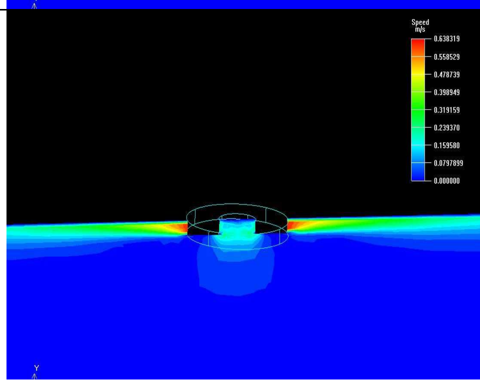
	Generic axial fan	Swirl diffuser
250 L/s		
50 L/s (25% turn down ratio)		

Figure 6a, 6b; 7a and 7b(top, left to right; bottom, left to right). Velocity profiles of generic axial fan with swirl (left) and swirl diffuser (right) at nominal flow rate of 250 L/s and 50 L/s (25% turn down ratio).

Comparatively, the swirl in the generic fan is helping the creation of radial flow by the tangential velocity at an angle from the ceiling because at the given flow rate and magnitude of swirl, an adequate upward force is taking place to give a lift to the flow. In a scenario where a turn down ratio of 25% is in effect such as in VAV distribution systems for compliance with Green Star IEQ-2's requirements to use the lowest turn down ratio, the swirl in the reduced flow rate of the generic fan has not enough lift resulting in a vertical trajectory downwash flow (Figure 7a). On the same note, the results of Allison and North (2011)'s swirl diffuser modelling derived from the standard fan construction showed a downward cascading of flow separating from the ceiling at a 45% turn down ratio, a situation coined as diffuser dumping which counter-intuitively in fact worked in ACE's favour due to the direct fresh air delivery to the measurement plane. The swirl diffuser in Figure 7b however, due to its physical characteristics, is still displaying the typical radial diffusion flow only at a much lower velocity.

Note that in the modelling, turbulence parameters in the k-ε turbulence model for the diffusers were defined by turbulent intensity taken to be 4% and the characteristic length (length scale) 0.07 times the diffuser diameter. These are considered acceptable for medium turbulent flow with Re (Reynolds number) around 20,000-25,000. The ASHRAE 55P-2003 and CEN Report-1998 recommend a turbulent intensity of less than 10% for Class A environment where higher than typical comfort standards are used, relevant in this case to the choice of lowest terminal velocity (0.25 m/s) for the diffuser and the high ACE aimed.

Coanda effect and fresh air short circuiting

Figure 8 shows the Coanda effects resulted from the radially diffused air flow from adjoining swirl diffusers. Room air is induced up into the supply flow pattern for superb mixing while the supply air is discharged as a swirling horizontal flow and diffused radially along the ceiling. Opposing flows of this type coming from adjacent diffusers collide and form a vertical layer down to then circulate and mix with the rest of the room air.

The inlet outlet configuration of the air distribution layout has to be given a consideration to minimise short circuiting of fresh air that will lower the effective delivery of it into the breathing zone of the room (measured at 1 m off the floor). This is particularly detrimental in the ceiling inlet - ceiling outlet setup because the fresh air coming out from the supply inlet is likely to exit out of the room through the return outlet before it gets mixed and delivered further into the room (see Figure 9). Several techniques that can be applied include using the minimum number of return air outlets placed strategically in the room for example against a wall.

This Coanda effect was lost at the lower flow rate (turn down ratio) of Allison and North (2011)'s diffuser CFD modelling results. Their approach led to somehow better ACE values due to diffuser dumping and defies the logic that the optimum ventilation efficiency should be achieved at maximum flow rates when the maximum amount of fresh air is supplied.

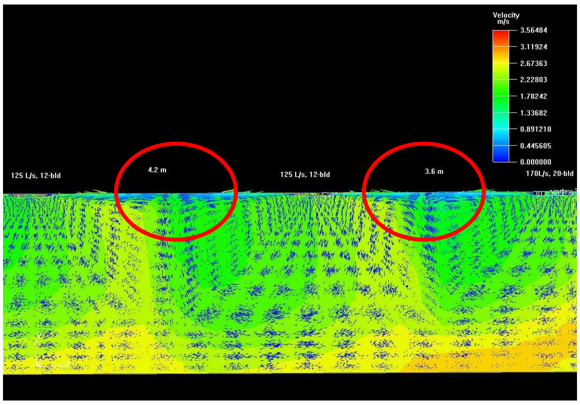


Figure 8 Coanda effects generated in radial diffusion flows from appropriate swirl diffuser modelling

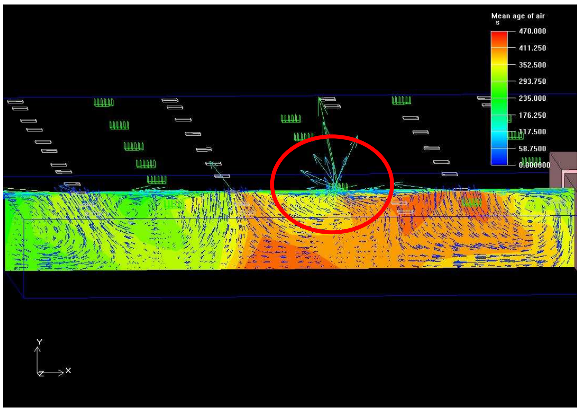


Figure 9 Short circuiting of fresh air from the ceiling inlet to outlet in entrainment flow air distribution system

CFD results compared with experimental data

The graphs below in Figures 10a, 10b and 10c show the CFD modelling results compared to experimental data supplied by the swirl diffuser manufacturer, Holyoake, for the CFP 600/24 series. The comparisons were made at 250 L/s flow rate for three different duct sizes: 250, 300 and 350 mm. The throw distances corresponding to 0.75, 0.5 and 0.25 m/s velocities obtained from their product brochure were compared with the CFD simulated results.

The modelling takes a simple approach whereby a rectangular box of 7 m x 7 m area with 2.7 m height equipped with a vortex (swirl) diffuser in the centre of the ceiling was used as the test chamber. The return air outlet was placed in the centre of the floor. Diffuser modelling specifications take the 3D radial diffusion model to match the performance data. Two discretisation schemes were tested in the CFD modelling for the dependent variables in pressure, temperature, velocity and the turbulence transports i.e. the first order and the higher accuracy second order. The turbulence k-ε model was employed with standard values used for its turbulent kinetic and dissipation rate initial values. For the first order model, only velocity was calculated whereas the second order calculation considered temperature in a cooling mode scenario where supply air temperature was set at 14ºC and room temperature was maintained at 24ºC.

The longer throw distances corresponding to the higher static pressures could be predicted by the second order to correlate reasonably well with the experimental measurements. The throw distances for 0.75 m/s velocity uniformly fall short of the measured values but for 0.5 m/s and 0.25 m/s the throw distances from the simulated results are in fair agreement particularly the latter. Although the second order discretisation scheme gives a more accurate prediction, it is envisaged that the computer resource-efficient first order scheme with only velocity calculation could still benefit designers to gain insights into the likely flow patterns and their interactions from different placements of inlets and outlets within an air distribution system and to some extent the assessment of ‘as design’ ACE to Green Star IEQ-2. All considered the practice of using CFD modelling to calculate the mean age of air for ACE assessment is by no means simple. The heat loads distribution, room layout, HVAC design, locations of supply and return air grilles, types and specifications of air inlets and outlets and the target ventilation effectiveness could make the modelling task quite exhaustive. Multiple iterations can sometimes be needed to achieve the target ACE whilst striking the right balance of all the design criteria above. It is thus very useful to have a simple and reliable tool and approach to perform all these requirements. In essence, the simulated results produced by the first order discretisation scheme represent reasonable characteristic airflow patterns and performance signature from swirl diffusers that provide a cost-effective means to optimise the air distribution within a room at design stage.

Other room or system related factors such as buoyancy effects in the air conditioning operating modes of heating or cooling (Fisk et al., 1997), short circuiting of fresh air as discussed above and its dependency on the configuration (displacement or entrainment) and layout of the air distribution system, the amount of re-circulated air in the return air path that gets channelled back into the supply air stream which affects the freshness (age of air) of the supply air are all important and should be considered in the ACE calculation. Although from a different perspective, Allison and North (2011) agree that the GBCA guidelines are inadequate, further stating the ACE benchmark value of 0.95 is unrealistically hard to achieve. It is then up to Green Star to establish a more rigorous set of guidelines on how to take into account or exclude with considerations all these other factors, a standard platform on which all ACE results can be compared and also the realistic benchmarks of ACE for these different air distribution systems used in real practice.

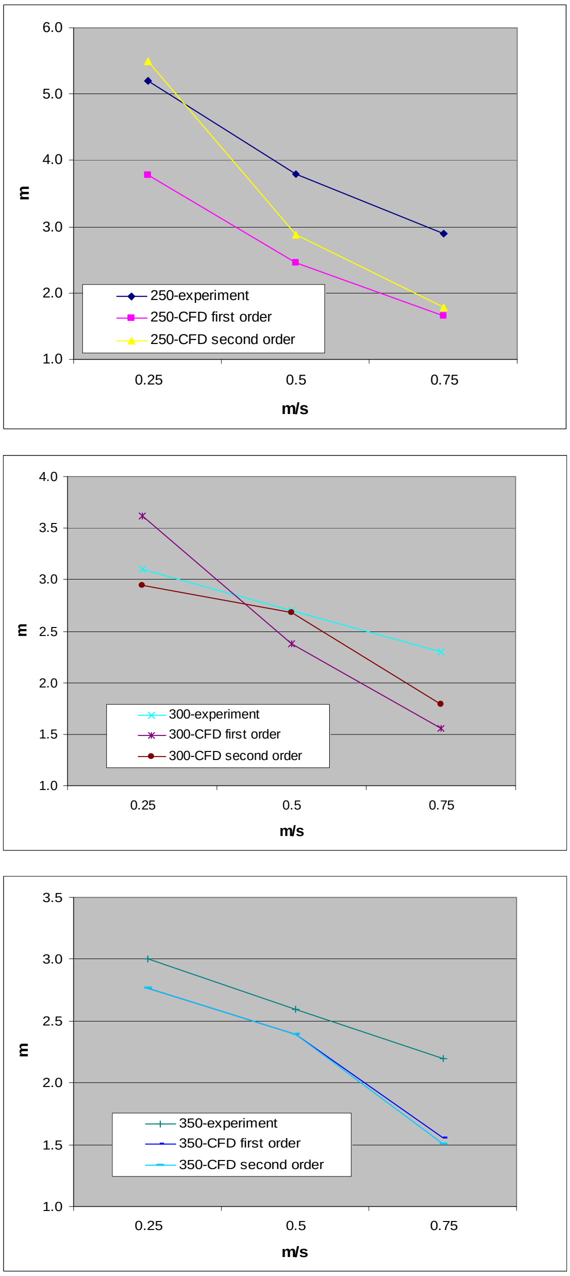


Figure 10a, 10b and 10c. Comparisons of simulated and experimental measurements of throw distances at the benchmark velocities 0.25, 0.5 and 0.75 m/s.

CONCLUSIONS

Several conclusions can be summarised from the studies as follows:

1. Accurate calibration and representation of swirl diffusers with proper modelling technique incorporating diffuser performance data need to be implemented in CFD modelling to assess ACE for Green Star IEQ-2.

2. Several fundamental attributes in swirl diffuser's virtual object accounting for its specific performance parameters to deliver its typical flow characteristics are different and can not be achieved by the use of generic axial fan in CFD modelling.

3. The flow pattern, air distribution and the age of air - used to calculate ACE - are significantly different between a generic axial fan and a swirl diffuser. The proper swirl diffuser modelling approach will give the correct airflow pattern with valid air distribution but not necessarily a favourable value of ACE.

4. The generic axial fan fails to maintain its radial flow pattern in a scenario with a low turn down ratio such as in VAV systems, a feature of the vortex (swirl) diffuser's that persists through out its operating range.

5. The radially horizontal diffusion along the ceiling from adjacent swirl diffusers forms coanda effects that collide and permeate down the room creating superior mixing typical of swirl diffuser air distribution pattern.

6. Optimising air distribution layout to allow best performance from diffusers include strategies in selecting the right type of diffusers with specific flow rates paying attention to load distribution of the room, spacing and distributing correctly and evenly across the ceiling and minimising short circuiting of supply air by strategically placing the return air outlets

7. CFD modelling has the capability to represent swirl diffuser performance data reasonably well. Comparisons with measured experimental data from diffuser manufacturer show throw distances corresponding with their terminal velocities are better represented with the second order discretisation scheme with some practical benefits to be gained from the cost-effective first order scheme.

8. More comprehensive guidelines should be developed for CFD modelling in ACE assessment to Green Star IEQ-2. Realistic passing thresholds for different air distribution systems and configurations should be formulated emphasising on inclusion of re-circulated air and operating modes in the ACE calculation for realistic representation of real HVAC systems.

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