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## Numerical investigation of diffuse ceiling ventilation in an office under different operating conditions

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**KEYWORDS:** *Diffuse ceiling ventilation, ventilation efficiency, ventilation, draught, CFD*

### **SUMMARY:**

*Diffuse ceiling ventilation is a novel air distribution device that combines the suspended acoustic ceiling with ventilation supply. A diffuse ceiling distributes the supply air above the acoustic tiles and has proven performance in both laboratory and class room experiments. This paper is a numerical study of the performance of a six person office equipped with diffuse ventilation ceiling. In total six extreme, yet realistic, operation scenarios were simulated to study the performance including different occupancy, ventilation rates and supply air temperatures. The performance was studied with regard to air change efficiency, air movements, temperatures and stratification. In all scenarios the ceiling performed satisfactorily.*

### **1. Introduction**

The main purpose of the ventilation system in a building is to supply fresh air to the occupants and secondary to remove excessive heat. The ventilation system should do so energy-efficiently and with minimal risk of discomfort in the occupied comfort zone. A promising concept for this is diffuse ceiling ventilation which is a novel air distribution device that combines the suspended acoustic ceiling with ventilation supply. The principle of diffuse ceiling ventilation for comfort ventilation is to inject the supply air into the plenum above a standard suspended acoustic ceiling. The plenum thereby works as a pressure chamber and air is distributed to the room below through cracks and perforations. The flow velocity into the room is very small and irregular, hence the term diffuse.

The reported research in this area mostly relies on laboratory experiments where results have been promising. Nielsen & Jakubowska (2008) published results where diffuse ceiling ventilation outperformed five conventional air distribution systems in a laboratory office environment. The findings were supported by Hviid & Svendsen (2013) who carried out experiments in a test facility resembling a small classroom as well as in a real classroom (Hviid & Terkildsen 2012). Fan et al. (2013) has documented good performance of the concept both experimentally and numerically. Numerical investigations by CFD (Computational Fluid Dynamics) were shortly performed on an office by Nielsen et al. (2010) to investigate the level of stratification in heating mode. Hviid & Petersen (2011) demonstrated that diffuse ceiling ventilation may improve the night cooling potential of a classroom.

The objective of the research reported in this paper is to assess the indoor climate performance of diffuse ceiling ventilation in an office environment under six different operating conditions that stresses the concept. The performance is assessed numerically by CFD in terms of thermal conditions, air movements, stratification, and air change efficiency.

## 2. Test case

The investigations have been performed on an office located in a large newly erected building in the harbour of Aarhus, Denmark. The numerical solution domain has been established from the geometric model of the irregular office room depicted in FIG 1. This room was selected because the large façade/floor ratio has high solar gain and high thermal losses, thus imposing maximum stress on the HVAC strategy. The floor area of the office is  $41.4 \text{ m}^2$  with a room height of 3.0 m. The room has two façades with the compass orientations  $177^\circ / 303^\circ$ . The windows are arranged as transparent bands with a height of 1.8 m. The parapet is 0.875 m high.



FIG 1. Plan view of office. Furniture layout reflects potential meeting room

### 2.1 Diffuse ceiling

The suspended acoustic ceiling used as diffuse inlet consists of horizontal aluminium lamellas attached to a carrier as depicted on FIG 2 with 20 mm of black acoustic mineral wool batts overlying the lamellas. The finalized ceiling is depicted on FIG 3. The main air paths are the small holes in the universal carrier as well as cracks between batts and suspension profiles originating from the installation. This means that some air enters the room as linear microjets along the carriers, and some air enters diffusively.

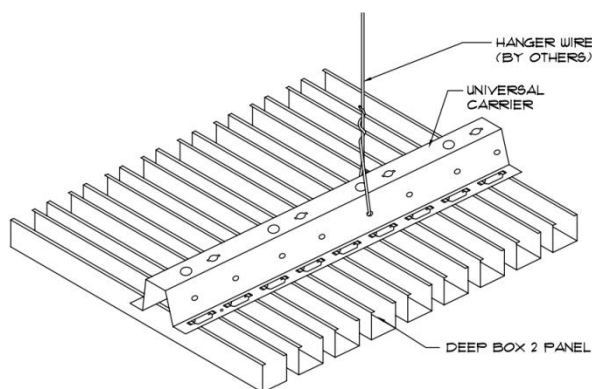


FIG 2. Aluminium lamellas with suspension system



FIG 3. Finalized ceiling

### 3. Method

#### 3.1 CFD model

For the simulations the commercial CFD code Ansys CFX version 14.5 was used. FIG 4 depicts the CFD model with six workstations, occupants and computers. The lighting fixtures are not represented as they are off in all scenarios due to daylight control. All internal components of the model are simplified with box-shaped geometric models.

Heat transmission through the façade is modelled by the following U-values and the scenario-dependent external temperature:

- Opaque façade U-value:  $0.14 \text{ W}/(\text{m}^2 \cdot \text{K})$
- Window U-value:  $0.8 \text{ W}/(\text{m}^2 \cdot \text{K})$ , g-value perpendicularly: 0.3, incident angle dependent

Solar radiation is modelled by applying convective heat sources on the windows, i.e. it is not directional. The amount of incoming direct, diffuse and reflected solar and sky radiation depends on the simulation scenario as well as time of year and day, and the g-value of the glazing. Radiation exchange between surfaces in the model is not included in the calculations. The radiator is modelled as a surface under the window with a given heat flux depending on the simulation scenario. Each occupant provides a heat gain of 90 W and each computer a heat gain of 60 W.

The heat gains of the domain is balanced by making the internal walls absorb excess heat which occurs when, for instance, the room is heated with warm supply air.

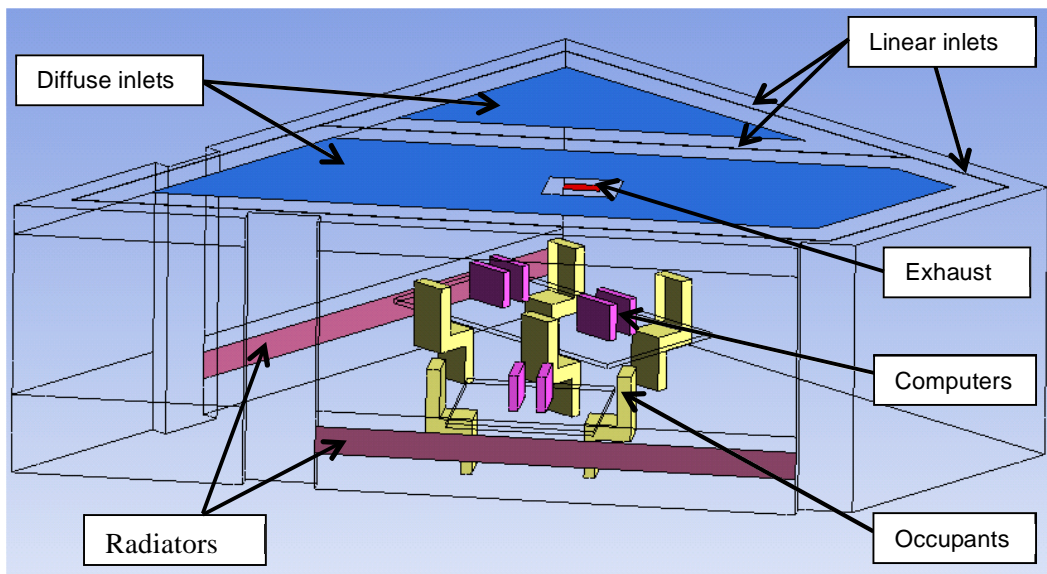


FIG 4. Solution domain with six workstations

The ventilation is supplied partly diffusively through the ceiling surfaces depicted on FIG 4, partly by microjets along the ceiling carriers (linear inlets). The latter are placed where the ceiling lamella finishes to the wall and every 3 m. The airflow distribution is assumed to be 70/30 with most air through the linear inlets. The width of the slot is adjusted to an inlet velocity of 0.23 m/s. The rest of the air is distributed over the diffuse surfaces.

The mesh comprises approx. 3.5 million cells and plane-parallel cell layers are implemented along the walls and heat sources to better capture the fluid flow. The Boussinesq approximation was chosen to model buoyancy driven forces as it provides faster convergence for many natural-convection flows. The applied turbulence model is the k-epsilon model, which is stable and with sufficiently accurate

results for most indoor applications. The turbulence model is combined with the CFX feature 'scalable wall laws', thus heat transfer between surfaces and air is made independent of grid size.

The convergence residual criteria were default values (0.0001) and no mass/heat flux imbalances were larger than 1 %. The reported results are transient since the temperature differences create weak unstable air currents which are unresolvable by steady-state calculations.

### 3.2 Investigated scenarios

The driving forces with diffuse ceiling ventilation are the thermal convective plumes that arise above any heat source. Thus, for air change efficiency we expect that the most critical scenarios are those with small temperature differences throughout the domain and small convective heat sources.

Regarding thermal environment and elevated air movements, we expect that large airflow rate with low supply temperature to be the most problematic scenario. Six scenarios are therefore investigated:

1. **Normal operation.** Overcast winter day with occupants and computers. There are no expected critical aspects.
2. **Pre-conditioning.** Winter morning without occupants and computers. The critical aspect is possibly worsened air change efficiency with no occupants.
3. **Air-heating.** Winter morning without occupants and computers. The critical aspect is stratification.
4. **Cooling by low inlet temperature.** Summer day with occupants and computers. The critical aspect is the risk of draught due to low supply temperature.
5. **Cooling by high flow rate.** Summer day with occupants and computers on. The critical aspect is the risk of draught due to high airflow rate.
6. **Night ventilation.** Summer night without occupants and computers. The critical aspect is air change efficiency, subsequently night cooling efficiency, without heat sources present.

The design indoor air temperature is 21 °C in winter and 26 °C in summer. The specific data for each scenario is summarised in TABLE 1.

TABLE 1. Input to investigated scenarios. The emboldened values indicate the critical aspects.

|   | Season         | Out. temp. | Flow rate               | Supply air temp. | $\Delta T$ air | Work-stations | Radiator power | Solar radiation |
|---|----------------|------------|-------------------------|------------------|----------------|---------------|----------------|-----------------|
| 1 | Winter day     | -5 °C      | 2 h <sup>-1</sup>       | 16 °C            | -5 °C          | 900 W         | 0 W            | 87 W            |
| 2 | Winter morning | -12 °C     | 2 h <sup>-1</sup>       | 21 °C            | <b>0 °C</b>    | 0 W           | <b>249 W</b>   | 0 W             |
| 3 | Winter morning | -12 °C     | 3 h <sup>-1</sup>       | 27 °C            | <b>+6 °C</b>   | 0 W           | 0 W            | 0 W             |
| 4 | Summer day     | 28 °C      | 4 h <sup>-1</sup>       | 14 °C            | <b>-12 °C</b>  | 900 W         | 0 W            | 1330 W          |
| 5 | Summer day     | 28 °C      | <b>6 h<sup>-1</sup></b> | 18 °C            | -6 °C          | 900 W         | 0 W            | 1330 W          |
| 6 | Summer night   | 18 °C      | 2 h <sup>-1</sup>       | 19 °C            | -2 °C          | <b>0 W</b>    | <b>0 W</b>     | <b>0 W</b>      |

### 3.3 Age of air

The age of air is a measure of the air quality. The air age (Sandberg & Sjöberg 1983) in a given point is a measure of how long a massless particle of air spent on transport from the inlet opening to that point. In the case of perfect mixing the air has the same age throughout the room. In locations with short circuiting or poor mixing, i.e. poor ventilation efficiency, the age differs significantly from the mean age. This age of air index is not dependent on the presence of specific pollutants or source location and eliminates the need for definitions of the occupied zone and breathing zone.

## 4. Results

### 4.1 Temperature performance

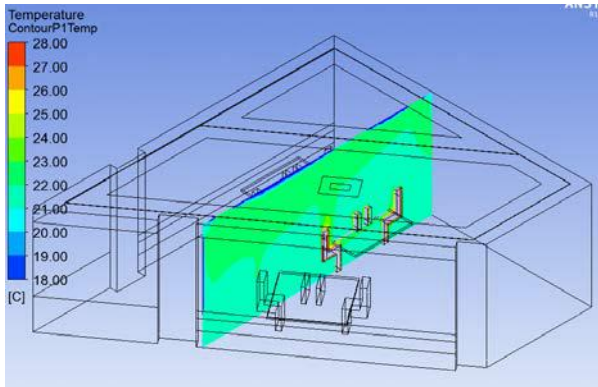


FIG 5. Thermal, scenario 1

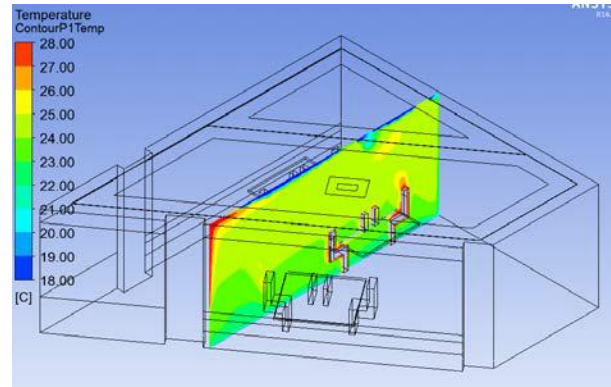


FIG 6. Thermal, scenario 4

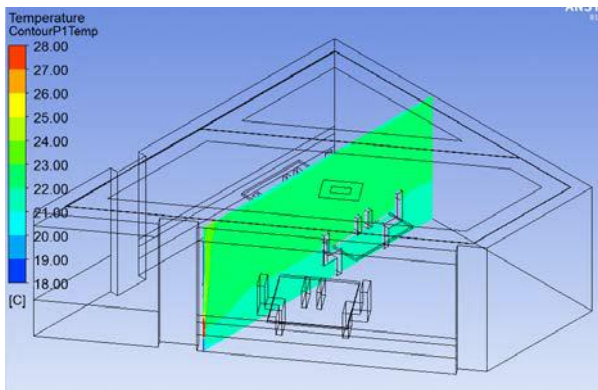


FIG 7. Thermal, scenario 2

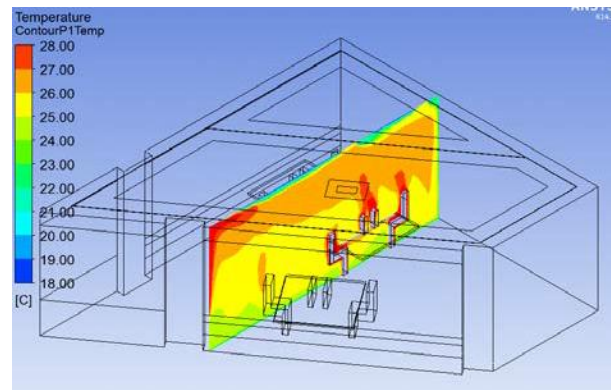


FIG 8. Thermal, scenario 5

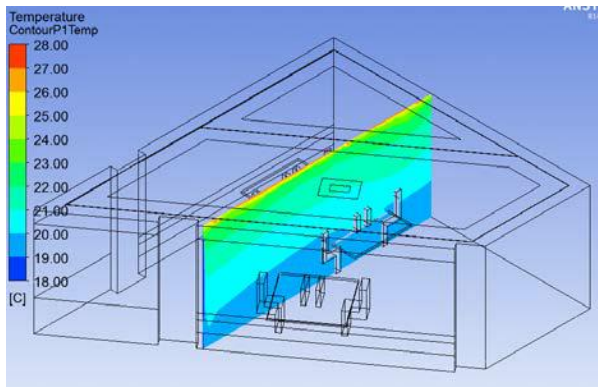


FIG 9. Thermal, scenario 3

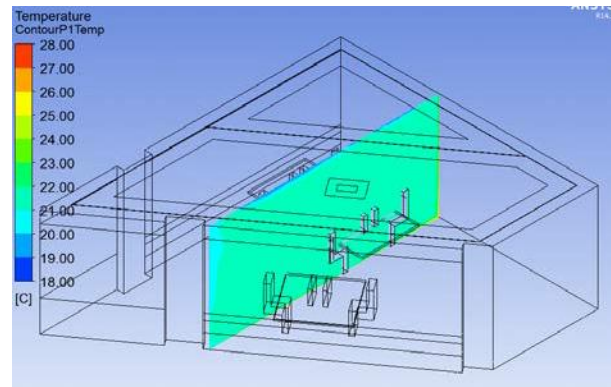


FIG 10. Thermal, scenario 6

The thermal performance of the scenarios, FIG 5-FIG 10, is in general satisfactory with very small stratification which is in good agreement with experiments (Hviid & Svendsen, 2012; Fan et al., 2013). Scenario 3 shows critical temperatures in the comfort zone partly due to cold downdraught from the window. Some stratification is present in scenario 5 but it is quite small, 1 K/m, and thus well within the limits of DS/EN ISO 7730.

## 4.2 Air movements

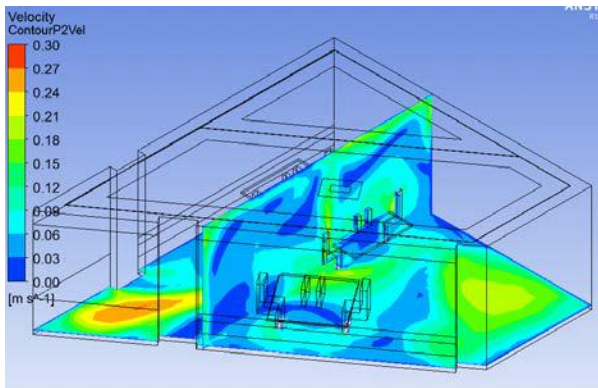


FIG 11. Air velocity, scenario 1

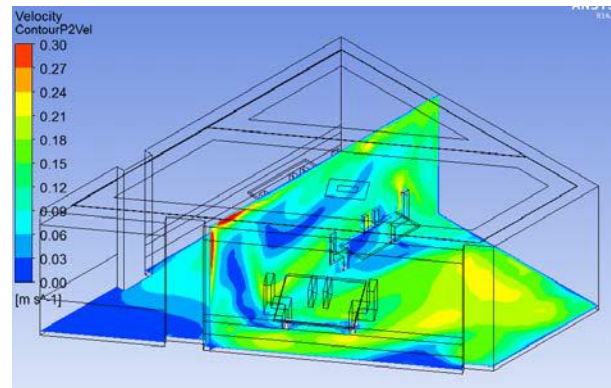


FIG 12. Air velocity, scenario 4

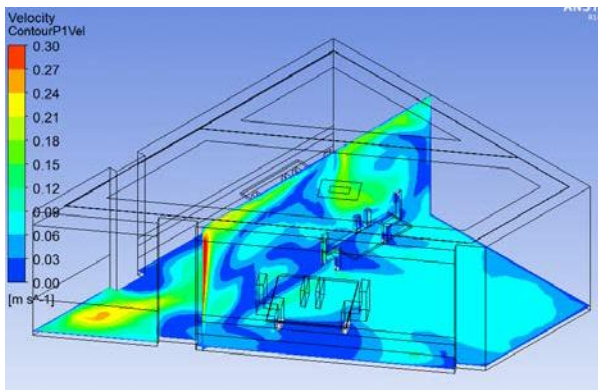


FIG 13. Air velocity, scenario 2

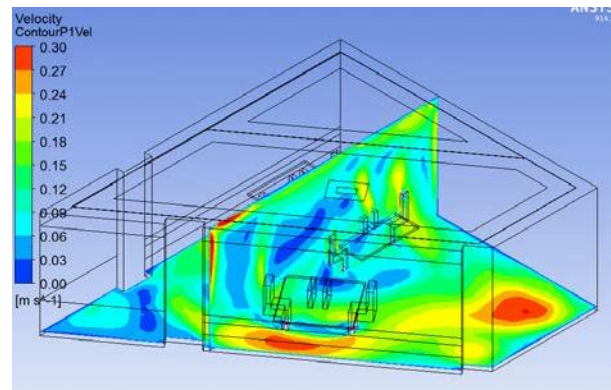


FIG 14. Air velocity, scenario 5

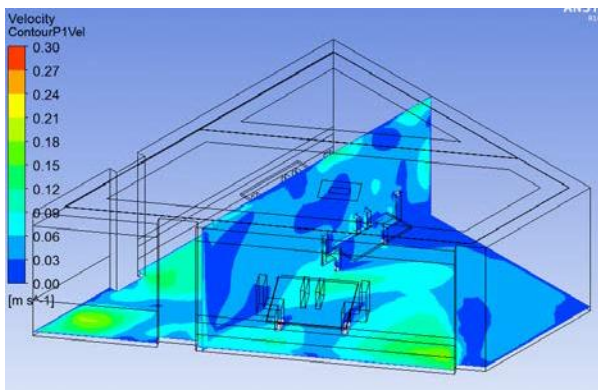


FIG 15. Air velocity, scenario 3

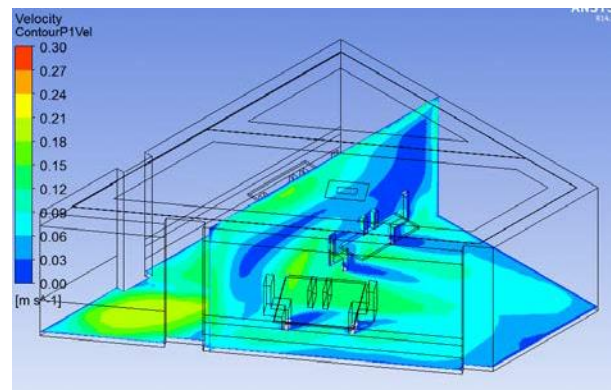


FIG 16. Air velocity, scenario 6

The air velocities in the winter scenarios, FIG 11 - FIG 16, are satisfactory, i.e. the air velocities in the comfort zone are below 0.15 m/s. In the summer scenarios, the two daytime scenarios (4 and 5) show elevated velocities (approx. 0.3 m/s) at ankle height in some areas. Analyses of these two scenarios show that cool, fluctuating ventilation air drops from the ceiling. This and the convective thermal plumes from occupants, computers, and the warm window create room size vortices along the floor. This behaviour has not been reported, neither by Nielsen et al (2010), nor by Hviid & Svendsen (2013) or Fan et al. (2013). However, a certain displacement effect is expected and this indicates the need for more investigations into the natural convection forces and transient behaviour of diffuse ventilation ceilings.

### 4.3 Air quality by age-of-air

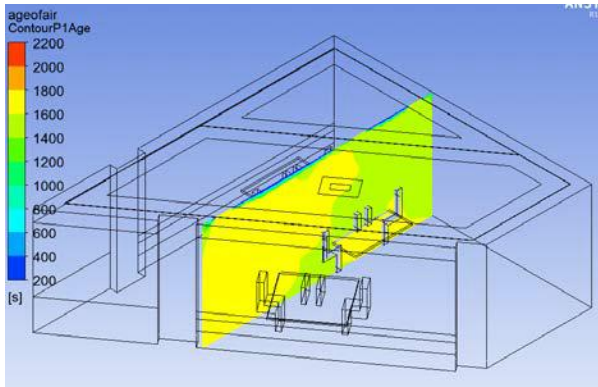


FIG 17. Age of air, scenario 1

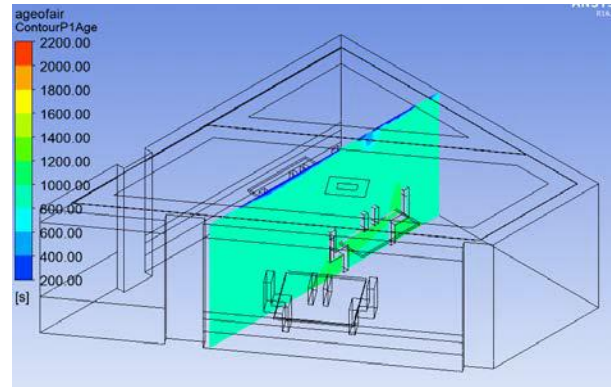


FIG 18. Age of air, scenario 4

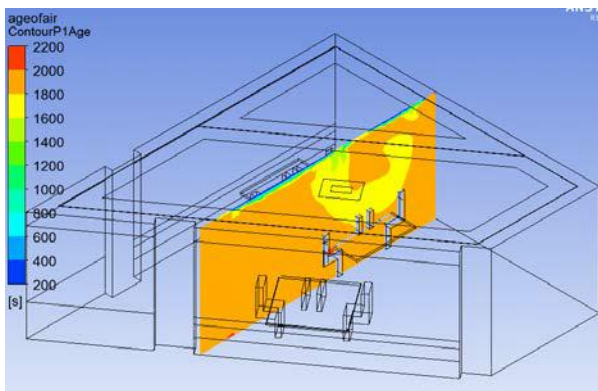


FIG 19. Age of air, scenario 2

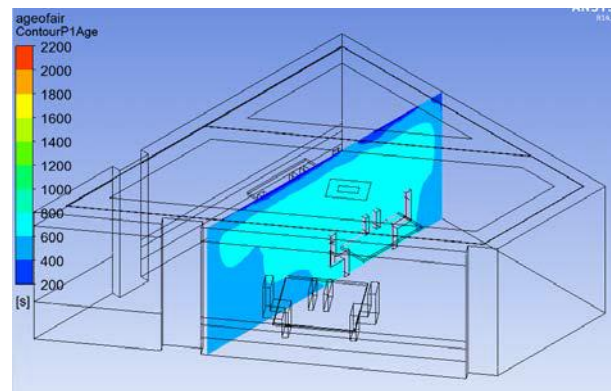


FIG 20. Age of air, scenario 5

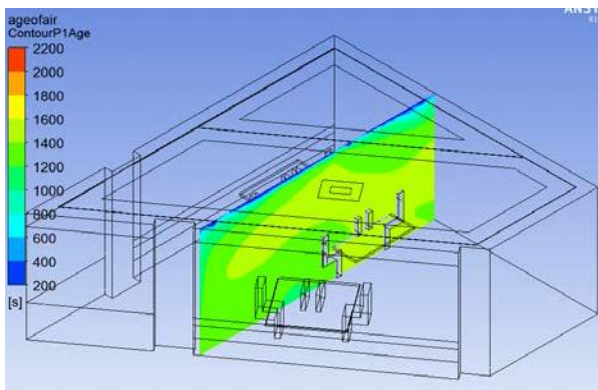


FIG 21. Age of air, scenario 3

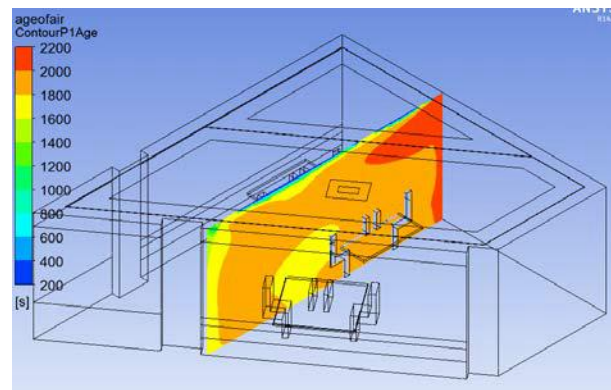


FIG 22. Age of air, scenario 6

The age of air shows the air change efficiency in different parts of the room in FIG 17-FIG 22. None of the scenarios show any sign of short-circuiting and stagnant zones. conventional mixing ventilation, which is also validated by experimental data from Hviid & Svendsen (2012), and Fan et al. (2013).

TABLE 2 compares the mean air change efficiency of the room with the analytically derived value of perfect mixing ventilation. The largest deviation is 9 %, hence the diffuse ceiling ventilation performs on par with conventional mixing ventilation, which is also validated by experimental data from Hviid & Svendsen (2012), and Fan et al. (2013).



TABLE 2. Comparison of simulated air change efficiency with perfect mixing

| Scen. | Description                                 | Perfect mixing | CFD-results @ outlet | Deviation |
|-------|---|----------------|----------------------|-----------|
| 1     | Winter day, normal operation                | 1800 s         | 1640                 | -9 %      |
| 2     | Winter morning, pre-conditioning            | 1800 s         | 1793                 | 0 %       |
| 3     | Winter morning, air-heating                 | 1200 s         | 1220                 | 2 %       |
| 4     | Summer day, cooling by low supply air temp. | 900 s          | 901                  | 0 %       |
| 5     | Summer day, cooling by high flow rate       | 600 s          | 576                  | -4 %      |
| 6     | Summer night ventilation                    | 1800 s         | 1755                 | -3 %      |

#### 4.4 Energy efficiency

In terms of energy efficiency, the pressure drop of diffuse ceilings was measured and reported by Hviid & Svendsen (2012) and Fan et al. (2013) to be approx. 0.5-3.5 Pa. Thus, in comparison with conventional mixing diffusers, the diffuse ceiling performs on par at 1/20 pressure drop. For current best-practice ventilation systems with a specific fan power of 1.1 kJ/m<sup>3</sup>, the pressure savings by the diffuse ceiling constitutes approx. 10% of the supply pressure drop.

## 5. Conclusion

In general, the diffuse ceiling ventilation concept has performed on par with conventional mixing ventilation where the momentum forces of the supply jet entrains and mixes with the room air. The air change efficiency (age of air) documented that no stagnant zones or short-circuiting were present.

The thermal performance does not show stratification in the investigated scenarios, except when used with air-heating. Possibly, a radiator below the window will create sufficient thermal plume to avoid stratification.

The air velocities were satisfactory but the investigations in this paper have disclosed cold down draughts with this type of diffuse ceiling in an office setting during summer. However, these issues have not been reported in the literature, indeed, laboratory experiments performed on similar types of ceilings have not identified any problematic behaviour. Consequently, it is relevant to investigate further the behaviour and consequences of the transient convective forces.

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