The impact of a uniformly distributed thermal load on mixed flow

Ingo Gores

Hermann-Rietschel-Institute for Heating and Air-Conditioning, Technical University Berlin, Germany

ABSTRACT

The influence of uniformly distributed cooling load is investigated on the air velocity in the occupied zone. The experiments have been carried out in a scaled test room of 6 × 6 × 2 m with a scale factor 1:1.5. The room is equipped with nine vortex inlets. The inlets are uniformly distributed; every inlet supplies a cubical volume to remove the maximum possible cooling load with a minimum airflow rate. The air supply rates based on floor area are set to 30 $m^3/(h\ m^2)$. The cooling load generated by mannequins is uniformly distributed and varies between 0 and 180 W/m^2. The study describes the observed airflow pattern and the relationship between the air velocity and the cooling load per m^2 for mixed flow.

INDEX TERMS

Mixing ventilation; Air movement; Air velocity; Diffuser; Draught; Test room; Thermal comfort; Ventilation rate

INTRODUCTION

Although displacement flow guarantees better ventilation efficiency, mixed flow supply is still a dominant ventilation strategy in most buildings. Usually, mixed airflow systems are equipped with high flow momentum inlets. In order to reduce room airflow velocities and to improve the thermal comfort low airflow rates should be adjusted (Fitzner, 1997). Basically three main phenomena influence the maximum air velocity in the room (Figure 1): (1) inlet air velocity and diffuser type, (2) interaction of air jets, (3) deflection and/or increasing of supply jets by buoyancy forces (heat load, walls, etc.).

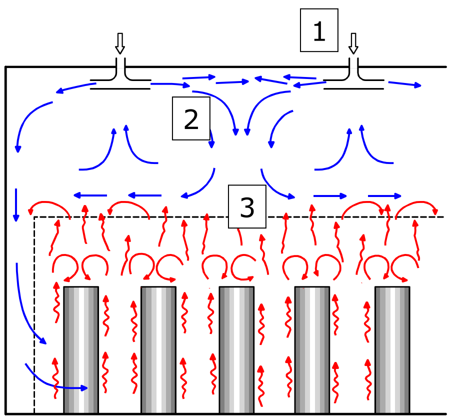

Figure 1 Inlet velocity, interaction of jets and buoyancy forces influence the maximum room air velocity.

Several investigations have described the behaviour of radial jets in rooms (e.g. Waschke, 1975; Regenscheit, 1981; Schaefers, 1982). Waschke, e.g. recommends a decreasing of velocity to

$$\frac{w}{w_0} \sim \left(\frac{r}{r_0}\right)^{-1.151},$$

where w and w_0 are the air velocity in radial direction and the air velocity at distance $r = 0$, r_0 and r are the radius of the radial diffuser and the radial distance to the inlet. Some authors examined the interaction of airflow jets (Conrad, 1972) and made suggestions for the supply air velocity reduction after the jets collide:

$$\frac{w_m}{w_o} = K \cdot \left(\frac{h_o}{m \cdot X \cdot L}\right)^{0.375} \cdot \left(\frac{L}{L-Y}\right)^q,$$

where Y is the vertical distance from the ceiling to the measurement place, L is the horizontal distance to the point of interaction and $K = 0.65$, $q = 0.65$ are empirical factors, if two air jets collide. In recent years, computers and machines have increased the internal cooling load. Investigations by Fitzner (1997) have illustrated that the widely spread radial mixed airflow systems are able to remove cooling loads up to 100 W/m² without draught. Behne (1995) has shown a relationship between cooling load based on floor area and the airflow velocity in rooms cooled by chilled ceilings without any supply or exhaust air (Figure 2).

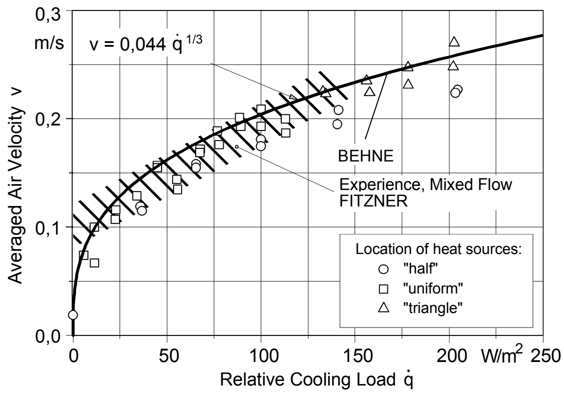

Figure 2 Averaged air velocity in relationship to relative cooling load in buoyancy forced flow and mixed flow, $H = 3$ m.

THE CUBE RULE

Investigations by Fitzner (1997) have proved optimal airflow patterns with lowest velocities and maximum cooling load can be removed, if every inlet supplies a cubical volume (Figure 3).

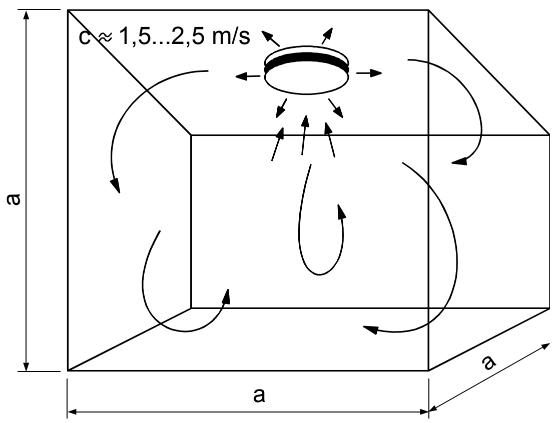

Figure 3 One radial inlet supplies a cubicle volume.

This leads to less but larger inlets in tall halls and to more but smaller inlets in normal office facilities. The diameter of the inlet is proportional to the height of the room.

We have applied the cube rule and have performed the examination in a test room consisting of nine similar 'cubes'. The set-up of the test room contains nine radial inlets to create one 'cube' in the middle of the room. The airflow pattern of this cube was not surrounded by walls. This is an important requirement for transferring the results to larger spaces. Unfortunately, it was impossible to set up an office room with 3 m height and a floor area of 3 × 3 cubes = 9 × 9 = 81m^2. The test chamber had to be reduced by a scaling factor M = 1.5. The dimensions result in $L \times W \times H = 6 \times 6 \times 2$ m. Non-isothermal flow in the model is similar to the full-scale flow if the Archimedes number and the Reynolds number is the same in both cases. The results will be sufficiently correct if the Ar-numbers of the prototype airflow and the model airflow have the same values in the far-wall area. The absolute temperature T and the temperature difference ΔT supply/exhaust air in the prototype and the model shall be equal. As a consequence the velocity w in the model is calculated from the Ar-numbers $\mathrm{Ar}_{\mathrm{model}} = \mathrm{Ar}_{\mathrm{prototype}}$ as

$$w_{\mathrm{model}} = \frac{w_{\mathrm{prototype}}}{M^{1/2}}$$

EXPERIMENTAL SET-UP

Cooled air is supplied to the test from nine swirl diffusers (Figure 4). The airflow rate has been set to 30 m³/(h m²). Exhaust air devices were located near the floor. Sixty-four mannequins were uniformly distributed on the floor to simulate heat sources like people in a theatre. During the investigation the loads varied between 0 and 180 W/m².

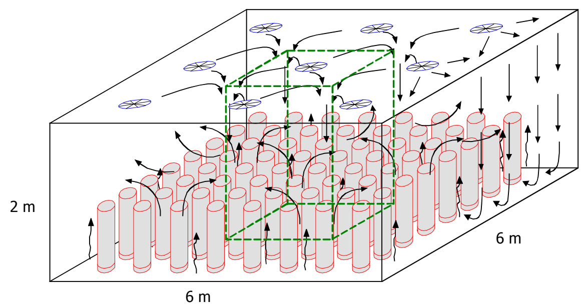

Figure 4 Test room set up with nine inlets, exhaust outlets near the floor and 64 mannequins to simulate cooling loads.

Wall and air temperatures were measured by thermocouples. The air velocities were measured by thermal anemometers. The test room has been located in an air-conditioned experimental hall. The mean test room air temperature and the experimental hall temperature were kept constant.

One of the main difficulties was to find the area of maximum velocities. For this reason it was necessary to visualize the airflow pattern by smoke before measuring.

A swirl diffuser generates characteristic airflow patterns. Although there are differences, the swirl diffuser flow is similar to the radial flow. Every radial inlet forms a rotating airflow pattern in a round-shaped way (Figure 5).

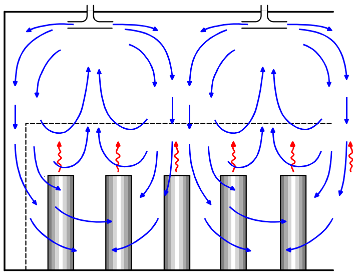

Figure 5 Expected airflow pattern between two inlets.

We expected a maximum value of the air velocity, where the radial airflow jets meet each other and be deflected downwards. But the observed airflow pattern shows a different behaviour. Significant down flow in the occupied zone is only observed in the geometric middle of four radial inlets (Figure 6).

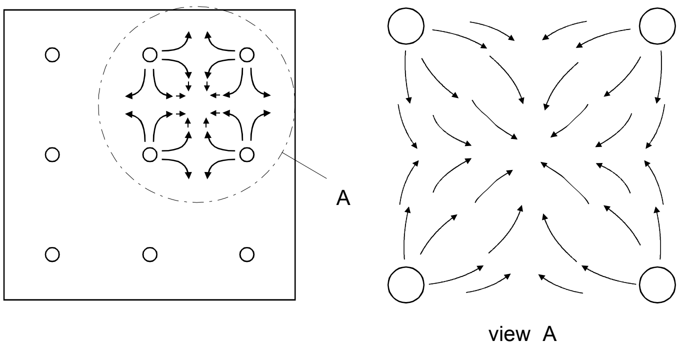

Figure 6 Observed airflow pattern: significant down flow in the geometric middle of four inlets.

As a result the author decided to measure the velocities with eight anemometers in line at the positions described in Figure 7 in a height of 0.07 m to 1.65 m in the scaled test room, which means a height of 0.1 m to 2.5 m in the prototype scale.

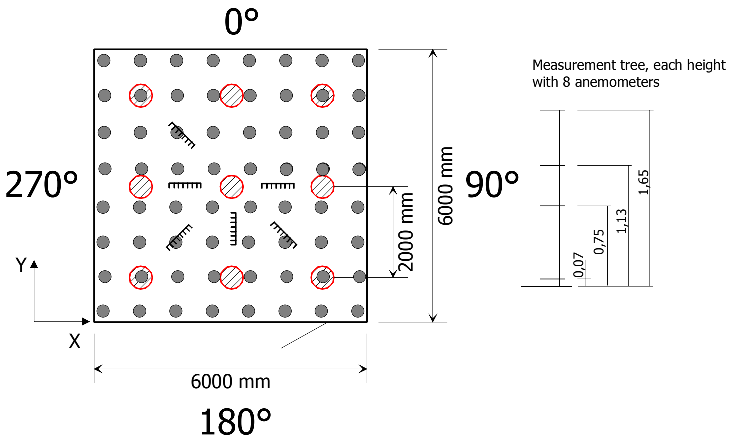

Figure 7 Velocity measurement set up in the model room.

RESULTS AND DISCUSSION

The velocities were measured at six different angles (Figure 7). At all places the maximum air velocity was increasing with the cooling load (Figure 8a and b). Maximum values were achieved in sections 225°, 315°. The maximum of average air velocities in section 180° were lower then the values of section 225°. The airflow jets between the 'shortest distance' inlets (90°, 180°, 270°) were colliding and deflecting not downwards but to the side. In section 180° most of the maximum values were measured at the bottom. The smoke tests have shown that the air jets have been deflected into the occupied zone only at the geometric centre of four inlets.

Most of the measured velocities were above the postulated curve by Fitzner. This could be due to the influence of the supply diffusers. Fitzner investigated the airflow pattern in a room without supply system, but with a chilled ceiling.

In addition to different maximum values a minimum of air velocity has been observed at the interaction point of four air jets at approx. 40 W/m² in section 225°. Figure 8(b) illustrates that the maximum of the average air velocity is increasing again, although the cooling load is adjusted to lower values then 40 W/m². For nearly all adjusted cooling loads the heated mannequins have generated a strong and stable buoyancy flow. Cooling loads less than 40 W/m² were no longer able to deflect the supply air jets. The chilled supply air entered the occupied zone undisturbed.

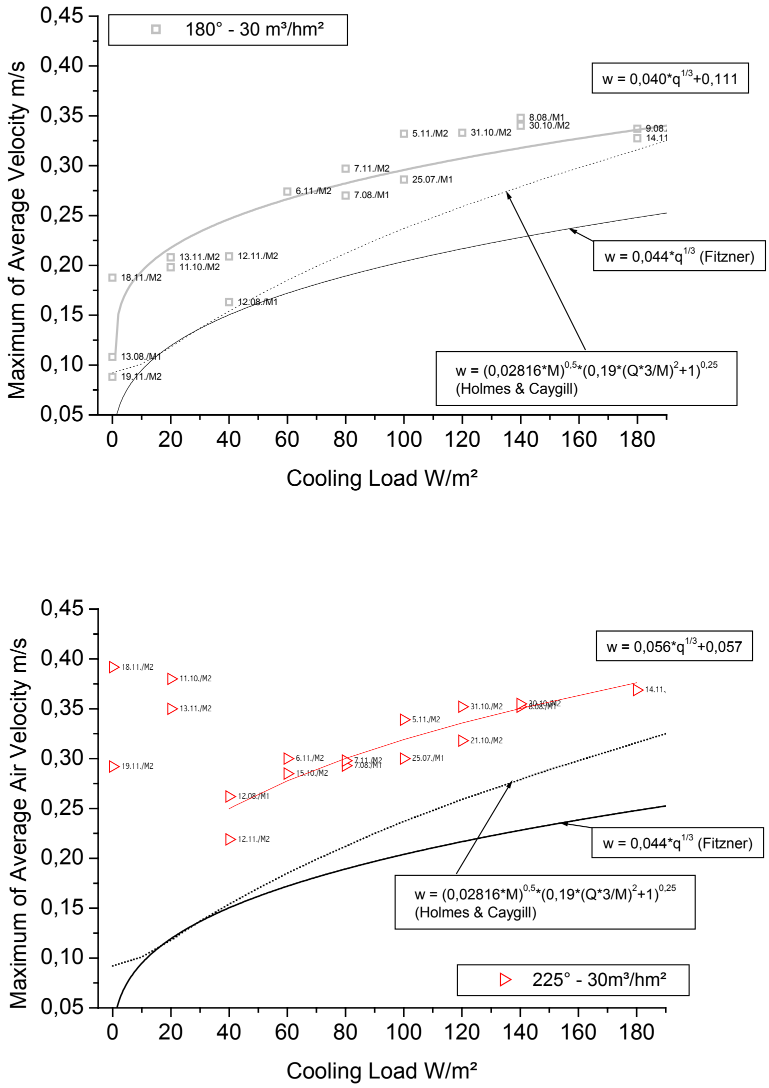

Figure 8 (a,b) Maximum of average air velocity at section 180°/225° in relation to the cooling load (airflow rate = 30 m³/h m²).

This leads to the assumption that there will be a definite cooling, which will be removed by a certain ventilation rate with a minimum of room air velocities, for every diffuser type and set up. Few authors have made suggestions for the relationship between cooling load and maximum room air velocity. Holmes and Caygill have described the velocity in relation to cooling load as follows:

$$\frac{w_r^2}{a^2 M_0} \cdot \left(\frac{L^2}{4} + H^2 \right) = 0.22 \cdot \sqrt{0.19 \cdot \left(\frac{Q \cdot H}{M_0} \right)^2 + 1}$$

where M_0 is the momentum, a is an empirical constant having a value of 1.2 for circular diffusers, Q is the room load (kW), H is the ceiling height (m), L is the room length and B is the room width.

Awbi (1991) has compared measurement and numerical data and has found the relation shown in Figure 9.

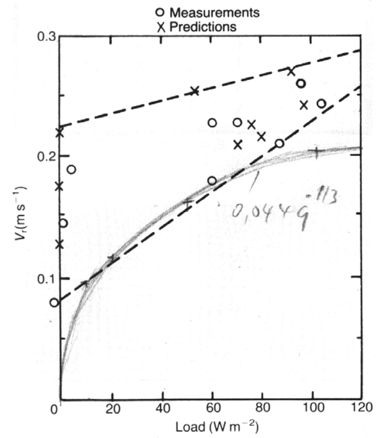

Figure 9 Effect of load on mean velocity in the occupied zone of a room (Awbi).

Figure 8(a) and (b) show that the measured values are not much above the curve suggested by Holmes and Caygill, but in relation to Awbi they are in the predicted range. Further experimental and numerical examinations will investigate the influence of the ventilation rate and the set up of the radial diffusers.

CONCLUSION AND IMPLICATIONS

Mixed airflow and uniformly distributed cooling load has been examined to find a relationship between air velocity and cooling load. The measurements lead to following results:

- The maximum of average room air velocity has enlarged with an increasing cooling load. The velocities follows the function: $w = a \cdot q^{1/3} + b$.
- The maximum has been found between two 'shortest distance' inlets, but in the geometric middle of four inlets.
- A cooling load with a minimum of room air velocities has been observed. This leads to the assumption that there will be a definite cooling load, which will be removed by a certain ventilation rate with a *minimum* of room air velocities, for every diffuser type and set-up. For nearly all adjusted cooling loads the heated mannequins have generated a strong and stable buoyancy flow. Cooling loads less than 40 W/m² were no longer able to deflect the supply air jets. The chilled supply air entered the occupied zone undisturbed.

REFERENCES

Awbi (1991). *Ventilation of Buildings*. London: Chapman & Hall.

Behne (1995). Temperatur-, Luftgeschwindigkeits- und Konzentrationsverteilungen in Raeumen mit Deckenkühlung. Dissertation, TU Berlin, Verl. für Wissenschaft u. Forsch. ISBN 3-930324-36-9.

Conrad (1972). Untersuchung ueber das Verhalten zweier gegeneinander stroemender Wandstrahlen. *Gesundheits-Ingenieur (gi) Heft* **10**, 303–309.

Fitzner (1996). Air velocities in spaces air-conditioned by cooling ceilings or mixed flow as a function of the cooling capacity. *Proceedings of ROOMVENT 1996*, Vol. 1, Yokohama.

Fitzner (1997). Anwendungsbereiche von Verdraengungs-, Quelluft- und Mischstroemung in Kombination mit Deckenkühlung. ki 3, pp. 110–113.

Holmes, M.J. and Caygill, C. (1973). Air movement in rooms with low supply air flow rates. BSRIA Rep. No. 83, BSRIA, Bracknell, UK.

Regenscheit (1981). *Isotherme Luftstrahlen: Klima und Kälte Ingenieur*, Extra 12, Karlsruhe: Verlag C.F. Mueller.
Schaefers (1982). Lueftung durch radiale Deckenstrahlen. Dissertation RWTH Aachen.
Waschke (1975). Ueber die Lueftung mittels isothermer turbulenter radialer Deckenstrahlen. Dissertation RWTH Aachen.