AIVC 11226

Hygienic Aspects in Air Conditioning Systems and Components

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1. Introduction

Two examples of investigations on hygienic questions of air conditioning systems for human comfort being performed in our institute shall be shown. They deal with two different subjects, one very well-known, the other beginning to be investigated:

- Air flow patterns for comfortable air velocities,
- Perceived indoor air quality assessed by trained panels.

The limits of accectable air velocities are well known and many national and international guidelines and standards are summarizing the relevant requirements. But here is not enough knowledge about the influencing parameters. In the field of indoor air quality nearly nothing is known and we are still discussing how to assess the indoor air quality. Standards are beginning to deal with this important subject.
This contribution will deal with the following two items:

1. Air velocities as a function of the cooling load, range of application of A/C systems.
2. Perceived Indoor Air Quality in office buidings, scale for trained panels.

2. Air Velocities as a Function of the Cooling Load, Range of Application of Air Flow Patterns

With respect to thermal comfort air velocities in ventilated spaces are limited to values depending on air temperature and degree of turbulence (1). The draft of the prestandard preNV 1752 of CEN/TC 156 WG 6 suggests acceptable velocities between 15 and 20 cm/s according to the categoires A to C. On the other hand velocities depend on specific cooling loads in a room. Figure 1 shows the velocity versus the relative cooling load in a room (height 3 m) with the cooled ceilings (closed line) and also the range of velocities found experimentally in mixed flows earlier (2). The velocity is the total cooling load related to the floor area, even distribution presumed. For comfort reasons air velocities should not exceed 0,15 to 0,18 m/s in the occupation zone with turbulent flow assuming a relative standard deviation (or degree of turbulence) of the air flow of about 50%. This limits applications of mixed flow or cooled ceilings to 100 W/m² according to Figure 1.

Moreover as a result of numerous air flow experiments with optimized diffusors (3) the range of application of mixed flows is limited to the triangular field shown in Figure 2 with the floor related cooling load as ordinate and the floor related air change rate as abcissa. The limit on the left is given by the proposed temperature difference between room and supply air of 12 K. The limit on the right is a result of tolerable air velocities according to Figure 1. The maximum cooling load is 100 W/m² combined with an air flow rate of almost 30 m³/(hmm2)or 8 l/(sm²). The above mentioned optimal distribution of diffusors is given when one radical outlet is used for one cube shaped room volume.

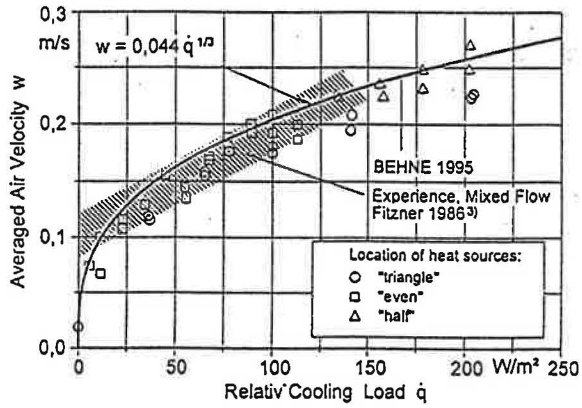

Figure 1: Air velocities in rooms with mixed flow or cooled ceilings as a function of the specific cooling load.

On the right hand side of Figure 2, a limitation line for displacement flow from the ceiling to the floor is given. Stable displacement flow exists on the right of this line and there is no limit of application up to specific cooling loads of 1 kW/m² and more. But this type of displacement flow requires high air exchange rates and is only applied in clean rooms.

In the case of flow from the floor to the ceiling the application range is different. As shown in Figure 3 there is a large area of source flow to which the capacity of a cooled ceiling up to 100 W/m² may be added. The only limitation of application is the specific air flow rate especially if the air is not supplied from the floor. At the limitation line on the right the type of air flow only changes from source flow to displacement flow but there is no upper limitation by draft problems.

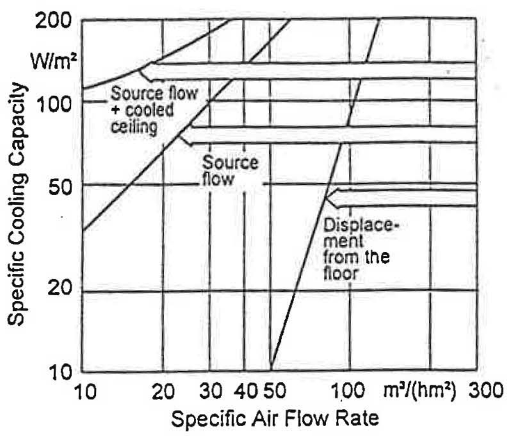

Figure 2: Field of application of mixed and displacement flow from the floor to the ceiling or cooled ceilings.

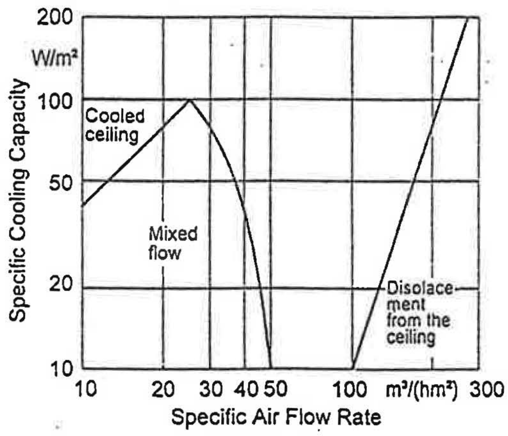

Figure 3: Range of application of flow from the floor to the ceiling

3. Perceived Indoor Air Quality

In the European Audit Project to Optimize Indoor Air Quality (4) and some other investigations we applied the method of assessing perceived indoor air quality (PAQ) by a trained panel. Some results and conclusions shall be mentioned here.

Figure 4 shows the assessed PAQ results averaged for the buildings of the various participating countries. The values for the outdoor air, supply air and room air are given. As expected the outdoor air has the best quality in all countries. An exception present the Danish results where the supply air is better than the room air. The differences in quality can clearly be assessed by the method. The values themselves are much higher than expected if compared with the data given in several standards (5,6). A paper of Wargocki (7) gives an explanation of this discrepancy. The low values of the standards are results assessed by untrained panels and the higher values of the European Audit are assessed by trained panels. Wargocki gives an s-shaped function by which both scales can be correllated very well. A first conclusion of our measurement results is to change values of the required indoor air quality categories of 1 1,4; 2,5 dp to 2, 4 and 6 dp for assesement by trained panels (8). This is in fairly good agreement with a Finnish proposal (9) suggesting 2, 4 and 5,5 dp. The differences between the categories are big enough to be able to differentiate between them by assessments of trained panels.

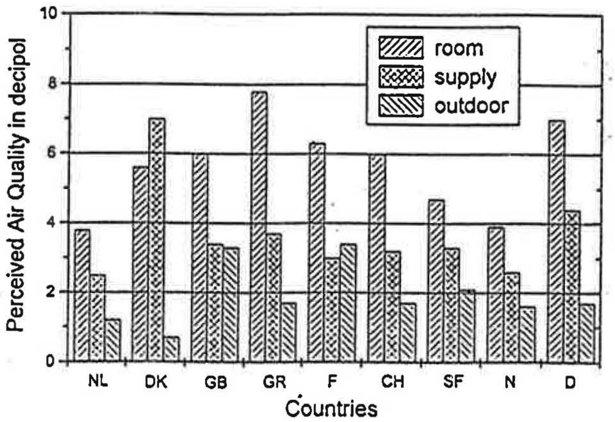

Figure 4: PAQ of outdoor, supply and indoor air averaged for all buildings of one country.

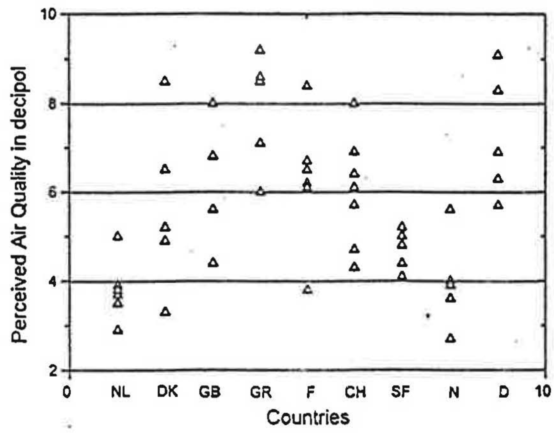

Figure 5: Assessed PAQ results averaged for each building of the various countries.

Figure 5 shows that the highest requirements in few buildings yet are realized. But as a first target one should try to realize PAQ category 6 and lateron 4 dp. The 4 dp building can be created with an outdoor air flow rate of 10 l/s/person if the outdoor air quality is 1 dp, and one additional olf of pollution added by the air conditioning system, the room and the person. This target will only be reached when low polluting materials are used for the air conditioning system and the room itself. But this is possible as shown by Figure 5.

Another investigation has been finished in our institute (10). It was focussing on the question whether the exchange of outdoor air in highly polluted environments has to be much higher than in low polluted ones according to equation (1) of the standards (5,6).

$$\dot{V} = 10 \frac{G}{(c_i - c_a)\varepsilon_v} \qquad (1)$$

$\dot{V}$ air flow rate
G source strength of pollutant
c_i perceived indoor air quality in dp
c_a perceived outdoor air quality in dp
ε_v ventilation efficiency

If one differentiates between different sources of pollution (index r room, p person, rlt air conditioning system) the expression changes to equation (2).

$$\dot{V} = 10 \left\{ \frac{G_r + G_p}{(c_i - c_a)\varepsilon_v} + \frac{G_{rlt}}{c_i - c_a} \right\} \qquad (2)$$

If one allows for example an indoor air quality of 4 dp, the outdoor air quality changes from 1 to 2 dp. In this case the air flow rate has to be increased by a factor of 1,5 according to equ. (1) or (2). This is not in agreement with experience. One explanation is that one cannot add decipols of various sources linearly if they cover a larger range. The scale of the perceived air quality is not linearly growing with the concentration of the pollutants. One finds an exponential function with different exponents for different substances. Figure 4 shows the perceived air quality versus the surface of a linoleum sample which is proportional to the concentration of the pollution. The perceived air quality is

$$c_m = 26{,}4\ (A_m)^{0{,}36} \qquad (3)$$

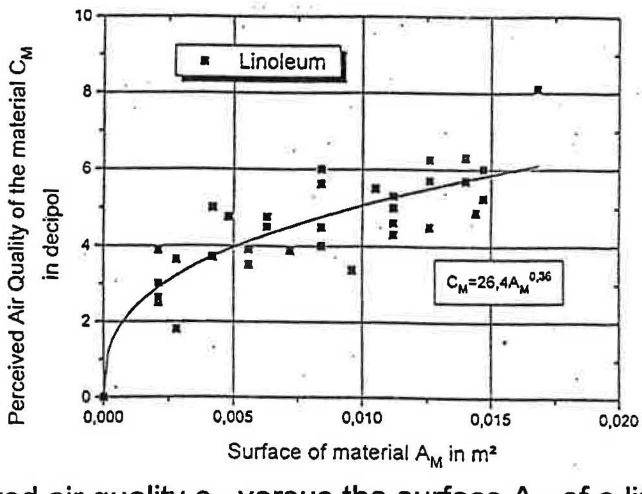

Figure 6: Perceived air quality c_m versus the surface A_m of a linoleum sample.

Measurements with several substances showed that an exponent of 0,5 is a rather good approach. This allows a rather simple addition of mixtures according to equ. (4) if the constant K_m is known.

$$c_m = K_m \, (c_s)^{0,5} \quad (4)$$

Combining equ. (4) and (2) results in the new equ. (5)

$$\dot{V} = 10 \left\{ \frac{G_r + G_p}{\sqrt{(c_i^2 - c_a^2)\varepsilon_v}} + \frac{G_{rlt}}{\sqrt{c_i^2 - c_a^2}} \right\} \quad (5)$$

The required outdoor air change rates calculated by equ. (1) and (5) are compared in Figure 7. The air flow rate for the three categories 2, 4 and 6 is given as a function of the outdoor air quality. The air flow rate is nearly independent of the outdoor air quality as long as both values are not too close together.

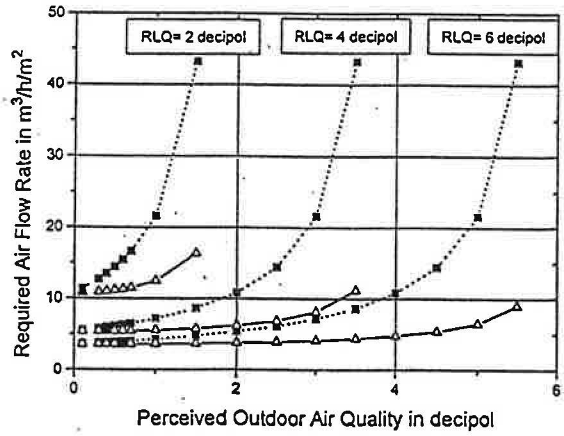

Figure 7: Required outdoor air exchange as a function of indoor and outdoor air quality for the linear and nonlinear case

The result shows, that the nonlinear addition is in better agreement with our experience. Nevertheless for simplicity reasons one should use the linear addition as long as low values have to be added.

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