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*Publication date:*  
2006

*Document Version*  
Publisher's PDF, also known as Version of record

[Link to publication from Aalborg University](#)

*Citation for published version (APA):*

Nielsen, P. V. (2006). *Comparison between Different Air Distribution Systems*. Paper presented at Slovene Klima Forum, Hidria Institute Klima, Godovic, Slovenia.

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# Comparison between Different Air Distribution Systems

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For presentation at

Slovene Klima Forum, Hidria Institute Klima, Godovic, Slovenija

# Comparison between Different Air Distribution Systems

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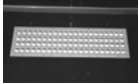



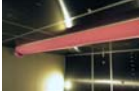
## Introduction

The aim of an air conditioning system is to remove excess heat in a room and replace room air with fresh air to obtain a high air quality. It is not sufficient to remove heat and contaminated air, it is also necessary to distribute and control the air movement in the room to create thermal comfort in the occupied zone.

Most air distribution systems are based on mixing ventilation with ceiling or wall-mounted diffusers or on displacement ventilation with wall-mounted low velocity diffusers. New principles for room air distribution were introduced during the last decades, as the textile terminals mounted in the ceiling and radial diffusers with swirling flow also mounted in the ceiling.

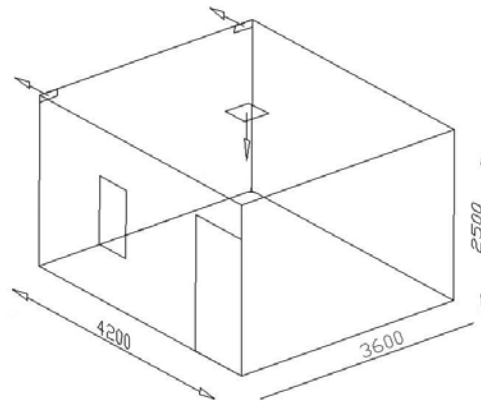
This paper addresses five air distribution systems in all, namely mixing ventilation from a wall-mounted terminal, mixing ventilation from a ceiling-mounted diffuser, mixing ventilation from a ceiling-mounted diffuser with a swirling flow, displacement ventilation from a wall-mounted low velocity diffuser and a low impulse system based on a textile terminal. All the systems are summarised in table 1.

*Table 1. Five different air distribution systems.*

System	Supply	Return	
Mixing ventilation	End wall-mounted	Return opening below supply terminal	
Mixing ventilation	Ceiling swirl diffuser	End wall-mounted below ceiling	
Mixing ventilation	Radial ceiling diffuser	End wall-mounted below ceiling	
Displacement ventilation	End wall-mounted	End wall-mounted below ceiling	
Vertical ventilation	Ceiling-mounted low impulse textile terminal	End wall-mounted at floor level	

## Test room

The systems are all tested in the same full-scale room, see figure 1. The dimensions of the room are in accordance with the requirements of the International Energy Agency Annex 20 work with length, width and height equal to 4.2 m, 3.6 m and 2.5 m.



*Figure 1. Sketch of a full-scale room with location of the diffusers and return openings in the experiments with a ceiling-mounted radial diffuser and a ceiling-mounted swirl diffuser.*

Figure 2 shows the furnishings and the heat load of the room (office layout). The heat load consists of two PCs, two desk lamps and two manikins producing a total heat load of 480 W. It is possible to make a direct comparison between the five different air distribution systems because they are tested in the same room.



*Figure 2. Furnishings and heat load in the full-scale room. The manikin designated A is located on the right side and the manikin designated B on the left side in the picture.*

The thermal load is constant in all experiments, and the flow rate has been raised from  $2.75 \text{ h}^{-1}$  up to  $10.45 \text{ h}^{-1}$  with a variation in the temperature difference  $\Delta T_o$  between 12 K and down to 3.5 K. The room temperature is close to  $22.9^\circ\text{C}$  in all the experiments.

### The design chart for the five air distribution systems

Three important parameters are considered in the design of room air distribution. The parameters are the air velocity (draught), the vertical temperature gradient and the asymmetry in the mean radiant temperature. The air velocity can either be the velocity  $u_{ocz}$  when it penetrates the upper boundaries of the occupied zone (1.8 m above the floor), the velocity  $u_{0l}$  close to the ankles of the manikins or  $u_{1l}$  at head height of the seated manikins.

The similarity principles show that any dimensionless velocity in the room can be given as a unique function of the Archimedes number if the flow in the room is a fully developed turbulent flow (high

Reynolds number flow), see Tähti and Goodfellow (2001). The similarity principle is used to obtain a formulation of the velocity level for  $u_{ocz}$ ,  $u_{01}$  and  $u_{11}$ .

$$u_{ocz}/u_o = func_1 (Ar) \quad (1)$$

$$u_{01}/u_o = func_2 (Ar) \quad (2)$$

$$u_{11}/u_o = func_3 (Ar) \quad (3)$$

The Archimedes number  $Ar$  is given by

$$Ar = \frac{\beta g d \Delta T_o}{u_o^2} \quad (4)$$

where  $\beta$ ,  $g$ ,  $d$  and  $\Delta T_o$  are the thermal expansion coefficient, gravitational acceleration, reference length and temperature difference between return and supply flow, respectively.  $u_o$  is the supply velocity given as a face velocity  $q_o/a_o$ .  $q_o$  is the flow rate to the room and  $a_o$  is a reference area of the diffuser.

Figures 3a and 3b show as an example the measured dimensionless velocities  $u_{ocz}/u_o$ ,  $u_{01}/u_o$  and  $u_{11}/u_o$  as a function of the Archimedes number for the radial diffuser and the diffuser with swirl. It is seen that the velocity at the upper surface of the occupied zone ( $u_{ocz}$ ) (outside the vertical plumes from the heat sources) in both cases is higher than the velocity at head ( $u_{11}$ ) and ankle height ( $u_{01}$ ) of the manikin. Although the velocity  $u_{ocz}$  at the height of 1.80 m is larger than  $u_{01}$  and  $u_{11}$  it will not often be a limiting design velocity because it only exists outside the thermal plume of the persons. The persons are thus protected from this velocity by their own thermal plumes in all the experiments given in figures 3a and 3b.

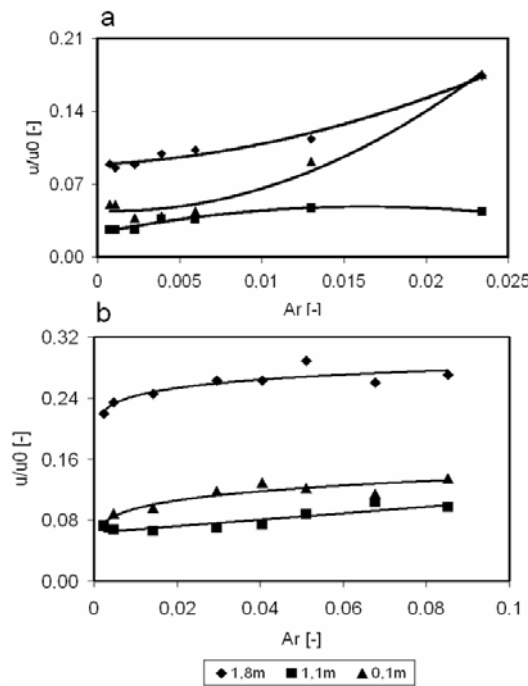


Figure 3. Flow from the radial diffuser is given in the upper figure (a) and flow from the diffuser with swirl is given in the lower figure (b). The figures show dimensionless velocities versus the Archimedes number for the room.  $u_{ocz}$  is the downward velocity between the two manikins (and

other heat loads) at the height of 1.8 m.  $u_{0I}$  and  $u_{0II}$  are velocities at ankle level and head level, respectively.

The experiments show that the velocity at ankle height ( $u_{0I}$ ) is higher than the velocity at head height ( $u_{0II}$ ). The velocity at ankle height is therefore the limiting parameter for the air distribution system.

One of the aims in this work has been to find possible flow rates into the room, and temperature differences between the supply and return temperatures where an acceptable comfort level with low draught and low temperature gradients is still obtained. To make the decisions more clear a design chart has been developed and used, see Nielsen (1980), Jacobsen et al. (2002), Nielsen et al. (2003), Nielsen et al. (2005), and Nielsen et al. (2006). By use of the design chart it becomes possible to compare different systems to find a system usable for the demands in a certain situation.

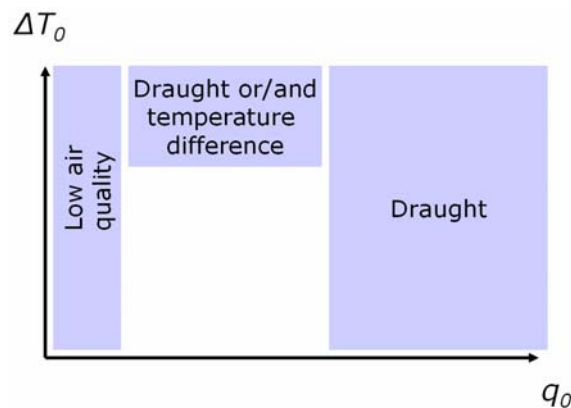


Figure 4. Design chart showing the restrictions on the flow rate  $q_o$  and on the temperature difference  $\Delta T_o$  between return and supply owing to draught, temperature gradient and air quality.

Figure 4 describes the idea behind a design chart for air distribution in rooms. The chart is based on the minimum and maximum allowable flow rate  $q_o$  to the room, and also on the maximum temperature difference between return and supply. The figure indicates that it is necessary to have a minimum flow rate of fresh air into the room to obtain a given air quality. It is also expected that there is a limit for the maximum flow rate to avoid draught in the room as well as a maximum flow rate in connection with the design of the duct system. It is typical of air distribution systems based on the supply of momentum flow that this flow generates draught when the flow rate is above a certain level. The systems, on the other hand, also generate a mixing in the occupied zone which is important for the creation of uniform conditions when the heat load is high. The temperature difference  $\Delta T_o$  between return and supply is also restricted. A too high temperature difference may either cause draught in the occupied zone or create a temperature gradient in the room which is too large. There is also a restriction on the temperature difference  $\Delta T_o$  in connection with design of the cooling system.

Figure 4 indicates an area for  $q_o$  and  $\Delta T_o$  which supplies sufficiently fresh air and ensures a draught free air movement in the occupied zone plus a restricted vertical temperature gradient. This area is the design area for a given air distribution system. The limits for this area in the design chart are often defined as a maximum air velocity of 0.15 m/s in the occupied zone, a maximum vertical temperature gradient of 2.5 K/m and a minimum flow rate of 10 l/s per person, but other reference values can of course also be selected.

## Comparison between five different air distribution systems

It is possible to make a direct comparison between the different air distribution systems because the experiments are made with the same office equipment; two computers, two desk lamps and two manikins, see figure 5.

The air distribution systems are:

- Mixing ventilation with a wall-mounted diffuser
- Mixing ventilation with a ceiling-mounted radial diffuser
- Mixing ventilation with a ceiling-mounted swirl diffuser
- Vertical ventilation with a ceiling-mounted textile terminal
- Displacement ventilation with a wall-mounted low velocity diffuser

The limiting design parameter for mixing ventilation generated by a wall-mounted diffuser is the velocity  $u_{ocz}$  of the jet when it penetrates the upper boundaries of the occupied zone in the case of non-isothermal flow. The maximum penetration velocity through the occupied zone  $u_{ocz}$  is equal to 0.4 m/s because this corresponds to a level where  $u_{01}$  and  $u_{11}$  are up to 0.15 m/s. In this case the vertical temperature gradient is not considered to be important.

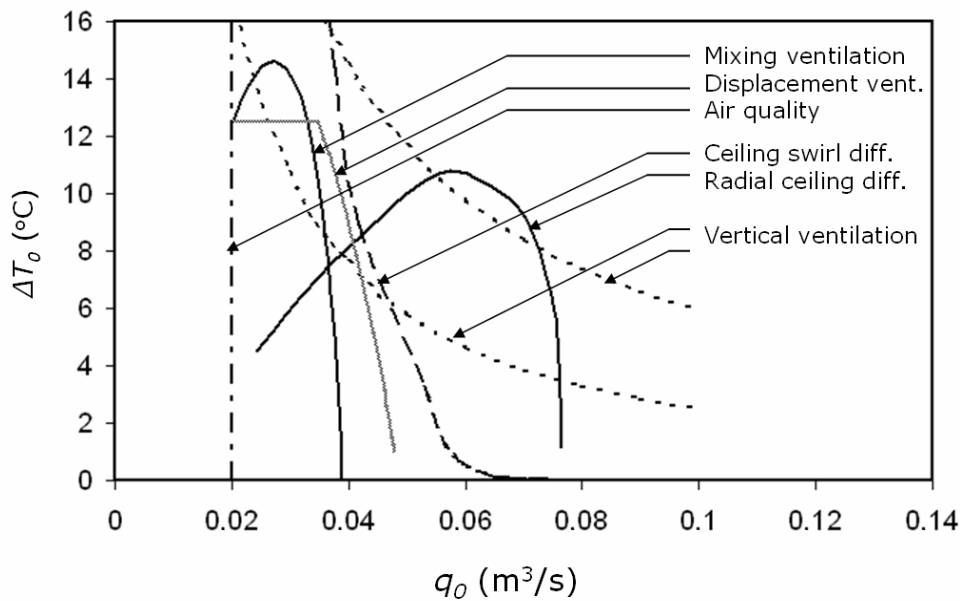


Figure 5. Design chart for the five different air distribution systems. Air quality requires that  $q_o$  is larger than  $0.02 \text{ m}^3/\text{s}$ .

The experiments with the radial diffuser show that  $u_{01}$  equal to 0.15 m/s corresponds to both a  $u_{ocz}$  value close to 0.3 m/s and a maximum  $\Delta T_o$  value of 10 K. It can thus be concluded that  $u_{ocz}$  equal to 0.3 m/s combined with  $\Delta T_o = 10 \text{ K}$  should be the maximum entering velocity in the occupied zone and the maximum temperature difference for a design with a radial diffuser.

The experiments with the swirl diffuser show that  $u_{01}$  equal to 0.15 m/s corresponds to a  $u_{ocz}$  value in the range of 0.3 – 0.4 m/s. It can thus be concluded that  $u_{ocz}$  equal to 0.35 m/s should be the maximum entering velocity in the occupied zone for a design with a ceiling-mounted diffuser with swirl.

The design area for the diffuser with swirling flow is located at lower air flow rates than the design area for the radial diffuser. A swirl diffuser with large supply openings could, if necessary, move the design area towards large air flow rates because an increase in the supply area decreases the corresponding velocity level in the occupied zone. There is no upper limits for  $\Delta T_o$  for this diffuser. The high mixing rate in the initial flow removes temperature differences in the flow and eliminates a large part of the gravity forces.

The limiting design parameter for vertical ventilation from a ceiling-mounted textile terminal is the velocity  $u_{ocz}$  of the downward directed jet established when the flow is non-isothermal. Draught free conditions around the manikins correspond to  $u_{ocz} \leq 0.2$  m/s. The vertical temperature gradient is not important for this type of flow because the thermal plumes from heat sources and the downward directed displacement flow from the terminal generate a large mixing of the room air.

Displacement ventilation has a high air velocity in the stratified flow at the floor. Reduced velocity in the occupied zone is therefore obtained by restricting the velocity in the stratified flow where it enters the vertical boundary of the occupied zone in front of the diffuser. This is expressed by the length of the adjacent zone  $l_n$  which is the distance from the diffuser to a given velocity level in the stratified flow, see Nielsen (2000) and Skistad et al. (2002). The vertical temperature gradient is also important in displacement ventilation and should be kept below a certain level.

The length of the adjacent zone  $l_n$  is given to 1 m with a reference velocity of 0.2 m/s in the case of displacement ventilation, and the temperature difference  $\Delta T_o$  is in principle restricted to 12.5°C in the room giving a gradient of 3 K/m. The curves in figure 5 show the corresponding area for fulfilment of thermal comfort in the  $\Delta T_o, q_o$  chart. It should be mentioned that there is a possibility to reduce the minimum air flow rate for the displacement ventilation system because the stratification of supply air improves the air in the inhalation zone when the contaminant source in the room is also a heat source. This effect has not been considered in figure 5.

Figure 5 indicates that the room to some extent has the same level of comfort (in terms of maximum velocity and temperature gradient) in the case of mixing ventilation with a wall-mounted diffuser or displacement ventilation with a wall-mounted low velocity diffuser. The figure shows also that the vertical ventilation (low impulse) systems are superior to both mixing ventilation based on a wall-mounted diffuser and to displacement ventilation. In connection with vertical ventilation it should be noticed that the layout with two work places in a small room may result in a  $q_o - \Delta T_o$  curve which is favourable compared with the situation in a large room with vertical ventilation. (The lower curve in figure 5 for vertical ventilation is obtained for one manikin in the room, corresponding to two persons in a larger room).

Mixing ventilation generated by ceiling-mounted diffusers is able to handle a higher heat load than any of the other systems.

It should also be emphasized that the design chart, figure 5, to some extent is dependent on room size, room layout and layout and design of the terminal units (number, location, etc.).

It is possible to use the method behind the design chart in larger rooms than the test room in this paper as well as for other types of diffusers. It is necessary to modify the flow elements (wall jet, penetration length, stratified flow, etc.) to the actual room dimension, which results in a specific design chart for the room and the diffuser.



## Local discomfort

The design models discussed in the previous section can only give the limits for the operation of the air distribution system. It is necessary to consider thermal comfort for all flow rates if the system has to be optimized.

The thermal environment often shows temperature gradients, velocity gradients, different turbulence levels and an asymmetric radiant temperature distribution. The local discomfort, which is the result of this environment, is found from measurements of the local values of air temperature, air velocity, turbulence level, and from measurements of surface temperatures or asymmetric radiant temperatures, see Fanger and Langkilde (1975), Olesen et al. (1979), Fanger et al. (1989), Toftum et al. (1997).

The number of dissatisfied because of draught, the draught rating ( $DR$ ), is used as a measure of local discomfort. The draught rating is defined as:

$$DR = e^{d(t_a - 24)}(34 - t_a)(u - 0.05)^{0.62}(0.37 u Tu + 3.14) \quad (5)$$

where  $t_a$ ,  $u$  and  $Tu$  are ambient air temperature, air velocity and turbulence intensity, respectively, see Toftum et al. (1997).  $d$  is a factor which is dependent on direction.

Equation (5) is especially based on a situation where the velocity is exposed to the back of the neck of the persons. In this paper the highest velocity is often found at ankle level, and the sensitivity at this position is slightly lower than that of the back of the neck. The draught rating given from equation (5) is therefore representing a conservative estimate.

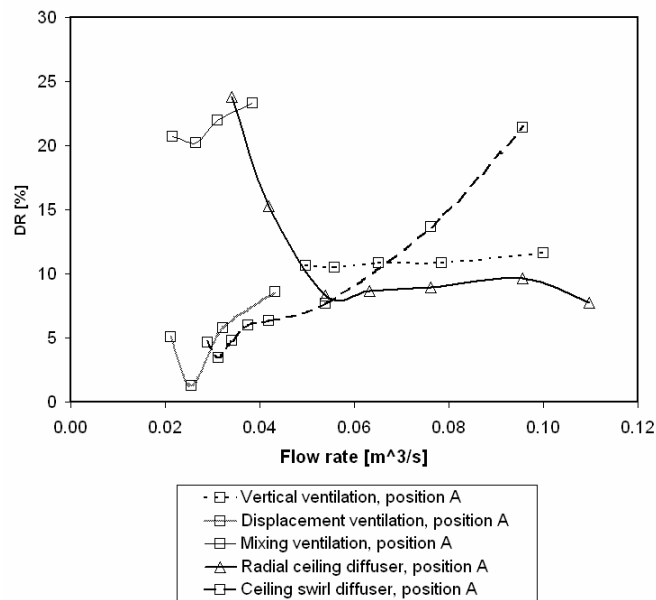


Figure 6a. Draught rating ( $DR$ ) versus flow rate  $q_0$  measured at position A for five different air distribution systems.

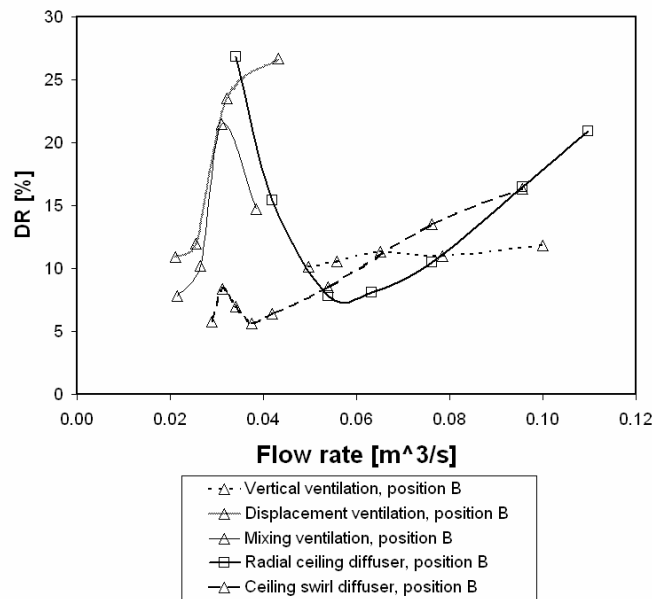


Figure 6b. Draught rating ( $DR$ ) versus flow rate  $q_o$  measured at position B for five different air distribution systems.

Figures 6a and 6b show the draught rating ( $DR$ ) for the persons at positions A and B. The draught rating is dependent on the location of the work place in mixing and displacement ventilation which, however, is not the case for the vertical ventilation system. Position A is the work place located farthest from the wall with the inlet openings, and position B is the work place closest to this wall. Generally, the best results are obtained for displacement ventilation with a work place far away from the diffuser, while mixing ventilation with a wall-mounted diffuser has a high draught rating. The vertical ventilation system gains from the high level of buoyant flow in the room, which in this case protects the manikins from draught. Mixing ventilation with a ceiling-mounted diffuser that generates flow with swirl has a draught rating close to the best result obtained by displacement ventilation at position A. The draught rating for mixing ventilation with a ceiling-mounted radial diffuser also shows a low value at both positions under the optimal conditions around  $0.055 \text{ m}^3/\text{s}$ . All the systems with ceiling-mounted diffusers (radial diffuser, swirl diffuser, textile terminal) are superior to systems with end wall-mounted diffusers.

Measurements show that the temperature gradient and asymmetric radiation are only important for displacement ventilation.

## Conclusions

Air distribution systems based on mixing ventilation and ceiling-mounted diffusers are able to generate comfortable conditions up to a thermal load which is slightly superior to systems with mixing ventilation from wall-mounted diffusers, displacement ventilation with a wall-mounted low velocity diffuser or vertical ventilation systems with ceiling-mounted textile terminals.

It is characteristic that a mixing ventilation system with a ceiling-mounted swirl diffuser does not give a restriction on the temperature difference between return and supply within the level of traditional design practice. This effect is seen in figures 5 and 7 as a lack of limitation of  $\Delta T_o$ . A similar effect can be seen for the vertical ventilation system which is without restrictions on both temperature difference  $\Delta T_o$  and flow rate  $q_o$ .

The draught rating is low for a mixing ventilation system with an optimal design and restricted heat load. A mixing ventilation system with an end wall diffuser and a displacement ventilation system with an end wall-mounted low velocity diffuser both have a higher draught rating under the design conditions.

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