Design methods for displacement ventilation: Critical review.

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Abstract
Displacement ventilation has been successfully applied for more than twenty years in Europe and it represents an opportunity for China.
Displacement ventilation (DV) is based on vertical stratification of temperature in the occupied zone of rooms. Buoyancy flows generated by heat sources govern the air distribution in rooms with DV. If properly designed DV has potential to provide higher ventilation effectiveness, i.e. better inhaled air quality for occupants and lower energy consumption than mixing ventilation.
In order to avoid thermal discomfort due to draughts and vertical temperature difference, the temperature distribution should be carefully predicted in the design stage. The REHVA and ASHRAE methods to design displacement ventilation systems for thermal comfort are introduced in this paper and critically reviewed.

Introduction

-Historical introduction-

Displacement ventilation for more than thirty years has been used in Europe in industrial premises with high thermal load present. In Scandinavian countries during the last twenty years, displacement ventilation has become popular in non-industrial premises too, i.e. office buildings, schools, theatres, etc.

-Advantages in regard to Energy saving and Indoor Air Quality-

In room ventilated by displacement a vertical temperature gradient is present. Temperature at floor level is lower than the air close to the ceiling. It is therefore possible to remove the exhaust air from the room with temperatures which are several degrees higher than the temperature in the occupied zone. Displacement principle permits an efficient use of energy because the supply temperature can be higher than the supply temperature in case of mixing ventilation (Nielsen, 1993). It can also save energy (longer periods with free cooling and less cooling needed for a given temperature in the occupied space (RHEVA, 2002).

It is well known that the indoor air quality in buildings affects occupants' health, comfort and performance. DV can provide a thermally comfortable environment and when contaminant sources are connected with heat sources it has the potential to achieve cleaner air in the occupied zone than mixing ventilation.

China in the recent decades has be protagonist of a high speed development. A huge quantity of buildings has been built and thus energy demand needed for their operation increases. 26% of the total energy consumption (primary energy) in China is used for commercial and residential building. Most of this energy is used for heating, cooling and air conditioning (Jiang Yi, 2005). The shortage of energy sources and problems related to the outdoor environment push the efforts of building designers towards a more rational utilization of energy.

Thus the displacement ventilation with its potential for providing high indoor air quality in energy efficient way may be a successful technology in China. However displacement ventilation may be advantageous only when carefully designed.

-Vertical temperature gradient and draught at the floor level-

The thermal stratification can generate a temperature difference between the head and the feet that may generate local discomfort, moreover, displacement ventilation system supply cold air directly in the occupied zone, it can easily generate draughts at the floor level (Nielsen, 1996).

In order to avoid local discomfort the vertical temperature distribution has to be watchfully predicted and suitable air diffusers should be chosen.

-Purpose of the paper-

In order to achieve the advantages related to displacement ventilation and to avoid thermal discomfort for occupants a deep understanding of the design methods is needed. This paper has the aim to introduce and critically discuss the methods used in Europe and US for design of displacement ventilation. In this paper will be not explained the design methods to fulfil the indoor air quality requirements.

Displacement ventilation: principle and application

-Principle-

Displacement ventilation is based on vertical stratification of temperature and pollution in the occupied zone of rooms.

When DV is used, the air flow pattern is mainly governed by the convection flows from different heat sources. The cool air is supplied with low velocity and low turbulence near the floor level and it is exhausted at ceiling level in the room as it is shown in Figure 1. The air transport within the room is done by the rising convection flows generated by the heat sources, which take air from the lower parts of the room into the upper parts (Mundt, 1996). The momentum flux of the flow from the diffusers is low and without any practical importance for the general flow in the room.

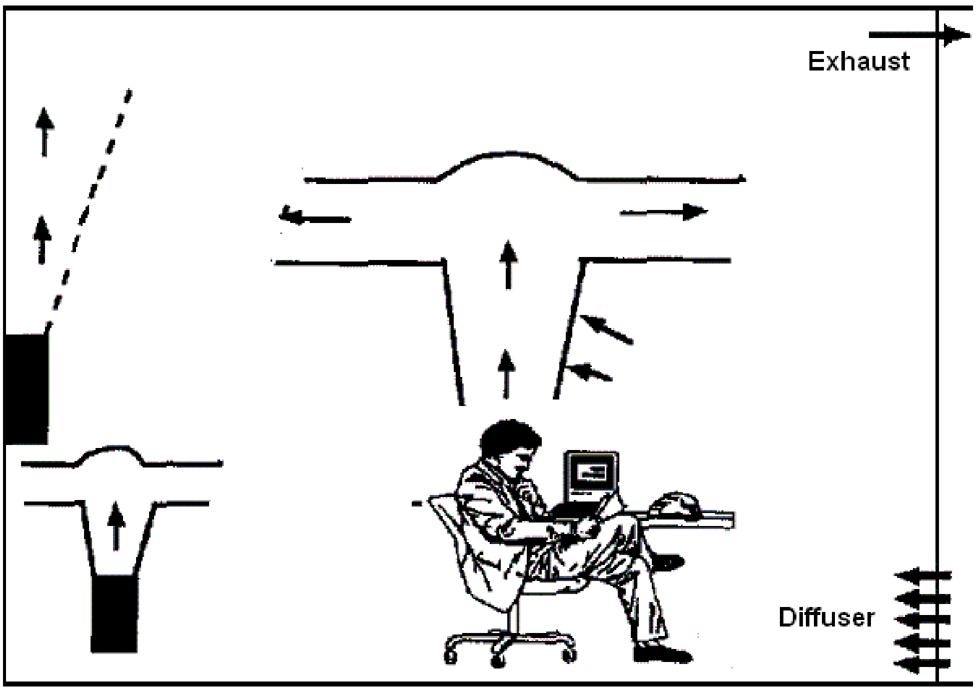

Figure 1 Displacement ventilation principle (Mundt, 1996)

In order to perform properly the supply air temperature must be several degrees (3-4 °C) lower than the one in the room.
The heat sources generate plumes and the amount of air transported in the convection flows depends on:

1. Geometry of the heat sources
2. Temperature of the heat sources
3. Temperature gradient

If the sources are both heat and contaminant sources a displacement ventilation system can have high ventilation effectiveness (Mundt, 1996). Otherwise it performance in regard to air quality in the occupied zone is similar to mixing ventilation, e.g. clean air supplied over floor carpet will bring the pollution upward and mix it with the room air.

-Where it can be applied-
Displacement ventilation can be applied in auditoriums, restaurants, offices, classrooms, meeting rooms, tall rooms as theatres, supermarkets, cinemas, theatre etc. Displacement ventilation is usually preferable when contaminants are warmer and/or lighter than the surrounding air or when great airflow rates have to be supplied in small rooms or in room higher than 3 metres (Skistad, 1994). Displacement ventilation is especially effective in the premises with high ceilings.

According to Wyon and Sandberg (1990) DV may preserve a comfortable environment if the cooling load does not overcome $40W/m^2$. However Chen and Glicksman (1999) suggested much higher value, up to $120W/m^2$. Higher cooling load can be reached if DV is applied together with chilled ceiling. In this case different design approach, not discussed in this paper is needed. The displacement ventilation system is ideal for cooling. The system should not be used for heating because the buoyancy and low air supply velocity will drive the hot supply air to the ceiling level. If heating is necessary in winter, an independent heating system (radiant panels, convectors, radiators or fan coil units at floor level, etc.) is needed.

Vertical temperature stratification
A temperature gradient is present in rooms ventilated by displacement. Figure 2 shows a simplified scheme of the vertical temperature profile where T_s, T_{af}, T_e are the air temperature at the inlet, at the floor level and at exhaust.

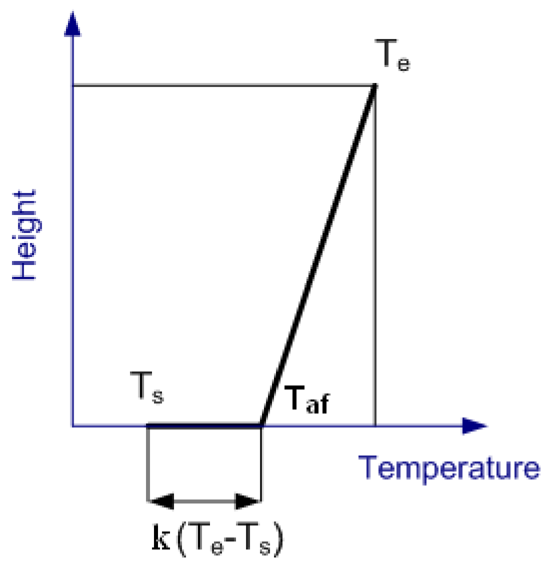

Figure 2 Simplified vertical temperature profile in rooms ventilated by displacement

The increase of the air temperature when introduced in the room is due to the entrainment of ambient air and the heat transfer by convection with the floor. According to Nielsen (1993) and Skistad (1994) it may be assumed that the air temperature vary almost linearly with room height. According to Chen et al. (1999) the temperature does not vary linearly from the floor to the exhaust in most cases. The air temperature is practically homogeneous in horizontal direction apart from the region close to diffusers. Dimensionless temperature of the air close to the floor (κ) is represented by equation (1). According to Nielsen (1996) and Brohus and Ryberg (1999), κ varies between 0.3 and 0.65. According to Chen et al. (1999) κ varies between 0.2 and 0.7.

$$\kappa = \frac{T_{af} - T_s}{T_e - T_s} \tag{1}$$

-From which parameter it depends on- The temperature distribution is influenced by the air flow rate, the type and vertical location of the heat source, the wall characteristics, space height and air diffusers.

-Air flow rate- Sandberg (1985), Chen et al. (1988) and Mundt (1990) showed that the dimensionless temperature decreases when the airflow increases. Mundt (1990) developed the equation (2) to predict κ as a function of the ventilation rate over the floor area. Ventilation rate and cooling load are correlated.

$$\kappa = \frac{T_{af} - T_s}{T_e - T_s} = \frac{1}{\frac{q_v \cdot \rho \cdot c_p}{A} \cdot (\frac{1}{\alpha_r} + \frac{1}{\alpha_{cf}}) + 1} \tag{2}$$

Where

c_p=	Specific heat of the air = 1004 J/kg K
q_v=	Air volume flow [L/s]
α_{cf}=	Convective heat transfer coefficient [W/m^2 K]. Usually equal to 4 W/m^2 K
α_r=	Radiant heat transfer coefficient [W/m^2 K]. Usually equal to 5 W/m^2 K
κ=	Dimensionless temperature of the air near the floor
ρ=	Air density = 1.20 kg/m^3 (θ = 21 °C)

The basic idea of the equation (2) is that the different temperature between the ceiling and the floor will generate a radiant heat transfer from the ceiling to the floor, i.e. the temperature of the floor will rise and will due to convection warm up the supply air. The model is an energy balance.

-Type of heat source- Nielsen (1996) pointed out that the temperature gradient is slightly dependent on the air flow and strongly dependent on the type of heat source (the surface temperature in particular). It is assumed that the primary flow is fully developed turbulent flow so that the dimensionless temperature can be given as unique function of the Archimedes number. He gave a design chart which gives the dimensionless temperature at the floor level (κ) depending from the Archimedes number for four different types of heat sources (distributed source, sedentary persons, ceiling light, point source) placed in the occupied zone.

-Vertical location of heat sources- When the heat sources are in the lower part of the room the temperature gradient is larger in the lower part and the temperature is more constant in the higher part. On the other hand, when the heat sources are located mostly in the upper part, the temperature gradient is smaller in the lower part and increases in the upper part as showed by Mundt (1996). In rooms where the heat sources are located at high level, displacement ventilation is efficient for keeping the occupied spaces cool (REHVA, 2002).

-Wall characteristics- The wall area is large, and so, even a small temperature difference between the walls and the room air may generate a considerable downward or upward flow. According to Jarmyr (1992) the vertical profile is affected by the change of the walls temperature due to the variation of the external temperature.

-Space height- Displacement ventilation is more suitable for high spaces. In the higher part of the tall spaces the temperature rises slightly. In concert hall with supply opening under the chairs the temperature rises rapidly from the supply air temperature at the floor level to the elevation where occupant are situated. Above the people only a slight temperature gradient is present (Skistad 1994). According to Skistad (1994), in industrial premises, where the room height is considerable, one third of the total temperature difference is even out at the floor level.

-Air diffusers- Nielsen et al (1988) showed that, in order to reduce the temperature gradient in the occupied zone, increasing the entrainment of room air is necessary, it could be done by higher efficiency air terminal supplier. The performance of the air supplier is fundamental to reduce the risk of draughts at the floor level.

Design methods

In this paragraph two design methods developed to guarantee thermal comfort for occupants are described. The first is the method recommended by REHVA (Federation of European Heating and Air-Conditioning Associations) and the second is the method developed by ASHRAE (American Society of Heating, Refrigeration and Air-Conditioning Engineers).

A ventilation system has to guarantee simultaneously thermal comfort and indoor air quality.
These requirements should be fulfilled in the occupied zone during all the different operating conditions in the most economical way without generating noise.
The important parameters in the design of an environment ventilated by displacement are:

- Supply air flow rate and temperature
- Air temperature at the floor level
- Air velocity at the floor level
- Vertical temperature gradient
- Pollution gradient

The physical parameters obtained during the design are the supply air flow rate and temperature. The calculation of the supply air flow rate and temperature for thermal comfort is based on the prediction of the vertical temperature distribution.

-REHVA method-

This method was developed in the Scandinavian countries (Skistad, 1994. REHVA, 2002), it is based on an experimental approach and it assumes a linear temperature distribution between the floor and the ceiling.
The method defines κ as equal to 0.5. Half of the temperature difference between the supply and the exhaust is evened out at the floor level and the rest of the difference is developed linearly in height.
κ is independent of cooling load, ventilation rate, type and location of the heat sources.
The design procedure is:

1) Select the target (aim) temperature at 1.1m above the floor ($T_{d,1.1}$), and the design (aimed) vertical temperature gradient in the occupied zone, s (°C/m), which is usually equal to 2°C/m
2) Calculate the heat surplus to be removed by the ventilating air
3) With κ =0.5 the total temperature difference is determined by the following equation (3):

$$\Delta T = T_e - T_s = 2 \cdot h \cdot s \qquad (3)$$

where h is the height of the room.

4) Calculate the air flow rate (q) for removal of the surplus heat (Φ_{tot}) using the energy balance equation (4):

$$q = \frac{\Phi_{tot}}{\rho \cdot c_p \cdot (T_e - T_s)} \tag{4}$$

5) Calculate the supply air temperature with the following formula:

$$T_s = T_{d,1.1} - s \cdot (1.1 + h) \tag{5}$$

6) Now when the supply air flow rate and temperature are known, re-evaluate the temperature at floor level (T_{af}) using the equation (2). Re-calculate the temperature distribution in the room
7) Select size, number, type and location of the diffusers according to the manufactures' data in order to avoid draughts.

-ASHRAE method-

The method was developed in US by (Chen and Glicksman, 1999) as result of the ASHRAE research project- RP-949. The method (Chen and Glicksman, 1999) was built on a validated CFD program. It was developed for typical US building. This method assumes linear temperature distribution in the range 0.1-1.1 m. It uses equation (2) to predict κ. The method is based on the idea that only part of the heat generated in the room affects the temperature gradient in the range 0.1-1.1 m. The design procedure is as follows:

1) Judge the applicability of displacement ventilation and select the target (aim) temperature at 1.1m above the floor ($T_{d,1.1}$)
2) Calculate summer design cooling load. Itemize the cooling load into:
   - the occupants, desk lamps, and equipment, Q_{oe} (W)
   - the overhead lighting, Q_l (W)
   - the heat conduction through the room envelope and transmitted solar radiation, Q_{ex} (W)
3) Determine the required flow rate of the supply air for summer cooling according the following equation:

$$q = \frac{Q_{oe} \cdot a_{oe} + Q_l \cdot a_l + Q_{ex} \cdot a_{ex}}{\rho \cdot c_p \cdot (T_{1.1m} - T_{0.1m})} \tag{6}$$

Where

q= Air volume flow [m^3/s]

c_p= Specific heat of the air = 1004 J/kg K

ρ= Air density = 1.20 kg/m^3 (θ = 21 °C)

$T_{1.1m}$-$T_{0.1m}$ = Design air temperature difference between head and feet of a sedentary occupant. Usually equal to 2°C.

a_{oe}, a_l, a_{ex}, = 0.295, 0.132, and 0.185 respectively are weighting factors. They define the fractions of the cooling loads entering the space between the head and the feet of a sedentary occupant

4) Calculate κ using equation (2). The radiant and convective heat transfer coefficient can be found from ASHRAE Handbook – Fundamentals. As a rough estimate, α=5 W/(m^2 K)
5) Calculate the supply air temperature with the following formula:

$$T_s = T_{d,1.1} - (T_{1.1m} - T_{0.1m}) - \kappa \cdot \frac{Q_{oe} + Q_l + Q_{ex}}{\rho \cdot c_p \cdot q} \tag{7}$$

8) Select size, number, type and location of the diffusers according to the manufactures' data in order to avoid draughts
9) Check the winter heating situation. For the building perimeter where heating may be necessary in winter, a separate heating system is necessary to offset the heating load from exterior walls and windows, when the displacement ventilation is used.

The weighting factors are defined as a result of a calibrated CFD that simulated several commons US buildings. The weighting factors bear the uncertainty of being the average of the results of many different conditions.

More detailed methods allowing for computational treatment of the radiation exchange were presented by Li et al. (1992), Mundt (1996) and Livtchak (2001). They are interactive and too complex to be used manually. They need to be coded into software.
CFD (computational fluid dynamics) is also possible. It is used for large spaces with complex environment. In such case it is necessary to pay attention in the description of the boundary conditions in heat and contaminants, radiant exchange, and supply air units.

Comparison of the two methods

The temperature distribution depends on the air flow rate, type and location of the heat source, the wall characteristics, the space height and the air diffusers. The two methods presented in this paper do not include the effect of all these parameters.
-Linearity of the vertical temperature distribution- According to Nielsen (1993), Skistad (1994) and Mundt (1996), the air temperature vary almost linearly in height. This is an assumption of the REHVA method. According to Chen et al. (1999) this supposition is not correct in most cases. The ASHRAE method assumes linear temperature only in the occupied zone, the method do not predict how the temperature varies in the upper part of the room. This assumption permit to take into account only the part of the heat load that affects the thermal sensation of the occupants.
-Air flow rate and cooling load- Both methods use the model of Mundt (equation (2)) to calculated the dimensionless temperature at the floor level. Equation (2) has two unknown parameters , κ and q. The REHVA method applies the equation only after having determined the supply air flow rate (q) and temperature (T_s). Thus the input parameters (q and T_s) are calculated asuming $\kappa = 0.5$. It is a meaningful limitation because Mundt's model is not used to calculate more accurate input parameters but just to predict how could be the temperature profile for the already known q and T_s. On the contrary, the ASHRAE method applies Mundt's equation (2) to calculate T_s because the air flow rate is already known. The air flow rate is obtained from the prediction of the temperature profile only in the occupied zone.
-Type and location of heat sources- The REHVA method do not take into consideration the type and the location of the heat source, and as pointed out by Nielsen (1996), it may lead to a significant error on the prediction of dimensionless temperature at the floor level. The ASHRAE method do not consider separately location and type of each heat source.The ASHRAE method itemizes the heat sources in three categories according to their type and location and defines their contribution to the heat load calculation by weighting factors. However the accuracy of determination of the factors needs to be verified.
-Wall characteristics- The REHVA method does not take into consideration the possibility that the walls are not adiabatic. The ASHRAE method does not model a non-adiabatic walls, but using the weighting factors, it does consider only the part of the heat, transferred through the walls, that affects the temperature in the occupied zone.
-Space height and air diffusers- None of the two methods takes into account space height and type of diffusers. Both methods underline that the choice of size, number, type and location of the air diffusers is a key factor in order to avoid draughts.

Conclusion

Two are the main methods for the design of displacement ventilation in non-industrial premises. The REHVA method, based on a experimental approach, is mainly used in Europe, the ASHRAE method, based on a validated CFD predictions, is primarily applied in US.
The ASHRAE method takes into account more parameters that can affect the vertical temperature distribution than the REHVA method.
The ASHRAE method was developed for typical US buildings and so it may not be directly applied in China.
Those two method are not suitable for the design of a tall/big space or if cooling/heating floor/ceiling systems are present. CFD simulation or more complex method should be used.
The development of a design method adapted to the specific features of China climatic conditions and building techniques is needed.

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References

Brohus, H. and Ryberg, H. 1999. The Kappa model- a simple model for the approximate determination of vertical temperature distribution in rooms. Department of building technology and structural engineering, Aalborg University.

Chen, Q. 1988, Indoor Airflow, Air quality and Energy Consumption in Building. Ph.D. Thesis of Delft University of Technology. Delft.

Chen, Q., Glicksman, L., 1999. Performance Evaluation and Development of Design Guidelines for Displacement Ventilation. ASHRAE Research project-RP-949.

Jamrym, R.. 1982. Displacerande inblasning – Negra erfarenheter. Tekn. Maddelanded o. 247, Inst. For Uppv. –o vent. Teknik. KTH, Stockholm.

Jiang Yi, Building Energy Efficiency Technology and Application, China Architecture and Building Press, Beijing, 2005, pp. 1. (In Chinese).

Li, Y., Sandberg, M. and Fuchs, L. 1992. Vertical temperature profiles in rooms ventilated by displacement: full-scale measurement and nodal modelling, IndoorAir, Vol. 2, pp. 225-243.

Livtchak, A., Nall, D., 2001. Displacement Ventilation. Application for Hot and Humid Climate, Submitted for Proceedings of Clima 2000/Napoli 2001.

Mundt, E., 1990, Convection flow above common heat source in rooms with displacement ventilation. Proceedings of ROOMVENT '90.Oslo.

Mundt, E., 1996, The performance of displacement ventilation system. Ph.D. thesis, Royal Institute of Technology, Sweden.

Nielsen, P.V, 1993, Displacement ventilation- Theory and design, Aalborg University. ISSN 0902-8002-U9306.

Nielsen, P.V, 1996, Temperature Distribution in a displacement ventilated room. In ROOMVENT '96. Proceedings of the 5th International Conference on Air Distribution in rooms, Yokohama, Japan, July 17-19, 1996/ Ed. By Murakami, S.-vol 3, pp. 323-330. ISBN: 4-924557-01-3.

Nielsen, P.V., Hoff, L., Pedersen, L.G. 1998. Displacement ventilation by different type of diffusers. Proc. Of the 9th AIVC conference. ISBN 0946075 409, Warwick.

REHVA, Displacement ventilation in non-industrial premises, REHVA Guidebook 2002, Håkan Skistad (ed), REHVA - Federation of European Heating and Air-Conditioning Associations.

Sandberg,M. 1985. Luftutbyteseffektivitet, ventilationseffektivitiet, temperatureeffektivitiet I cellkontor. System med luft som energibaerare. Statens institut foer byggadsforskning, Meddelande M85:24.
Skistad, H., 1994, Displacement ventilation. Research Studies Press, John Wiley & Sons, Ltd., west Sussex. UK.
Wyon, D.P. and Sandberg, M. 1990. Thermal manikin prediction of discomfort due to displacement ventilation. ASHRAE Transaction, 96 (1).

Definitions

Archimedes number

The Ar is the ratio between the buoyancy forces and the inertia forces. Several phenomena in a ventilated room can be described by the Archimedes number. (Temperature gradient, velocity levels in the stratification flow and ventilation effectiveness). In it is original form it is defined as:

$$Ar = \frac{\Delta\rho \cdot g \cdot L}{\rho \cdot v^2}$$

It can be expressed in several ways, e.g. using the temperature difference instead of the density difference. Larger numbers mean that the buoyancy forces are dominant, smaller numbers mean that the inertia forces (velocity are dominant).

In this thesis is used in the following form:

$$Ar = \frac{\Delta\vartheta \cdot g \cdot H \cdot \beta}{(q_s / A)^2}$$

Occupied zone

Volume of air, which is confined by horizontal and vertical planes. The vertical panels are usually parallel with the walls of the room. Typical definitions for the occupied zone are given in Table 1.

Table 1 Typical and standard dimensions of the occupied zone

Element	Distance from the inner surface of the elements [m]	
	Typical range	Default value (CR 12792)
External windows, doors and radiators	0.5-1.5	1.0
External and internal walls	0.25-0.75	0.5
Floor-lower boundary	0.00-0.2	0
Floor-higher boundary	1.3-2.0	1.8

中文翻译：

置换通风设计方法综述

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关键词：置换通风，热舒适，设计方法，室内空气品质
摘要：

置换通风系统已在欧洲成功应用有二十多年，现在它也成为中国的空调系统的一种新型方式。置换通风系统（Displacement ventilation: DV）是建立在工作区温度垂直分层的基础上的通风形式。热源所产生的空气流动影响置换通风系统房间内的空气流动。合理设计的置换通风系统与混合通风相比可以提供较高的通风效率，为室内人员提供较好的空气品质，并且能耗较低。

为了避免气流和垂直温差引起的不舒适，在设计阶段要仔细预测温度分布。本文对设计置换通风系统中为满足热舒适需求采用的 REHVA 和 ASHRAE 方法，做了介绍并给予了综述。

1.引言：

1.1 背景介绍

置换通风系统应用于欧洲的高热负荷工业建筑中已有 30 多年。近 20 年置换通风系统在北欧国家中的非工业建筑中也逐渐流行起来，如写字楼、学校、剧院等。

1.2 置换通风系统节能和空气品质的优势

在房间中，垂直温度梯度导致了置换通风。由于地板处的空气温度要低于天花板处空气的温度，所以工作区和天花板之间的几度温差就使经过工作区的空气可以通过天花板处排走。置换通风的规律可以使得能源有较高的利用率，因为置换通风的送风温度可以比混合通风的送风温度要高[10]。而且它可以节约能量（较长的 free cooling 使用时间和工作区较少的冷量需要[13]）。

众所周知，室内空气品质直接影响到其中人员的健康、舒适和工作成效。置换通风系统可以提供一个舒适的环境。与混合通风相比，对于与热源有联系的污染源，置换通风系统更容易使工作区的空气清洁。

中国在近些年正在高速的发展，已经建设并仍在建大量的建筑。当然，对能源的需求也随着增加。据统计，中国 26%的能源消耗来自于商业建筑和住宅，这其中大部分是由空调消耗的[5]。能源的短缺和自然环境的恶化使得建筑设计者为更为理性的利用能源而努力。
所以，置换通风系统由于其可以提供高品质室内空气和节能的特性，可能会为中国提供一种成功的空调形式。然而置换通风系统必须经过仔细的设计才可以将其优越性发挥出来。

1.3 垂直温度梯度和地板处的气流

温度分层可导致头和脚之间的温差，这可能会引起局部的不舒适。而且置换通风系统直接将冷空气提供到工作区，这很容易在地板处产生气流[11]。为了避免局部不舒适，应仔细预测垂直温差，且要合理选择送风口。

1.4 本文目的

为了使置换通风系统发挥出优势，避免人员的不舒适，就需要对设计方法有深入的了解。本文讨论了欧洲和美国的置换通风系统设计方法。文章将不解释设计中满足室内空气品质的问题。

2.置换通风系统的原理和应用

2.1 原理

置换通风系统是根据垂直温度分层和工作区的空气污染而提出的。

当使用置换通风系统时，空气流动方式主要是由各个热源的对流决定的。冷空气被低速低扰动的送到地板高度的区域，然后由天花板处排出，如图 1 所示。空气流动是由热源上升的对流所产生的，对流将空气从低处带到高处[9]。从出风口流出的空气的动量通量较低，所以对整个房间的空气流动没有太大的影响。

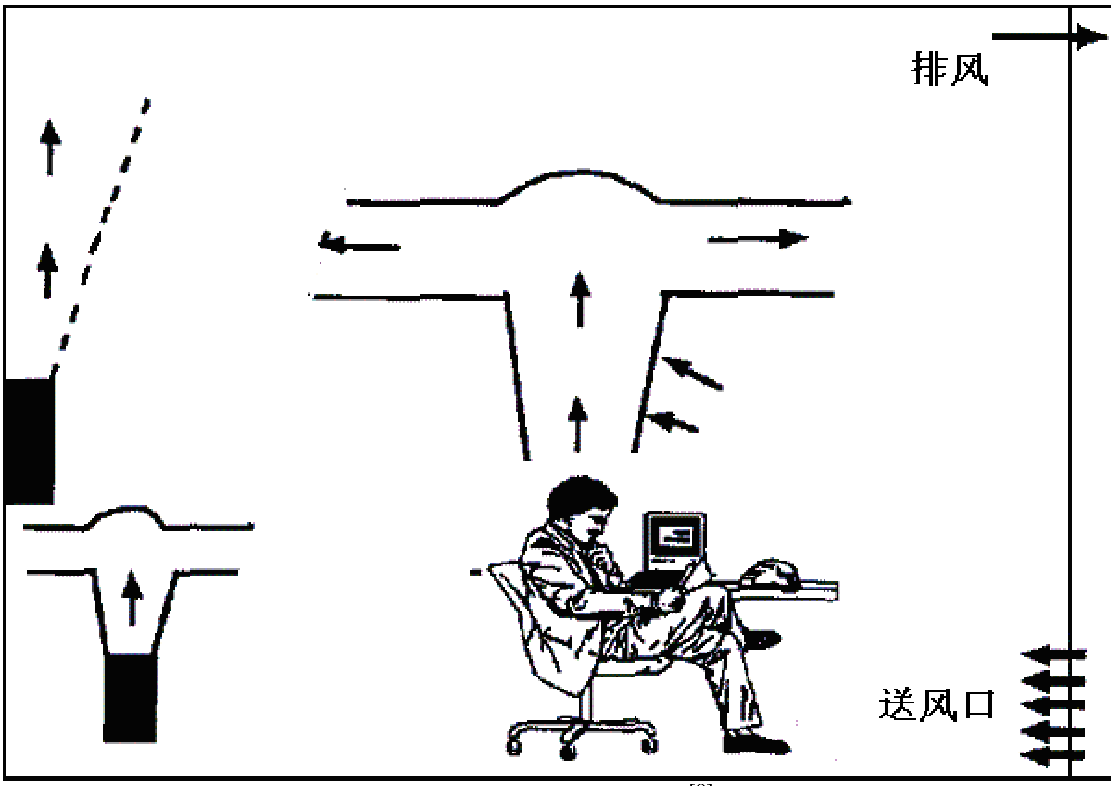

图 1 置换通风原理图[9]

为了较好的运行，送风温度要比室内空气温度低 3-4 °C。

热源产生的气流和对流所导致的空气流量依据下列条件：

1) 热源几何学
2) 热源温度
3) 温度梯度

如果热源也是污染源的话，置换通风系统可以有很高的通风效率[9]。否则对于工作区空气品质，置换通风系统与混合通风很相似。比如：送出的新鲜空气流过地毯后将把污染物带起来与室内空气混合。

2.2 应用场所

置换通风系统可以被应用于礼堂、餐厅、办公室、教室、会议室，还可应用于高大建筑，比如剧院、超市、电影院等等。置换通风系统通常比较适合于污染物比周围空气温度略高并且/或者略轻的情况，还有在较小的房间或高度大于 3 米的房间内有较高送气速率的情况[15]。置换通风系统对于有较高天花板的房间尤为适合。

根据 Wyon 和 Sandberg 的研究[16]，如果冷负荷不超过 40 W/m^2，置换通风系统可以维持一个舒适的环境。然而 Chen 和 Glicksman 的研究[3]则表明冷负荷可为更高的值，达 120W/m^2。如果置换通风系统和冷辐射吊顶一起使用，则可适用于较高的冷负荷。所以，在这种情况下，就需要不同的设计方法（本文未讨论）。虽然置换通风系统的制冷效果很理想，但这种系统无法进行制热，这是因为浮力和低送风速率导致送的热空气流向天花板。如果在冬季必须要制热，则需要独立的制热系统（辐射板、对流式暖器、暖气和地板处的风机盘管等）。

3.垂直温度分层

温度梯度体现在置换通风的房间内。图 2 为垂直温度分布简化图。T_s, T_{af}, T_e 分别为送风口空气温度、地板处空气温度和排气口处空气温度。

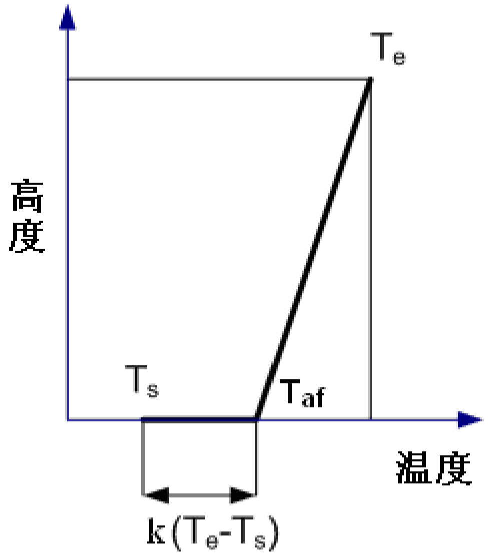

图 2 置换通风房间内垂直温度分布简化图

送入房间的空气温度升高是由于周围空气的掺混和地板对流换热所产生的热传递。根据 Nielsen[10]和 Skistad[15]的研究，空气温度随房间高度几乎线性变化。但是根据 Chen 等的研究[3]，大部分情况空气温度不随房间高度线性变化。事实上，除了送风口附近的区域，空气温度在水平方向上均匀一致。地板附近空气的无量纲温度 κ 由式（1）表示。根据 Nielsen [11]和 Brohus 与 Ryberg[1]的研究，κ 值在 0.3 到 0.65 之间。而 Chen 等人的研究[3]表明 κ 在 0.2 到 0.7 之间。

$$\kappa = \frac{T_{af} - T_s}{T_e - T_s} \qquad (8)$$

3.1 温度分布决定因素

温度分布受空气流动速率、热源类型、热源垂直位置、墙的特性、空间高度和送风口影响。

3.2 空气流动速率

Sandberg[14], Chen 等[2] 和 Mundt 的研究[8]表明当空气流动速率增加时，无量纲温度降低。Mundt 改进了 κ 的公式[8]，如式 2。Mundt 的公式将 κ 作为了地板上方通风速率的函数，将通风速率和冷负荷联系了起来。

$$\kappa = \frac{T_{af} - T_s}{T_e - T_s} = \frac{1}{\frac{q_v \cdot \rho \cdot c_p}{A} \cdot (\frac{1}{\alpha_r} + \frac{1}{\alpha_{cf}}) + 1} \qquad (9)$$

其中，
c_p为空气比热，1004 J/kg K
q_v为空气体积流量[L/s]
α_{cf}为对流传热系数 [W/m^2 K]， 通常为 4 W/m^2 K
α_r为辐射传热系数 [W/m^2 K]， 通常为 5 W/m^2 K
κ为地板附近空气无量纲温度
ρ为空气密度，1.20 kg/m^3 (θ = 21 °C)

式（2）的基本含义是天花板与地板之间的温差导致天花板到地板之间的辐射热传递。这也就是说地板温度将会升高，对流换热将使送风温度升高。这个模型是一个能量平衡方程。

3.3 热源形式

Nielsen 指出温度梯度受空气流动的影响较小[11]，受热源类型尤其是热源表面温度的影响较大。如果主流是充分发展的湍流，无量纲温度可以表示为阿基米德数的单一函数。Nielsen 给出了一个设计图表[11]，它为工作区的四种热源形式（分布源、坐姿人员、棚顶灯、点源）列出了由阿基米德数决定的地板处无量纲温度 κ。

3.4 热源垂直分布

Mundt 指出当热源在房间较低处时，温度梯度在低处较大，而在高处更加均匀[9]。当热源在房间较高低处时，温度梯度在低处较小，在高处逐渐增加。在热源处于高处的房间，置换通风可以有效的保持工作区凉爽[13]。

3.5 墙的特性

如果墙的面积较大，即使墙和室内空气有一个小的温差也会产生一个相当大的向上或向下的空气流动。垂直温度分布曲线受到由外温变化而导致的墙温度变化的影响。

3.6 空间高度

置换通风更适合于高大空间。高大空间的上部，空气温度只有轻微的升高。在有座椅下送风口的音乐厅，空气温度由地板处的送风口到人员的位置会显著的上升。而在人员上方，只有很小的温度梯度[15]。根据 Skistad 的研究结果，在房间高大的工业建筑中，垂直的温差中的 1/3 都是在地面附近产生的[15]。

3.7 送风口

Nielsen 等人指出，为了减少工作区的温度梯度，增加室内空气的混合程度是必要的[12]。这可以由高效的送风末端做到。送风口对于降低地板附近较大气流的风险是很重要的。

4.设计方法

这一节介绍了在保证室内人员热舒适性方面比较成熟的两种设计方法。第一种是由 REHVA（欧盟采暖与空调协会）推荐的，第二种是由 ASHRAE(美国采暖、制冷与空调工程师协会)发展起来的。

一个通风系统必须能同时保证热舒适和室内空气品质。工作区的这些要求应该是在不同的运行条件下，以最经济且不产生噪音的方式来满足。在置换通风的设计中重要的参数有：

1) 送风速度和送风温度
2) 地板面的空气温度
3) 地板面的空气速度
4) 垂直温度梯度

5) 污染物（浓度）梯度

在设计中已知的物理参数是空气的送风速度和送风温度。保证热舒适性的送风速度和送风温度是基于预期的垂直温度分布而计算得到的

4.1 REHVA 方法

这种方法是从北欧国家中发展起来的[13][15]，它是基于实验的方法并假设从地板到天花板的温度分布是线性的。

这种方法定义 κ＝0.5，认为送排风温差的一半在地面附近平稳过渡，剩余部分的温差沿高度方向线性变化。κ 值与冷负荷、通风速率、类型和热源位置无关。

设计步骤如下：

10) 设定在离地面 1.1m 处的目标温度($T_{d,1.1}$)和工作区的设计垂直温度梯度 s（单位是°C/m），s 一般取 2°C/m。
11) 计算需通过通风排出的余热量
12) κ =0.5，则总温差可以由以下公式（3）决定：

$$\Delta T = T_e - T_s = 2 \cdot h \cdot s \tag{10}$$

这里的 h 指房间高度

13) 通过能量平衡公式（4）计算排出余热 Φ_{tot} 所需的空气流量 q

$$q = \frac{\Phi_{tot}}{\rho \cdot c_p \cdot \left(T_e - T_s\right)} \tag{11}$$

14) 通过以下公式计算空气的送风温度：

$$T_s = T_{d,1.1} - s \cdot (1.1 + h) \tag{12}$$

15) 此时送风速度和温度已知，用公式(2)复算地面处的温度(T_{af})以及房间内的温度分布。
16) 为了避免较大的气流，需要依照样品数据来选择送风口的尺寸、数目、类型及安装地点。

4.2 ASHRA 方法

这种方法是 1999 年在美国由 Chen and Glicksman 基于 ASHRAE 研究项目 RP-949 结果的基础上发展起来的[3]。Chen and Glicksman,提出的这种方法建立在一个经验证的 CFD 程序的基础上，是针对典型的美国建筑的。假设在 0.1-1.1 m 的范围内的温度分布是线性的，通过公式(2)来预测 κ 值。这种方法的出发点在于认为仅仅有一部室内产热会影响 0.1-1.1 m 高度范围内的温度梯度。设计步骤如下：

6) 判断置换通风的适应性（即此种条件适不适合用置换通风）、选定距地面1.1m高处的目标温度($T_{d,1.1}$)。
7) 计算夏季的设计冷负荷，详细的冷负荷项包括：
    - 室内人员、台灯及设备，Qoe (W)
    - 头顶照明，Ql (W)
    - 通过房间维护结构的导热及穿透的太阳辐射热, Q_{ex} (W)
8) 由以下公式确定负担夏季冷负荷所需的送风量：

$$q = \frac{Q_{oe} \cdot a_{oe} + Q_l \cdot a_l + Q_{ex} \cdot a_{ex}}{\rho \cdot c_p \cdot \left(T_{1.1m} - T_{0.1m}\right)} \tag{13}$$

q：空气的体积流量[m^3/s]

c_p：空气的定压比热= 1004J/kg K

ρ：空气密度= 1.20 kg/m^3 (θ = 21 °C)

$T_{1.1m}$-$T_{0.1m}$：室内久坐者头脚处的空气温差，一般为2°C.

a_{oe}, a_l, a_{ex},：为权重系数，分别为0.295, 0.132及.185，表示三种冷负荷进入久坐者头脚区域内的份额。

9) 通过公式(2)计算κ值。辐射和对流换热系数可以从ASHRAE手册中查到，作为粗略估计取α=5 W/(m^2 K)

10) 用以下公式计算送风温度：

$$T_s = T_{d,1.1} - (T_{1.1m} - T_{0.1m}) - \kappa \cdot \frac{Q_{oe} + Q_l + Q_{ex}}{\rho \cdot c_p \cdot q} \tag{14}$$

11) 为了避免较大的气流，需要依照样品数据来选择送风口的尺寸、数目、类型及安装地点。

12) 校核冬季供暖工况，因为建筑外区冬季需要供暖，所以当设置置换通风时还需要一个单独的供暖系统来承当从外墙和外窗渗透的热负荷。

这些权重系数是由一个通过对几个普通的美国建筑的模拟而被校准过的 CFD 程序计算出来的。所以这些权重系数是在不同情况下的平均结果，必然带有不确定性。

Li，Mundt 和 Livtchak 等人提出更为详细的方法，将辐射换热的计算也考虑在内了[6][7][9]。这些方法由于相互式的耦合及过于复杂难以用于手工计算，只能编成电脑软件。

CFD(计算流体力学)的方法也是可行的，它应用于大空间的复杂环境。在这种情况下就必须注意热、污染物、辐射换热以及送风口等边界条件的描述。

5.两种方法的比较

温度的分布取决于空气流动率，热源的种类和位置，墙面的特性，空间高度和空气扩散器的状况。本文所举的两种方法没有考虑上述因素的影响。

5.1 垂直温度线性的分布

根据 Nielsen, Skistad 和 Mundt 所说，空气的温度随高度成线性的改变[9][10][15]。这就是 REHVA 方法的一个设想。但根据 Chen 等人所说，这个假定在大多数情况下是不正确的[3]。ASHRAE 的方法只是在可触及的范围内假定了温度的线性性，但无法预知房间上面部分的温度是如何不同的。这个假说只是考虑了影响居住者温度感觉的那一部分热负荷。

5.2 空气流动率和冷负荷

两种方法都使用了 Mundt (式(2))模型来计算地平面无量纲的温度。方程式 2 有两个未知的参量，κ 和 q。REHVA 的方法只有在已经测定供给空气流动率(q)和温度(T_s)时才能使用这个方程式。因而代入的参量 (q 和 T_s) 可计算出大致 κ =0.5。但这样有一个很大的局限性，因为 Mundt’s 的模型不是用来计算出精确的代入参数的，仅仅是用来预测一下在已有空气流动率和温度下温度的大致轮廓。正相反，ASHRAE 方法应用 Mundt 的式(2)来计算温度，因为空气流动率已经知道了。而空气流动率只是根据以占用地带的温度曲线图的预测中得来的。

5.3 热源的种类和位置

REHVA 的方法没有把热源的种类和位置考虑进去，正如 Nielsen 所指出的那样，它可能导致使对地板处地无量纲温度的预测产生重大错误[11]。ASHRAE 的方法没有单独考虑每个热源的位置和种类。ASHRAE 的方法依照种类和位置把热源详列为三个种类，并通过权重因数详细说明了它们对热负荷平衡计算中所起的作用。然而测定的因数其准确性还是需要核实的。

5.4 墙壁的特性

REHVA 的方法没有考虑到墙壁不隔热的可能性。ASHRAE 的方法没有模拟非绝热的墙壁状况，但通过权重因数，它确实考虑到只是通过墙壁转移的那部分热，影响了使用空间的温度。

5.5 空间高度和送风末端

两种方法都没有考虑空间高度和空气扩散器的因素。两种方法都强调了对送风末端尺寸，数量，种类和位置的选择是避免干燥的关键因素。

6.结论

两种方法是在非产业前提下取代通风装置设计的主要方式。REHVA 的方法，以实验的步骤为依据，主要用于欧洲，而 ASHRAE 的方法，以被认证了的 CFD 的预测为基础，主要应用于美国。

ASHRAE 的方法考虑了比 REHVA 方法更多的能影响温度垂直分布的因素。

ASHRAE 的方法是为典型美国建筑而发展开来的，所以或许不可以直接应用在中国。这两种方法用在高大空间，或带有冷（暖）地面（屋顶）系统的设计中是不合适的。应该采用 CFD 仿真或更复杂的方法。

建立适于中国气候状况特征的设计方法是十分需要的。

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参考文献

[1] Brohus, H. and Ryberg, H. 1999. The Kappa model- a simple model for the approximate determination of vertical temperature distribution in rooms. Department of building technology and structural engineering, Aalborg University.

[2] Chen, Q. 1988, Indoor Airflow, Air quality and Energy Consumption in Building. Ph.D. Thesis of Delft University of Technology. Delft.

[3] Chen, Q., Glicksman, L., 1999. Performance Evaluation and Development of Design Guidelines for Displacement Ventilation. ASHRAE Research project-RP-949.

[4] Jamrym, R.. 1982. Displacerande inblasning – Negra erfarenheter. Tekn. Maddelanded o. 247, Inst. For Uppv. –o vent. Teknik. KTH, Stockholm.

[5] Jiang Yi, Building Energy Efficiency Technology and Application, China Architecture and Building Press, Beijing, 2005, pp. 1. (In Chinese).

[6] Li, Y., Sandberg, M. and Fuchs, L. 1992. Vertical temperature profiles in rooms ventilated by displacement: full-scale measurement and nodal modelling, IndoorAir, Vol. 2, pp. 225-243.

[7] Livtchak, A., Nall, D., 2001. Displacement Ventilation. Application for Hot and Humid Climate, Submitted for Proceedings of Clima 2000/Napoli 2001.

[8] Mundt, E., 1990, Convection flow above common heat source in rooms with displacement ventilation. Proceedings of ROOMVENT ’90.Oslo.

[9] Mundt, E., 1996, The performance of displacement ventilation system. Ph.D. thesis, Royal Institute of Technology, Sweden.

[10] Nielsen, P.V, 1993, Displacement ventilation- Theory and design, Aalborg University. ISSN 0902-8002-U9306.

[11] Nielsen, P.V, 1996, Temperature Distribution in a displacement ventilated room. In ROOMVENT ’96. Proceedings of the 5th International Conference on Air Distribution in rooms, Yokohama, Japan, July 17-19, 1996/ Ed. By Murakami, S.-vol 3, pp. 323-330. ISBN: 4-924557-01-3.

[12] Nielsen, P.V., Hoff, L., Pedersen, L.G. 1998. Displacement ventilation by different type of diffusers. Proc. Of the 9th AIVC conference. ISBN 0946075 409, Warwick.

[13] REHVA, Displacement ventilation in non-industrial premises, REHVA Guidebook 2002, Håkan Skistad (ed), REHVA - Federation of European Heating and Air-Conditioning Associations.

[14] Sandberg,M. 1985. Luftutbyteseffektivitet, ventilationseffektivitiet, temperatureeffektivitiet

I cellkontor. System med luft som energibaerare. Statens institut foer byggadsforskning, Meddelande M85:24.
[15] Skistad, H., 1994, Displacement ventilation. Research Studies Press, John Wiley & Sons, Ltd., west Sussex. UK.
[16] Wyon, D.P. and Sandberg, M. 1990. Thermal manikin prediction of discomfort due to displacement ventilation. ASHRAE Transaction, 96 (1).

定义：

阿基米德数:

Ar 被定义为浮力与惯性力之比。通风房间的一些现象可以用阿基米德数来解释。（温度梯度，层流速度水平和通风效力速度水平）

$$Ar = \frac{\Delta\rho \cdot g \cdot L}{\rho \cdot v^2}$$

阿基米德公式可以有多种表达形式：例如，可以用温度差替换密度差。大数值意味着浮力占主导作用，小数值意味着惯性力（即风速）占主导作用。

这篇文章中，我们采用了以下的表达式：

$$Ar = \frac{\Delta\vartheta \cdot g \cdot H \cdot \beta}{(q_s / A)^2}$$

工作区：

空气的体积受水平和垂直位面的限制。垂直层通常指平行于房间墙面。工作区的典型定义由表 1 给出。

表 2 工作区典型和标准的维度

元素	由元素内表面测量的距离 [m]	
	典型量程	默认值 (CR 12792)
外窗、门和散热器	0.5-1.5	1.0
外墙和内墙	0.25-0.75	0.5
地板附近低处的边界线	0.00-0.2	0
地板附近高处的边界线	1.3-2.0	1.8