Ventilation in Commercial and Residential Buildings

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Publication date:
1999

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. (1999). *Ventilation in Commercial and Residential Buildings*. Dept. of Building Technology and Structural Engineering, Aalborg University. Gul serie Vol. U9903 No. 43

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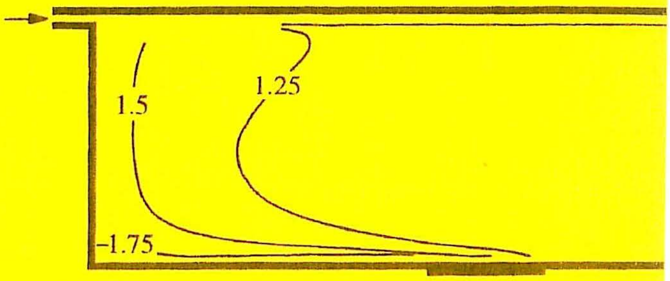
Ventilatio

Ventilation in Commercial and Residential Buildings



Peter V. Nielsen

In: Ventilation Systems and Air Quality, Lecture Series at
von Karman Institute for Fluid Dynamics, Belgium, May
11-13, 1998

ISSN 1395-8232 U9903

Ventilation in Commercial and Residential Buildings

Peter V. Nielsen

Printed at Aalborg University

VENTILATION

Removal of heat generated by

- Persons
- Equipment
- Solar radiation
- ...

Removal of vapour, gases and particles as

- Moisture
- Tobacco smoke
- VOC's
- CO_2
- Odour
- Particles and fibres
- ...

Supply of fresh air

Ventilation

A building requires supply of fresh air and removal of heat, gases and particles which are emitted in the building. Heat is generated by persons in the building, different types of equipment (PCs, television, lighting, etc.) and by solar radiation into the room. Vapour, gases and particles are also generated inside buildings and emitted to the room air. Moisture is generated by people, cooking and drying of e.g. clothes. Tobacco smoke is by far the most important source of chemical pollution of indoor air but a large group of Volatile Organic Compounds (VOC) coming from building materials, furniture, household maintenance products, cosmetics and office materials are also a very important pollution source (Molina et al., 1989). Other gaseous substances are carbon dioxide (CO_2), carbon monoxide (CO), nitrogen dioxide (NO_2) and ozone (O_3) which also may pollute the air in buildings. CO_2 is generated by people and it occurs naturally in the atmosphere. CO and NO_2 can be produced by incomplete combustion and tobacco smoking whereas O_3 is produced by photocopying machines and laser printers (Molina et al., 1989).

Odours from gases and vapours will give rise to sensory discomfort and should be removed from the building. Particles and fibres will also influence the air quality and should consequently be removed by the ventilation system or by cleaning.

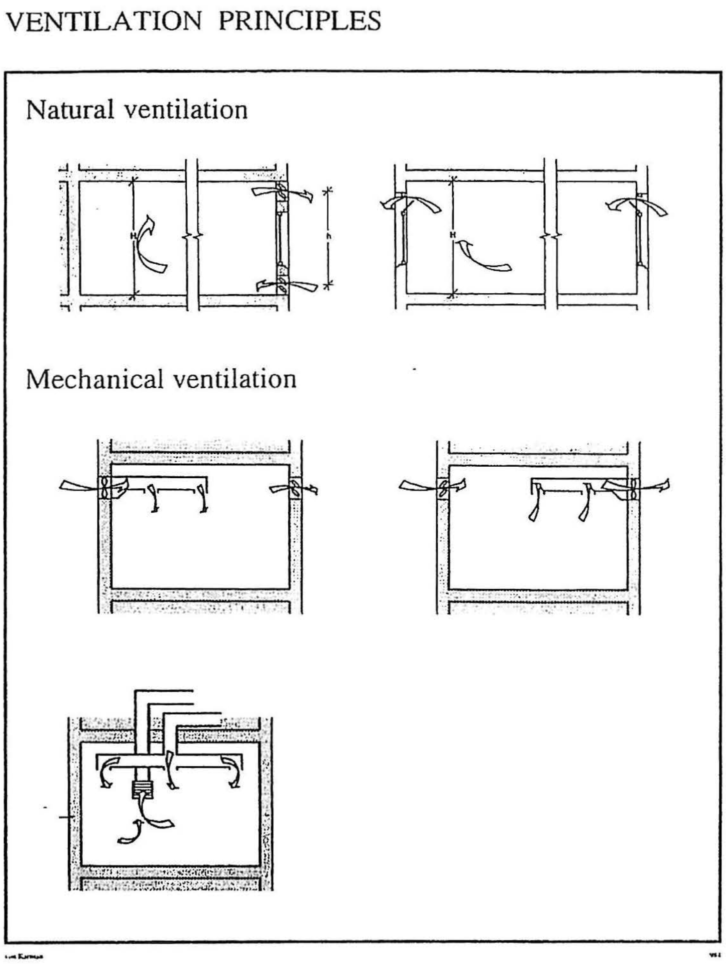
VENTILATION AND INDOOR AIR QUALITY

- Ventilation principles
- Components in a ventilation system
- Air diffusion and contaminant distribution in a room
- Ventilation effectiveness and air exchange efficiency
- Minimum airflow rates

Ventilation and indoor air quality

A number of areas have to be considered in connection with indoor air quality and ventilation. The selection of ventilation principle and components in the ventilation system will have influence on the indoor air quality and this subject will be discussed on the following pages.

The main object of this lecture notes is to discuss the design of the air diffusion system with respect to indoor air quality and to give recommendations for minimum air flow rates.



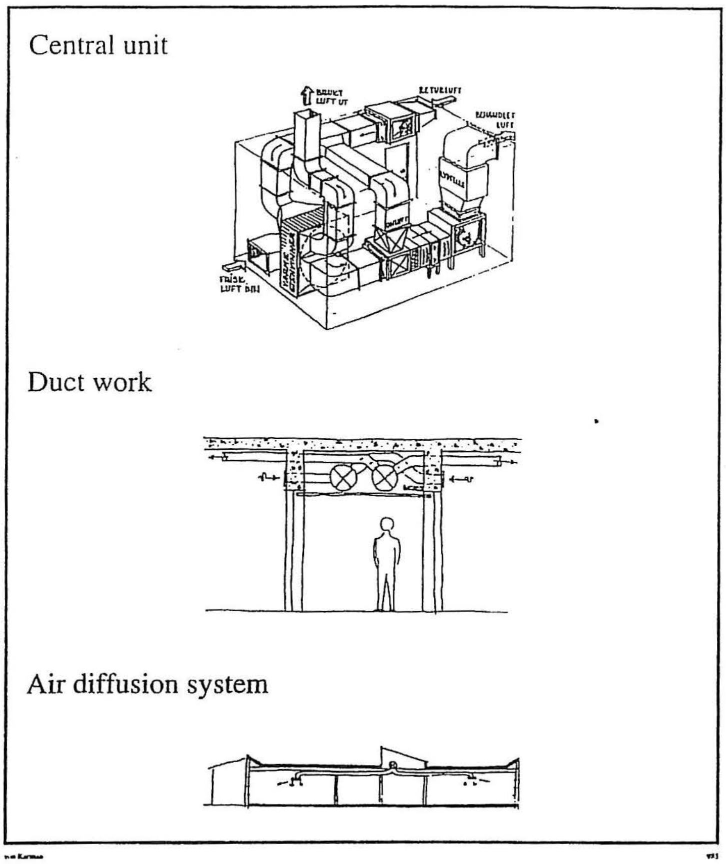
Ventilation principles

The upper figures show two ventilation configurations in the case of natural ventilation. For a single sided double opening (shown in the left figure) the height between the openings will define the thermal stack height which can be maximized by separating them in the wall. Temperature-driven natural ventilation will be proportional to this height. The wind will also generate ventilation due to a probability of pressure difference between the two openings separated at a vertical distance (Barker, 1995). Cross ventilation (right figure) is very effective for wind-generated pressure differences with a useful depth of at least three times floor to ceiling height. It is difficult to control the air quality in areas with natural ventilation.

The mechanical system can either be a pressurized system, an exhaust system or a balanced system. The pressurized system (upper left figure) is appropriate for fresh air supply in wintertime when the air can be preheated and uncontrolled draught from infiltration is eliminated due to overpressure of the interior. It is possible to filter the supply air or locate the system inlet to the building in low polluted areas (high above street level). The exhaust system (upper right figure) is appropriate in rooms with pollution sources where the exhaust openings are located close to the sources. It is an inexpensive system because few exhaust fans can be combined with a large number of wall openings as used in livestock buildings. It is difficult to control the air quality.

The lower figure shows a balanced system which is the configuration usually adopted to provide full air condition and at the same time permitting recirculation and heat recovery. This system gives the possibility of a controlled pressure distribution in the building (high and low pressure in different rooms) and thus a control of the contaminant flow in the building.

ELEMENTS IN AN AIR-CONDITIONING SYSTEM



Elements in an air-conditioning system

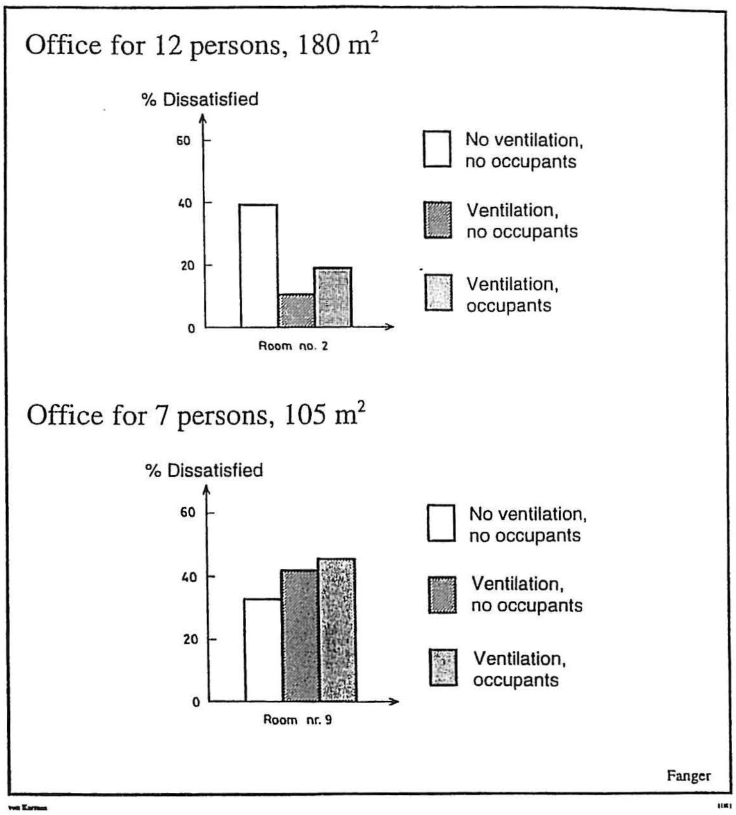
The central unit, often located in the cellar of the building or on the roof (rooftop unit), prepares the outdoor air for the distribution in the building. The outdoor air is filtered, heated or cooled and heat recovery from the exhaust air or recirculation may take place. The fans for the air condition system will also be located in the central unit.

Each component in the Heating, Ventilation and Air-Conditioning system (HVAC system) can be a sink or a source of pollution depending on the quality of the air and the strength of the source (Clausen et al., 1996). It has been documented that the cleaning of the duct work has a significantly positive influence on reducing the odour emission because the dust layer seems to be good storage material for the odour.

Certain diseases as legionnaire's disease, pontiac fever, extrinsic allergic alveolitis and humidifier fever may be associated with air conditioning or ventilation systems, but the risk of severe diseases appears to be extremely small (Bartlett and Pickering, 1983).

The air diffusion system (air distribution in the room) is also very important for the indoor air quality. This aspect will be discussed in the following chapters.

AIR QUALITY INFLUENCED BY THE AIR CONDITIONING SYSTEM



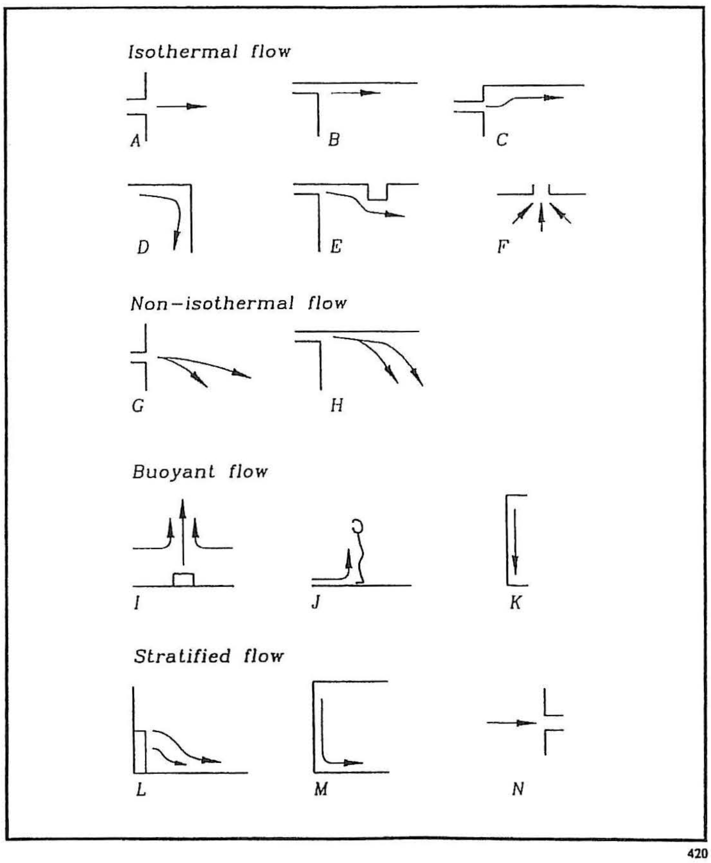
Sensory panel

A sensory panel can be used to determine the perceived air quality. The panel may be untrained (naïve) or trained. An untrained panel comprises a group of persons without any bearing on the space to be examined. The panel members are asked to judge whether the air quality is acceptable or not. In this way the percentage of dissatisfied provides the perceived air quality (ECA, 1992). The necessary number of panel members depends on the required statistic confidence but it will typically be in the range of 20 - 50 persons (Pejtersen and Mayer, 1993).

Air quality influenced by the air conditioning system

The perceived air quality is obtained by an untrained panel. The situation is acceptable in the building with a 180 m^2 office (upper figure). There is a high percentage of dissatisfied when the ventilation is turned off (measured in weekends). The ventilation system will decrease the percentage of dissatisfied when it is turned on, and occupants will of course increase the level because people are one of the important pollution sources.

The lower figure shows a situation which is unacceptable. The ventilation system *increases* the contaminant level which means that the central unit and the duct work act as contaminant sources.

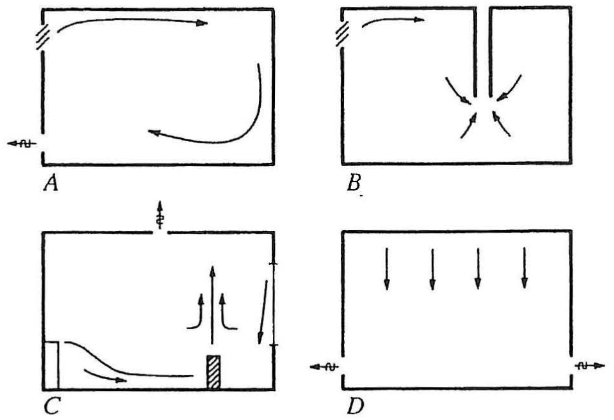


Flow elements

Air movement in a room may often be divided into a restrict number of flow elements which can be treated independently of the surrounding flow and dimensions (Nielsen, 1994A). The figures A to C show different kinds of supply jets, and the figures D and E show disturbance of jets as e.g. deflection at an end wall and flow around an obstacle. The flow around an exhaust opening is indicated in figure F.

Non-isothermal flow elements consist of a free jet with deflection due to gravity, figure G, and a wall jet with restricted penetration depth, figure H. Buoyant flow is illustrated in the figures I to K. Figure I shows a thermal plume above a heat source, figure J shows natural convection around a human body and figure K shows cold downdraught at a wall. Stratified flow from a low impulse opening and from cold downdraught is illustrated in the figures L and M. The last figure shows the stratified exhaust flow in the case of a vertical temperature gradient (displacement ventilation), figure N. This flow is different from the isothermal situation, figure F, because the air will be removed as a horizontal layer from the room due to the temperature gradient.

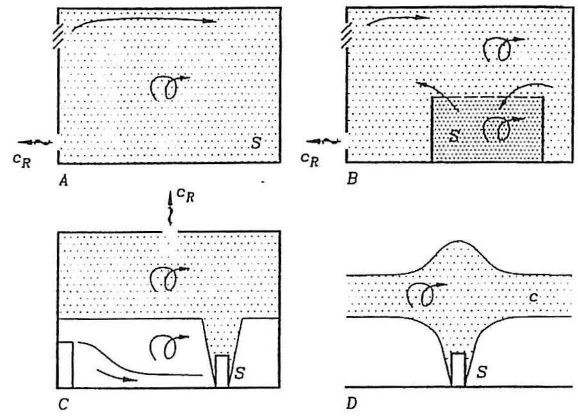
Different air distribution systems composed of a variety of flow elements.



Different air distribution systems

Different air distribution systems can be composed of a variety of flow elements. Figure A shows a mixing ventilation system with three main flow elements, namely a wall jet below the ceiling, deflection of the jet at the end wall and an exhaust flow close to the return opening. The name "mixing ventilation" indicates that full mixing of room air and contaminants is assumed in this situation - corresponding to a single zone. It can be shown that this is not always the case in practice. Figure B shows the situation in the case of industrial ventilation where the return opening is used as local exhaust in connection with an emission source. Displacement ventilation is obtained by the combination of four flow elements as shown in figure C. Air is supplied from a wall-mounted low velocity diffuser, and the plume above the heat source will move the air and the contaminant to the upper part of the room where it will be exhausted by the return opening. Cold downdraught is indicated in the figure. The last figure, D, shows the flow in a clean room with unidirectional flow from the ceiling.

Different air distribution systems can compose a variety of zone models.

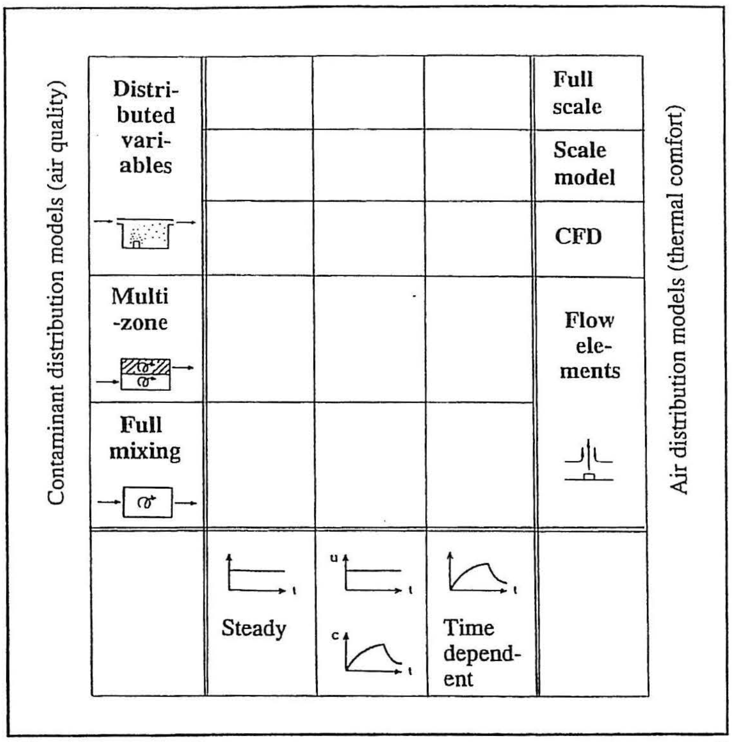


Zone models

The main principle of zonal models is to divide the indoor air volume into several macrovolumes of an assumed uniform concentration and (usually uniform) temperature distribution, and to solve the mass balance, the concentration balance and the energy balance in each zone in order to compute the temperature, the concentration and the flow fields.

Different air distribution systems (which are composed of a variety of flow elements) can be considered as zonal models with macrovolumes of assumed uniform concentration. Figure A shows a room ventilated by mixing ventilation which indicates - as an initial approximation - that the contaminant concentration is constant everywhere in the room corresponding to a single zone. The room in figure B is also ventilated by mixing ventilation but the contaminant source is located in an enclosure with restricted air exchange to the room. A two-zone model can in this case describe the concentration distribution. Figure C shows a room ventilated by displacement ventilation. This type of flow will generate two zones in the room: a lower zone with cold and fresh air and an upper zone with hot and contaminated air (if the heat source is a contaminant source also). Figure D shows a hot and contaminated plume which is stratified (at neutral level) in a vertical temperature gradient. The stratified layer can be considered as a zone with high concentration level surrounded by another zone with smaller concentration.

MODELS FOR CONTAMINANT DISTRIBUTION AND AIR MOVEMENT



Models for contaminant distribution and air movement

Different combinations of typical models are shown in the figure. Left side of the figure shows assumptions which are important for the calculation of contaminant distribution, namely full mixing, multizone models and distributed variables. The right side of the figure shows assumptions and methods which are important for the calculation of velocity and temperature distribution, namely flow elements, Computational Fluid Mechanics (CFD) and full scale and scale-model technique. The horizontal axis is divided into three areas: steady flow and a constant contaminant source which are often considered in a design situation, constant flow and a time-dependent contaminant source which may be a useful assumption in practical situations and fully time-dependent air distribution and emission of pollutant which are the real situation in buildings.

All the examples discussed in the following chapters can be identified as a combination of the variables shown in *Models for contaminant distribution and air movement.*

MIXING VENTILATION

Air distribution and maximum velocity in the occupied zone

- Jet Flow
- Throw of isothermal jet
- Maximum velocity in the occupied zone
- Room with furniture and obstacles
- Non-isothermal flow

Contaminant distribution

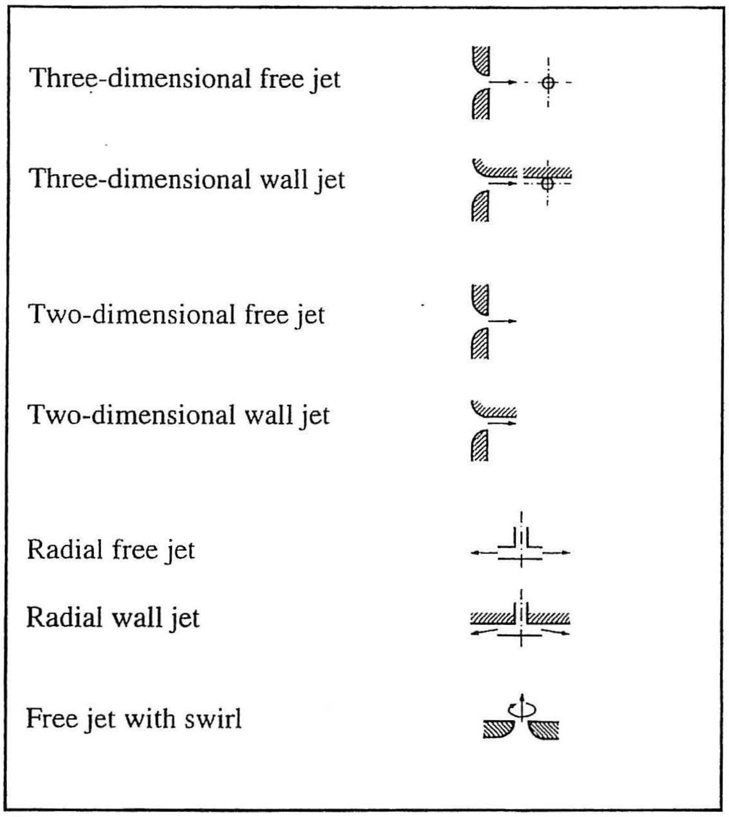
- Fully mixed steady flow
- Fully mixed steady flow and unsteady emission
- Concentration distribution and mixing ventilation

Mixing ventilation

The following chapter discusses different aspects of mixing ventilation. Air distribution and contaminant distribution can not be treated separately, and it is therefore important to consider the air distribution and the velocity in the occupied zone as part of a design procedure which has focus on contaminant distribution and air quality.

Different design methods for the air distribution system are mentioned, and different models for distribution of pollution are discussed ranging from a steady fully mixed zone to the concentration distribution obtained by more detailed research (CFD and full-scale experiments).

JET FLOW



Jet flow

Mixing ventilation is based on jet flow and it is controlled by the momentum flow from the jets. A buoyancy effect is assumed to be of minor importance in the general situation, but due to this effect high thermal load may change the flow pattern to the design limits.

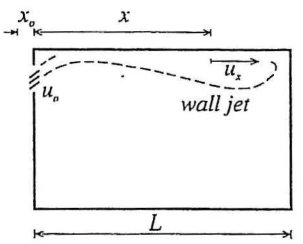
The flow from a supply opening will generate one of three different patterns, namely three-dimensional (axial) flow, two-dimensional flow or radial flow. The velocity in the three-dimensional flow and in the radial flow is inversely proportional to the distance from the opening, and the velocity in the two-dimensional flow is inversely proportional to the square root of the distance. Furthermore, the flow may be divided into free jets and wall jets.

The supply momentum flow will in practice be preserved in the jets.

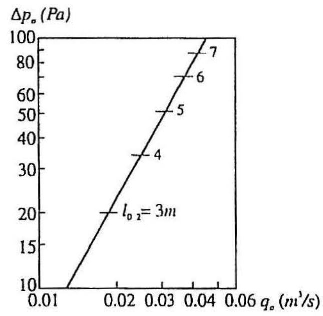
A free jet with swirl will have a high velocity decay which makes the diffuser suited for variable air volume systems (VAV-systems). A detailed description of the jet flow applied in room air distribution is given by Nielsen (1995 and 1996A).

DESIGN ACCORDING TO THROW OF ISOTHERMAL JET

The throw ℓ_{Th} is the distance to a location where the maximum velocity is equal to a reference value u_{Th}. Experience shows that the maximum velocity u_{rm} in the occupied zone is up to 0.15 m/s when ℓ_{Th} is equal to the room length L and u_{Th} is equal to 0.2 m/s.



Design chart



Design according to throw of isothermal jet

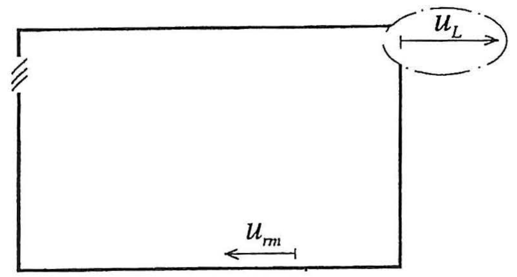
A length ℓ_{Th}, called the throw, is defined as the distance from the supply opening to a location where the maximum velocity is equal to a reference value u_{Th}. u_{Th} is often equal to 0.2 m/s in comfort ventilation and slightly higher in industrial ventilation.

It is the purpose of the design procedure to obtain an air distribution in the room with a maximum velocity in the occupied zone u_{rm} which is up to 0.15 m/s. (European Prestandard prENV 1752 (1994) indicates 0.15 m/s and 0.18 m/s as the design criteria in winter and summer situations, respectively). General experience shows that this is achieved when the throw is equal to room length L and the reference velocity u_{Th} is equal to 0.2 m/s. The theory is based on isothermal flow.

The throw is half the length between two air terminal devices with opposite positions or the length between opening and wall in a room with several diffusers. Other definitions of ℓ_{Th} may be used to compensate for different designs of air terminal devices and different room geometry. A throw of $L + H$ - 1.8 m is e.g. used when the room is high.

Producers of air terminal devices are presenting the throw ℓ_{Th} in a design chard as a function of the flow rate q_o m^3/s and the pressure difference Δp_o Pa.

MAXIMUM VELOCITY IN THE OCCUPIED ZONE



The maximum velocity in the occupied zone u_{rm} is connected to the velocity u_L in an undistrubed wall jet of the length L.

Two-dimensional isothermal flow $\frac{u_{rm}}{u_L} \sim 0.7$

Three-dimensional isothermal flow $0.3 < \frac{u_{rm}}{u_L} < 0.7$

Maximum velocity in the occupied zone

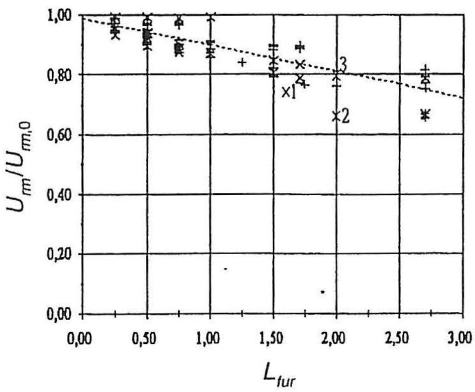
The maximum velocity in the reverse flow u_{rm} is located close to the floor at a distance of ~ 2/3 L from the supply opening. This velocity is also the maximum velocity in the occupied zone in cases where the jet flow below the ceiling and at the end wall is outside the occupied zone. Experiments with isothermal flow show that u_{rm} is a simple function of a reference velocity u_L, which is the velocity in an undisturbed wall jet of the length L from the actual air terminal device. The velocity u_L contains information on supply velocity, distance from inlet and geometrical details which will influence the initial flow, as e.g. type of Air Terminal Device, adjustable blades and distance from the ceiling.

The ratio u_{rm} / u_L will contain both information on velocity decay in the deflected jet due to the end wall height and geometry and information on the velocity level in the recirculating flow due to the entrainment into the jet below the ceiling. The velocity ratio will especially be influenced by both the deflected jet in a short room and the level of return entrainment flow in a deep room. The ratio u_{rm} / u_L as well as the level of u_L can therefore be used as elements in a design procedure (Nielsen, 1982, 1991, 1995, 1996A).

$\ell_{Th} = L$ and $u_{Th} = 0.2$ m/s give $u_{rm} = 0.2 \cdot 0.7 = 0.14$ m/s for $u_{rm} / u_L = 0.7$.

Advanced Air Terminal Devices which create a semiradial jet below the ceiling may obtain a high velocity decay (low u_L) and a u_{rm} / u_L ratio higher than 0.7.

ROOM WITH FURNITURE AND OBSTACLES



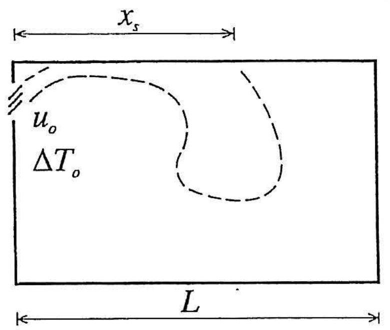
Furniture and obstacles will redistribute the velocity field in the occupied zone. The maximum velocity u_{rm} will be reduced compared to $u_{rm,0}$ in an empty room in the case of isothermal flow. L_{fur} in the graph is the length of the area with furniture (room length L is 5,4 m).

Room with furniture and obstacles

Furniture and obstacles will redistribute the velocity field in the occupied zone. The velocity will increase in some areas and decrease in other areas. The maximum velocity will be reduced compared to the value in an empty room in the case of isothermal flow (Nielsen et al., 1997).

Office furniture as e.g. table, computer and chair will typically cover the length of 2 m. The graph shows that the isothermal flow will be redistributed and that the maximum velocity will be reduced to 80% of the value in the empty room.

NON-ISOTHERMAL FLOW



The wall jet will separate from the ceiling at a distance x_s in case of cooling. The jet will penetrate the occupied zone and have a velocity which is larger than the isothermal velocity u_{rm}.

$$\frac{x_s}{\sqrt{a_o}} \sim 1/\sqrt{Ar}$$

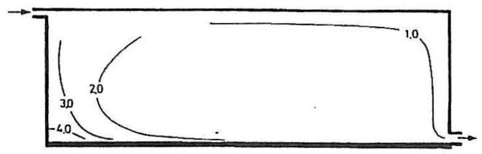
where a_o and Ar are supply area and Archimedes number respectively.

Non-isothermal flow

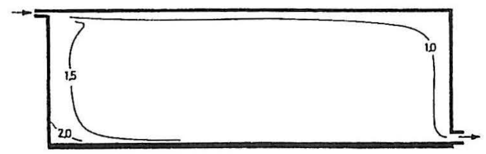
An undisturbed wall jet will penetrate the ventilated room in the case of isothermal flow, and it will entrain air from the occupied zone to induce recirculating air movement in the room. This picture will change when a thermal load is supplied to the room. The supply temperature will be reduced and the load may reach a level causing the wall jet to separate from the ceiling at a distance x_s from the supply opening and flow down into the occupied zone.

Situations with a short penetration length are undesirable because the jet may have a high velocity and a low temperature when it flows into the occupied zone. A calculation of the penetration depth is therefore a part of the design procedure of the air distribution system. An acceptable length is dependent on the actual product. It is often recommended that the penetration length x_s / L should be equal to or larger than 0.5. Air terminal devices with a high diffusion may work with a very short penetration length, see references (Nielsen, 1995, 1996A), (Nielsen and Möller, 1987).

Contaminant distribution given as c/c_R in a room with two-dimensional flow and an emission source at the floor (full length). Concentration c_R in the return opening is equal to S/q_o, where S is emission and q_o is air flow rate.



h/H = 0,056, L/H = 3,0



h/H = 0,01, L/H = 3,0

Fully mixed steady flow

$$c/c_R = 1 \qquad c_R = S/q_o$$

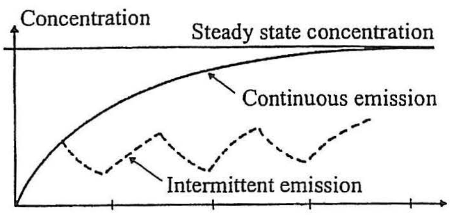
The figure shows that the distribution is close to a fully mixed flow in the case of a small supply opening h/H.

Contaminant distribution

The basic idea of mixing ventilation is to create a recirculating flow which will absorb the contaminant from different sources in such a way that the local concentration is low everywhere in the room. A transport of contaminant must, on the other hand, both involve a turbulent diffusion and a convection which will give rise to concentration gradients.

The figure shows the concentration distribution in a room with two-dimensional isothermal flow and a contaminant source which covers the whole floor. The local concentration is normalized by the concentration c_R in the return opening. The concentration in the left side of the room below the supply slot is large compared with the concentration in the return opening, so it is obvious that there will be significant gradients in rooms ventilated by mixing ventilation (Nielsen, 1981). The figure shows that the distribution is close to a fully mixed flow in the case of a small supply opening h/H and isothermal flow. It is the degree of entrainment and recirculation in the room which is important for the mixing process. The process is independent of the supply velocity.

DYNAMIC CONCENTRATION MODEL



$$c = \left(c_1 - \frac{S}{q_o}\right) e^{-(t-t_1)/\tau_n} + \frac{S}{q_o}$$

c	Concentration at time t	$(mg/m^3 \ or \ cm^3/m^3)$
c_1	Initial concentration	$(mg/m^3 \ or \ cm^3/m^3)$
S	Emission	$(mg/s \ or \ cm^3/s)$
q_o	Ventilation flow rate	(m^3/s)
t_1	Initial time	(s)
τ_n	Mean age in exhaust $(= V/q_o)$	(s)
V	Volume of air space	(m^3)

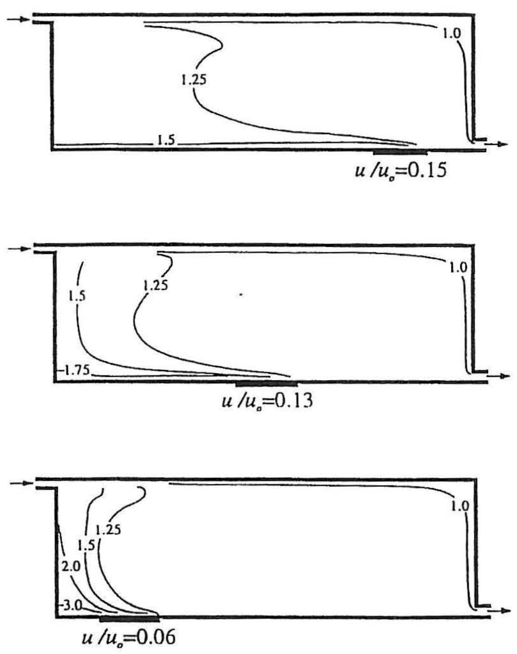
Dynamic concentration model

This model can be used in situations where the flow is considered to be steady and fully mixed with a time-dependent contaminant source. A typical example will be a room ventilated by a constant flow rate and occupied by people in intervals where people are considered to be the contaminant source.

A closer examination of the exponent $-(t-t_1)/\tau_n$ shows that it takes five hours before the steady situation is obtained in the case of a small air change rate of 1 h^{-1} , whereas it will only take one hour to obtain a steady situation in the case of an air change rate of 5 h^{-1} . Therefore, it can be concluded that many contaminant loads as e.g. different levels of people present in a room should be treated by a dynamic concentration model.

The CO_2 emission in a meeting room with 50 people is a typical example. If the room has a volume of 600 m^3 and an air change rate of 2 h^{-1} the model will show that the concentration will be 778 cm^3/m^3 after half an hour and 918 cm^3/m^3 after one hour. A dynamic concentration model seems to be the right model for this situation.

LOCATION OF CONTAMINANT SOURCE



Different locations of the contaminant source
$(h/H = 0.01 \; ; \; L/H = 3.0)$.

Location of contaminant source

The figure shows an example of steady concentration distribution from a line source. It can be seen that the location of the contaminant source is important for the concentration distribution and the concentration level. The line source is located close to the area with the maximum velocity in the occupied zone as shown in the upper sketch, and the maximum value of the concentration has a level of 1.25 - 1.5 c_R in the area below the supply slot. The concentration will increase to a level of 3.0 c_R if the sources are located below the supply slot in the stagnant air as shown in the lower sketch which emphasizes the importance of source location in a room ventilated by mixing ventilation.

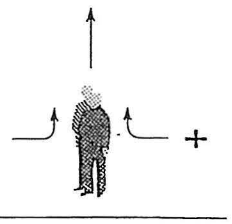
Variables as for example a dimensionless concentration c / c_R at a given location are independent of the Reynolds number in the case of fully developed isothermal flow or fully developed flow with constant Archimedes number (Nielsen, 1981).

The concentration level is not only a function of source location. It is of course also a function of the room geometry and the air distribution system. c_R is equal to S / q_o which means that both the emission source and the ventilation flow rate are important for the dimensioned concentration c.

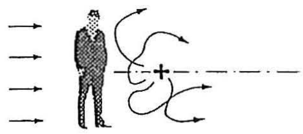
INTERACTION BETWEEN PERSONS AND CONTAMINANT SOURCES

Persons in a ventilated room with contaminant sources present will influence the contaminant distribution.

- The human thermal boundary layer



- Local disturbance of airflow



- Persons' movements
- Respiration
- Persons as heat sources and contaminant sources

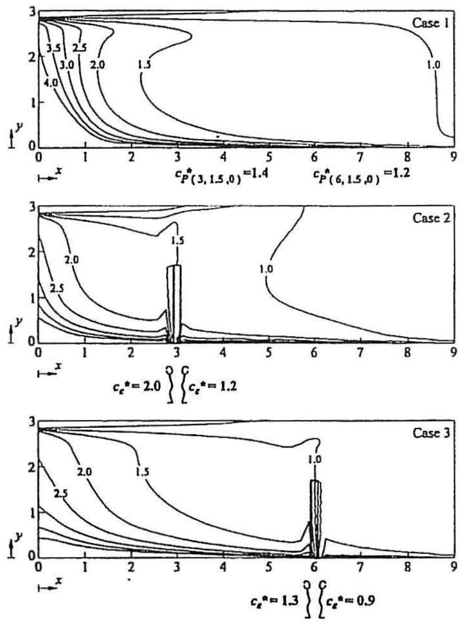
Interaction between persons and contaminant sources

A person will influence the contaminant distribution if concentration gradients are present in the room. The thermal boundary layer around a human body will add a vertical movement to the field, and movement as well as flow restriction from the body may give rise to a vertical and a horizontal flow of contaminants, even in the opposite direction of the main flow.

People and interaction between people will have a great influence on the indoor air quality. This has been studied for some years at Aalborg University, and reference should be made to some of the reports on the subject taking both mixing ventilation and displacement ventilation into consideration as well as interactions between persons and moving persons (Brohus and Nielsen, 1996A, 1996B, Brohus et al., 1996, Brohus, 1997, Bjørn et al., 1997 and Bjørn and Nielsen, 1996).

ROOM WITH MIXING VENTILATION AND ONE PERSON

Contaminant distribution given as $c^* = c/c_R$ in a room with an emission source along the floor. Different exposures are obtained for the person at different positions.



Henrik Brohus, 1997

Room with mixing ventilation and one person

This figure shows one example of the information which can be obtained when a person is added to the flow field in a room. The room is of the same type as shown in the figure *Contaminant distribution*. The contaminant is emitted from the floor surface. The upper figure shows the concentration distribution without a person in the room as well as the concentration c_P^* at the two locations of the person. The two lower figures show that the exposure c_e^* will vary from 0.9 to 2.0 which is outside the variation of c_P^*. It is obvious that both location and orientation of the person are important (Brohus, 1997).

DISPLACEMENT VENTILATION

Air distribution and maximum velocity in the occupied zone

- Principle
- Temperature distribution
- Velocity distribution

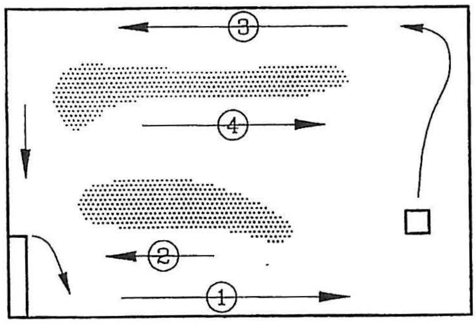
Contaminant distribution

- Two-zone model
- Room with contaminant distribution

Displacement ventilation

The following chapter will discuss different aspects of displacement ventilation. Air distribution and contaminant distribution can not be treated separately, and it is therefore important to consider the vertical temperature gradient and the velocity in the occupied zone as a part of a design procedure which has focus on contaminant distribution and air quality. A design method for the air distribution system is mentioned, and different models for distribution of pollution are discussed ranging from a steady fully mixed two-zone model to the concentration distribution obtained by more detailed research (CFD and full-scale experiments).

FLOW IN A ROOM WITH DISPLACEMENT VENTILATION



Vertical flow

Vertical flow at the diffuser
Plume above the heat source
Cold downdraught

Horizontal (stratified) flow

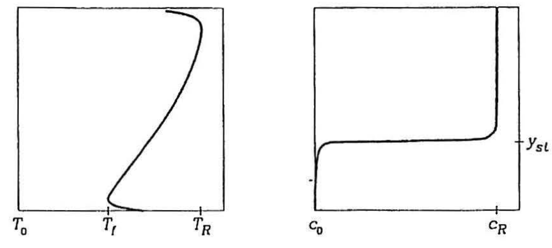
① Radial flow from the diffuser
② Entrainment flow in front of the diffuser
③ Radial flow below the ceiling
④ Entrainment flow into the plume

Flow in a room with displacement ventilation

The airflow to the room is supplied directly into the occupied zone by floor or by wall-mounted diffusers. The plumes from hot surfaces, from equipment and from persons entrain air from the surroundings in an upward movement and the airflow is extracted from the room by return openings in the ceiling. The figure shows a more detailed picture of the flow. Three areas with vertical flow are indicated. The flow from the diffuser has a downward direction because cold air is supplied at the full height of the diffuser. The vertical flow above the heat source obtains momentum from the buoyancy effect on the heated air. A vertical cold downdraught exists at the walls due to a gravity effect on the cooled air close to the surface. The downward flow at the wall may be connected to a detrainment effect where movement in the outer part of the boundary layer will stop at density levels equal to the surrounding density. The temperature distribution will force the remaining flow in the room to be horizontal and stratified. The figure shows four areas with horizontal flow. A stratified radial flow from the diffuser exists at the floor (1), and above this area a return flow (2) is connected to the entrainment in front of the diffuser. The plume above the heat source will generate a stratified radial flow below the ceiling (3). A flow (4) is present in the middle of the room and it is connected to the entrainment into the plume. This flow covers a large part of the room height and it has a temperature which is increasing with the height. Areas with low velocity are shown between the flows of the opposite direction (2), (4) and (3), (Nielsen, 1993 and 1996B).

TEMPERATURE AND CONCENTRATION DISTRIBUTION

Vertical temperature and concentration distribution in a room with displacement ventilation.



Temperature and concentration distribution

The figure shows the vertical temperature distribution and a simplified vertical concentration distribution in cases where some of the heat sources are contaminant sources. The right sketch in the figure shows that the concentration in a lower part of the room has the level c_o corresponding to the supply concentration. The plumes in the room will entrain fresh air (concentration c_o) up to a height where the total vertical volume flow is equal to the supply flow q_o. This height is called the stratification height y_{st}. The plumes continue above this height, and the entrainment will generate a full mixing in the upper region with a concentration c_R corresponding to the concentration in the return flow. More complex profiles with stratified concentration peaks can be obtained when the contaminant sources are connected to weak heat sources (Bjørn and Nielsen, 1996).

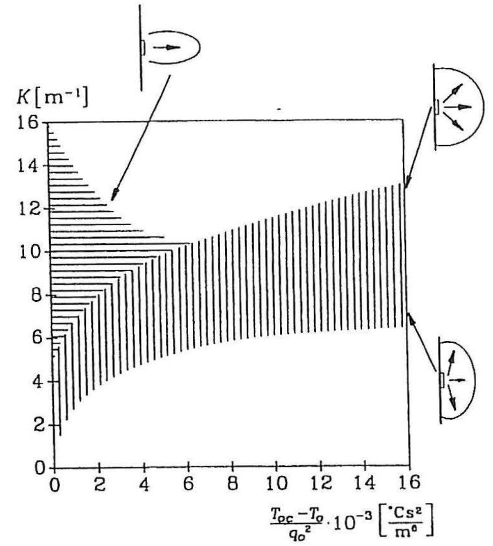
The temperature distribution is described by the energy transport equation, the radiation and the conduction through the surfaces. The temperature distribution influences the flow via the buoyancy term in the vertical momentum equation. The energy transport equation and the transport equation for the contaminant are identical in structure, and it is therefore possible to study the influence of radiation, conduction and buoyancy by comparing the two curves in the figure. The temperature close to the floor T_f is high in comparison with the equivalent concentration distribution. The high level of temperature is due to radiation from the ceiling, and the gradients close to the floor and the ceiling indicate the corresponding heat transfer by convection. The vertical temperature distribution varies almost linearly with height compared with the concentration distribution. This may be the result of an influence from the detrainment at the walls (Nielsen, 1996B).

A PRACTICAL DESIGN METHOD

Velocity distribution

$$u_x = q_o K \frac{1}{x} \qquad [m/s]$$

K- factor versus Archimedes number.



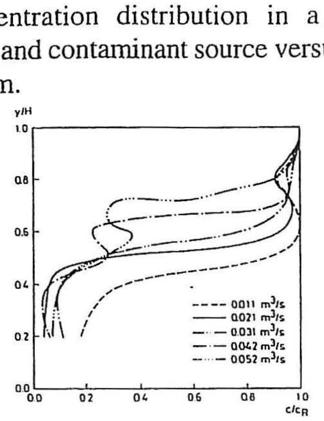
A practical design method

This figure shows a practical method for the calculation of the air velocities in the radial flow from a wall-mounted diffuser. $(T_{oc} - T_o)$ is the temperature difference between the temperature at the height 1.1 m and the supply temperature. The ratio $(T_{oc} - T_o)/q_o^2$ can be regarded as part of the Archimedes number.

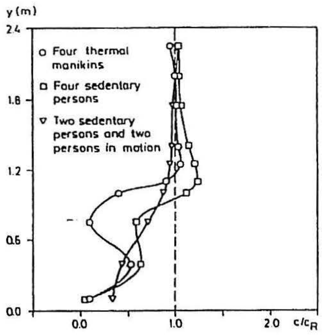
The figure indicates that the first generation of diffusers located in the upper part of the shaded area had a radial distribution of the flow and a high level of the K-value. Some diffusers had even an axial distribution of the flow at low Archimedes number which in that situation gave a further increase in K, but the gravity effect turns the flow into a radial pattern at higher Archimedes numbers. The new generation of diffusers has a distribution with high velocity parallel to the wall and lower velocity perpendicular to the wall (x-direction). This will give the low K-values shown in the lower part of the figure. The upper part of the shaded area in the figure is therefore typical of diffusers with radial/axial distribution of the velocity, and the lower part of the shaded area is typical of a "flat" velocity distribution.

The figure is based on diffusers with a height up to 1.0 m and an area up to 0.5 m^2, and the equation for u_x will be valid at a distance of 1 - 1.5 m from the diffuser (Nielsen, 1992 and 1994B).

Vertical concentration distribution in a room with a combined heat and contaminant source versus volume flow rate to the room.



Concentration distribution in a room with thermal manikins, sedentary persons and persons in motion.



Concentration distribution

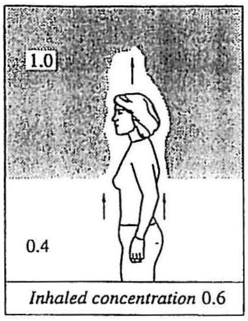
The upper figure shows the vertical concentration distribution in a room where the heat sources also work as a contaminant source (Heiselberg and Sandberg, 1990). The location of the stratification height is clearly indicated by the measurements, and both the lower and the upper zone have a well-mixed concentration distribution. The tracer gas concentration in the upper zone is close to the concentration in the return opening, c_R, and the concentration in the lower zone is 0.1 to 0.3 times c_R which is typical of many situations in highly loaded rooms. It is obvious that the stratification height is a function of the volume flow rate.

Two of the concentration profiles in the upper figure show a small concentration peak just below the main stratification height. This increase is generated by cold downdraught at the wall which has a neutral buoyancy at the height of $y / H \sim 0.5$.

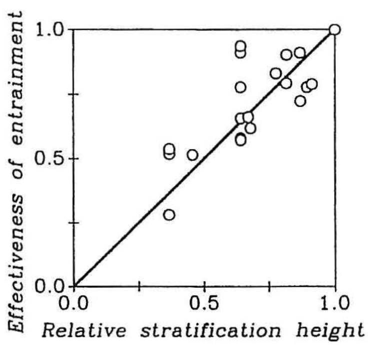
It is important to preserve the stratification in the room when persons are present and in motion. The lower figure shows the vertical concentration distribution with four thermal manikins. A CO_2 tracer gas is released in the plumes above the manikins. The relative concentration distribution for four persons is also shown and the CO_2 concentration is, in this case, the values obtained from the presence of persons in the room. It is shown that two persons in motion are able to smooth the vertical gradient slightly, but it is in all situations possible to observe a stratification of CO_2 (Brohus and Nielsen, 1994).

EXPOSURE IN STRATIFIED SURROUNDINGS

Example.



Effectiveness.



Exposure in stratified surroundings

In principle it is necessary with a high volume flow rate to raise the contaminated upper zone to a level above the breathing zone, but new experience shows that it is possible to accept a reduced stratification level. The figure of the concentration distribution close to a person shows that the convection boundary layer around the person will move air with less contaminant up into the breathing zone. The inhaled concentration is low $\left(c_e^* = 0.6\right)$ compared with the surrounding concentration of 1.0.

The lower figure shows that it is possible to formulate an effectiveness of entrainment which is a linear function of the relative stratification height (Brohus and Nielsen, 1996A).

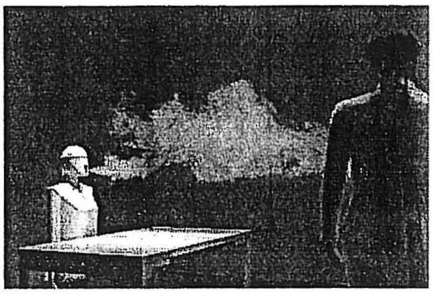
$$\eta_e = \frac{c_p - c_e}{c_p - c_f} \sim \frac{y_{st}}{y_e}$$

In rooms with persons in mainly sedentary working position a stratification height of minimum 1.1 m can be recommended (Nielsen, 1993).

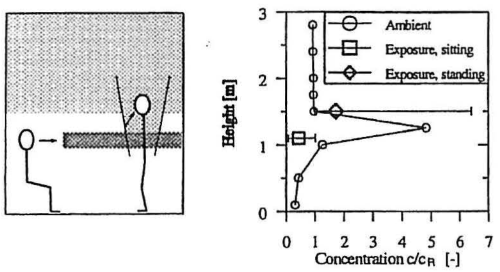
The movements of the persons will to some extent destroy the protecting effect of the boundary layer around the body (Bjørn et al., 1997).

PASSIVE SMOKING IN A DISPLACEMENT VENTILATED ROOM

Visualization of stratified exhaled air



Smoke observations and tracer gas measurments



Passive smoking in a displacement ventilated room

In these experiments it is demonstrated that the exhalation flow may stratify because of density differences, that high concentration can occur in the breathing zone and that the exposure may be much larger than expected. The convective boundary layer close to the human body modifies the concentration field locally so that the inhaled concentration is different from the ambient concentration at the same height. If the exhalation is to give rise to unusually high levels of exposure the stratified layer must be situated below breathing height. Many heat sources will generally give rise to effective mixing. The vertical temperature gradient will play a role with regard to the stratification height of the exhalation (Bjørn and Nielsen, 1996).

Ventilation effectiveness Air exchange efficiency

Efficiency of the air distribution system

The following chapter will discuss the ventilation effectiveness and the air exchange efficiency.

The ventilation effectiveness shows how fast contaminant is removed from a room and how certain areas are influenced by contaminant sources in the room. The ventilation effectiveness is dependent on the air distribution in the room and, therefore, on the type of air terminal devices, geometry of the room, location of any heat source and location of the contaminant source.

The air exchange efficiency shows how fast air is exchanged in a ventilated room. It is dependent on the type of air terminal devices, geometry of room and location of heat sources, but not on the location of the contaminant sources.

Measurements of air exchange efficiency are useful in connection with the general evaluation of an air distribution system, while the ventilation effectiveness can be used for optimizing of a contaminant source location.

Ventilation effectiveness in the occupied zone

$$\varepsilon_{oc} = c_R / c_{oc}$$

Local ventilation index

$$\varepsilon_P = c_R / c_P$$

Mean ventilation effectiveness

$$\bar{\varepsilon} = c_R / \bar{c}$$

Personal exposure index

$$\varepsilon_e = c_R / c_e$$

Tracer gas or relevant contamination in the air supply to the room must be subtracted from the level in the room.

Example:

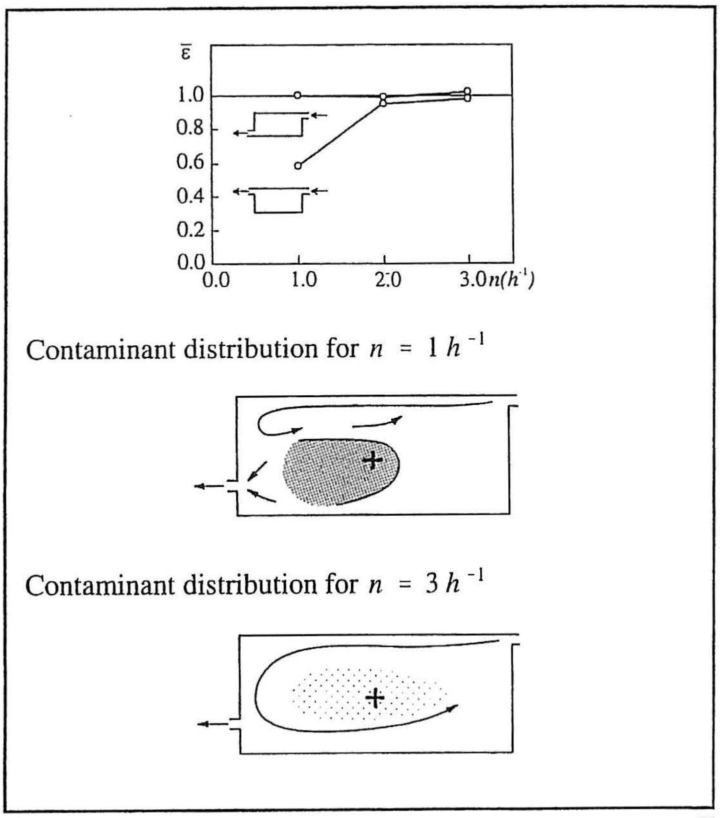
$$\varepsilon_{oc} = \frac{c_R - c_o}{c_{oc} - c_o}$$

Ventilation effectiveness

Four different definitions are commonly used. The ventilation effectiveness in the occupied zone ε_{oc}, the local ventilation index ε_P, the mean ventilation effectiveness $\bar{\varepsilon}$ and the personal exposure index ε_e (Nielsen, 1995, 1996A and Brohus and Nielsen, 1996A).

The number of necessary measurements is different for ε_{oc}, ε_P, ε_e and $\bar{\varepsilon}$. Calculation of the ventilation effectiveness ε_{oc} in the occupied zone includes measurements in several points in the occupied zone as well as measurements of c_R in the return channel. The ventilation index and the personal exposure index can be calculated from two measurements. The mean ventilation effectiveness can also be calculated from two measurements. The return concentration c_R is found during steady state tracer gas release, and the mean value in the room is found by making a complete mixing of room air with a fan after the tracer gas and ventilation system have been shut off.

VENTILATION EFFECTIVENESS AND LOCATION OF RETURN OPENING



Ventilation effectiveness and location of return opening

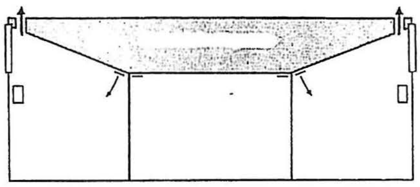
It is well known that the return opening only has a very small influence on the velocity distribution in a room, but it may have large influence on the ventilation effectiveness as shown in this example. The upper figure shows the mean ventilation effectiveness in a room with a wall-mounted supply nozzle and two different locations of the return opening. The flow is isothermal and it is obvious from the measurements that a high location of the return opening will decrease the effectiveness at low air change rates. The flow in the room is not fully developed and the lower part of the room will only obtain air movement if the return opening is located in this region.

The two lower figures are obtained with visualization and they show the flow in the case of small and large air change rates in the room with a low location of the return opening. The contaminant distribution around the source (in the middle of the room) is moved towards the return opening as a displacement flow when the air change rate is small, and recirculation takes only place in the upper part of the room. The contaminant is fully mixed in the room when the air change rate is large as indicated in the lower figure. Both situations will give a high mean ventilation effectiveness (Heiselberg and Nielsen, 1987).

A ROOM WITH LOW VENTILATION EFFECTIVENESS (LANGAGERVEJ)

A room with two persons, a filled ashtray and a door frame newly tightend with sealant.

Situation:



Experience:

20 - 40 persons, 10 - 20 ashtrays and 50 - 100 m of sealant.

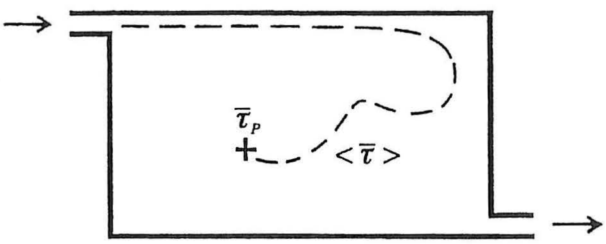
A room with low ventilation effectiveness

The following example shows the importance of a high ventilation effectiveness and it demonstrates that a high indoor air quality is not only a question of low emission but also of good ventilation.

The examined room is heated by an air heating system with supply opening located in the upper part of the room. The return opening and a small radiation (to prevent cold downdraught from a high located window) are also located in the upper part of the room. The consequence of this layout is the fact that all air movement and air exchange take place in the upper part of the room in the heating situation and it is possible to measure a ventilation effectiveness of 0.05 - 0.1 in the occupied zone.

Two persons in this room with - for example - a filled ashtray and a newly sealed door frame will have the *experience* of 20 - 40 persons, 10 - 20 ashtrays and 50 - 100 m of sealant at a very small floor area. It is the consequence of an insufficient ventilation but the situation may often be wrongly expressed as high emission from the sealant etc.

THE AGE CONCEPT AND AIR EXCHANGE EFFICIENCY



$\bar{\tau}_P$	Local mean age of air in point *P*
$\langle\bar{\tau}\rangle$	Room mean age
$2\langle\bar{\tau}\rangle$	Air change time
τ_n	Nominal time constant

Air exchange efficiency

$$\varepsilon_a = \frac{\tau_n}{2\langle\bar{\tau}\rangle}$$

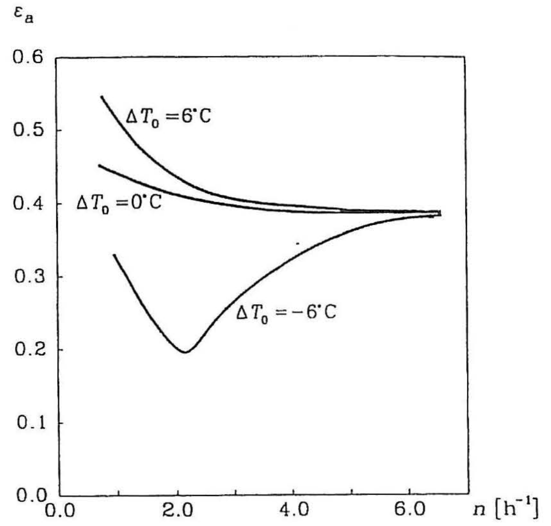
The age concept and air exchange efficiency

The figure illustrates the age concept (Sandberg and Sjöberg, 1983). The air has the age zero when it enters the room. The age distribution in point *P* is given by a statistical distribution and the local mean age is called $\bar{\tau}_P$. The mean value is a function of the location of *P*. The mean age of all air present in the room can be found from the distribution of $\bar{\tau}_P$ and it is called $\langle\bar{\tau}\rangle$.

The average time it takes to replace the air present in the room is equal to twice the mean age of all air in the room, $2\langle\bar{\tau}\rangle$. The mean age of the air when it leaves the room is always equal to a nominal time constant τ_n of the ventilated space and it is defined as room volume *V* divided by the ventilation flow rate q_o. The air exchange efficiency ε_a is defined as the ratio between the nominal time constant τ_n of the ventilated space and the air exchange time $2\langle\bar{\tau}\rangle$. The nominal time constant τ_n is the theoretically shortest exchange time for all air in the room in the case of a piston flow and $2\langle\bar{\tau}\rangle$ is the real exchange time in a given situation. The air exchange efficiency ε_a will therefore have values below 1.0 for all practical air distribution systems, and it can be shown that a system with perfect mixing will have a value of 0.5.

AIR EXCHANGE EFFICIENCY

Mixing ventilation.



Air exchange efficiency versus air change rate in case of heating, cooling and isothermal flow.

Air exchange efficiency

The air exchange efficiency ε_a is useful for a global study of a air distribution system. The figure shows the outcome of measurements in a room ventilated by mixing ventilation and a wall-mounted air terminal device. The experiments are performed up to a very high value of n (outside the comfort range) in order to study the effect of a fully developed flow. The air exchange efficiency seems to have an asymptotic value of ~ 0.4. This is in good agreement with the fact that 0.5 is the highest possible value to obtain in case of full mixing. The temperature difference ΔT_o is the difference between return temperature and supply temperature. ΔT_o equal to 6°C corresponds to the cooling case.

The air exchange efficiency will always be lowest for heating because the return opening has a high location which is confirmed by the experiments. The figure also shows that the flow obtains a semiisothermal character at high air change rates corresponding to a situation with a small Archimedes number.

MINIMUM AIR FLOW RATE

European Prestandard CEN prENV 1752.

- Design criteria for spaces in different types of buildings given as ℓ/s (m^2 floor) for three categories. Low-polluting building materials and furnishing are assumed.
- Design criteria determined from the calculation of perceived air quality.

Minimum air flow rate

Minimum air flow rates are given in the European Prestandard CEN prENV 1752 (1994) drawn up by the Technical Committee CEN/TC 156.

The design criteria for the required ventilation are based on the assumption that low-polluting building materials and furnishings are systematically selected, that an assumed occupancy is used, that smoking is not permitted, that the ventilation effectiveness is one and that outdoor air of excellent quality is available. If smoking occurs additional ventilation is required.

The categories A, B and C correspond to 15, 20 and 30 percentage of dissatisfied with respect to indoor air quality.

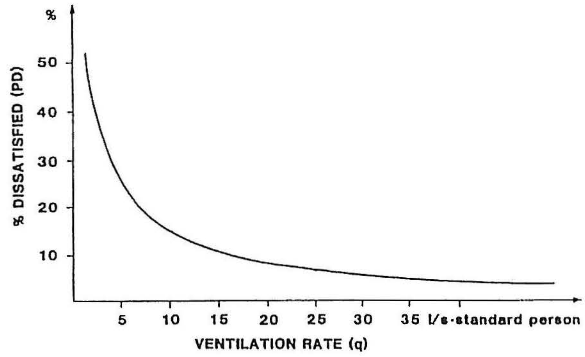
The design criteria can also be determined from the calculation of perceived air quality.

Perceived air quality concept allows to take into account the effect of different pollution sources on occupants. The strength of these sources is evaluated as person-equivalence referring to the sensory load of one persone (1 olf).

Perceived air quality may be expressed as:

the percentage of Persons Dissatisfied just after entering a space (PD)

The figure shows PD as a function of the ventilation rate per standard person.



Fanger

Perceived air quality

Humans perceive the air by two senses. The olfactory sense is situated in the nasal cavity and is sensitive to several hundred thousands of odorants in the air. The general chemical sense is situated all over the mucous membranes in the nose and the eyes and is sensitive to a similarly large number of irritants in the air. It is the combined response of these two senses that determines whether the air is perceived as being fresh and pleasant or stale, stuffy and irritating, ECA (1992).

Perceived air quality may be expressed as the percentage of dissatisfied, i.e. those persons who perceive the air to be unacceptable just after entering a space. For air polluted by human bioeffluents the figure shows the percentage of dissatisfied as a function of the ventilation rate per standard person (average sendentary adult office worker feeling thermally neutral). The pollution generated by such a standard person is called one olf.

The strength of most pollution sources may be expressed as person equivalents, i.e. the number of standard persons (olf) required to make the air as annoying (causing equally many dissatisfied) as the actual pollution source.

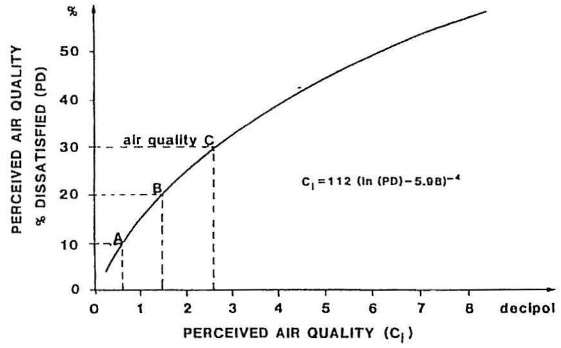
The curve is given by the following equations:

$$PD = 395 \cdot \exp(-1.83 \cdot q^{0.25}) \quad \text{for } q \geq 0.32 \ \text{l/s} \cdot \text{olf}$$

$$PD = 100 \quad \text{for } q < 0.32 \ \text{l/s} \cdot \text{olf}$$

Perceived air quality in decipol (dp)

$$1\ \text{dp} = \frac{1\ \text{olf}}{10\ \ell/s} \qquad c_i = \frac{G}{q_o}$$



Fanger

Perceived air quality

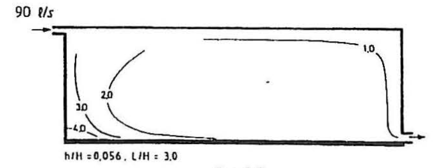
Perceived air quality may be expressed in decipol (dp), where 1 dp is the air quality in a space with a pollution source strength of 1 olf, ventilated by 10 l/s of clean air, i.e. 1 dp = 0.1 olf / (l/s) (Fanger, 1988).

The figure shows three categories of air quality

Category A	10%	PD	0.6 dp
Category B	20%	PD	1.4 dp
Category C	30%	PD	2.5 dp

$$c_R = 10 \frac{G}{q_o}$$

c_R, G, and q_o are perceived air quality (decipol), pollution source strength (olf) and ventilation rate (ℓ/s), respectively.



$$\varepsilon_P = c_R / c_P \qquad c_P = 10 \frac{G}{q_o \varepsilon_P}$$

Local air quality in a room with more pollution sources.

$$c_P = c_{od} + \frac{10}{q_o}\left(G_{eq} + \frac{G_1}{\varepsilon_{P1}} + \frac{G_2}{\varepsilon_{P2}} + \ldots\right)$$

c_P, c_{od} and G_{eq} are local perceived air quality (decipol), outdoor air quality (decipol) and pollution source strength from ventilation equipment (olf).

Indoor air quality

The first equation shows the prediction of the return concentration c_R in the general situation or the room concentration when full mixing is assumed. A source of 9 olf and a ventilation flow rate of 10 l/s will thus give a room concentration of 1 dp in the case of full mixing of the room air.

The figure shows the pollution distribution in the real situation in a room ventilated by mixing ventilation and a source distributed along the floor. The variation of the pollution concentration is between 1 and 4 dp in the occupied zone. It is possible to find the local air quality from several pollution sources if it is assumed that the sensory sources are additive as shown in the last equation where the individual ventilation effectiveness $(\varepsilon_{P1}, \varepsilon_{P2}, \ldots)$ has been taking into account.

POLLUTION LOAD CAUSED BY OCCUPANTS

	Sensory pollution load olf/occupant
Sedentary, 1-1.2 met	
0% smokers	1
20% smokers	2
40% smokers	3
100% smokers	6
Physical exercise	
Low level, 3 met	4
Medium level, 6 met	10
High level (athletes), 10 met	20
Children	
Kindergarten, 3 - 6 years, 2.7 met	1.2
School, 14-16 years, 1-1.2 met	1.3

ECA (1992)

Pollution load caused by occupants

It should be realized that smoking may increase the pollution load by a factor of 6 and that a high level of physical activity may increase the load by a factor of 20.

Design criteria determined from the calculation of perceived air quality

The following example shows the calculation of the required ventilation rate (category C) for an office building. Low-polluting building materials and furnishings are systematically selected providing a pollution load of 0.1 olf / (m^2 floor). Efficient mixing ventilation $(\bar{\varepsilon} = 1.0)$ is assumed and smoking is not permitted. The outdoor air quality (0 dp) is excellent. The occupancy is 0.1 person/ (m^2 floor).

Sensory pollution load from occupants and the building is $G = 1 \cdot 0.1 + 0.1 = 0.2$ olf (m^2 floor).

Category C corresponds to a perceived air quality c_p of 2.5 and the equation $q_o = 10\, G / c_p$ gives the following air flow rate:

$$q_o = 10 \cdot 0.2 / 2.5 = 0.8 \text{ l/s } (m^2 \text{ floor})$$

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ISSN 1395-8232 U9903

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