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Ventilation '85, edited by H.D. Goodfellow, 1986
Elsevier Science Publishers B.V., Amsterdam — Printed in The Netherlands

VENTILATION BY DISPLACEMENT - CHARACTERIZATION AND DESIGN IMPLICATIONS

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ABSTRACT

Ventilation by displacement is described in terms of ventilation efficiency and quantyfied by means of a two-zone flow and diffusion model. The practical procedure is by means of adequate diffusers firstly, to "hit" the persons with the ventilation air and secondly, to displace both air and contaminants out of the zone of occupation and avoid recirculation. This procedure has a firm basis in the research work on ventilation efficiency that is carried out in Norway and Sweden during the last years as well as in current theory practice. The paper describes practical design implications as well as calculation procedures based on the two-zone model. A system evaluation model is also treated.

INTRODUCTION

Ventilation for contaminant controlinvolves, as a first step, the use of elimination technics, i.e. to apply any practical and economical method to prevent contaminant exposure, say:

- o Using building materials that do not emit contaminants.
- o Close up processes and depressurize the enclosure.
- o Change the process.
- o Change to less toxic solvents.
- o Apply hoods and other local exhaust methods, etc.

When above are done there is still some residual contaminant emission to be handled by the ventilating system.

The minimum outdoor air requirement is, however, equal to the flow rate of make-up air required by the elimination solutions. The right area for supplying the ventilation air is in the space occupied by the people. In principle the procedure is firstly, to "hit" the persons by the ventilation air and secondly, to displace both air and contaminants out of the zone of occupation and avoid recirculation of contaminated air. In this way, the residence time for the air in the room is minimized and the ventilation potential maximized.

In other words, the main objects of ventilation for occupants in a building are to replace "old" and contaminated air in the zone of occupation with "new" fresh air as quick as possible and to remove generated contaminants as quick as possible. The words "quick", "new" air and "old" air are related to time and can be quantified through time parameters.

The air renewal process and the contaminant removal process are generally not identical. Research work in Norway and Sweden has proved that criteria for effective ventilation can be defined through the age concept (1,2).

THE PHYSICAL MEANING OF VENTILATION EFFECTIVENESS

The air exchange efficiency

One way of characterizing the ventilation process is to observe the frequency of air change in a ventilated enclosure. Efficiency can then be assessed as comparing the actual exchange frequency with an ideal one. The inverse of frequency is a time constant (turnover time, transit time, residence time or mean age). The room volume, V, devided by the air flow, $\dot{V}$, expresses the time it on average takes for the inflowing ventilation air to flow from the air supply unit to the exhaust grill. This time is called the transit time for the ventilation air flow through the room, τ_n:

$$\tau_n = \frac{V}{\dot{V}} \tag{1}$$

This is a real and measureable quantity in any ventilating system. It is system invariant and thus independant of the air flow patterns in the room. However, the quantity represents the shortest possible average residence time for the total air mass in the room.

The average residence time for the air in the room, τ_r, which is different from the transit time for the air flow is closely related to the flow patterns for the air in the room and is hence a suitable system parameter,fig.1. The ventilation air is a carrier of the contaminants out of the room and the relative ventilation potential increases as the residence time decreases. A physical explanation for this is that the residence time for the room air increses from the ideal case when air short-circuits to the exhaust and the short-circuited air does not replace older, contaminated air in the room.

If there is complete mixing of the room air, the probability for a "new lump" of air entering the room to stay there, and hence replace an "older lump" of air, is the same as the probability for the "new lump" of air to leave the room without replacing any "old lump" of air. The result is that the average residence time for the room air is two times the transit time for ventilation air flow. This can be exactly calculated through a statistical analyses(1,2) and can also be exactly measured in real situations. If there is an ideal unidirectional flow (plug flow), no shortcircuiting takes place and the residence time for the room air is exactly the transit time(ideal case). Tendency to unidirectional flow is called desplacement flow.

In Scandinavia (3) the average air exchange efficiency is defined as the ratio between the transit time for the ventilation air flow and the average resi-

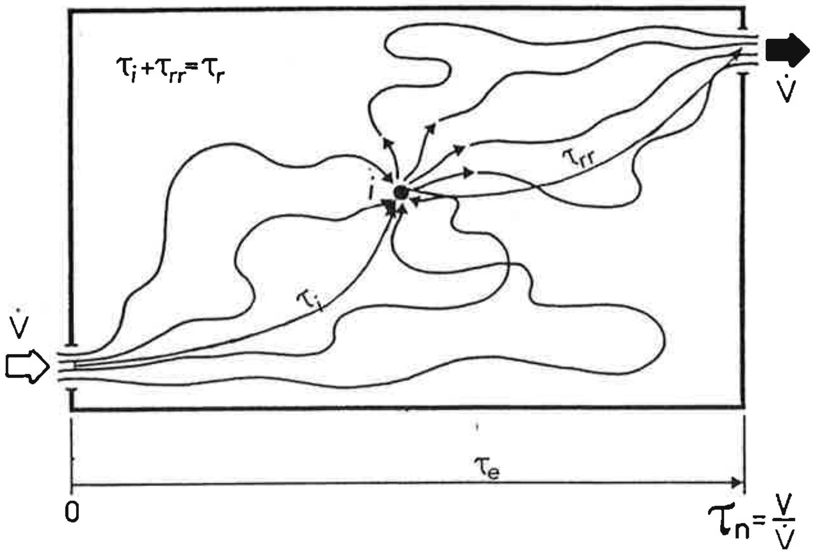

Fig.1. Time parameters for ventilating systems. The ventilation air-flow and the room air.

dence time for the room air:

$$\langle\eta_a\rangle = \frac{\tau_n}{\tau_r}\ 100\ \% \qquad (2)$$

Stagnant flow: $0 < \langle\eta_a\rangle < 50\ \%$

Complete mixing: $\langle\eta_a\rangle = 50\ \%$

Displacement flow: $50\ \% < \langle\eta_a\rangle < 100\%$

Unidirectional flow: $\langle\eta_a\rangle = 100\%$

Now, this average air exchange efficiency does only reflect the gross flow patterns. The conditions in the zone of occupation should also be considered. If the air arrives at the zone of occupation quicker than other parts of the room the mean age of the air there is lower than the room average mean age. If it arrives later the age is higher than the average, fig.1 (age is the same as arrival time). Measureing the local mean age, τ_i, and comparing it with the average age of the total air mass reflects the relative ventilation potential in the zone of occupation. The average age of the room air, $\langle\tau_i\rangle$, is always half the residence time. The ratio between the average and the local age is in Scandinavia called the local air exchange indicator, ε_a:

$$\varepsilon_a = \frac{\tau_r}{2\ \tau_i}\ 100\ \% \qquad (3)$$

The local efficiency is then: $\langle\eta_a\rangle\ \varepsilon_a$

All time parameters can be determined using a tracer gas technique. Instead of using the traditional slope of the curve method from decay tests, the parameters are calculated from the area above the concentration curves from a step-up test, and from the area under the curve from a step-down test.

Ventilation effectiveness

For contaminants it is necessary to have additional evaluation parameters. Some contaminant sources are more or less evenly distributed throughout the room and distributes approximately like the ventilation air, but most contaminants develop their own flow patterns which are superimposed by the ventilation air flow pattern.

The average transit time for the contaminant flow through the room, τ_t^c, can be compared with the transit time for the ventilation air flow, fig.2. The ratio between those two times happens to be the ratio between the concentration of

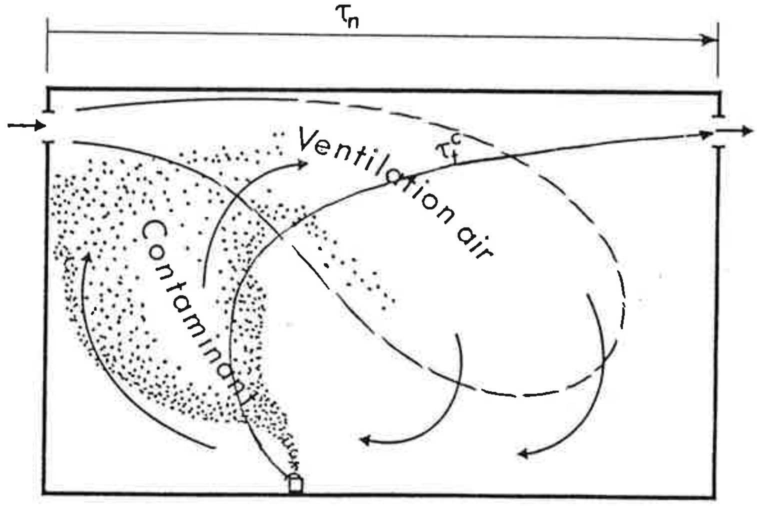

Fig.2. Time parameter for the contaminant flow. Ventilation air and contaminants have different flow patterns.

contaminants in the exhaust air and the average concentration of the contaminants in the room at steady state. This ratio is in Scandinavia called the average ventilation effectiveness:

$$\langle \varepsilon_v^c \rangle = \frac{C_e(\infty)}{\langle C_i(\infty) \rangle} = \frac{\tau_n}{\tau_t^c} \qquad (4)$$

Complete mixing results in $\langle \varepsilon_v^c \rangle = 1$. A design goal should be to achieve a value greater than one. It is now important to bear in mind that if complete mixing is achieved, the probability for the contaminants to stay in the room is equal to the probability for the contaminants to leave the room. For this reason it is reasonable to say that the efficiency is 50 %, not 100 %. When the transit time for the contaminant flow is zero the average ventilation effectiveness is infinit. In this situation the probability for the contaminants to leave the room is 1 and to stay it is zero. In terms of contaminant removal efficiency it should be 100 %. This may be expressed as:

$$\langle \eta_v^c \rangle = \frac{\langle \varepsilon_v^c \rangle}{1+\langle \varepsilon_v^c \rangle}\ 100\ \% \qquad \text{or} \qquad \langle \varepsilon_v^c \rangle = \frac{\langle \eta_v^c \rangle}{100 - \langle \eta_v^c \rangle} \tag{5}$$

Complete mixing: $\langle \eta_v^c \rangle = 50\ \%$

Above should also clearly explain the difference between effectiveness and efficiency.

Instead of using the efficiency concept, it is in Scandinavia (3) introduced a quantity called the air quality index, defined as the ratio between the concentration of contaminants in the exhaust duct and the concentration of contaminants in the zone of occupation:

$$\varepsilon_v = \frac{C_e(\infty)}{C_i(\infty)} \tag{6}$$

This parameter can be used as a design guideline. A design goal is to maximize the parameter. It is obvious that creating a tendency to unidirectional flow decreases τ_t^c and hence increases both the ventilation effectiveness and the ventilation index. This is also a practical experience.

Surplus heat is a contaminant and can be treated in the same way as described above. Instead of concentrations the temperatures can be used. Doing this, one has to subtract the supply air temperature in the expressions.

VENTILATION BY STRATIFICATION AND DISPLACEMENT

Ventilation by stratification and displacement means to design ventilation systems to utilize thermal or density stratification to create a tendency to unidirectional flow i.e. to create displacement flow.

Design Implications

General. The displacement flow principle is the most efficient design principle (4,5,6) for ventilating system for two main reasons:

1. It improves the air renewal and contaminant removal speed.

2. It assists in maintaining favourable concentration gradients of the contaminants generated in the room.

There are several ways of accomplishing displacement ventilation in a ventilated room.

Piston flow. The most obvious way of creating unidirectional flow is to supply air through one surface and to extract it at the opposite. This principle requires that disturbances like bouyancy forces and momentum fluxes from contaminant sources have to be overcome by the piston flow. Practical experience has shown that this requires a piston velocity from .25 m/s and up. The principle is therefore air consuming and of high cost. Areas of application of this principle are clean rooms in hospitals, electronic and space craft industry etc.. Further discussion of this technique is left out here because the design principles should be fairly well known.

Thermal stratification. Practical design principles for displacement ventilating systems for normal use is, rather than to overcome natural forces, to utilize buoyancy, momentum fluxes from contaminant sources etc. The displacement direction can either be vertical-up or vertical-down. Vertical-up flow direction is accomplished by supplying ventilation air to the zone of occupation with a lower temperature than the temperature in this zone, and to extract it at ceiling level. Vertical-down flow direction is accomplished in the opposite way, i.e. by supplying ventilation air under the ceiling heated to a temperature above the temperature in the zone of occupation and extract it at floor level.

In applying vertical up displacement the air is filling the room from below due to gravity and "older" air is displaced upwards. Any heat source in the zone of occupation creates convective currents and contributes to carrying the air to the upper zone. In this way a temperature stratification will be formed, creating two more or less distinct flow regions. The "new" air should spread through the zone of occupation before being carried to the upper zone (zones).

To maintain the best stratification effect, all convective plumes should be feeded by the supplied "new" air to a height equal to the height of the zone of occupation. Conditions impairing the air exchange efficiency and the ventilation effectiveness are downward convective currents along surfaces that are colder than the air in the room. To supply less air than the necessary make up air for the upward convective currents causes the same effect. Downdraught from the upper zone reduces the effectiveness, and at the same time, the necessary requirement of ventilation air to keep a certain stratification height. This means among other things that keeping a certain stratification height requires generally less supply of "new" air in the winter than in the summer. Unfortunately, the effectiveness is consequently lower in the winter.The ventilation effectiveness is at its highest when all plumes carrying contaminants (originating from a contaminant source) are flowing directly to the exhaust.

In applying vertical-down displacement direction it is important to bear in mind that this principle has to overcome all convective currents from the heat sources. The principle is advantages only when the main contaminant sources are denser than the room air, are located mainly below the breathing zone and there is a minor effect from heat sources in carrying the contaminants upwards.

If there is weak or negative contaminant buoyancy, where also the thermal conditions in the room are taken into account, and there is little contaminant production in the upper zone, are situations where vertical down displacement should be utilized. Properly designed the zone of occupation, i.e. the breathing zone, may then be located in the upper zone.

Calculation procedures

Two-zone model. It has been justified (7) to base calculations for designing displacement ventilating systems on a two-zone flow model, fig.4. An important prerequisit for the calculations is that it is assumed that the air and contaminants are well mixed within each zone. Between the zones the air recirculation is characterized through the air exchange parameter β_{12}. This parameter quantyfies the relative air flow from zone 1 to zone 2. The absolute value of the air flow is $\beta_{12}\dot{V}$. The numerical values of the effectivenesses thus calculated are generally conservative, i.e. they are lower than in the practical case.

The basic equations and formulas for calculating concentrations and effectivenesses are based on the following mass balance equations:

$$
\begin{aligned}
d\bar{C}_1/dt &= a_{10} + a_{11}\bar{C}_1 + a_{12}\bar{C}_2 \\
d\bar{C}_2/dt &= a_{20} + a_{21}\bar{C}_1 + a_{22}\bar{C}_2
\end{aligned}
\qquad (7)
$$

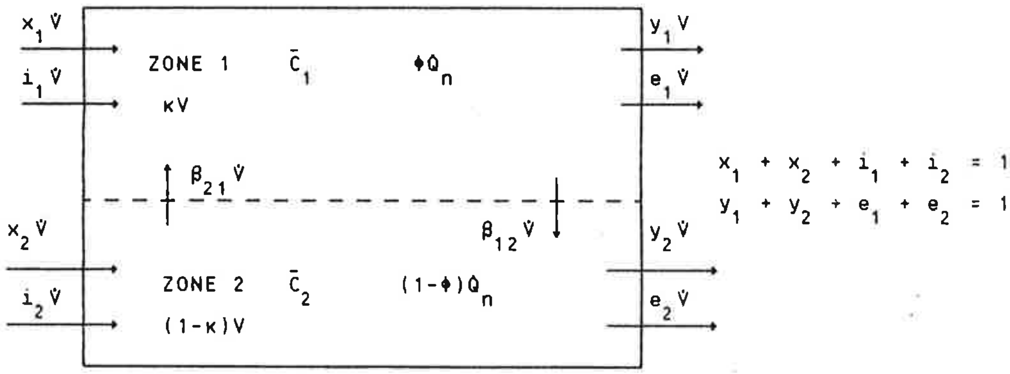

Fig.3. The two-zone flow and diffusion model.

Constants used in equation 7:

$$a_{10} = \frac{x_1 C_s}{\kappa} n \;;\; a_{11} = -\frac{y_1 + e_1 + B_{12}}{\kappa} n \;;\; a_{12} = \frac{x_1 + x_2 - (y_2 + e_2) + B_{12}}{\kappa} n$$

$$a_{20} = \frac{x_2 C_s}{1-\kappa} n \;;\; a_{21} = \frac{B_{12}}{1-\kappa} n \;;\; a_{22} = -\frac{x_2 + i_2 + B_{12}}{1-\kappa} n$$

$\dot{Q}_n$ = Net load to the room of either chemical contaminants or surplus heat.
ϕ = Fraction of the load that is released in zone 1.
κ = Fraction of the room volume belonging to zone 1

i = infiltration
e = exfiltration
x = mechanically supplied ventilation air
y = mechanically exhausted air
Subscript 1 = belonging to zone 1
Subscript 2 = belonging to zone 2
Subscript e = exhaust air
Subscript i = internal air (total air volume for the room)
Subscript s = supply-air
< > = room average
Overbar - = time mean
Subscript 12 = from zone 1 to zone 2
Subscript 21 = from zone 2 to zone 1
Superscript d= decay from an initially well-mixed room , step-down
Superscript s= step-up, tracergas injected well mixed to mechanically supplied ventilation air.

The solution for a step-up or a step down in contaminant production or tracer gas supply is:

$$\bar{C}_1(t) = K_1 e^{n\sigma_1 t} + K_2 e^{n\sigma_2 t} + \bar{C}_1(\infty)$$
$$\bar{C}_2(t) = K_1 {}^1k_2 e^{n\sigma_1 t} + K_2 {}^2k_2 e^{n\sigma_2 t} + \bar{C}^2(\infty) \quad (8)$$

$$\bar{C}_1(\infty) = \frac{a_{10}a_{22} - a_{20}a_{12}}{a_{12}a_{21} - a_{11}a_{22}} \;;\; \bar{C}_2(\infty) = \frac{a_{20}a_{11} - a_{10}a_{21}}{a_{12}a_{21} - a_{11}a_{22}} \quad (9)$$

$$\sigma_1 = \frac{1}{2n}\left[(a_{11} + a_{22}) + \{(a_{11} - a_{22})^2 + 4a_{12}a_{21}\}^{1/2}\right]$$

$$\sigma_2 = \frac{1}{2n}\left[(a_{11} + a_{22}) - \{(a_{11} - a_{22})^2 + 4a_{12}a_{21}\}^{1/2}\right]$$

$${}^1k_2 = \frac{n\sigma_1 - a_{11}}{a_{12}} \;;\; {}^2k_2 = \frac{n\sigma_2 - a_{11}}{a_{12}}$$

$$K_1 = \Delta\bar{C}_1 \frac{{}^2k_2 - \dfrac{\Delta\bar{C}_2}{\Delta\bar{C}_1}}{{}^2k_2 - {}^1k_2} \qquad K_2 = \Delta\bar{C}_1 \frac{\dfrac{\Delta\bar{C}_2}{\Delta\bar{C}_1} - {}^1k_2}{{}^2k_2 - {}^1k_2}$$

$$\Delta\bar{C}_1 = \bar{C}_1(0) - \bar{C}_1(\infty) \quad ; \quad \Delta\bar{C}_2 = \bar{C}_2(0) - \bar{C}_2(\infty)$$

The average concentration in the total exhaust air, included exfiltration, is calculated from the following expression:

$$\bar{C}_e(t) = (y_1 + e_1)\bar{C}_1(t) + (y_2 + e_2)\bar{C}_2(t) \qquad (10)$$

Time constants are calculated either from the time function of concentrations of the real contaminants or of tracer gases. One necessary condition for calculating time constants for the ventilation air is that the starting condition should either be zero concentration (step-up) or uniform concentration (step-down). When tracer gas is injected in the supply air it should be uniformly mixed with the ventilation air upon entering the room.

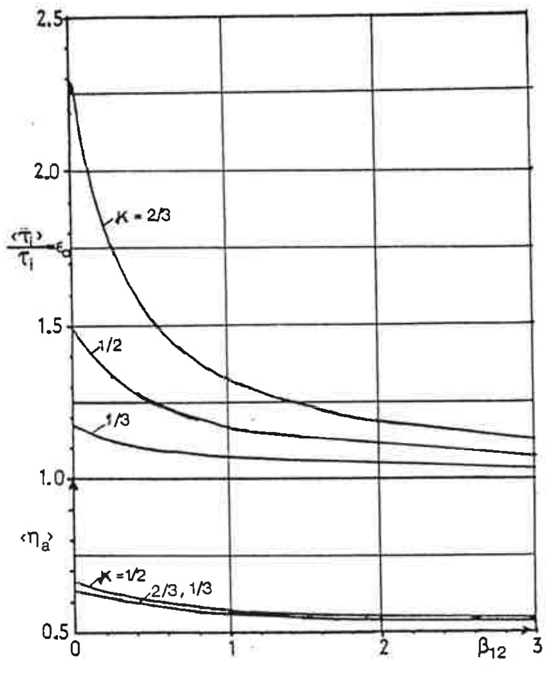

Fig.4. Two-zone model. Curves showing the air exchange efficiency. $x_1 + i_1 = 0$; $y_1 + e_1 = 1$

$$\tau_e \text{ or } \tau_n = \begin{cases} \int_0^\infty (C_e^d(t)/C_e^d(0))\,dt & \text{(step-down)} \\ \int_0^\infty (1 - C_e^s(t)/C_e^s(\infty))\,dt & \text{(step-up)} \end{cases}$$

$$\tau_i \quad = \text{as above only subscript e is excanged with subscript i} \qquad (11)$$

$$\langle\tau_i\rangle \quad = \Big(\sum_1^n \tau_i\Big)/n \qquad \text{or}$$

$$\langle\tau_i\rangle \quad = \begin{cases} [\int_0^\infty (C_e^d(t)/C_e^d(0))t\ dt]/\tau_n & \text{(step-down)} \\ [\int_0^\infty (1-C_e^s(t)/C_e^s(\infty))t\ dt]/\tau_n & \text{(step-up)} \end{cases}$$

The parameters that determine the air exchange efficiency are β_{12} and κ. The concentrations and the ventilation effectiveness are, in addition to above parameters, determined by Φ and $\dot{Q}_n$. The air exchange efficiency and the ratio beween average age and local age are shown in fig.4 as a function of β_{12} for different values of κ. The ventilation effectiveness and the ratio between average and local concentrations (local means the zone of of occupation, in general zone 2) are given in fig.5 as a function of β_{12} for different values of κ and Φ. These figures show the importance of determining the air exchange between the zones, as well as the size of the zones and the load distribution. One main task for the designer is concequently, in addition to determine the load $\dot{Q}_n$, to determine β_{12}, κ and Φ. According to what is previously mentioned, this problem mainly consists of determining the convective currents and the source characteristics.

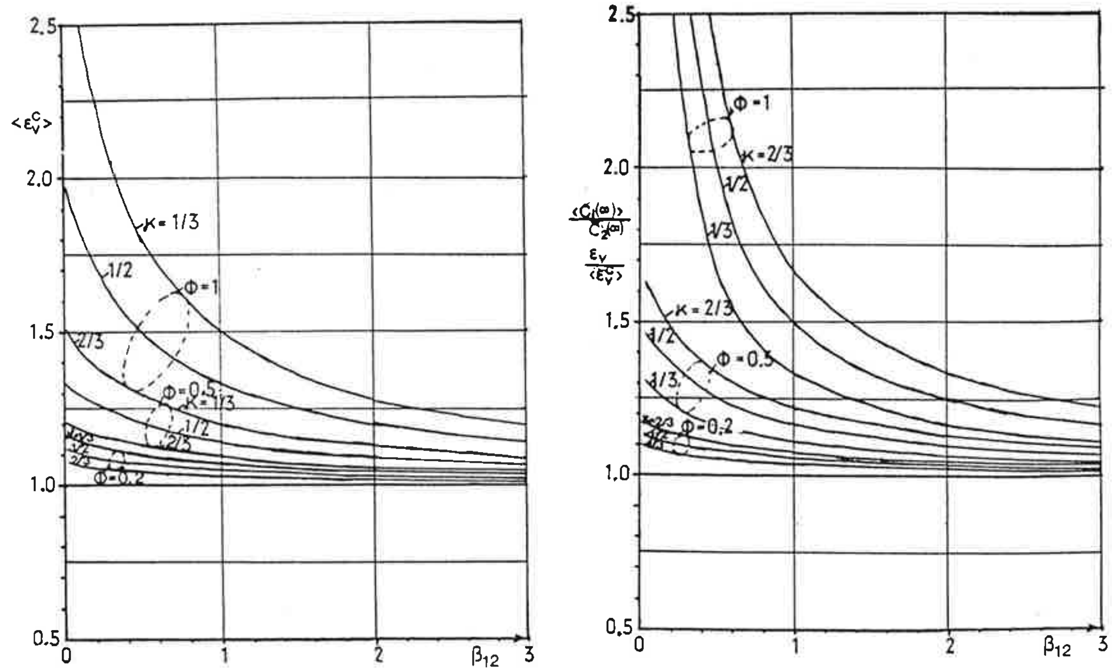

Fig.5. Two-zone model. Curves showing the ventilation effectiveness. Average and local performance. $x_1+i_1 = 0$; $y_1+e_1 = 1$.

<u>Convective flows.</u> Relevant formulas for calculation of flow rates, velocities and excess temperatures in convective flows are, for the purpose of evaluating β_{12}, stratification heights, etc., the following (10):

Point heat sources:

$$\dot{V}_k = 0{,}19[g/(\varrho c_p T)]^{1/3}\, Q_k^{1/3}(x + x_p)^{5/3} \qquad (m^3/s) \qquad (12)$$

$$u_m = 4{,}27[g/(\varrho c_p T)]^{1/3}\, [Q_k/(x + x_p)]^{1/3} \qquad (m/s) \qquad (13)$$

$$\Delta T_m = 7{,}56(T/g)^{1/3}[Q_k/(\varrho c_p)]^{2/3}(x + x_p)^{-5/3} \qquad (K) \qquad (14)$$

Line heat sources:

$$\dot{V}_k = 0{,}47[g/(\varrho c_p T)]^{1/3}\, \dot{q}_k^{1/3}(x + x_p) \quad (m^3/s \text{ og m source length}) \qquad (15)$$

$$u_m = 2{,}22[g/(\varrho c_p T)]^{1/3}\, \dot{q}_k^{1/3} \qquad (m/s) \qquad (16)$$

$$\Delta T_m = 2{,}6(T/g)^{1/3}[\dot{q}_k/(\varrho c_p)]^{2/3}(x + x_p)^{-1} \qquad (K) \qquad (17)$$

Convective currents along cooled/heated surfaces:

$$\dot{V}_k = 0{,}011\ (g\ \Delta T_f/T)^{0.4}\ z^{1.2} \quad (m^3/s \text{ og m width}) \qquad (18)$$

$$u_m = 0{,}54(g\ \Delta T_f z/T)^{1/2} \qquad (m/s) \qquad (19)$$

Q_k = convective heat output (kW)
$\dot{q}_k$ = convective heat output (kW/m)
x = vertical distance from source (m)
x_p = vertical distance to virtual line or point origin (m)
z = height of heated/cooled surfaces (m)
g = acceleration due to gravity (9,81 m/s^2)
T = temperature (K)
ΔT_f = difference in temperature between air and surface (K)
ϱ = density (kg/m^3)
c_p = specific heat capacity for air, constant pressure (kJ/kg K)

A formal condition for above equations is that there are no vertical temperature gradients in the area of calculations. In practice it is O.K. to use the average temperature over the height if the gradients are not too large.

The formulas can also be applied to distributed sources. It is necessary to classify them as point- or line sources. The main problem is to estimate the distance from the top of the source to the virtuel point or line origin.

In addition to convective currents come momentum fluxes from contaminant sources (which may be identical to convective currents above persons, hot industrial processes etc., normal air jets, leakage from pressurized vessels etc.). Also turbulence created by persons, machinery, vehicles and other moving objects has to be quantified. Calculation procedures for "normal" air jets are assumed to be well known.

Coming to the evaluation of Φ, several conditions have to be considered. One main thing is to locate the zone where the contaminants are actually spread. This may not be the zone where they are produced. A few good examples to be mentioned here are contaminants carried in convective plumes, above persons and processes, welding plumes etc., or carried upwards in a jet caused by the contamination source. In these cases most of the contaminants are carried to the upper zone and hence Φ approaches 1.

The most difficult situation, evaluating Φ, is in cases where the contaminants are emitted scattered or the sources cannot be properly defined,like for instance radon, formaldehyde or other types of outgasing from building materials. In many of these cases Φ may be set to 0,5. Neutral emission in the zone of occupation of contaminants having a weak buoyancy or even a negative one, may result in Φ approaching zero. In some cases, however, the ventilation air currents, properly designed, may carry the contaminants rather direct to the upper zone, resulting in Φ approaching 1. The process lay-out may considerably influence on the outcome.

Weak or negative contaminant buoyancy, where also the thermal conditions in the room are taken into account, are situations where vertical down displacement should be utilized. Properly designed the zone of occupation, i.e. the breathing zone, may then be located in the upper zone.

Methods of Air Supply

Through a porous floor. This is the best way with respect to aerodynamics and comfort. Supply-air velocities are very low. And the displacement effect is at its best. The method is in most cases prohibited of either practical or economic reasons.

A jet-type supply through nozzles, slots or grilles in the floor. These methods are easy to design and apply. Common air-jet theory applies. An important criterion is that the vertical throw (penetration height) should be no higher than the height of the zone of occupation. Note that the jets will daccelerate due to gravity.

A necessity is that the jet zone is defined as being outside the zone of occupation. In some cases, however, the jet zone may be used as local cooling air douches. The temperature gradients outside the jet-zone are generally small and favour comfort. The method causes intense mixing turbulence and may, depending on the strength and stability of stratification, impair the ventilation effectiveness,including the heat removal effectiveness with regard to surplus heat.

Diffuse air supply through diffusers at floor level with horizontal direction of out-flow, at the walls or other convenient locations. A characteristic feature with this method is that the negative buoyancy in the outflow cuses a downward acceleration which increases the air velocities at floor level and may cause draught. Usually the supply velocity is lower than the upper limit for comfort. Another feature is a vertical temperature gradient that may be unfavourable.

Calculation procedures for such diffusers are under development. Some manufacturers in Scandinavia provide design data.

If the convective plumes above the heat sources are dumped in the upper zone the temperature gradients in the zone of occupation would be small. The convective plumes may loose its momentum flux at unequal heights, creating a larger temperature gradient.

The outlet part of the diffuser may either be a porous plate like a filter mat or a perforated plate. In the close proximity of the floor there may be a wedge of cooler air, depending on the cooling load imposed by the system. Using perforated plates evens out this wedge. The same effect is obtained, entraining large quantities of room air (8), applying special mixing and/or induction devices. In addition, such devices also even out the temperature gradients in the zone of occupation. The last effects improves comfort but unfortunately, it also generally impaires the system performance. The reason for this is that the increased turbulence increases the entrainment from the upper zone.

Performance documentation of air diffusion devices should contain the following information as a function of size and flow rate:

- Max. velocity in the near zone of the diffuser as a function of the temperature difference between the supply air and the air temperature 1,1 m above floor level.

- The size of the near zone including isovel envelopes.

- The relative temperature increase in the near zone.

- Pressure drop and noise generation data.

Evaluation Model for Displacement Ventilating Systems

In order to choose the right strategy for solving a ventilation problem there is a need for some kind of model to compare the effect of different strategies. The model should also serve as an input to a cost - benefit analyses.

The model proposed here has the same features as a recirculation model presented by Olander (9) . The main difference is that presented model is adjusted to the two-zone calculation model presented in this paper. Schematics of the model is shown in fig.6.

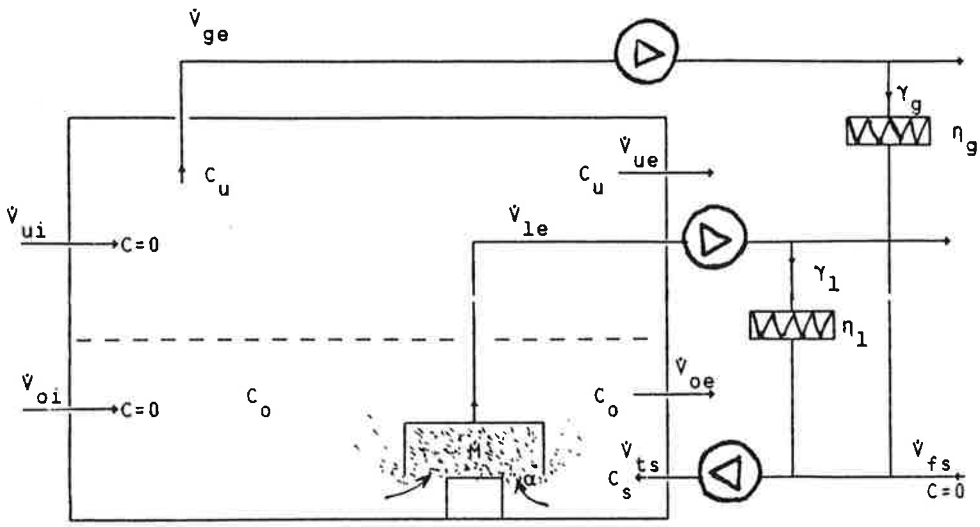

Fig.6. Evaluation model for ventilation, schematics.

$\dot V_{ge}$ = Genaral exhaust air flow
$\dot V_{ts}$ = Total supply-air flow
$\dot V_{fs}$ = Fresh supply-air flow
$\dot V_{ue}$ = Upper zone exfiltration air flow
$\dot V_{oe}$ = Zone of ocupation exfiltration air flow
$\dot V_{ui}$ = Upper zone infiltration air flow
$\dot V_{oi}$ = Zone of occupation infiltration air flow
$\dot V_{le}$ = Local exhaust air flow
$\dot M$ = Total production rate of contaminants
C_u = Upper zone concentration of contaminants
C_o = Concebtration of contaminants in the zone of occupation
C_s = Concentration of contaminants in the supply-air
α = Captureing efficiency of local exhaust
γ_g = Recirculated fraction of general axhausted air
η_g = Efficiency of recirculation filter, general exhaust
γ_l = Recirculated fraction of local exhausted air
η_l = Efficiency of recirculation filter, local exhaust

$C_u/C_o = \varepsilon_v^g$ = General ventilation index

<u>Mass ballance:</u>

Mass in: $\dot M(1-\alpha) + \dot V_{ts} C_s$

Mass out: $\dot V_{ge} C_u + \dot V_{ue} C_u + \dot V_{le} C_o + \dot V_{oe} C_o$

$$C_s \dot V_{ts} = (\alpha \dot M + \dot V_{le} C_o)\,\gamma_l(1-\eta_l) + \dot V_{ge} C_u \gamma_g (1-\eta_g) \qquad (20)$$

Mass in = Mass out, with the expression for $C_s \dot V_{ts}$ inserted, results in the following expression for total exhausted air flow:

$$\dot V_{ge} + \dot V_{le} = \frac{\dot M}{C_o}\;\frac{[1-\alpha(1-\gamma_l(1-\eta_l))](1+x)}{\varepsilon_v^g\,[1+r-\gamma_g(1-\eta_g)] + x[1+y-\gamma_l(1-\eta_l)]} \qquad (21)$$

$r = \dot V_{ue}/\dot V_{ge}$; $x = \dot V_{le}/\dot V_{ge}$; $y = \dot V_{oe}/\dot V_{le}$; $xy = \dot V_{oe}/\dot V_{ge}$

With the knowledge of permissible concentration level in the zone of occupation and knowledge of contaminant production rate total air flows in the system can be calculated. The formula may form the basis for assessing technical and economic consequences for different strategies of solving the actual problem.

In Norway recirculation is not encouraged. Instead it is recommended to use heat recovery devices.

CONCLUSIONS

- o Suply and diffuse ventilation air in the zone of occupation.
- o Displace contaminants the shortest way out of the zone of occupation. The ideal aim is unidirectional flow.

To achieve this:

- o Use seperate heating and ventilating systems.
- o Suplly air temperature should be equal to or lower than the air temperature in the zone of occupation.
- o Use preferably low velocity diffusers for supply air.
- o Apply heat recovery through heat exchangers and do not use recirculation unless efficient and secure return-air cleaning devices can be used.

Benefits:

- o Optimal out-door air requirement.
- o Energy economic solutions.
- o Optimal and energy efficient conditions for removal of excess heat.

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