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INDUSTRIAL VENTILATION - MODEL TESTS AND GENERAL DEVELOPMENT IN NORWAY AND SCANDINAVIA

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ABSTRACT

Ventilation studies using small-scale water models have, especially in Norway gained new and important knowledge in industrial ventilation the last 15 years. The outcome has been twofold. First of all it has resulted in important experience in making small-scale model tests for ventilation studies. Secondly, it has increased the knowledge in where to apply and how to design efficient displacement ventilation systems. In most industries the displacement principle is outtanding compared to other principles for general ventilation. The displacement direction is generally vertical-up, providing for supply of fresh air directly to the workspace. In most cases thermal stratification can be utilized to optimize the effectiveness for the systems. Effectiveness is sensitive to the diffuser design for the supply air. It is also necessary to avoid using the ventilation system for space heating (covering the transmission losses).

Research results and practical applications have developed hand in hand and have also resulted in new understanding and practical definitions of the ventilation effectiveness as well as calculation methods, the two-zone model, and design guide-lines for displacement ventilation systems.

INTRODUCTION

Many problems in industrial ventilation can be solved by using small-scale physical flow models.

For many years it has been worked on numerical solutions of the Navier-Stokes equations for 3-D turbulent flow in closed spaces like in industrial premises. In principle the problem is solved using finite difference methods coupled with turbulence models of the type k-ε (1, 2). However, to-days computers have difficulties in handling problems with complex boundary conditions. The reason for this is that data capacity may be too small and the calculation time too long for practical applications. Another thing is that it is not proved that the turbulence models have the neccessary accuracy for solving industrial ventilation problems.

To have the nature solving hydro- or aero-dynamical problems through the use of small-scale models has been a reliable method within many disiplins for decades ranging from flow in ducts to ship and aeroplane design.

Also ventilation flow models are reported in the literature (1, 3). However, the use of such models has never been systematicaly developed to a generally applied design tool in this area. Probably, small-scale models are regarded as being a too expencive method to be generally applied.

Approximately 15 years ago, it was in Norway decided to renew the legislation on working environment. Before that, legislation limiting pollution from industry to the outer environment had been approved of. Because of knowing that more strict working environment legislation was under way and because of good profits in industry at that time, promoted investments for improving the work environment by improved ventilation. However, the knowledge in improveing the ventilating methods in industry was rather limited and unsatisfactory at that time.

The river and harbour laboratory at the Norwegian Institute of Technology had experience in making model test of waterfalls, harbours etc. and they were asked if they could make models of industrial plants in water as well. This was a a challenge and the beginning of a ten year active period in ventilation model testing. Unfortunately most of the reports are in Norwegian or are marked confidential, but some English versions are available (4).

SMALL-SCALE PHYSICAL MODELS

The types of models used were so called Archimedien models. A characteristic feature of an Archimedien model is that the basis for scaling are the Archimedien(Ar)-number together with geometrical similarity. The model law is to keep the Ar-number in the model equal to the full scale value. The physical interpretation of the Ar-number is that it expresses the ratio between buoyancy and inertia forces and it can mathematically be expressed as the Grashof(Gr)-number devided by the Reynolds(Re)-number squared:

$$Ar = Gr/Re^2 = gl\beta\Delta T/u^2 = \frac{\text{Buoyancy forces}}{\text{Inertia forces}} \tag{1}$$

g = Accelleration due to gravity (m/s^2)
l = characteristic length (m)
β = coefficient of htermal expansion (K^{-1})
ΔT= characteristic temperature difference (K)
u = characteristic velocity (m/s)

Building a physical correct small-scale model involves keeping both the Re- and Gr-number in the model equal to the full scale or original value, respectively. This limits rather drastic the reduction possibilities of the lenght scale. By keeping the above Ar-number constant the possibility of reducing the lenght scale becomes much greater. Because this envolves only to keep the specific ratio between the Gr- and Re-number constant. Generally the flow conditions in industrial premises are turbulent. Consequently the value of the Re-and Gr-number are limited according to the turbulence criteria i.e. the flow structure criteria. There are generally three levels of limitation to be applied.

o Level one is not to reduce the Gr-number for the boundaries more than keeping the boundary layers turbulent.

o Level two is not to reduce the Re-number for the ventilation air supply openings more than keeping the supply air flow conditions turbulent.

o Level three is to keep the general flow conditions in the room turbulent. The boundary layers not being included.

Generally the first condition is the most strict and the third one the least strict. One should bare in mind that the more the Re-number is reduced from the original value, whatever this is, the less accurate the quantitative predictions like velocity, temperature differences etc. at points in the room becomes.

Qualitative predictions like gross flow patterns, stratification, large scale eddies etc. is less influenced by reduced Re-numbers as long as the general flow conditions are turbulent.

Air models versus water models

Equation (1) leads to the following relationships:

$$\frac{(u/\Delta T^{1/2})_M}{(u/\Delta T^{1/2})_F} = \left(\frac{l_M}{l_F}\right)^{1/2} \left(\frac{\beta_M}{\beta_F}\right)^{1/2} \tag{2}$$

$$\frac{Re_M}{Re_F} = \left(\frac{l_M}{l_F}\right)^{3/2} \left(\frac{\beta_M}{\beta_F}\right)^{1/2} \frac{\nu_F}{\nu_M} \left(\frac{\Delta T_M}{\Delta T_F}\right)^{1/2} \tag{3}$$

Additional symbols:

ν = kinematic viscosity

M = refers to model scale

F = refers to full scale

Inserting relevant values for viscosity, ν, and thermal expansion, β, at atmospheric pressure and 290 K ambiant temperature, the following comparison can be made between air and water models:

1. The ratio $(u/\Delta T)^{1/2}$ must be significantly smaller a in water models compared to the original value and also compared to air models. The value ranges between 1:4 for legth-scale 1:1 to 1:20 for length-scale 1:20. Using air models the same ratio is much greater ranging from 1:1 for length-scale 1:1 to 1:4,5 for length-scale 1:20.

2. The ratio between the Re-number in the model and full scale will be the same for water models and air models because the temperature differences must be limited to a few K in water. This means that water and air models can be made to the same scale.
   The Re-number ratio is approximately $2(l_M/l_F)^{3/2}$ which results in 0,18 for length-scale (1:5), 0,063 for length-scale 1:10 and 0,023 for length-scale 1:20.

However, there are nevertheless some advantages with water models, compared to air models. Observing eq. 2 and the statements in point 1 above, the velocities becomes very low in the water model. The time scale is accordingly increased, meaning that what one sees in the model is in slow motion compared with the original. Secondly, it is easy to visualize the flow patterns with dye-liquids in whatever colour one wants.

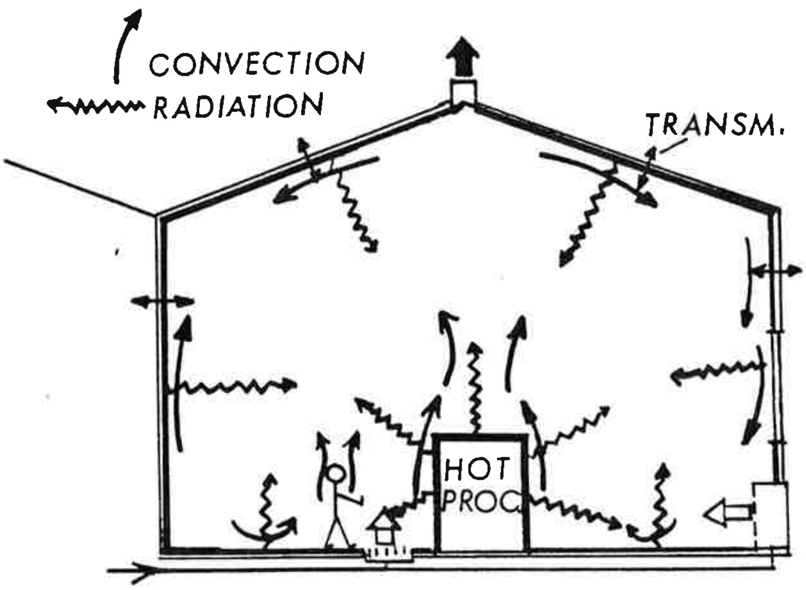


Fig. 1. Heat exchange, schematics.

In air models the conditions are generally the oposite,i.e. everything is speeded up. The visualization methods (generally smoke is used as a tracer) are less developed. Some of the problems may be overcome by using film or video play-back in slow motion. Nevertheless, the main reason for using water models in Norway was that the istitution at which model tests were initiated was a hydraulic laboratory.

The boundary conditions using water models has to be specifically mentioned. In the full scale the surfaces takes on certain temperatures as a result of the balance of convective and radiative heat transfer surface to air and surface to surface,fig. 1. This means that heat balance between the enclosure and the air is in the end convective.In other words, any heat in- or output in the full scale is transferred partly convective and partly radiative, which means that a surface with no heat in- or output in it self gains or looses heat also due to thermal radiation. In a water model, radiation between surfaces is not present meaning that the resultant convective heat balance has to be known and has to be artifically modelled. Among other things, the radiative part of the process heat has to be modelled as heat input to the other surfaces. A heat balance calculation is therefore necessary and the heat conduction through the enclosing surfaces must be included in these calculations in order to model the convective boundary conditions right.

Using air in the model the heat balance is adjusted by itself when the total heat is being set where it appears, and if the right U-values for the enclosing

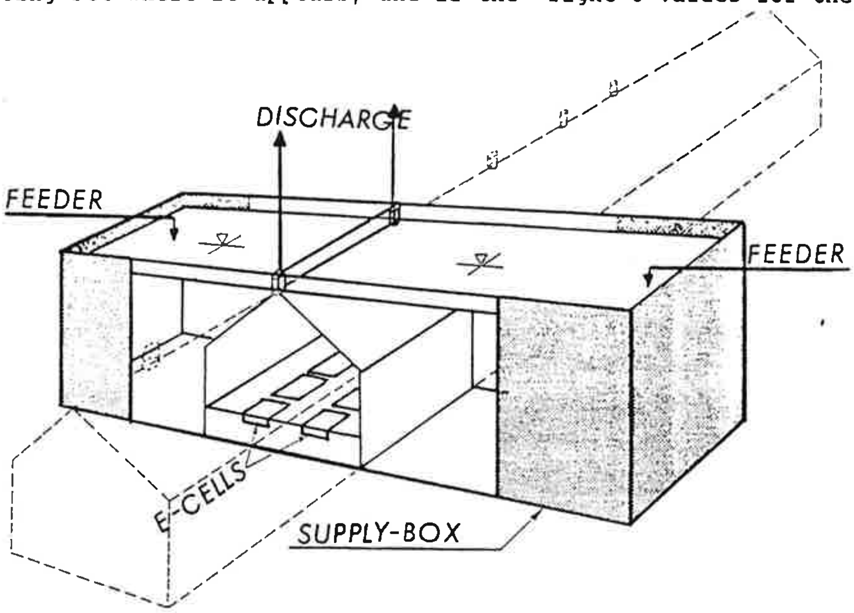


Fig. 2. Water model set-up. Aluminium electrolyzing plant.

structures are being used. However, because of the reduced Gr-number the aerodynamic boundary conditions are not automatically properly reproduced in the models.

<u>Other flow media.</u> Looking at equation 3, other flow media than air or water may have physical properties meeting the Re- and Gr-number requirements in a better way. This was not persued in the test program. Air and water is clean and cheap liquids to handle and there is little need for any special or complicated technical solutions to be applied.

<u>Water model set-up</u>.

A typical water models set-up is shown in fig. 2.

Heat input is usually done using insulated electric resistance heating cables.

Symmetry is utilized by slicing out a part of the plant. The easiest way of running a water model is to put it in a larger water tank. In this way wind forces can be simulated by adjusting the water levels on each side of the building. A closed circuit single tank model can also be used, fig. 3.

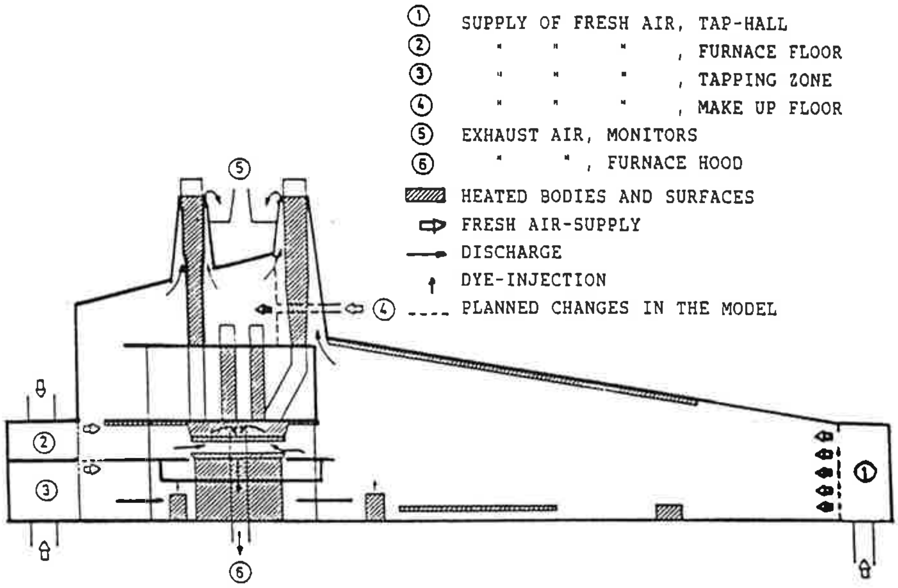


Fig. 3. Single tank model of a Fe-Si plant.

As to the modelling equations the parameter values inserted in them have to be consistant. For characteristic velocities any velocity may be used, but, generally, the velocity taken as the ventilation air flow divided by a characterisitc cross section, say the supply air openings, is used. As a characteristic temperature difference the change in temperature of the ventilation air flow from supply to exhaust is generally taken. In this way the Archimedien model law can be supplemented by the conservation equation for the heat input. This gives us 4 parameters to determine i.e. velocity, temperature, length scale and heat input. Now, having two equations, any two parameters may be calculated when the two others are chosen. For instance if the velocity and the length scale is chosen then the temperature scale and the heat input scale is calculated.

IMPORTANT GENERAL RESULTS

Model lenght scale.

The largest reduction in length scale that was made, giving meaningful test results, was 1:30.

Meaningful tests could be run even in models of very hot process industries down to a lenght scale of 1:20. With this scaling one are approching the limits with respect to avoiding laminar general flow conditions. Laminar boundary layers are, however, formed even before a lenght scaling of 1:10 is reached.

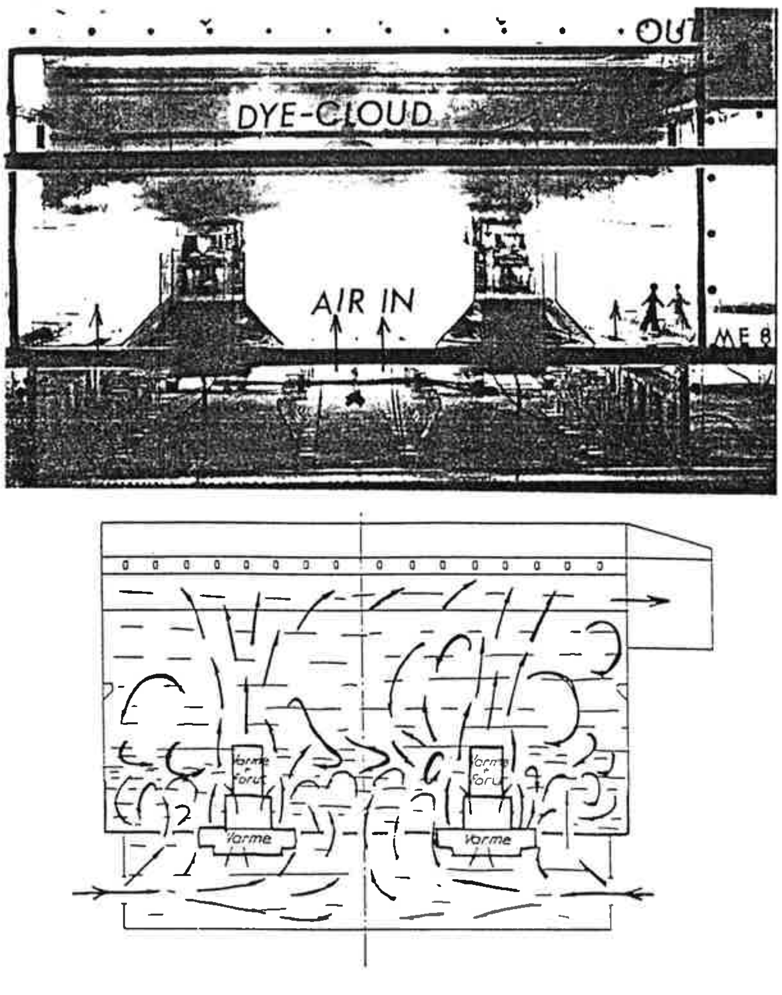


Fig. 4. Thermal stratification. Aluminium electrolyzing plant.

Important qualitative predictions.

In premises containing significant heat producing processes it was easy to create thermal stratifications separating two distinct flow regions. One characteristic feature for heat producing processes is that heat and contaminants are dissipated through the same plumes. It could easily be demonstrated that heat and contaminants are dissipating to and despersing in a warm air cushion below the ceiling, fig. 4, leaving an uncontaminated pool of fresh supply air above the floor, sufficiently high to enclose the workers. A necessity is that the air is supplied in the lower fresh air-pool. Any other way of supplying air results in a more contaminated working zone.

Now, most of the premises for heavy process industries in Norway (and also other countries) like aluminium electrolyzing plants, ferro silicon, steel and iron melting industry, foundries, forges and rolling mills etc. are utilizing natural ventilation involving low wall or floor air inlets and roof top exhaust with or without fans. The model tests revealed that the existing arrangements were operating far from in an ideal way. Some important findings were:

o The supply of ventilation air flow must balance against the convective air flow volumes at the desired height (stratification height).

o For every air supply method or diffuser type the turbulence becomes too strong when the air flow rate increases above a certain level, i.e. the air supply velocity increases above a certain value, causing too much down mixing from the upper contaminated zone. There is a marked stratification up to a certain velocity. The optimal air supply volume flow is the airflow at which the stratification starts to brake down. The optimal stratification height increases with increasing air supply cross sections. Increased supply-air turbulence and entrainment decreases the optimal supply air velocity.

o Typical high velocity horisontally directed air-jets in the lower zone destroys the stratification. One example is opening a window. The jet thus formed breaks down the stratification, increasing both the temperature and the concentrations of contaminants.

o A correct distribution of the supplied air is important. Cross wind increases the air supply on the up-wind side and decreases it on the lee side, when utilizing natural ventilation. This has to be overcome by using adequate throttling devices. Wind independency can better be achieved by using fan supply of air with a pressure drop over the supply system well above the prevailing wind pressure. In this way preheating of ventilation air is made more feasible. In many cases recovery of surplus heat is feasible. Generally, one should look for this possibility when preheating is necessary.

EXPERIENCE FROM PRACTICE

General conditions

Practical experience generally confirms the stratification effect found in the models. However, generally the air exchange between the two zones is larger than revealed in the models. This may be partly due to more disturbances in the real plants than in the models and also the laminarization effect in the models. Another reason could be that some effects like convective downdraught along vertical surfaces, due to walls and roofs colder than the room air, was not taken into account properly in the models. Also disturbences from the activety and process equipment in real plants was not properly modelled either. However, all differences could not be explained by these facts.

Diffusers.

The next thing of importance could be difference in the air supply devices or air supply diffusers. Details here seems to be very important. Distributing the supply-air evenly over the total inlet flow area is a number one requirement.

One main problem seems to be that the low Re-numbers in the models causes in general a laminar outflow that results in relatively more gentle outflow than through a geometrically similar diffuser in the real plant. The experience is that all diffusers in model scale have more or less the same flow quality as if the air was supplied through porous medias like rubber foam or filter cloths or mats in the full scale. Such divices causes a quasi laminar outflow.

The largest discrepances were found when applying perforated plates. In many industries the practical difficulties are too large and the costs too high using filtermaterial in the diffusers because of a too large content of particles in the supply air. A perforated plate is then a more practical device. The functional difference between such plates in small scale compared to full scale is quite large. These differences are specifically related to the turbulence generated (laminarization) and the entrainment capacity of room air.

Much of the difficulties can be overcome by using two or more stages with a small percentage or perforation for the first plate in the flow direction to create a pressure drop satisfactory for even air distribution. The last plate should be rather open i.e. have a high percentage of perforation, but have a sufficient pressure drop to prevent backward entrainment. The bad experience using single perforated plates was greatest when the location of the diffuser was close to the transition region between the two zones. It is no doubt that the quasi laminar low velocity outflow as through filter mats creates the best ventilation effectiveness in real plants.

An other important thing with diffusers is that they play an important role with respect to thermal climate, air temperature, draught etc. and should be carefully designed to meet the requirements.

Air intakes

Great care should be payed avoiding resirculation of exhaust air to the air intake regions. Exhaust air comtaminated wakes are easily formed if the exhaust air are not dumped in the undisturbed flow region around a building. It is not always safe to use the guide lines in the manual (5). However, we have also experienced problems with untight buildings in the way that exfiltration air from the upper zone has contaminated the intake air through the building wakes. Wind tunnel model tests can help in mapping and solving these problems.

NEW TRENDS

Extrapolation of the model test results to other industries

Encouraged by the water model test and practical experience applying the results it gradually became mature to apply the principles in other types of industries than tried out in the model tests. Welding shops was a natural new target area. The experience of such plants was encouraging. Stable stratification could be formed even with just a few K temperature difference. In these plants, it was even more important to choose the right diffuser. An other experience was that in the cold season, a combination of air heating and radiative heating was favourable. The air heating was arranged as aerotempers blowing horizontaly at a height slightly above the desired stratification height, fig. 5. Radiative heating could be done by ceiling mounted radiators. The aerotempers prevented cold draught and also stabilized the contaminated air cushion.

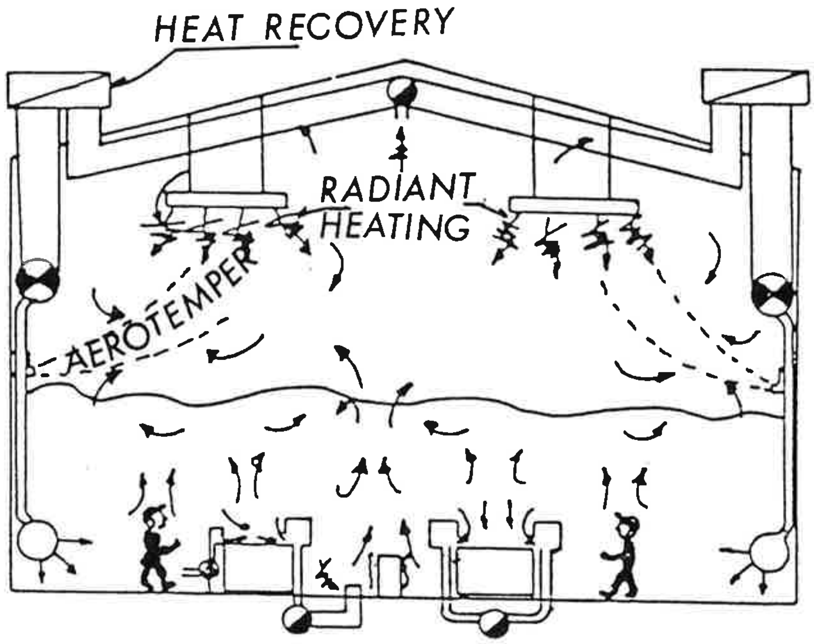


Fig. 5. Displacement ventilation in workshops.

The stratification effect and the use of radiative heating allows for keeping a lower air temperature in the zone of occupation compared to other solutions, maintaining a comfortable operative temperature. A typical value for the ratio between the upper and lower zone concentration of contaminants is 3.

The state of the art at present is that in most premises the heat production is great enough to generate stratification. There is an increasing tendency to use stratified ventilation although conventional systems are still being designed. However, the understanding of the advantages with ventilation for the work spaces is rapidly increasing. On the other hand to leave the jet principle in ventilation seems to encounter a lot of resistance.

Theoretical models and calculations

Most of the model tests were evaluated by measuring temperatures and by visually observing the dye patterns. Of course this is not always satisfactory, a more quantitative evaluation procedure is necessary. However, velocity measurements were difficult to make because of the very low velocity levels in the model. Dye concentrations were also difficult to measure and were not made.

The model tests created a lot of insight into the ventilation process. What was then more natural than trying to quantifyee the performance of the systems and also to develop calculation procedures for sizing the systems. To evaluate the ventilation efficiency is another important matter. Knowing that the stratification concept was reproduced in practice, motivated for using the two-zone flow concept to develop definitions of ventilation efficiency. Norway and Sweden has been leading this work in Scandinavia(6, 7). The calculations established the mathematical proof of the outstanding effectiveness of the displacement ventilation principle. It very soon also became clear that ventilation efficiency could not be expressed by one single number only. Further it became clear that a couple of numbers had to be introduced both for characterizing the ventilation air flow and the contaminant flow. A final suggestion (8) was to base the definitions on age distribution analysis. In this way general expressions may be developed without any coupling to a certain flow or calculation model.

Age analysis is the same as studying the time history for all small "parcels" of ventilation air supplied to the room and of all small "parcels" of contaminants that is emitted to the room. The definitions of efficiency are based on aiming at making the residence time for the air molecules in the room and the transit time for the contaminant flow from the place of emission to the exhaust as short as possible. The first is based on the fact that the air is a carrier of contaminants out of the room and the second one that air and contaminants generally do not have coinciding flow patterns. The outcome of this has been that different effectiveness expressions has been suggested and the following quatities are in the process of being approved in the Nordic countries.

o Average air exchange efficiency, characterizing the average residence time of the air in the room, ranging from zero to one (the ratio between the transit time for the ventilation air flow and the total average residence time for the room air).

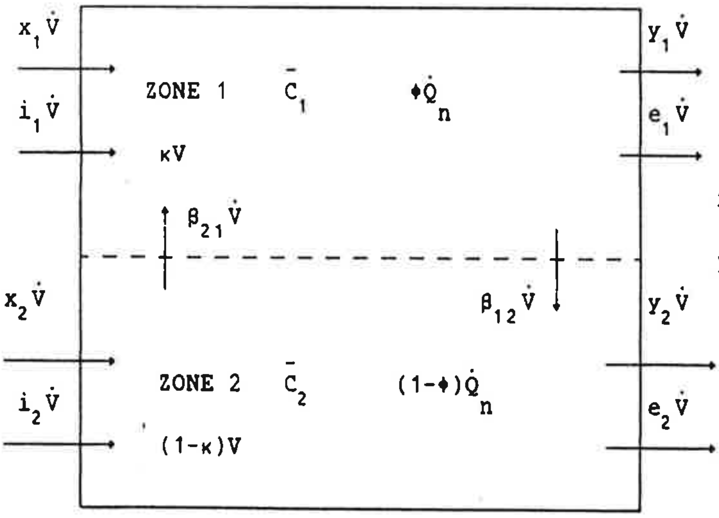
o Local air exchange indicator, characterizing local conditions, ranging from zero to infinity (the ratio between the average age of the room air and the local age).

o Average ventilation effectiveness, characterizing the transit time for the contaminants and also the average concentrations of contaminants, ranging from zero to infinity (the ratio between the transit time for the ventilation air flow and the transit time for the contaminant flow). Can be transfered to an efficiency ranging from zero to one.

o Local ventilation index, caracterizing local concentrations of contaminants ranging from zero to infinity (the ratio between the concentration of contaminants in the exhaust air and the concentration of contaminants at a work station).

It is refered to (8) and (9) for more detailes.

Definitions above fits very well into a two zone calculation model, the feature of which is shown in figure 6. A key to use this model is to quantify



$$x_1 + x_2 + i_1 + i_2 = 1$$

$$y_1 + y_2 + e_1 + e_2 = 1$$

Fig. 6. The two-zone flow and diffusion model, characeristic features.

V = Room volume (m^3)
$\dot{V}$ = Ventilation air flow (m^3/s)
$\dot{Q}_n$ = Net load to the room of either chemical contaminants or surplus heat. (Kg/s, KJ/s)

ϕ = Fraction of the load that is released in zone 1.
κ = Fraction of the room volume belonging to zone 1
C = Concentration (Kg/m^3, KJ/m^3)

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
Overbar - = time mean
Subscript 12 = from zone 1 to zone 2
Subscript 21 = from zone 2 to zone 1

all the loads and to quantify and calculate convective flows and other air flows between the two zones. More detailes in (10).

Present research work is aiming at combining the complete solution of the Navier-Stokes equations with the two-zone calculation model.

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