# Velocity and Temperature Distribution in Flow from an Inlet Device in Rooms with Displacement Ventilation

Jacobsen, T.V. ; Nielsen, Peter V.

*Publication date:*
1992

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

INSTITUTTET FOR BYGNINGSTEKNIK
DEPT. OF BUILDING TECHNOLOGY AND STRUCTURAL ENGINEERING
AALBORG UNIVERSITETSCENTER • AUC • AALBORG • DANMARK

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PAPER NO. 28

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SEPTEMBER 1992 ISSN 0902-7513 R9253

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T.V.Jacobsen and P.V.Nielsen
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## SUMMARY

Measurements are performed in a full-scale test room with displacement ventilation with focus on the velocity and temperature field in the region close to the inlet device.
Investigations based on these detailed measurements have been made in order to see if it is possible to describe the velocity decay and the shape of velocity and temperature profiles in front of the inlet device by traditional jet theory, by stratified flow theory or by a combination of the two theories. The velocity decay in the radial flow from the inlet device show accordance with an equation based on stratified flow theory. The jet practice of applying universal profiles is working satisfactory for the measured velocities while the temperatures show significant deviation.
The strong influence of the Archimedes number is discussed and finally results of numerical simulation with a k-$\epsilon$ turbulence model are presented and suggestions for improvement are made.

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## INTRODUCTION

The extended use of the displacement principle in room ventilation has been the incentive to a number of investigations concerning the restrictions on inlet flow and heat load with respect to human comfort. The recommendations which have been put forward are essential in the design phase but they do not explain the physical processes in details. An improved understanding of the underlying physical phenomena is important for the development of design tools and the recommendations which are offered today.
Displacement ventilation is characterized by a flow field which is driven mainly by buoyancy forces. Cool air is supplied at floor level and upward directed thermal plumes from heat sources give rise to a vertical temperature gradient. The stratification diminishes vertical air movement and diffusion and apart from the region above heat sources the flow field is divided into more or less distinct horizontal layers. The outlet is located in the upper part of the room ,e.g. at the ceiling, and it ensures a relatively efficient removal of excess heat. One of the main comfort problems in a room with displacement ventilation is the relatively high velocities in front of the inlet device. To compensate an increasing heat load it is necessary to increase the cooling capacity either by increasing the inlet flow or by increasing the temperature difference between inlet and outlet. Both adjustments will tend to increase the maximum velocities which are found a few centimeters above floor level. An altered relation between momentum and buoyancy forces at the inlet has a profound effect on the magnitude of the initial acceleration due to gravity, the rate of entrainment and the flow field in general. Provided that a specific inlet device is considered and that the variation in the volumetric expansion coefficient and in the gravitational acceleration is negligible, a reduced Archimedes number can be defined to describe this relation.

$$Ar_r = \frac{\Delta T_0}{U_o^2} \tag{1}$$

$\Delta T_0$ is the temperature difference between the occupied zone (y=1.10 m) and the inlet, $\Delta T_0=T_{1.10m}-T_{inlet}$ and $U_0$ is the velocity at the surface of the inlet device calculated from the inlet flow and the area of the inlet device.
It is the objective of the experimental work presented here to show how $Ar_r$ affects not only the maximum velocities but also the initial spread as well as the velocity distribution in front of the inlet device.

## TEST ROOM AND MEASUREMENTS

The air supply device, exhaust openings and heat sources are arranged in such a way that the middle plane of the room coincides with the middle plane of the inlet device constituting a symmetry plane for all boundary conditions.

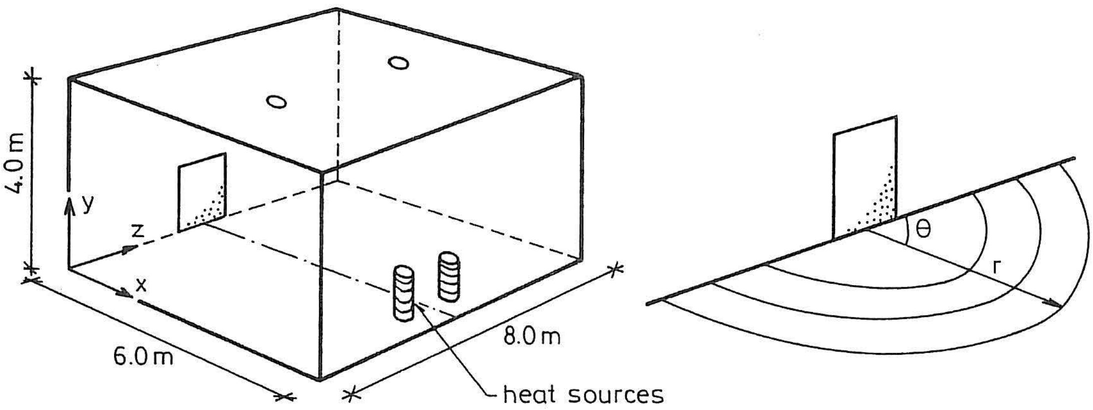



Fig. 1 Test room arrangement and illustration of radial flow pattern in front of inlet device.

Smoke is added to the inlet in order to visualize the flow pattern and choose a suitable line of procedure for the measurements. The smoke experiments reveal important features of the flow pattern. Under isothermal conditions it is observed that the flow from the diffuser penetrates horizontally about 1 m into the room where the smoke is dissolved due to entrainment. Even at small temperature gradients ($Ar_r \approx 300$ °$Cs^2/m^6$) the inlet air drops to the floor where it is deflected and it spreads out radially in a thin layer which flows to the surrounding walls. At $Ar_r > 500$ °$Cs^2/m^6$ the transition between the cool layer at the floor and the room air becomes even more distinct - presumably because of a more stable stratification. The apparently fully radial flow along the floor has a layer thickness of 20-30 cm with a slight decrement for increasing $Ar_r$.
Observations show that the streamlines are emanating from a centre point in front of the inlet device and that they are straight lines corresponding to a radial flow along the floor. The vertical velocity and temperature profiles are consequently measured along the streamlines (given $\theta$-values, $1.0\text{m} \leq r \leq 3.5\text{m}$), see fig. 1.
The measurements are made at steady state conditions using hot sphere anemometers for the velocities and thermocouples for the temperatures.

## DECAY OF MAXIMUM VELOCITIES

The maximum velocities are located 2-5 cm above the floor. The decay of maximum velocities with distance from the air terminal device has earlier been described by *Sandberg et al. 1991* and *Nielsen 1990.* A simple expression suggesting exponential decay (*Nielsen 1990*) is suitable in the following form:

$$\frac{U_{max}(r)}{q_0} = K\ (\theta,Ar_r)\ \frac{1}{r} \tag{2}$$

$K(\theta,Ar_r)$ is a factor of proportionality, $\theta$ and r are angle and radius respectively and $q_0$ is the inlet flow.

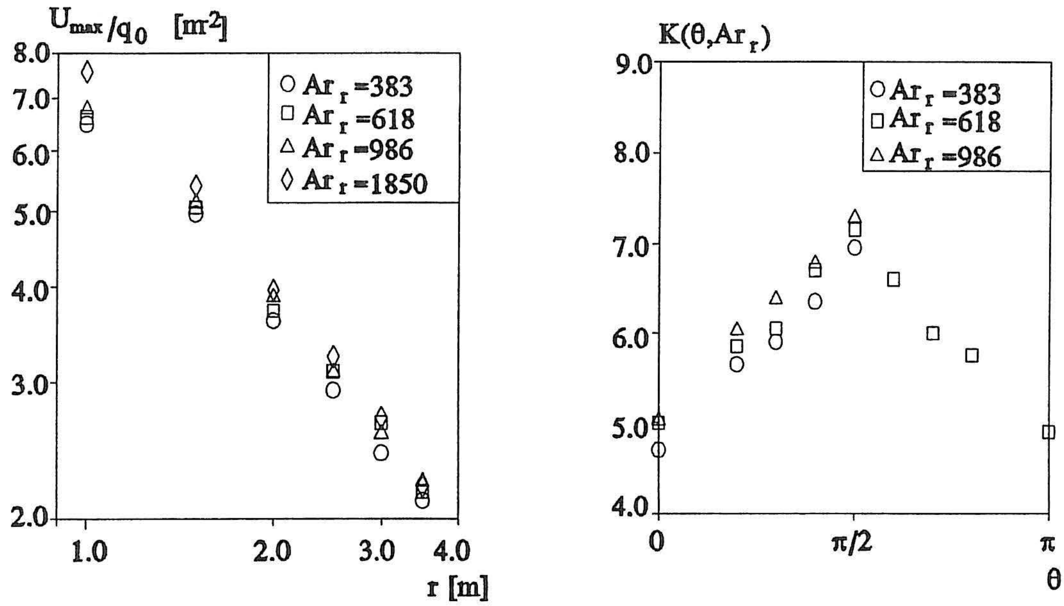



Fig.2 Measured velocity variation in the centre line ($\theta=\pi/2$) with distance from inlet for different $Ar_r$-numbers and $K(\theta,Ar_r)$ variation with streamline angle for different $Ar_r$-numbers.

Fig. 2 shows that eq.2 is a good approximation to the velocity decay for a given $Ar_r$-number. The velocity level is highest in the centre line and it is seen that an increment in $Ar_r$-number will increase the velocity level slightly.
Furthermore, fig. 2 shows that the K-values calculated for each streamline are distributed symmetrically around the centre line of the inlet device and vary with the direction. The distribution of K-values with streamline direction is approximately similar for the chosen $Ar_r$-numbers. This might change for larger Ar-numbers where the initial spread of inlet air is different from the cases of the current study.

## VELOCITY AND TEMPERATURE PROFILES

In the field of turbulent jets velocity profiles are often assumed to be similar. The measured velocity profiles in front of the inlet device shows not only similarity in shape but they are also much like the profiles for a wall jet. This opens up the prospect of applying a similarity function of the jet type.

*Mathisen 1990* applies a similarity function for half a free jet but here a wall jet expression is preferred. Verhoff's (*Verhoff 1963*) solution takes into account the bondary layer occuring near the surface and the free jet boundary to the surrounding fluid.

$$\frac{U(y)}{U_{max}} = A\left(\frac{y}{b}\right)^{\frac{1}{7}}\left[1-erf\left(B\left(\frac{y}{b}\right)\right)\right] \tag{3}$$

where the constants A and B equals 1.48 and 0.68, repectively, erf is the error function and b is the height of the profile defined as the level where the velocity is 0.5 $U_{max}$.
Relations for heat transfer can often be deduced from the momentum transport. For turbulent jets a ratio between temperature profile and velocity profile can be applied. It is confirmed by regression that using the second power in eq.4 actually gives the best obtainable fit when all T-profiles are included.

$$\frac{U(y)}{U_{max}} = \left(\frac{\Delta T(y)}{\Delta T_{max}}\right)^2 \tag{4}$$

$\Delta T(y)$ is the temperature difference between room air and a point at y distance from the $\Delta T_{max}$ location.

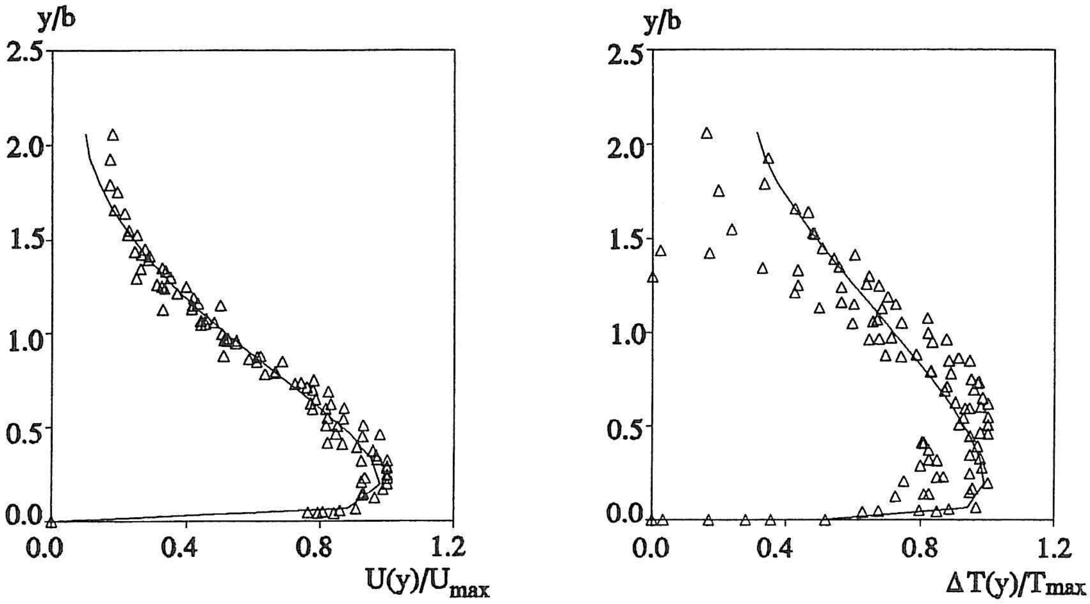



Fig.3 Measured velocities and temperatures compared with theoretical profiles, eq.(3) and eq.(4),($Ar_r$=618 °$Cs^2/m^6$).

The agreement between eq.3 and the measured velocities in fig.3 is obvious and the slight divergence at $y > 1.5$ y/b is probably caused by the recirculating flow in the domain above the inlet flow.
The temperatures have a more scattered distribution and they do not fit into the theoretical expression in the same convincing way as the velocities.
The divergence does not occur solely at single points but also the entire profile shape is altered which implies caution in using eq.4 as an approximation. A closer study reveals a slowly increasing discrepancy with distance from the inlet device. The heat flux at the floor and the heat sources distort the temperature distribution and make the approach of universal profiles less obvious.
The conclusions based in fig.3 can be extended to the $Ar_r$-number range which is considered (~300-1900 °C$s^2/m^6$). The measured values of velocity and temperature become relatively more scattered as the angle between the centre axis of the inlet and streamline increases where the flow is influenced by the sidewalls. The use of equations (2),(3) and (4) is therefore restricted to the area outside the initial zone where the profiles are not yet developed, and outside the zones near to the walls where deceleration occurs.

## EFFECTS OF STRATIFICATION

As mentioned earlier buoyancy forces are important to the overall flow pattern. The stratification effects generate an interface between the layer of supplied cool air close to the floor and the surrounding room air. The flow development in the dense current along the floor is strongly influenced by the interfacial mixing. On microscale the entrainment rate of light air into the dense layer is closely connected to the interfacial turbulence and on macroscale it is highly dependent on density and velocity profiles. The density difference at the interface tends to counteract the diffusion at the interface while the velocity gradient tends to increase it.
*Sandberg et al. 1991* elaborate the subject and suggest that the flow field in front of the inlet device should be considered as divided into a sub- and supercritical flow domain. The concept of a sub- and supercritical domain has its origin in the field of hydraulics where the transition between the two domains is defined by the densimetric Froude number, $F_\Delta$, but here it is more convenient to use a local Arkimedes number, Ar.

$$Ar = \frac{1}{\sqrt{F_\Delta}} = \frac{g\beta\Delta Tl}{U^2} \qquad (5)$$

$$Ar < 1 \Rightarrow supercrit.$$
$$Ar > 1 \Rightarrow subcrit.$$

In this case a local Archimedes number $Ar_l$ is used.

$$Ar_l = \frac{g\beta\Delta T_{max} l}{U^2_{max}} \qquad (6)$$

It is possible to calculate entrainment rates for the flow as the increment of volume flow through sections of the cool air layer. Provided that the flow from the inlet device can be regarded as being fully radial and that the air is entrained through the face, $A_e$, an entrainment velocity can be estimated as shown in fig.4.

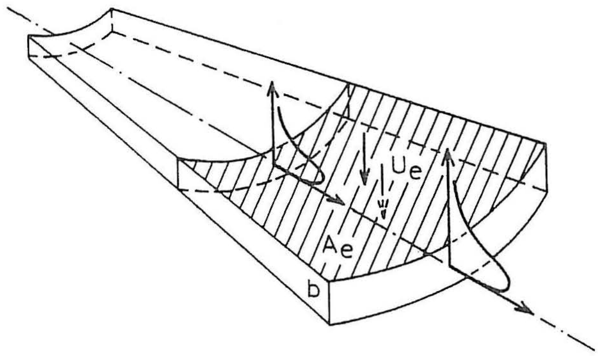
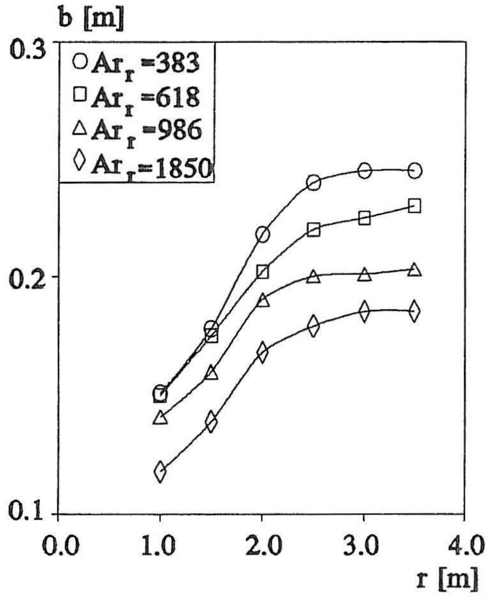



Fig.4 Entrainment velocity in radial flow from inlet device.



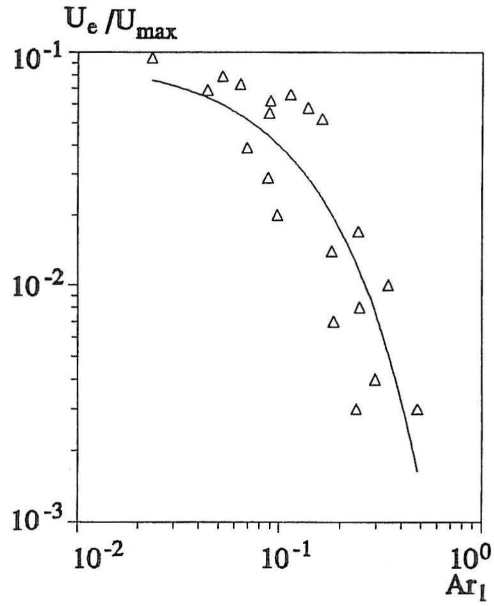




Fig.5 Growth of cool layer thickness with distance from inlet device and decay of entrainment velocities with local Archimedes number.

In the zone close to the inlet device the entrainment velocities are of the same order of magnitude as an isothermal jet. As the inlet flow mixes and spreads out the $Ar_l$ increases and the entrainment velocity falls rapidly as it is seen in fig. 5. The entrainment velocity becomes extremely small for $Ar_l > 0.1$ ($r > 2.0$-$3.0$ m). *Pedersen (1986)* and *Turner (1979)* describe a similar abrupt decrement of entrainment rate as the flow becomes subcritical for dense bottom currents in the field of hydraulics. The measurements seem to support the theory on a two domain approach.
It should be kept in mind that the room geometry might play an important role in the flow development along the floor. Theoretically two flow domains still occur when the room geometry is altered but the location of the transition between flow domains will probably be displaced.

## NUMERICAL MODELLING

This section presents the introductory considerations which are made for modelling displacement ventilation and it shows the results achieved up to now.
The reliability of predictions made by a turbulence model depends on the theory itself in terms of a mathematical description, the precision of the computer algorithm and the boundary conditions employed.
For example *Skovgaard and Nielsen 1991* show that provided a proper method for describing especially inlet boundaries is applied the predictions made with the k-$\epsilon$ model in case of a isothermal jet in a room agree to a large extent with the experimental reality. Even though the k-$\epsilon$ model is less reliable than more advanced models it is widely accepted as applicable in room air simulation.
If acceptable predictions shall be expected by extending the use to the non-isothermal field and in this case displacement ventilated rooms, the measurements suggest that additional properties are to be implemented. It is of crucial importance that *buoyancy forces, stratification effects and heat flux at boundaries* are included. The influence of *radiation* is essential to the temperature distribution in particular. In the process of building a model which complies with these demands the first step is to include buoyancy forces by adding a term to the momentum equation of the vertical velocity component $U_2$.

$$U_2 \frac{\partial U_i}{\partial x_i} = -\frac{1}{\rho}\frac{\partial P}{\partial x_i} + (\nu_l + \nu_t)\left(\frac{\partial^2 U_2}{\partial x_i^2}\right) - g\beta\Delta T \qquad i\in[1..3] \tag{7}$$

$P, \rho, \beta, g, \nu_l$ and $\nu_t$ are pressure, density, volumetric expansion factor, gravitational acceleration, laminar and turbulent kinematic viscosity, respectively.
The turbulent quantities are also modified due to buoyancy thus the k- and $\epsilon$ equations are extended by a generation term. This extension results in lower turbulent kinetic energy, k, higher dissipation, $\epsilon$, and a reduced turbulent viscosity, $\nu_t$ when the flow is stratified. Equations for $U_1$,$U_3$-momentum,continuity,temperature and turbulent quantities can be found in *Chen 1988* or *Davidson 1989.*
At the boundaries the dependent variables and the heat transfer are predicted by means of empirical logarithmic wall functions.
The surface radiation in the room causes a net heat flux to take place from the warmer upper part of the room to the colder lower part. The vertical temperature gradients are approximately linear except close to the floor and close to the ceiling. An advanced radiation model e.g. a discret transfer model, can be used but in this case a simple approach based on experimental results is applied. A linear vertical temperature variation is prescribed at the walls :

$$T(y) = \frac{y}{H}(T_e - T_f) + T_f \tag{8}$$

where $T_e$ is the exhaust temperature and $T_f=0.55(T_i+T_e)$ is the floor temperature.

When a simple inlet profile is applied it it possible to obtain a solution for the flow domain which has the characteristic velocity decay in a region in front of the inlet device, and a radial spreading.

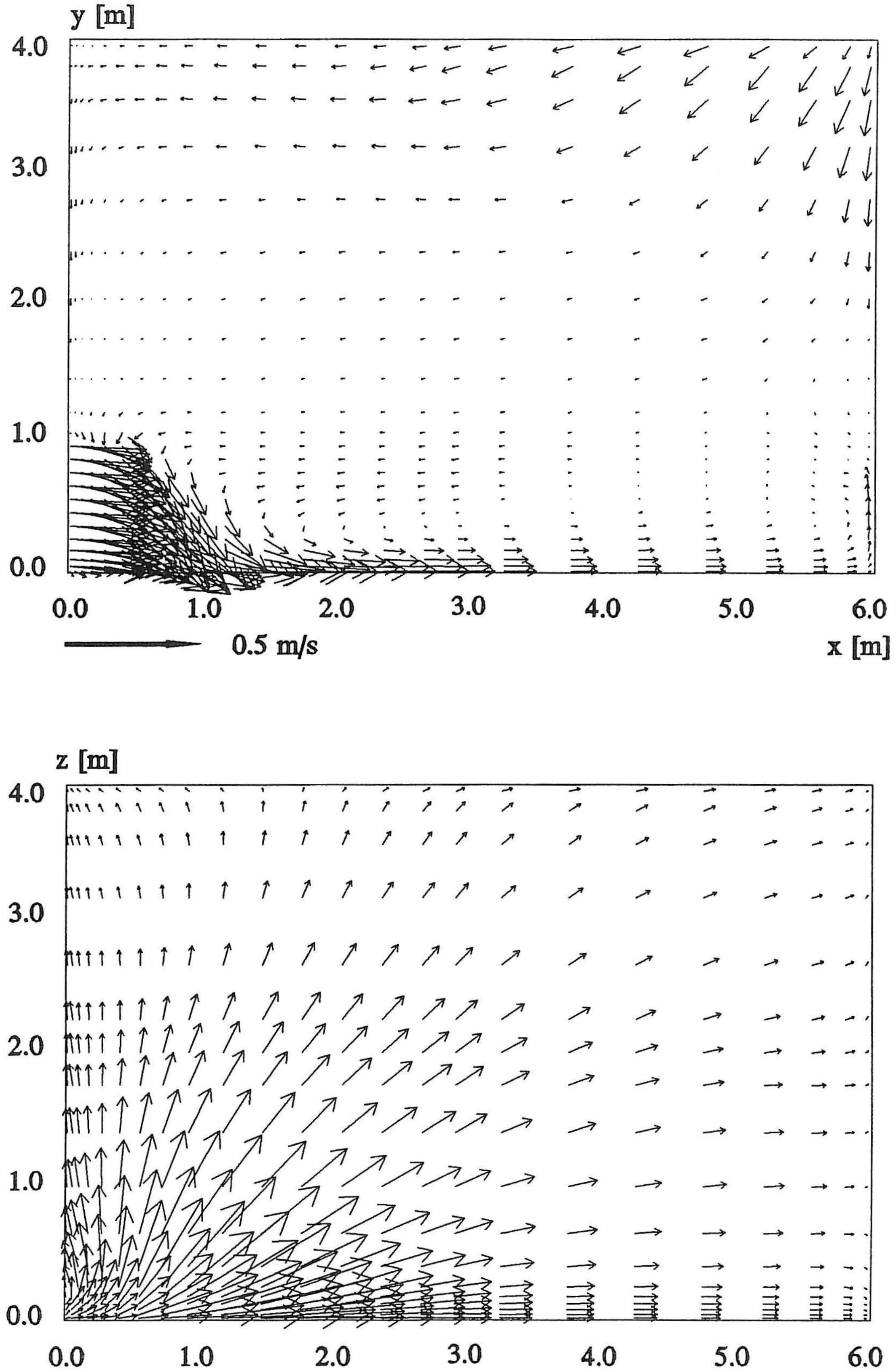



Fig. 5 Velocity vectors in an xy-plane (z=0.0 m) and in an xz-plane (y=0.03 m).

A realistic flow pattern is predicted but the heat flux at the surfaces described by logarithmic wall law and eq.8 results in a net heat removal from the room. The $Ar_r$-numbers in the measurements are larger than the $Ar_r$-numbers obtained in the numerical simulation and this causes a reduction in the predicted initial acceleration and consequently too low maximum velocities along the floor.
To avoid this discrepancy it is necessary to improve the calculation of heat flux at the walls and to obtain qualified estimates of the internal heat exchange by radiation. Furthermore it is probably required to apply a more advanced inlet boundary condition. This will be the scope of future development of the model.

## CONCLUSION

This paper deals with aspects of the air flow in rooms with displacement ventilation with reference to detailed full-scale measurements. The results show that it is possible to describe the decay of maximum velocities along the floor by relatively few parameters, and that theories concerning the vertical temperature and velocity profiles for turbulent jets can be applied in displacement ventilation as well. While the similarity of profiles is obvious for the measured velocities it can be regarded as an acceptable approximation in case of temperature profiles.
Even though the measured velocity and temperature profiles show resemblance to isothermal wall jet profiles the flow is clearly stratified. This is illustrated by the variation of entrainment velocities which indicates that the flow in front of the inlet device can be divided into two different flow regimes.
The application of numerical modelling for displacement ventilation seems promising with respect to simulating the fundamental characteristics. A further refinement of the CFD method in terms of improved handling of heat flux at walls, radiation and inlet boundary conditions can contribute to the introduction of numerical modelling of displacement ventilation in the design phase.

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