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THE DISTRIBUTION OF AIR VELOCITY IN LARGE ROOMS WITH SMALL SIDE-WALL MOUNTED SUPPLY OPENINGS

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1. INTRODUCTION

The ventilation of a large room is often achieved by supplying inlet air from a small side-wall mounted opening. Fig. 1 shows the velocity distribution in a typical room with a small circular inlet opening close to the ceiling. The supplied air forms a wall jet below the ceiling which is easy to describe in terms of velocity distribution, entrainment etc. The jet is deflected at the end wall opposite the supply opening and the resultant flow in the lower part of the room - the occupied zone - has a rather complicated structure. It is the purpose of this paper to discuss the results of a calculation procedure which can predict the turbulent three-dimensional flow in all parts of the room including the occupied zone, ref. /1/. The procedure is an extension of an earlier method determining the two-dimensional flow which takes place when the supply opening is of the linear diffuser type, ref. /2/, /3/ and /4/. A similar general calculation procedure has also been applied to room air distribution by Hjertager and Magnussen /5/.

2. CALCULATION PROCEDURE

The velocity distribution in the room is obtained by solving time-average differential equations for the flow by a computerbased numerical method.

The differential equations consist of three momentum equations (Navier-Stokes equations) and the continuity equation. A turbulent model described by differential equations for the turbulent kinetic energy and the dissipation of turbulent kinetic energy completes the set of time-averaged equations. The two-equation turbulence model has been developed for the present method by Launder et al /6/ and it has been extensively used for recirculating flow, recently by Pope and Whitelaw /7/.

This paper will only deal with isothermal flow, but it is easy to extend the method to include non-isothermal flow by adding the energy equation to the description.

The flow domain is divided into nodes where the differential equations are expressed in a finite difference form. The numerical method solves the finite difference equations by a line by line iteration as described in ref. /1/ and, in more detail, in a two-dimensional version by Gosman and Pun /8/.

Turbulent boundary layer profiles describe the flow in the vicinity of walls and this description is used to link the difference equations to the wall by means of the wall functions given in ref. /1/.

An economy of grid nodes is achieved by describing the inlet conditions as a developed wall jet in an area in front of the opening ($0 < x/H < 1.14$ in fig. 1). This precludes the need for a large number of grid nodes to describe the flow in the immediate vicinity of the inlet opening.

3. VELOCITY DISTRIBUTION

The measurements and all calculations discussed in this paper were made in a room with a length, L, three times the height, H, and a width, W, equal to the height. The inlet opening has an area, a, of the relative size $a/A = 0.00126$ where $A = H \cdot W$, except in fig. 4.

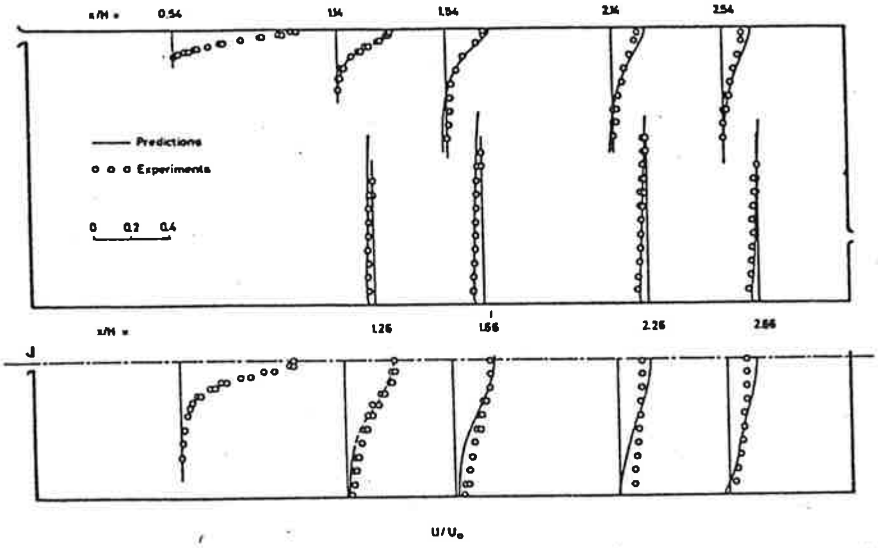

Fig. 1 - Measured and calculated velocity profiles in a room with a small side-wall mounted supply opening. L/H = 3.o, W/H = 1, and a/A = o.oo126.

Fig. 1 shows comparisons between calculated velocity profiles and the measurements of Blum /9/ which were obtained with a Pilot-tube. Velocities correspond to a Reynolds number of Re = 93ooo.

The upper figure shows a vertical section, and the lower a horizontal section through the centre line of the inlet opening. The results indicate a discrepancy of around lo % in the horinzontal spreading rate of the jet adjacent to the ceiling and near to the side walls at x/H = 2.14. The decay of maximum velocity is also slightly underpredicted, with an associated difference of 5 % in the centre line at x/H = 2.54. The general agreement is, however, satisfactory and in the reserve flow, for example, the discrepancies are below 1 % of the maximum flow velocity.

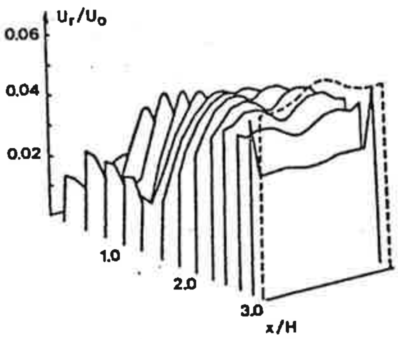

Fig. 2 - Distribution of maximum velocities through the occupied zone of the room.

The complicated three-dimensional structure of the flow in the occupied zone establishes the need for a condensed depiction of the velocities as shown on fig. 2. U_r is the maximum velocity on a vertical line through the occupied zone. The dotted curve shows the velocity distribution from the deflected jet close to the end wall. These values, and the values close to the side walls, are not normally included in the occupied zone. Fig. 2 therefore shows that U_r has a maximum in the area $x/H \sim 2.0$ with $U_{rm}/U_o = 0.056$ and a minimum at $x/H \sim 1.0$ with $(U_r/U_o)_{min} = 0.007$. The average velocity in the occupied zone is $(U_r/U_o)_{avg} = 0.035$.

The velocity U_r is normalized with U_o and it is independent of the Reynolds number owing to the level of turbulence in room air flow. A supply velocity of 5 m/s therefore means a maximum velocity in the occupied zone of 0.28 m/s and a minimum velocity of 0.04 m/s.

Closer examination of the computed velocities shows the highest values in the vicinity of surfaces. Those velocities are given on fig. 3 close to the side wall, the floor and close to the end wall opposite the supply opening. The max. velocity U_{rm} is found in an area with strong reverse flow deflected from both the end wall and side walls.

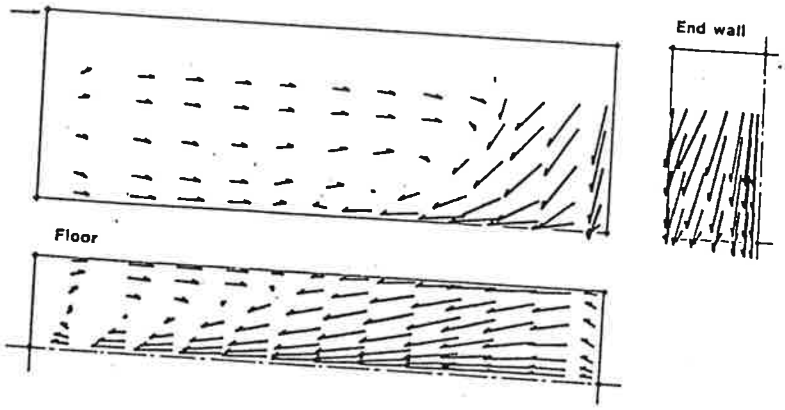

Fig. 3 - Velocity distribution close to side walls, end wall and floor of the room.

The minimum velocity $(U_r/U_o)_{min}$ is connected to two vortices rotating about near vertical axes in the area below the supply opening.

The influence of the size of the supply opening has a practical relevance for designers. The results of fig. 4 demonstrate this influence and relate to an opening with a small aspect ratio (square, round etc.) and to a room of square cross section (W/H = 1) and a length to height ratio of 3.0. The maximum velocity in the reverse flow, U_{rm}, is plotted against the relative area of the inlet, a/A, and it is clear that increasing the size of the supply opening also increases the maximum reverse velocity if the inlet velocity U_o is kept constant. Closer examination of fig. 4 indicates that, for small openings, U_{rm}/U_o tends to vary as $(a/A)^{0.5}$, i.e. the maximum reverse velocity is approximately proportional to the square root of the momentum flow rate at the inlet opening $\sqrt{a} \times U_o$. This is in accord with the findings of Skåret /10/ and Jackman /11/ for rooms with side-wall mounted diffusers.

In practice the flow rate is commonly determined by air refreshment requirements, i.e. aU_o must be assumed constant; then U_{rm} increases with decreasing a, and tends to vary as $a^{-0.5}$ for small a/A ratios. In figure 4 the results of the two-dimensional calculations have been included, ref. /3/, and it is clear that although U_{rm} tends to vary as $a^{-0.5}$ in both cases for constant mass flow rate, the two-dimensional values are higher (10 % at a/A = 0.003 and 30 % at a/A = 0.01).

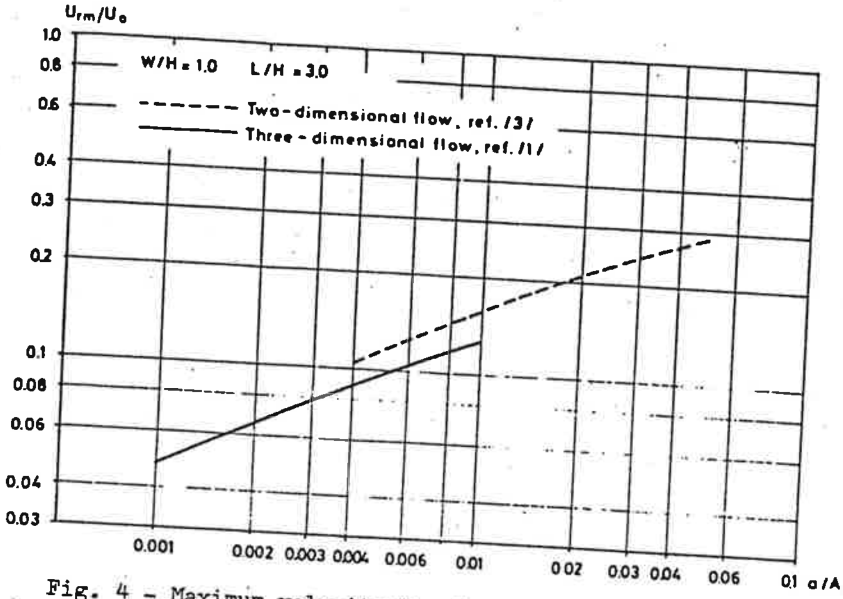

Fig. 4 - Maximum velocity U_{rm} in the occupied zone as a function of supply area a/A for openings with small aspect ratios (three-dimensional flow) and for linear diffusers (two-dimensional flow).

4. ENVIRONMENTAL QUALITY

The environmental quality can be expressed by the maximum velocity in the occupied zone U_{rm} , or by the average velocity $(U_r)_{avg}$, but it is more relevant to evaluate the velocities in connection with the temperatures in the room and the activity levels and clothing of the occupants.

Fanger /12/ has developed a thermal comfort index which will be used to calculate the distribution of "Predicted Percentage of Dissatisfied" (PPD) in the room of fig. 1 and 2. Fanger defines a dissatisfied person as one who would vote "cool, cold" or "warm, hot" in the seven point psycho-physical ASHRAE scale. He shows that even with a perfect environmental system which creates absolutely uniform conditions in the occupied zone one cannot attain a PPD value lower than 5 % for similarly clothed people in the same activity. This is due to the variance in the thermal sensation of a group exposed to the same environment (2.5 % are cold-dissatisfied and 2.5 % are warm-dissatisfied).

The assumptions for the following figures are summer conditions and sedentary activity which is expressed as light summer clothing of o.5 clo and a metabolic rate of 1 met (= 58 W/m²). It is also assumed that the flow is isothermal corresponding to a small thermal load of the room.

The curves on the left side of fig. 5 show the distribution of PPD in the occupied zone in the case of a supply velocity of 5 m/s and an air temperature T of 25°C. The percentage of dissatisfied follows the velocity level, see fig. 2, because the temperature is low. The maximum value of PPD is 22.5 % and the minimum value is 7.4 % corresponding to a velocity below o.1 m/s. The mean value, PPD_{avg} , for the room is 14.6 % and this may be used to express the level of thermal comfort of the whole occupied zone in the given situation. The curves on the right side of fig. 5 show the distribution of PPD at the temperature level of 28°C. The lower PPD values are obtained in areas with high velocities because cooling by forced convection compensates for the high temperaturelevel. The minimum, maximum and mean values of PPD are respectively 7.7 %, 18.6 % and 12.1 %. Although the mean value, PPD_{avg} , only differs a little in the two cases, fig. 5 shows that the complaints involve quite different areas of the occupied zone.

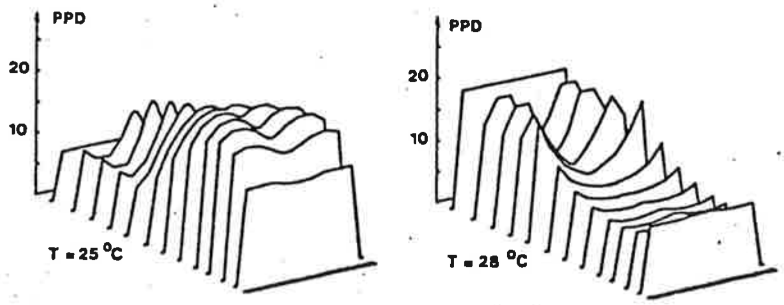

Fig. 5 - Distribution of "Predicted Percentage of Dissatisfied" (PPD). Supply velocity U_o = 5 m/s.

It is obvious from fig. 5 that there exists a minimum value of PPD_{avg} between 25°C and 28°C. This is quantified in fig. 6 where PPD_{avg} is plotted against the room air temperature T and it has a minimum value of 5.8 % for T ~ 26.5°C (U_o = 5 m/s). This minimum value is the "Lowest possible Percentage of Dissatisfied" (LPPD), ref./12/, and the difference between LPPD and 5 % dissatisfied is a figure of merit for the nonuniformity of the room which characterizes the ventilation system of the room.

The difference between PPD_{avg} and LPPD may characterize the thermostat setting and the capability of a control system. It is, for example, possible to analyze the consequences of using a floating temperature level during the day to achieve energy savings. Also the influence of proportional band and on-off difference for various types of controllers can be evaluated from fig. 6.

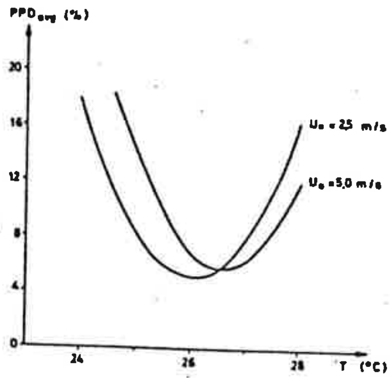

Fig. 6 - PPD_{avg} as a function of air temperature T and supply velocity U_o.

The curve plotted for U_o = 2.5 m/s shows a decrease in LPPD (LPPD slightly above 5 %), and a general displacement to lower air temperatures. Those findings are due to the fact that a lower velocity means a decrease in velocity differences giving a more uniform environment and it also means a decrease in the cooling of the human body allowing a slightly lower room temperature.

The curves on fig. 6 conclude that low supply velocity creates optimal conditions in the special situation where the thermal load of the room is very small. But this cannot be concluded in the more general case where the air conditioning systems are dealing with a thermal load. In this case we may expect a high PPD value at low supply velocities due to large temperature differences and also a high PPD value at high supply velocities due to the velocities in the occupied zone. It may therefore be assumed that there exists an optimal supply velocity between the two situations.

It should also be mentioned that the calculation procedure gives the level of turbulence which is also of importance to the environmental quality, and which may result in further increase in the number of dissatisfied persons.

5. CONCLUSIONS

The present numerical solution of three-dimensional equations shows in general a good agreement with measurements and it gives the necessary information for the evaluation of thermal comfort in the case of isothermal flow, i.e. air velocity and turbulence intensity.

The calculations show that the maximum velocity in the occupied zone is approximately proportional to the square root of the momentum flow rate at the inlet opening at various supply areas.

It is further shown how the thermal quality of a space, expressed by PPD functions and LPPD values, can be predicted.

NOMENCLATURE

a	area of supply opening, m^2
A	area of end wall with supply opening, m^2
H	height of room, m
L	length of room, m
LPPD	Lowest Possible Percentage of Dissatisfied, %
PPD	Predicted Percentage of Dissatisfied, %
T	air temperature, deg. C
U_o	supply velocity, m/s
U_r	maximum velocity on a vertical line through the occupied zone, m/s
U_{rm}	Maximum velocity in the occupied zone, m/s
W	width of room, m
x	coordinate in the length of the room, m

Indices:

avg	average
min	minimum

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LA DISTRIBUTION DE VITESSE DE L'AIR EN SALLES GRANDES AVEC PETITS BOUCHES DE LA VENTILATION MONTEES SUR LES PAROIS.

RESUME: La solution numérique des équations tridimensionelles présentée donne des résultats bien comparable a des résultats d'essais disponibles et pourvoit l'information nécessaire d'évaluation du comfort thermique en cas d'un courant isothermique, c.-a-d. vitesse de l'air et l'intensité de turbulence.

Les calculs indiquent, que la vitesse maximum dans la zone d'occupation est approximativement proportionelle a la racine carré du courant de quantité de mouvement a la bouche de la ventilation.

De plus il est manifesté, que la qualité thermique, d'un salle, exprimée par les fonctions PPD et les valeurs LPPD, peut etre prédite.