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MEASUREMENTS ON BUOYANT WALL JET FLOWS IN AIR-CONDITIONED ROOMS

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INTRODUCTION

Side-wall-mounted diffusers placed in the vicinity of the ceiling in a ventilated room will often generate a flow of the wall jet type. The jet follows the ceiling, entrains air from the occupied zone and generates a recirculating flow in the whole room.

This paper will deal with the flow in the ceiling region. The wall jet flow is especially influenced by diffuser design and surrounding details such as distance to the ceiling and the ceiling structure. The flow is lesser influenced by other parameters in the room such as length, width, height and furnishings.

It is important to study the conditions and locations where the flow can be described as a wall jet. This description is useful when different diffusers are compared, and it is the background for calculation of "throw" and "penetration depth", see (1). It is also convenient to use the wall jet description of inlet conditions in computer predicted flow in rooms. This description makes it possible to ignore details at the diffuser as e.g. vanes which means reduction of computer storage and increased computation speed, see (2).

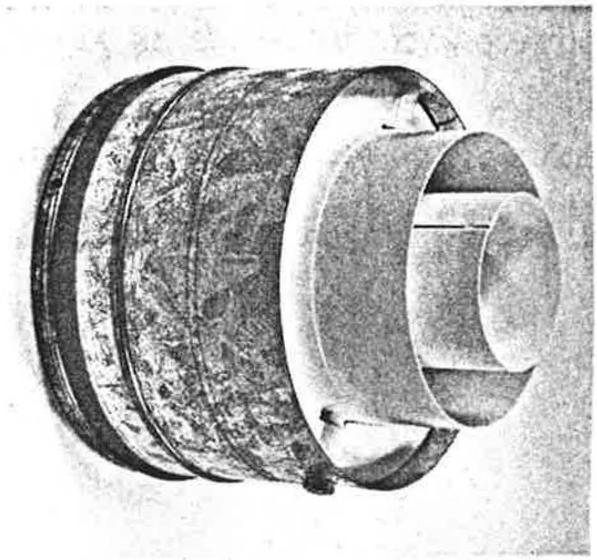

Fig. 1. Diffuser - named product A - for the experiments in the paper. Nozzle dimension 14 cm^{Φ}, and outer dimension 20 cm^{Φ}.

DIFFUSER

Fig. 1 shows the diffuser used for the measurements in the paper. It can be described as a kind of nozzle (14 cm^{Φ}) and it is normally applied to ventilation of large areas with a location far from surfaces, in which case it generates a free jet in the room.

The diffuser is selected for the experiments in this paper because its nozzle characteristics are different from other commercial diffusers. It is located as close as possible to the ceiling (3 cm from ceiling to nozzle edge) and it will generate a three-dimensional wall jet along the ceiling.

WALL JET FLOW

A three-dimensional wall jet flow is a flow established close to a surface where air is supplied parallel to the surface from a concentrated opening (round, square etc.). The wall jet conditions are fulfilled when the velocity profiles can be expressed in a single universal profile normalized with a local length and a local velocity and when growth rate and velocity decay can be expressed by the following equations, see Rajaratnam (3) and Sforza and Herbst (4).

$$\frac{\delta_y}{\sqrt{a_o}} = D_{ay} \frac{x+x_{oy}}{\sqrt{a_o}} \qquad (1)$$

$$\frac{U_x}{U_o} = K_a \frac{\sqrt{a_o}}{x+x_o} \qquad (2)$$

δ_y is wall jet thickness perpendicular to the surface at the velocity $U_x/2$, where U_x is the local maximum velocity. x is the length from the opening and a_o the area of the supply opening. x_{oy} and x_o are virtual origins of the wall jet growth and velocity decay, respectively, while D_{ay} and K_a are constants and U_o is the supply velocity.

x_{oy} and x_o are small compared to the distance x in the wall jet, and equation (1) shows that the growth of the wall jet thickness is in practice proportional to the distance from the opening with the growth rate D_{ay}. Equation (2) shows that the velocity ratio U_x/U_o is proportional to 1/x, which is typical of a three-dimensional wall jet, and it expresses that the momentum flow is preserved in the jet.

A thermal three-dimensional wall jet can be described by equations (1) and (2), and by the following equation for the temperature distribution

$$\frac{T_x - T_R}{T_o - T_R} = K_{aT} \frac{\sqrt{a_o}}{x+x_o} \qquad (3)$$

where T_x is the wall jet extreme temperature at the distance x from the diffuser, T_o the supply temperature and T_R the room or return temperature. K_{aT} is a constant for the wall jet.

The theory behind the wall jet assumes that the flow has a fully developed turbulent level, which also means that the normalized flow (velocity, turbulence) is independent of the Reynolds number. The general turbulent flow in ventilated rooms is often fully developed, see Hanel and Scholz (5), but diffusers may in some cases be used at low supply velocities giving a jet which is not fully developed. The equations (2) and (3) are in this case used with K_a and K_{aT}-factors which are functions of the supply velocity U_o or functions of the Reynolds number for the diffuser, see (6).

A Reynolds number for the diffuser can be expressed by

$$Re = \frac{U_o \sqrt{a_o}}{\nu} \qquad (4)$$

where ν is the kinematic viscosity. This expression for the Reynolds number makes it possible to compare different sizes of the same type of diffuser, but it is not possible to compare two different designs.

When the air distribution system has to remove heat from a source in the room it will result in a flow with a thermal wall jet. The flow in the wall jet will in this case be dependent on the diffuser, the Archimedes number Ar and it might be dependent on the Reynolds number at small supply velocities but independent at a higher velocity levels, see Müllejans (7). The Archimedes number is given by

$$Ar = \frac{\beta g \sqrt{a_o}\,\Delta T_o}{U_o^{2}} \tag{5}$$

where β, g and ΔT_o are the coefficients of thermal expansion, graviational acceleration and temperature difference between return and supply, respectively. The Archimedes number will in this paper be expressed by the following factor

$$\frac{\Delta T_o \sqrt{a_o}}{U_o^{2}} \qquad \left(\frac{Cs^2}{m}\right) \tag{6}$$

MEASUREMENTS AND DISCUSSION

All the measurements are made in a room which fulfill the Swedish regulation SP VVS 17 1973 on tests of supply and return openings. The room dimensions are 12 m × 12 m × 3 m and the diffuser generates a jet which is located 8 m from one side wall and 4 m from the other side wall. The diffuser is mounted close to the end wall, at a distance of 3 cm between ceiling and nozzle edge.

ISOTHERMAL WALL JET

The velocity profiles are measured at different distances from the diffuser and U_x and δ_y are obtained from the measurements. K_a is then established from a depiction of the maximum velocity U_x as a function of the distance x from the diffuser, and fig. 2 shows the level of K_a assuming $x_o = 0$ in the equation (2).

The measured K_a-faktor of 8 is in good agreement with other measurements on nozzles where the K_a-factor is in the range from 8 to 10, and it is different from many diffusers which have values in the range from 3 to 5. A high K_a-factor means a high velocity at a given distance x and thus a small entrainment of air into the jet.

It is not possible to show any systematic influence from the Reynolds number on the K_a-factor at the measured velocities down to U_o = 2,9 m/s. The scatter in fig. 2 can only be explained as some unsteadiness of the flow in the different tests.

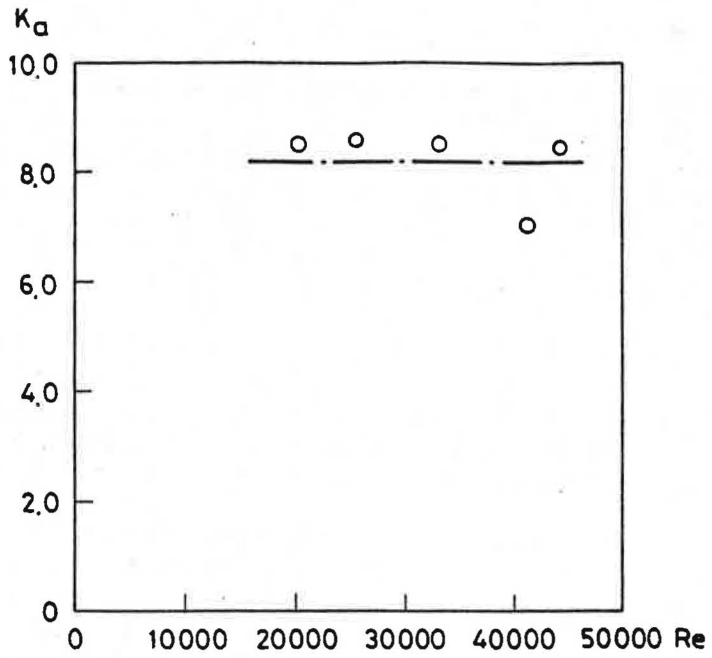

Fig. 2. K_a (equation (2)) versus Re for isothermal flow. $x_o = 0$.

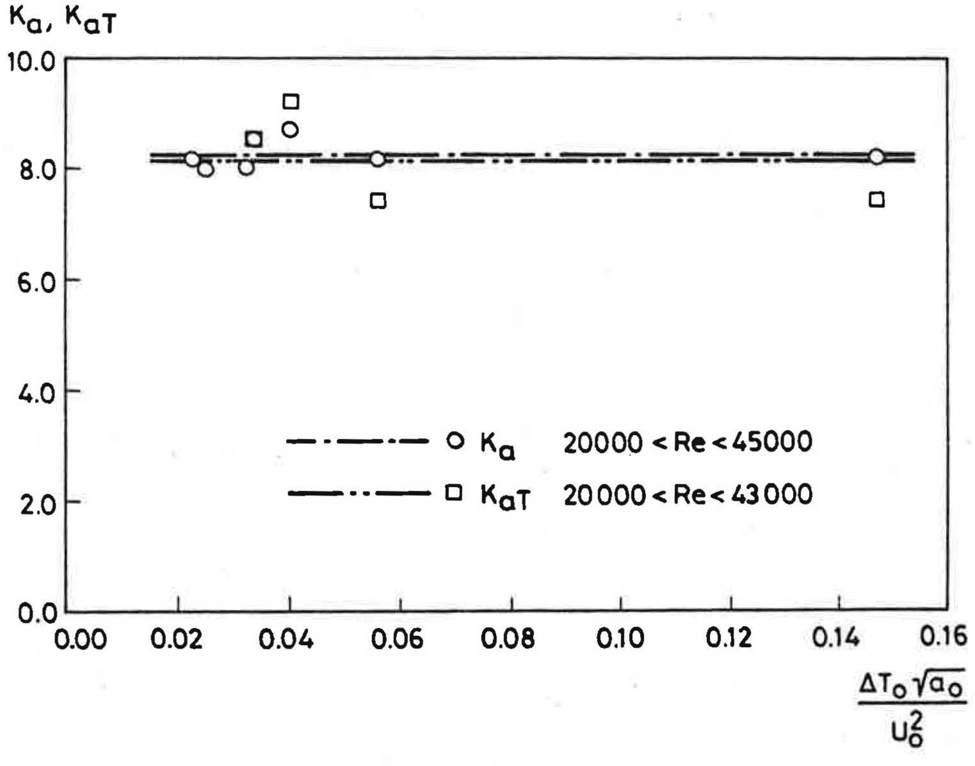

Fig. 3. K_a (equation (2)) and K_{aT} (equation (3)) versus Ar. $x_o = 0$.

THERMAL WALL JET

Experiments with thermal wall jets are obtained by supplying cool air to the diffuser and heat the floor area in the test room. The K_a-factor is established from depictions of maximum velocity U_x as a function of the distance x and the K_{aT}-factor from depictions of minimum temperature T_x as a function of the distance x from the diffuser. It is assumed that virtual origin in the equations (2) and (3) is equal to zero.

Fig. 3 shows that the K_a- and K_{aT}-factors are independent of the Archimedes number at the measuring conditions (R_e > 20000). The K_a-factor is close to the isothermal value and the mean value of the K_{aT} is slightly smaller than the K_a-factor, which is also measured earlier in free jets from nozzles, see (8).

Measurements of growth rates at isothermal and non-isothermal flow do not show any significant differences. Fig. 4 shows measurements of velocity profiles at a distance of 3 m from the diffuser. It is obvious from the figure that the increase in thickness at non-isothermal flow is unimportant for the comfort conditions in the occupied zone.

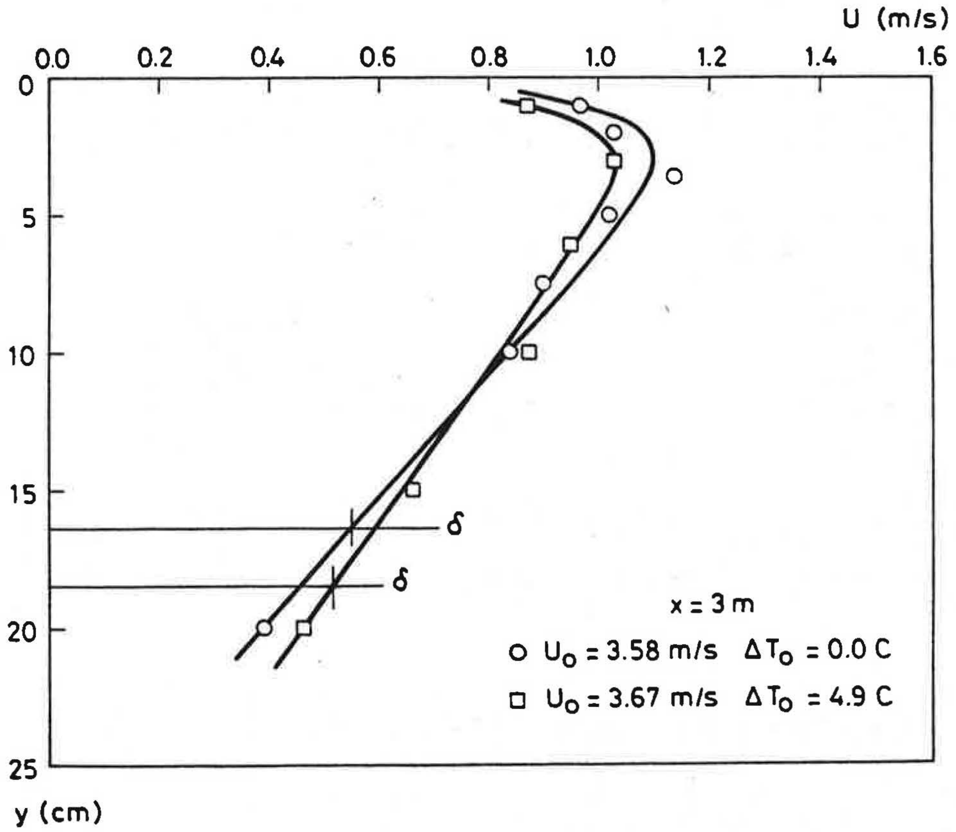

Fig. 4. Typical velocity profiles in wall jets at isothermal and non-isothermal flow.

PENETRATION DEPTH

An undisturbed wall jet will penetrate the ventilated room in the case of isothermal flow and 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 such that the wall jet will separate from the ceiling at a distance x_s from the diffuser and flow down into the occupied zone. Situations with a short penetration depth are undesirable, because the jet may have a high velocity and a low temperature when it flows into the occupied zone, and a calculation of the penetration depth is thus a part of the design procedure of the air distribution system.

Grimitlin (8) and Schwenke (9) have shown that the penetration depth for a cold three-dimensional wall jet is proportional to $1/\sqrt{Ar}$, where Ar is the Archimedes number. The measurements in fig. 5 confirm the linear relationship between $x_s/\sqrt{a_o}$ and

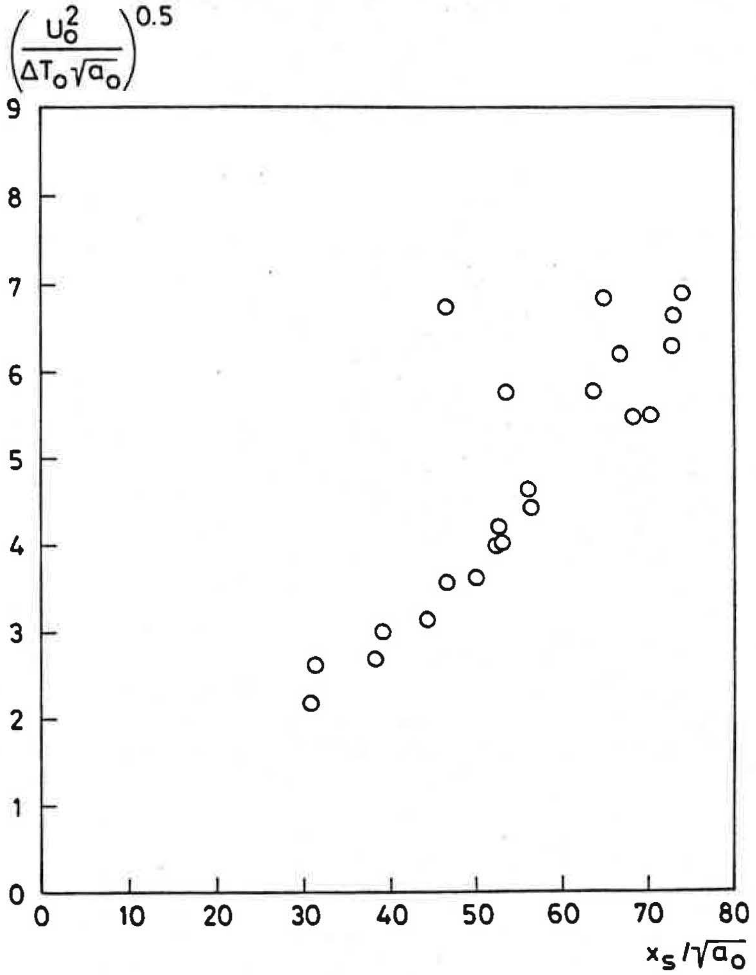

Fig. 5. Measurements of the normalized penetration depth $x_s/\sqrt{a_o}$ versus $1/\sqrt{Ar}$ for a wall jet established by the diffuser shown in fig. 1.

$1/\sqrt{Ar}$. The measurements may be influenced by the room length L for values of $x_s/\sqrt{a_o}$ exceeding 70, $L/\sqrt{a_o} = 102$.

An analysis of the forces acting on a non-isothermal wall jet leads to the following equation

$$\frac{x_s}{\sqrt{a_o}} \sim K_a \frac{1}{\sqrt{K_{aT}\, D_{ay}}} \frac{1}{\sqrt{Ar}} \tag{7}$$

If it is assumed that K_{aT} is proportional to K_a and D_{ay} is proportional to $1/K_a$ (or growth in the square root of the cross-sectional area of jet is proportional to $1/K_a$) it is possible to write the following equation, see Grimitlin (8) and Schwenke (9)

$$\frac{x_s}{\sqrt{a_o}} = K_{sa}\, K_a \left(\frac{U_o^2}{\Delta T_o \sqrt{a_o}}\right)^{0.5} \tag{8}$$

where K_{sa} is a constant dependent on parameters outside the wall jet, such as room dimensions, location of thermal load etc.

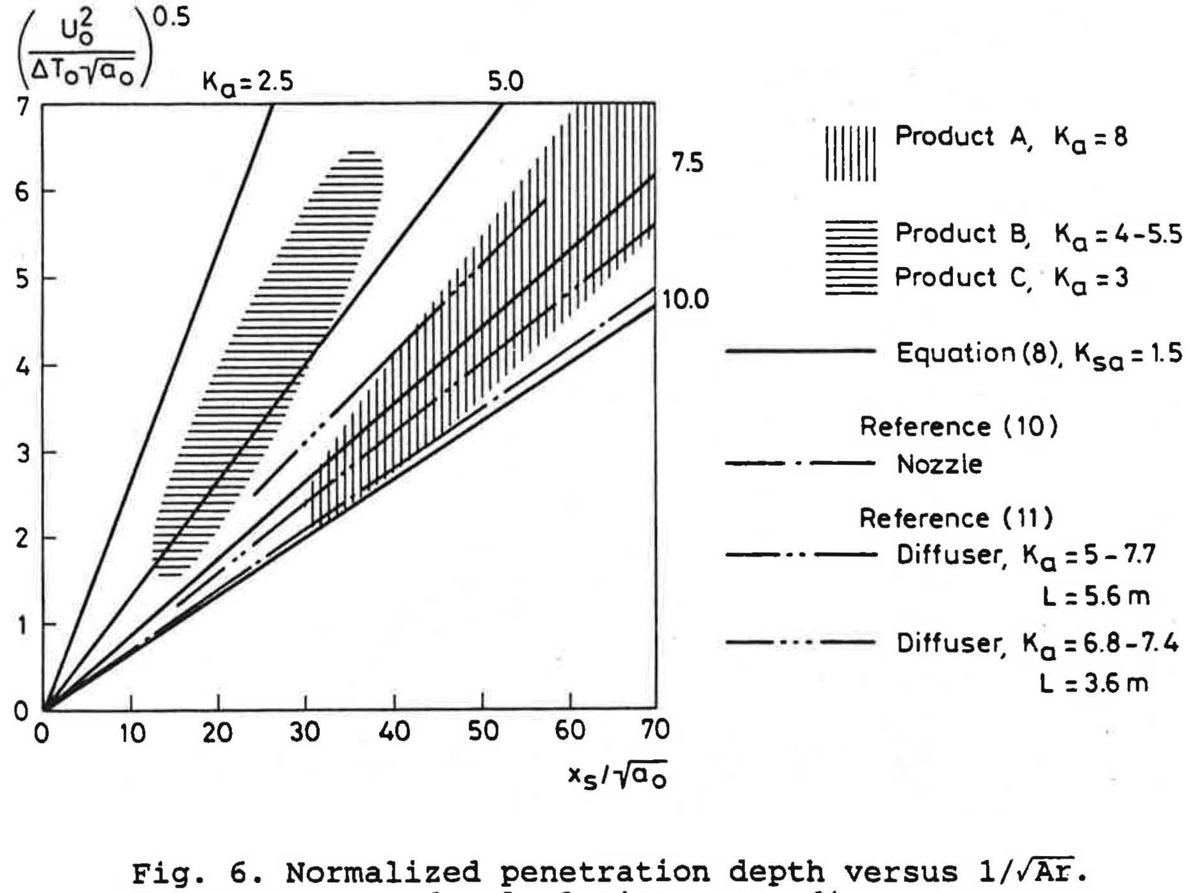

Fig. 6. Normalized penetration depth versus $1/\sqrt{Ar}$. Measurements and calculations according to equation (8).

The influence from the K_a-factor on the penetration depth x_s is obvious in fig. 6 where the measurements are compared with earlier measurements from reference (6). The diffuser in fig. 2 is named product A and the two diffusers in reference (6) are named B and C. Comparisons between measurements (product A, B and C) and values calculated from equation (8) for $K_{sa} = 1.5$ show a fairly reasonable agreement with respect to the K_a-variation. This is also supported by measurements by Sharp and Vyas (10) and Hestad (11) on wall jets from a nozzle (K_a = 8-10) and air diffusers (K_a = 5-7.7), respectively.

The discussion of the measurements in fig. 6 are complicated by the fact that the jet air flow from the diffusers has different flow pattern in the area where it leaves the ceiling region and penetrates into the occupied zone. Fig. 7 shows the main characteristics of the different flow pattern and the individual definitions of the penetration depth x_s.

The penetration depth x_s for product A is measured as the distance up to the area where the entrainment flow meets the jet at the ceiling surface (seperation line). The jet travels a further distance Δx_s before it flows down into the occupied zone, where Δx_s is typically about 2 - 2.5 m. There is a tendency for downward flow at the edge of the wall jet outside the symmetry plane as indecated by dotted lines in fig. 7, but the velocity is very low.

The penetration depth for product B in reference (6) is defined as the point of intersection between ceiling and a line through maximum velocity in the measured velocity profiles. This definition results in a short penetration length and a long distance Δx_s before the jet reaches the occupied zone.

The wall jet for product C leaves the ceiling very abruptly and the distance x_s is therefore independent of the different definitions.

The agreement between measurements and calculations in fig. 6 will be further improved if the penetration depths for products B and C are defined as the distance to the separation line. The measuring points for product B will be located at higher x_s-values at a distance from the measuring points for product C. This will also correspond to a smaller Δx_s for product B and it supports the use of equation (8) as a formula for estimates of the penetration depth for different grilles and diffusers.

Fig. 7 shows a variety of flow patterns. Large Δx_s-values, steep flow into the occupied zone and downward flow at the edge of the jet will all mean different perception of thermal comfort at a given penetration depth. A design procedure ensuring a given penetration depth at maximum thermal load will therefore only give a rough assurance of thermal comfort in the area.

The penetration depth is dependent on the room dimensions and the thermal load distribution and the measurements in fig. 5

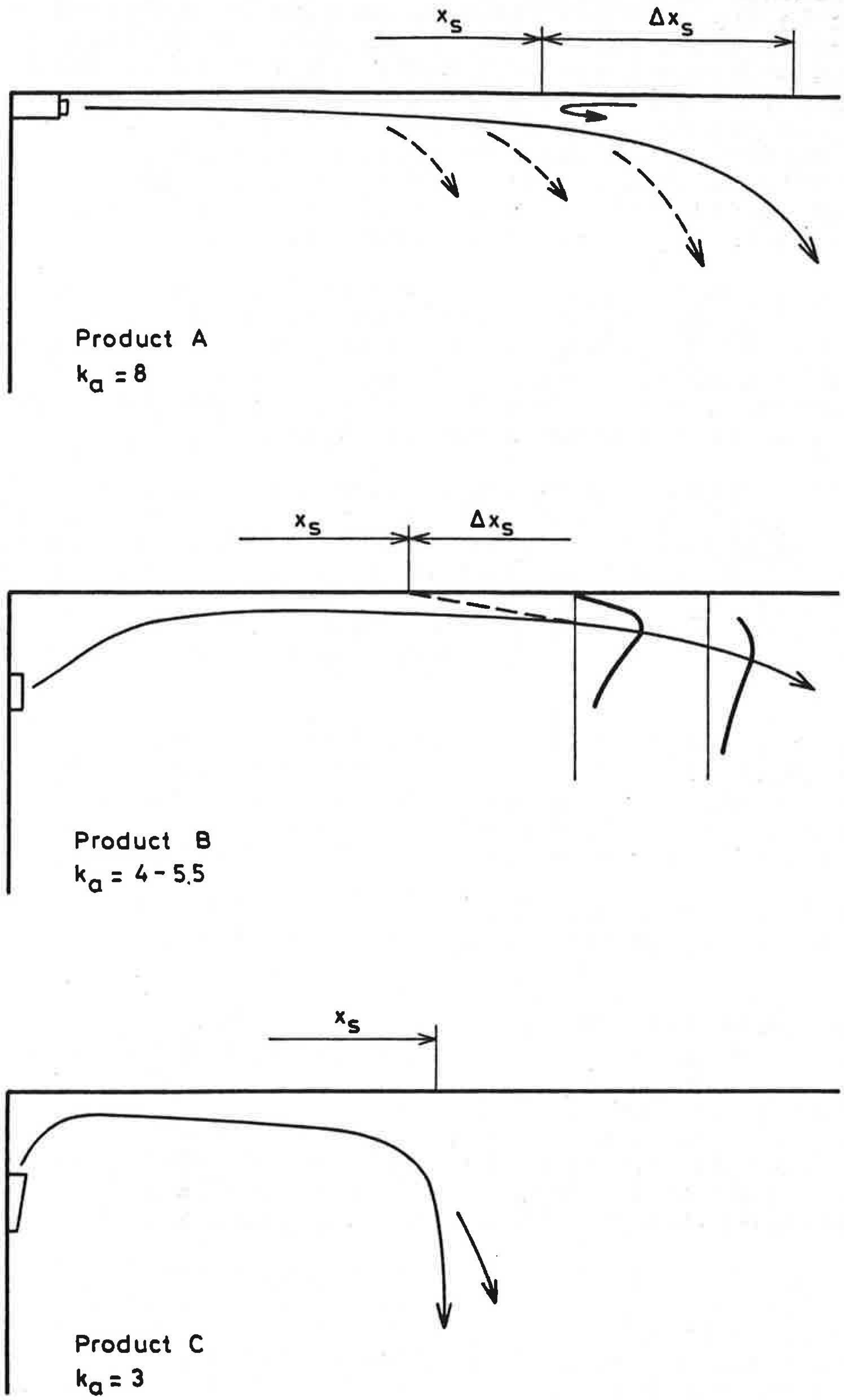

Fig. 7. Sketches of wall jets, penetration depth and flow for different diffusers.

and 6 are therefore only strictly valid for large rooms with an evenly distributed heat source over the whole floor area.

CONCLUSIONS

Measurements on a nozzle-like diffuser located close to the ceiling show that the established velocity distribution can be described as a wall jet. The flow is independent of the Reynolds number for all the tests ($U_o \geq 2.9$ m/s). Temperature distribution in the flow can also be described by wall jet equations, and it is independent of the Archimedes number for all practical supply velocities and temperature differences.

The penetration depth is a function of the Archimedes number. Measurements show that the penetration depth x_s is proportional to $1/\sqrt{Ar}$ and, furthermore, comparisons with earlier measurements show that the penetration depth x_s can be assumed to be proportional to the diffuser coefficient K_a.

It is shown that the character of the jet flow after separation changes with different grilles and diffusers.

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