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Stratified flow in ventilated rooms – a model study

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STRATIFIED FLOW IN VENTILATED ROOMS - A MODEL STUDY

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Introduction

The physical principale of ventilation by displacement is based on stratified flow with a stable interface between an upper zone with light air and a lower zone with denser air. This stable interface acts as a lock that will hinder the transport of contaminants between the zones. The need for better understanding the physics involved is the motive for carrying out this study. We want to stress that the experimental objectives were *not* to try to compare results obtained from the model with results from a full size room. In fact water was selected as operating fluid in order to suppress the radiative transport of heat that occurs in rooms. In water the temperature field is closely connected to the flow field and consequently to the stratified flow that occurs, while this is not the case in air. It is well known that, at the same Rayleighs number in both air and water, the temperature profile in air is more linear than in water and therefore does not reflect the stratified flow that occurs.

This paper is a continuation of the paper presented by Sandberg and Lindström (1987). The experiments presented in this paper have been carried out in a new model. This new model was equipped with double walls with a gap between then where water was circulated to keep the walls at a constant temperature. The objective of this new study was to study the influence on the stratification in the room from the supply and extraction of heat through the walls. Another goal was to record the flow rate in plumes and boundary layers with a simple experimental technique based on the continuity principle. Finally the vertical temperature profiles were recorded with better resolution than previously.

Factors that influence the interface

Let us first consider a situation where a plume is generated by a heat source (buoyancy source) in a room. The buoyancy gives rise to plume flow, q_p, that increases as a function of height. At the same time heat transfer may occur at the walls which gives rise to boundary layer flows at the walls. Continuity dictates that the total net flow at any height, z, is equal to the supplied ventilation flow rate q_v.

$$q_v = q_p(z) \pm q_b + u(z)A \tag{1}$$

A = area of model

q_b = flow rate in the boundary layer or layers occurring at the walls and which passes through the interface
(+ direction of boundary layer flow is upwards)
(- direction of boundary layer flow is downwards)

u = velocity in the "core" region (ambient) between the plume and the boundary layers

The velocity u is the velocity of the front, see Fig. 3.

It follows from (1) that at the height, z_{stat}, where

$$q_p \pm q_b = q_v \tag{2}$$

the velocity u is zero.

If there are no other "boundary layer" flows than the plume flow then we obtain

$$q_v = q_p(z_{stat}) \tag{3}$$

This is the level of the interface given by the plume alone. Continuity means that an additional supply of heat at the walls will lower the level of the interface while extraction of heat through the walls will increase the height.

Relation (3) shows that by varying the supply flow rate, q_v, and observing the level of stratification, z_{stat}, we can record the flow rate in the plume as a function of height. This is the basis of the method of recording the flow rate in the plume as a function of height. When our work started it was not known to the authors that this method had been proposed by Baines (1983).

We will make a more thorough analysis of one situation obtained in the model, namely an axisymmetric plume. Entrainment theory states that the flow rate from an axisymmetric plume is:

$$q_p(z) = B^{\frac{1}{3}} 3^{\frac{1}{3}} \left(\frac{6}{5}\alpha\right)^{\frac{4}{3}} \pi^{\frac{2}{3}} z^{\frac{5}{3}} \tag{4}$$

The magnitude of the flow rate is characterized by the *coefficient of entrainment*, α, and the *specific buoyancy flux B* of the source:

$$B = \frac{g\beta E}{\rho C_p} \tag{5}$$

C_p = specific heat at constant pressure

E = total power output from heat source output

g = gravitational acceleration

z = vertical distance from the source (virtual origin)

β = coefficient of thermal expansion

ρ = density

By experimentally determining the flow rate as a function of height we may determine the coefficient of entrainment from relation (4).

Due to heating and cooling of the walls, heat is either supplied to or extracted from the model. The power of the heat transfer at the walls is expressed as a buoyancy flux B_w and the flow rate occurring at the walls is denoted q_w. Molecular diffusion, quantified by the diffusivity D (m^2/s), is also involved. In addition to these parameters there are also the width b of the plume flow, height of model H, height of inlet, h_{in}, and height of heat source, h_s. By partly following Baines (1983) and carrying out a dimensional analysis of all variables included (A, B, b, D, E_w, H, h_{in}, h_s, q_v) we can write the height z_{stat} of the interface as:

$$z_{stat} = f\left(\frac{q_v^{\frac{3}{5}}}{B^{\frac{1}{5}}}, \frac{q_v^{\frac{3}{5}}}{B_w^{\frac{1}{5}}}, \frac{A}{b}, H, \frac{D}{u}, h_s, h_{in}, \text{inlet},...\right) \tag{6}$$

The first term is the volume flux of the plume flow while the second term is the flow rate in the boundary layer flows at the walls. Note that the expression above for the flow in the boundary layer is only a formal one. A boundary layer flow does not have the above functional form; however, it does give rise to the correct dimension. The third term is the ratio between area of model and area of plume flow (b is the width of the plume). If the area of the room is reduced the inertia of the flow in the core region is increased and this should tend to prevent the interface formation. In the fourth term the height of the model appears because the flow in the plume increases as a function of height. With a given cros-section A, increasing the height of the model will give rise to an increased inertia of the flow in the core region. This again will prevent the interface formation. The fifth term gives the thickness of the interface, δ, due to a balance between molecular diffusion and convection. We do not expect this term to directly affect the level of the interface. The thickness of the interface will be discussed below

If the height of the source is small then the flow generated by the source may be swept away by the air coming from the inlet. Numerical simulations carried out by Davidsson (1989) have shown that a high inlet velocity of air affects the plume from a heat source located at floor level. Furthermore the inlet conditions may affect the large scale motions in the ambient and this in turn may affect the rate of the entrainment into the plume. Therefore there are several reasons for including the last three terms.

In our experiments the size of model, height of inlet and fluid were fixed. The quantities we could vary were:

- Type of heat source
- Power supplied (buoyancy) to the heat source
- Ventilation flow rate
- Power supplied to or extracted from the walls

Experiments

The tests were accomplished in a model cube with side length equal to 0.5 m. The top and bottom of the model were made out of 15 mm plexiglas. The walls consisted of two layers of 10 mm plexiglas with a 10 mm gap between. For heating or cooling the walls water was circulated in the gap with the inlet at the bottom of the wall and the outlet at the top. For adiabatic conditions the walls of the model were insulated with 80 mm polystyrene foam (U value 0.030 $W/m^2\ ^oC$) while the top and bottom were insulated by a 50 mm thick insulation. Water was used as the operating fluid. Tap water was supplied to the model, via a constant head tank, through a slot which spanned the whole width of the model. The water left the model through a slot at the top. In order to prevent calcium carbonate deposition in the model the tap water was treated with an ion exchange type water softener. Temperatures were recorded with thermocouples (copper-constantan) calibrated to within $\pm$ 0.2 ^oC.

In order to visualize the flow, a dye (methyl hionini chloridum) was added to the flow.

Plume generated by a short vertical cylinder

An axisymmetric plume was generated by resistance heating of a vertical cylinder of height 105 mm and diameter 30 mm.

Fig. 1 shows the recorded temperature when the power supplied to the cylinder was kept constant at 800 W ($B = 33\ 10^{-8}\ m^4/s^3$) and the ventilation flow rate was varied from 1 room volumes/h (the lowest curve) up to 3.8 room volumes/h (the upper curve).

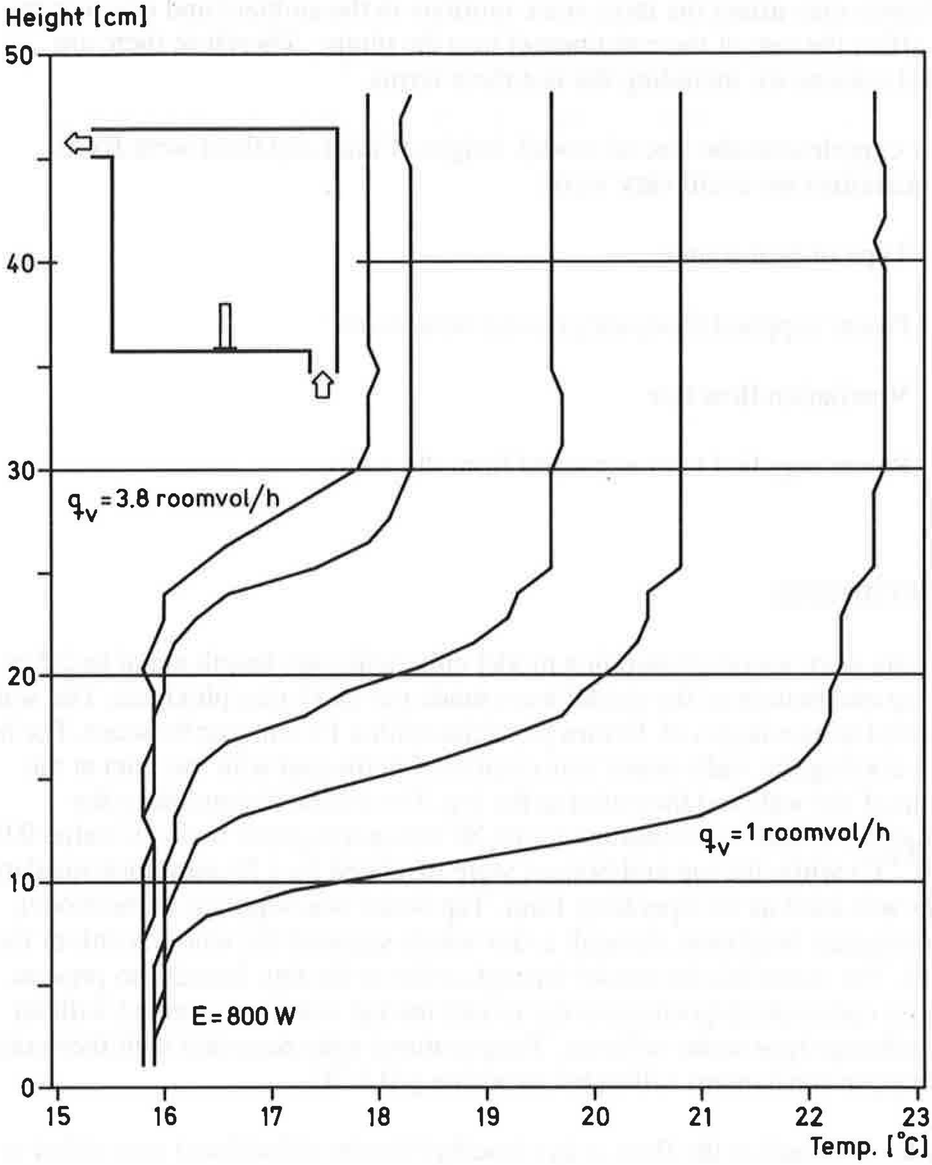

Figure 1. Temperature in the core region at constant power supply and constant wall temperature but varying ventilation flow rate

We notice from Fig. 1 that there is an interfacial boundary layer that matches the constant temperature in the lower half of the model with the constant temperature in the upper part of the model.

We see that interface consists of a linear region and at the edges of this region there are large jumps in temperature. When dye was added to the plume flow a sharp interface appeared. The height of this interface corresponded approximatively to the center of the temperature interface. There are several alternative mechanisms for explaining the thickness of the interface. We may for example imagine that there is a balance between thermal diffusion and convective turbulence above the interface. The convective turbulence is characterized by a time scale, τ_t. This gives us the thickness

$$\delta \sim (D\tau_t)^{1/2} \qquad (7)$$

However, our recorded thickness gives an unrealistic long timescale for the turbulence and presumedly the turbulence level in the upper zone is very low. Alternatively we may as Baines (1983) did assume a balance between the transport by the mean velocity field, u, and the thermal diffusion.

The mass conservation equation is

$$\frac{\delta}{\delta x}(uT) = D\frac{\delta^2 T}{\delta x^2} \tag{8}$$

where

T = Temperature

x = Distance from the center of the interface, $x \equiv z - z_{stat}$

In order to be able to solve equation (8) we linearize the velocity field u (see equation (1)) around the velocity at the center of the interface

$$u(x) = -\left(\frac{1}{A}\frac{d}{dz}q_p\right)_{z_{stat}} \cdot x \tag{9}$$

After introducing the variable

$$\eta(x) = \left(\frac{1}{2 \cdot AD}\left(\frac{d}{dz}q_p\right)_{z_{stat}}\right)^{1/2} \cdot x$$

the solution of equation (8) becomes

$$T(\eta(x)) = 0.5\,(T_u + T_b)\,(1 - \mathrm{erf}\,(\eta(x)) + T_u\,\mathrm{erf}\,(\eta(x)) \tag{10}$$

where T_u is the constant temperature of the region above the interface while T_b is the temperature of the region below the interface. The temperature at the center of the interface is $0.5\,(T_u + T_b)$.

Alternatively we may carry out a scale analysis of equations (8) and (9). We obtain that the thickness, δ, of the interface is proportional to $\delta \sim (DA/(dq_p/dz))$. By using the exact solution (10) we can be more precise and the thickness of the *linear part* becomes approximately

$$\delta = 2.5\left(\frac{DA}{\frac{d\,q_p}{d\,z}}\right)^{1/2} \tag{11}$$

Expression (11) indicates that the thickness should decrease with increasing dq_p/dz, that to say with increasing height. However, from Fig. 1 we see that the with increasing height the linear region steepens. As a consequence of this the thickness of the linear region became constant (5.6 cm). According to (11) the thickness should lie in the range 2.5-3.8 cm. For air in a typical office room expression (11) predicts a thickness in the range 0.25-0.45 m.

The next two figures shows the effect, on the temperature in the core, of cooling and heating of the walls. In both cases the supply temperature of the ventilation flow was about 20°C. The heat supply to the cylinder amounted to 400 W. Fig. 2 shows the effect of reducing the temperature in the supply to the wall from 20°C to 6°C. Due to the cooling of the walls approximately 50% of the heat supply was transferred through the walls. Observation of the dyed flow showed a weak masstransport from the upper to the lower region.

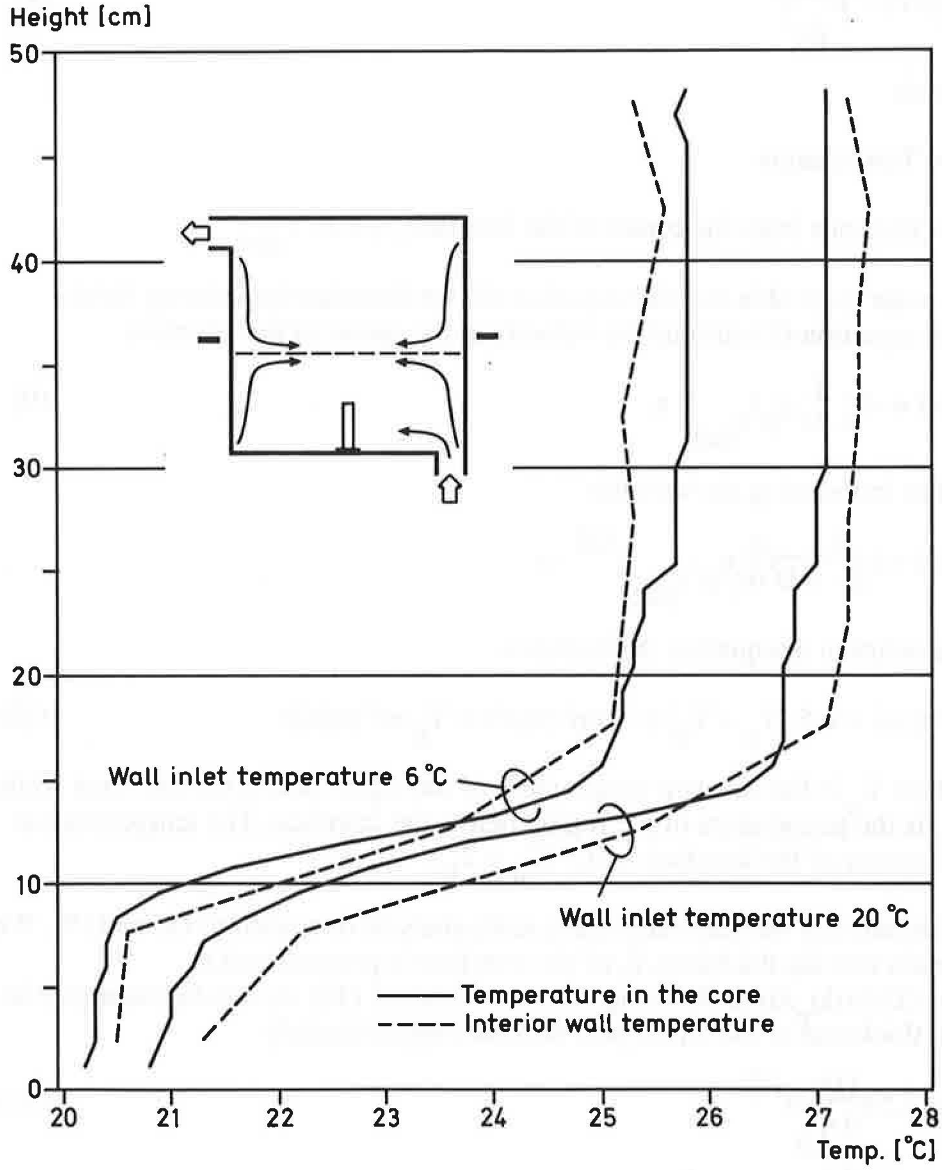

Figure 2. The effect of reduction in wall temperature. Constant power supply (400 W) and ventilation flow rate

By reducing the wall temperature the temperature drops in the whole model and the height of the interface increases. However, the change in height was very small. This may be ascribed to the circumstance that there is only a very small fraction of the falling boundary layer flow that passes through the interface. The mayor part of the flow is deflected at the interface. As result of this there is an almost pure translation to the left of the temperature curve. The drop in temperature is larger in the region above the interface because this region is exposed to a larger fraction of the wall surface than the region below the interface. When the wall temperature is reduced the shape of temperature profile is changed. At cooling of the walls the temperature profile is rounded off. This is a possibly an effect of horizontally directed flow above the interface. Fig. 3 show the effect of warming the walls.

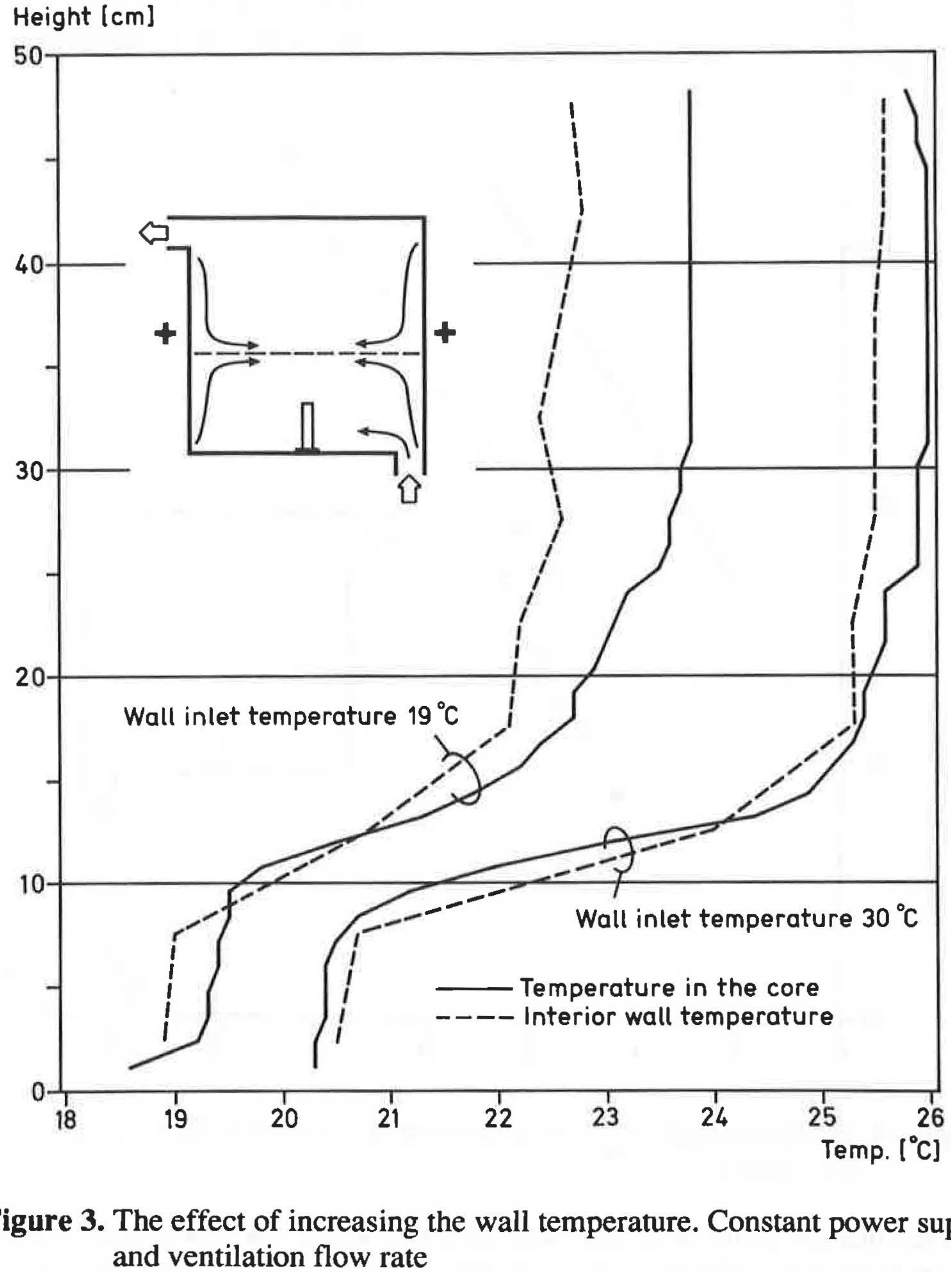

Figure 3. The effect of increasing the wall temperature. Constant power supply and ventilation flow rate

In this case we cannot see any change in height of the interface. This may be explained by fact that the power supply to the core region of the model from the wall now is very small and therefore the boundary layer flow at the walls do not pass the interface.

The next figure, Fig. 4, shows the flow rate in the plume obtained by varying the ventilation flow rate. The coordinates of the horizontal axis are based on relation (4) so that a straight line would appear with a constant coefficient of entrainment. Fig. 2 indicates the slope that would appear with a coefficient of entrainment equal to 0.0833, which is the mean value of several values reported in the literature (Rodi W (1982)).

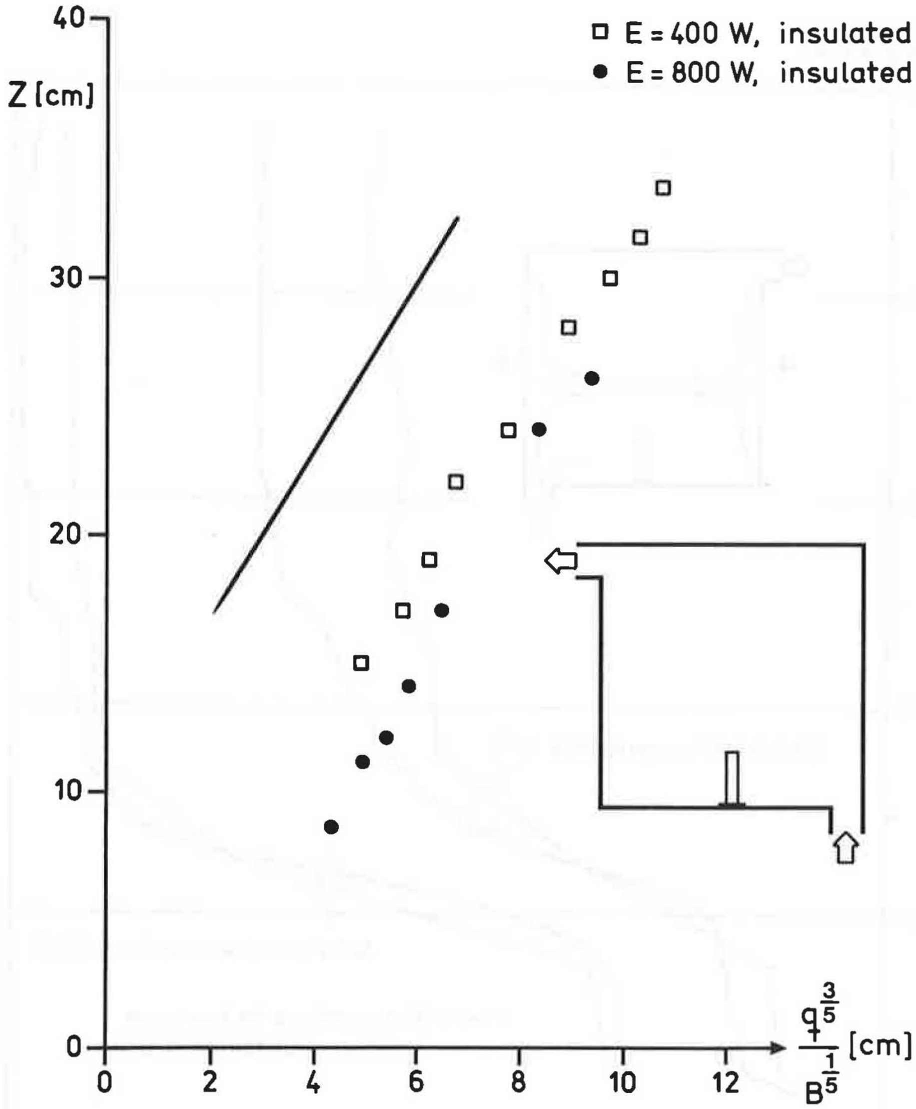

Figure 4. Interface height, z_{stat}, as a function of volumetric flow rate in the plume

We see that the points from tests with different source buoyancis (power supply) do not occur on the same curve; however, they do run parallel. One explanation of this behavior is that we have a heat source with a certain extension and the virtual origin of the plume is changed when the power supply is changed.

The coefficient of entrainment was determined by using the information presented in Fig. 4 and relation 4. The value of the entrainment coefficient became 0.083.

Boundary layer flow from a vertical slender cylinder

By a slender cylinder we mean a cylinder whose radius is less than the thickness of the boundary layer flow. The vertical cylinder had a diameter of 8 mm and spanned the whole height of the model, see Fig. 5.

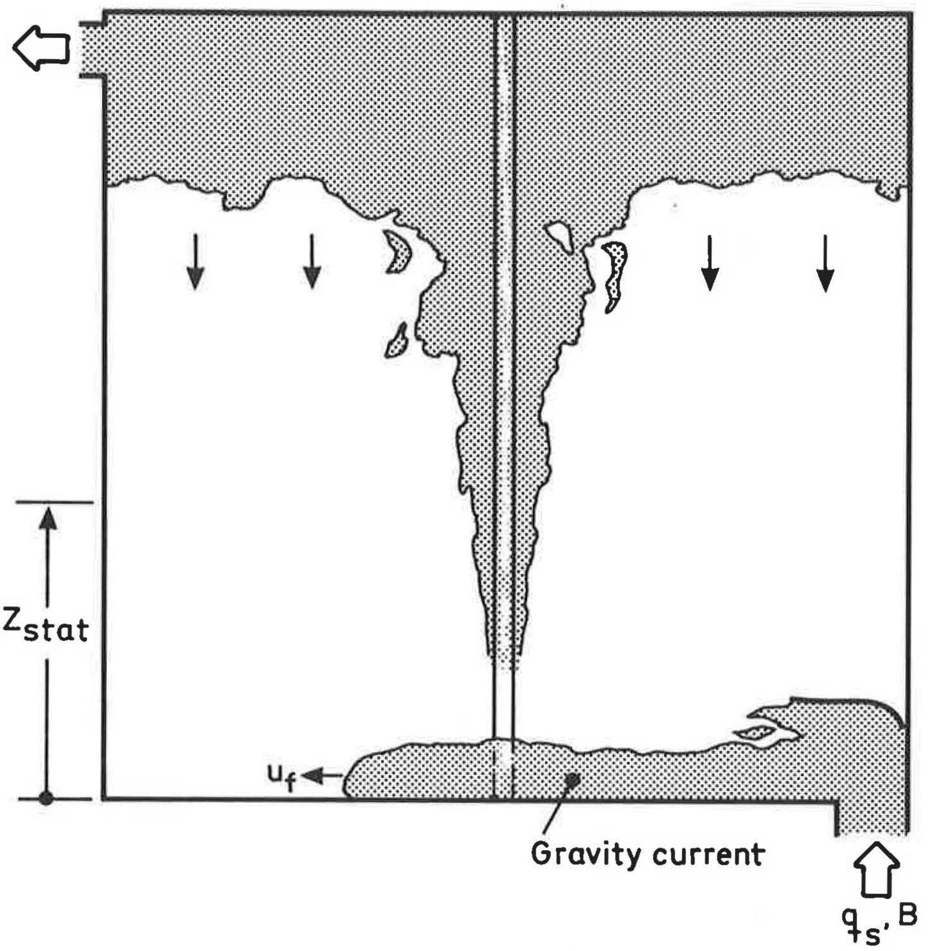

Figure 5. Sketch of flow with a slender cylinder in a room

The next two figures show how the vertical temperature profile is developed in the model. The initial (t=0) temperature was uniform; then the heat source was turned on and inflow of water was started. The inflowing water had a lower temperature than the ambient. Fig. 6 shows a visualization of the inflowing water.

In Fig. 6 an advancing front appears with the typical raised head of a two-dimensional gravity current. When the gravity current arrives at the opposite side of the model it is reflected. A wave is now formed which propagates back towards the supply opening. The next figure, Fig. 7, shows the recorded temperature in the ambient (core region). The vertical temperature profiles were recorded for two hours with an interval equal to 4 minutes. The temperature profile is built up from below by the inflowing water and from above by the downward moving front of warm water generated by the heat source, see Fig. 5. The final, almost linear temperature profile, is probably build up by detainment from the boundary layer flow around the cylinder in contrast to a case with a buoyancy source at a fixed height.

Detrainment must occur with a continuous vertical buoyancy source that spans the whole height of a room and therefore one should not expect entrainment theory to be applicable.

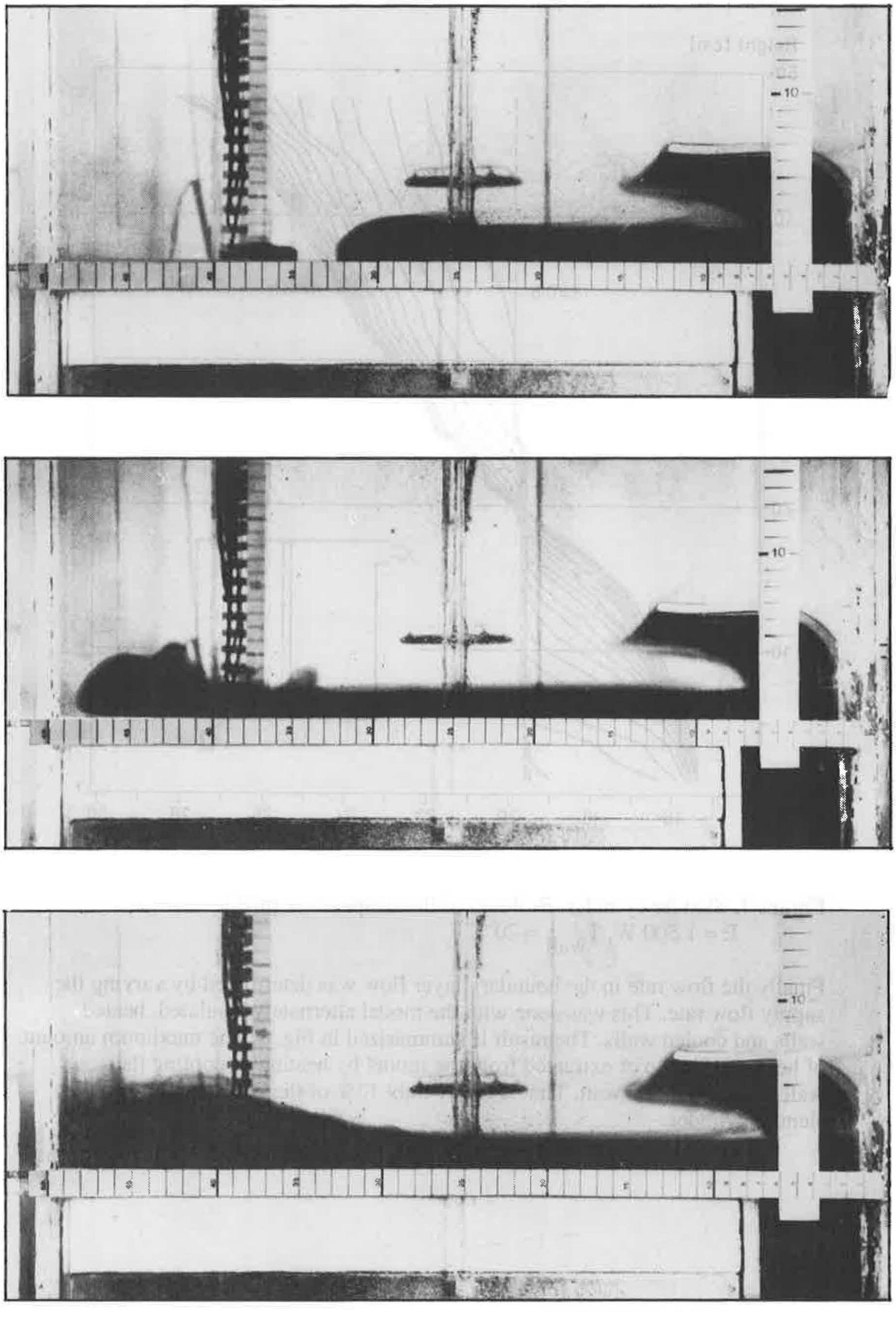

Figure 6. Flow visualization of incoming water

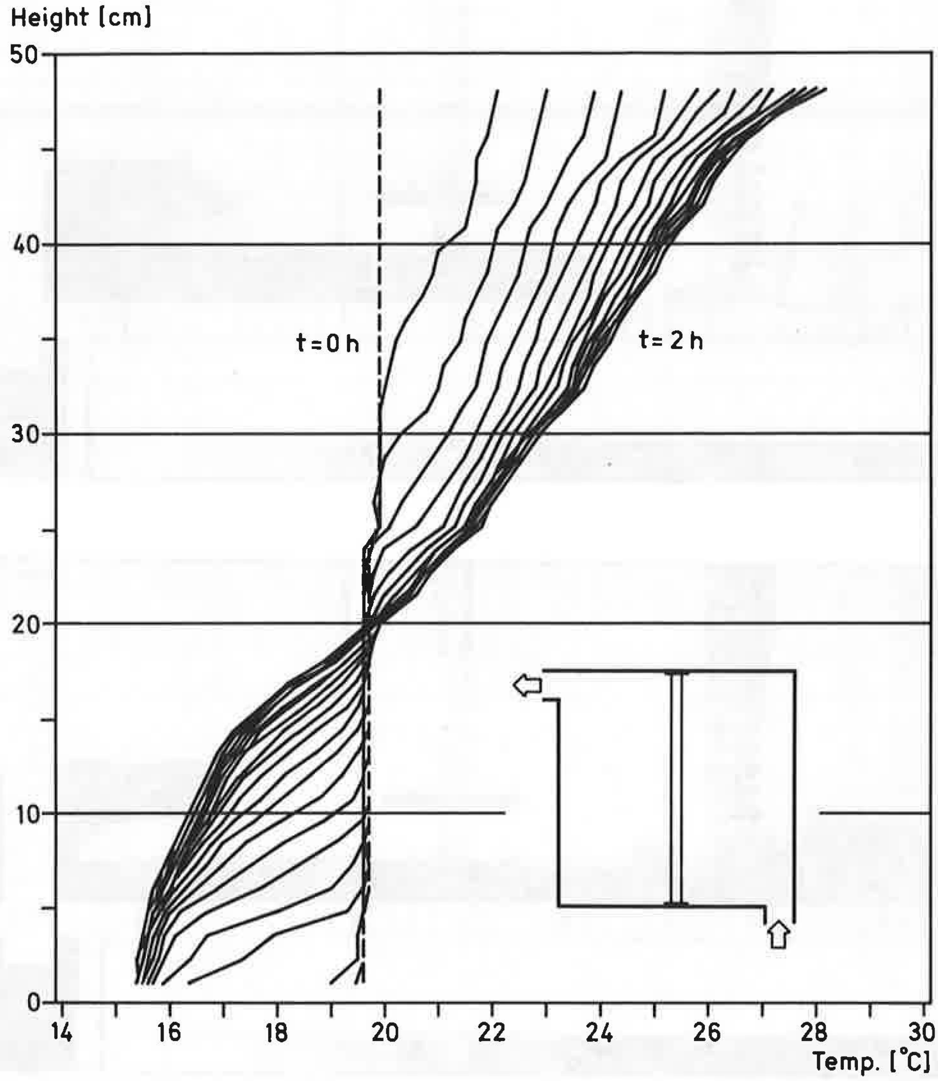

Figure 7. Slender cylinder. Build up of the temperature profile
$E = 1\ 500$ W, $T_{wall} = 20^{o}C$

Finally the flow rate in the boundary layer flow was determined by varying the supply flow rate. This was done with the model alternately insulated, heated walls and cooled walls. The result is summarized in Fig. 8. The maximum amount of heat supplied to or extracted from the model by heating or cooling the walls was about 200 watt. That is to say only 13% of the heat supply from the slender cylinder.

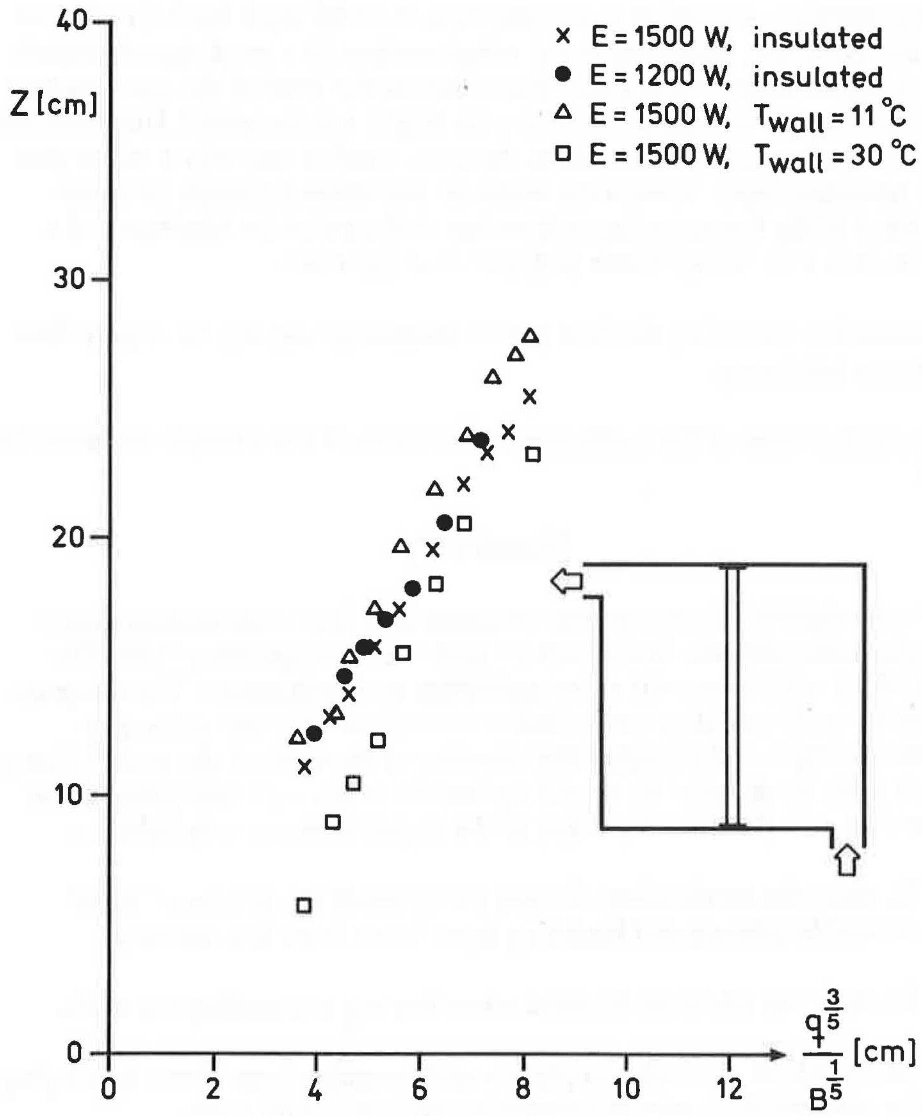

Figure 8. Slender cylinder. Recorded flow rate in the boundary layer flow

We see from Fig. 6 that we obtain the expected behavior. Heating of the walls lowers the level of the interface while cooling increases the level of the interface. However, the effect on the height of the interface is very small. This must be a consequence of the relatively small heating or cooling effect from the walls.

Conclusion

The experiments showed that the upper and lower zones in the room are separated by quite a thick interfacial boundary layer. This layer is somewhat thicker than what can be expected assuming that the interface is a region of one-dimensional convection-diffusion control. The thickness of the interface did not change with height in contrast to what the theory predicts by assuming a convection diffusion control.

Additional supply on extraction of heat at the walls gives rise to a well defined interface, according to the observation of the dyed fluid. Cooling of the walls gave rise, according to the visualizations, to a weak mass transport from the upper to the lower zone. The effect on the level of the interface was as anticipated. At cooling of the walls the height was increased. However, the effect was very small. This is due to the circumstance that only a minor part of the boundary layer flows at the walls are transferred through the interface. Most of the boundary layer flows are deflected at the interface and a circulation is only set up below and above the interface.

The method of recording the flow rate in plumes by varying the supply flow rate seems promising.

The recorded value of the coefficient of entrainment for a plume amounted to 0.083.

Summary

Results are reported from experiments conducted in a ventilated enclosure (plexiglas cube with side length 0.5 m) with water as operating fluid. The layout of the ventilation was as in ventilation by displacement. The temperatures of the walls could be controlled by circulation of water in the gap between two layers of plexiglas that constituted the walls of the model. Temperatures were recorded in the model and on the walls. Dye was added to the flow to visualize the flow. The aims of the experiments were as follows:

- To study the mechanisms for and evolution of the interface "front" created by plumes and boundary layer flows from heat sources.

- To study the effect on the front when heating and cooling the walls.

- To record the flow rates in plumes and boundary layer flows by varying the supplied flow rate and observing the level of the front.

Acknowledgement

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