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EXTRACT

The objective of this research project is to study the effect of low air velocity supply direct to the zone of occupation in small rooms with regard to effective ventilation and thermal comfort. A two zone mixing model is used to describe the concept of, and to define the effectiveness of ventilation.

This simple stratified model predicts generally high ventilation effectiveness for ventilation systems using the displacement principle, taking advantage of stratification. Laboratory tests in an office room for 1 - 2 persons (16 m^2, 2.8 m ceiling height), using this principle, are reviewed.

Tracer gas was supplied from a simulated person in the room. Very high effectiveness was obtained during these tests. The temperature effectiveness was some lower, but compared to complete mixing it is good. Thermal comfort is obtained for a wide range of heat loads and air flow rates.

3 INDEXING TERMS: NORWEGIAN / ENGLISH

NORWEGIAN	ENGLISH
Ventilasjonseffektivitet	Ventilation effectiveness
Luftkvalitet	Air quality
Lagdeling	Stratification

Jann H. Langseth

PREFACE

The objective of this research project was to study the effects of low air velocity supply direct to the zone of occupation with regard to the effectiveness of the ventilation.

This report describes and collates work carried out in cooperation with siv.ing. Gaute Flatheim A/S. The work was sponsored by NTNF and OED.

We also wish to express our appreciation for the help, cooperation and suggestions of the people who made it possible to carry out the laboratory tests and to write this report. Special thanks to Bjørn Hesthag for help with the statistical data processing, Brit Hellem Haugen who typed the manuscript and Diana Holm who proof-read the manuscript.

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SUMMARY

The objective of this research project was to study the effects of low air velocity supply direct to the zone of occupation with regard to the effectiveness of the ventilation.

A two zone mixing model is used to describe the concept of, and to define the effectiveness of ventilation. This simple stratified model, experimentally verified by laboratory tests, predicts generally high ventilation effectiveness for ventilation systems using the displacement principle, taking advantage of stratification. Tests in an office room for 1 - 2 persons (16 m^2, 2.8 m ceiling height), using this principle, are reviewed.

Stratification was secured by supplying the ventilation air with a temperature always lower than the air temperature in the zone of occupation. Necessary heating of the room was provided for by using panel heaters under the windows. The air supply was located at the opposite wall.

For summer conditions, the effectiveness was very high under conditions without direct solar radiation at the floor. However, it was dependent on the heat load in the room and the air flow rate.

For winter conditions, the stratification was weaker and also more or less impaired by the convective air circulation created by the panel heaters and cold windows. The effectiveness was, for all tests, higher than for complete mixing.

Required supply air temperature for cooling is higher when using the described model instead of complete mixing. The improved effectiveness means that the fresh air supply to the room could be decreased without reduced air quality, compared to complete mixing. Parameters concerned with the thermal comfort were also examined.

1. INTRODUCTION

The primary aims for ventilation are to remove pollution generated in an occupied space and to supply fresh (clean) air in order to maintain air quality in agreement with applied standards, which normally means to eliminate any health risk related to air quality in the zone of occupation. In addition, ventilation is also needed for eliminating fire hazards and building damage resulting from flammable gases, humidity and corrosive gases.

Requirements are very often expressed as air exchange rates for the different rooms, which by no means is satisfactory for assigning air quality for the zone of occupation. In addition, there is a tendency to decrease the air exchange rates, mainly for the purpose of saving energy and reducing the cost of heating and ventilating. For these reasons we are now approaching the safety limits for air quality, and the effectiveness of the ventilation system becomes very important.

The ventilaton process is of a complicated nature. However, introducing the concepts of ventilation efficiency or ventilation effectiveness, these can be used as a guide for designing more efficient ventilation systems with respect to both energy usage and air quality.

Earlier investigations (Skåret,Mathisen 1983) and theory indicate that a stratified model with two communicating zones of circulation and air supply to the lower zone gives good ventilation effectiveness, i.e. lower concentrations of contaminants in the zone of occupation, when air is exhausted from the upper zone. This way of ventilating has also been used for several years in industrial plants in Norway. To verify this method for small rooms and to find a way to supply the air, a test room was built in the laboratory. The test room was a full scale model of an office room in a new airport building at Sola. Two different heat loads were supplied to the room but, mainly, summer conditions without direct solar radiation were examined. The air supply was examined with regard to air velocity and temperature gradients in the room. In another test room, winter conditions had been closer examined earlier (Skåret, Mathisen 1983). High wall air inlets have also been examined earlier (Skåret, Mathisen 1983) and the present results are compared with this.

In the first part of the report the theory and definition of ventilation effectiveness are examined. The results from the measurements of ventilation effectiveness, the results of the measurements of air velocity and temperature gradients are then shown.

2. CONCLUSIONS

In the theoretical part of the report it is shown that ventilation effectiveness can be defined in several ways. These different methods supplement each other.

Steady state effectiveness: Tracer gas or contaminants are supplied continuously from a source. The effectiveness is defined as the ratio between the concentration in the exhaust opening and the zone of occupation.

Transient effectiveness: The test room is first filled with tracer gas. The decay of tracer gas is measured without mixing. The effectiveness is defined as the ratio between the apparent air exchange rate and the nominal air exchange rate.

Temperature effectiveness: The effectiveness is defined as the ratio between the heat removed by the ventilation air from the room and the heat supplied to the occupied zone.

There are also several other ways of defining ventilation effectiveness but they are not used in the measurements.

Theoretical considerations also demonstrate that a model with supply and exhaust opening in different zones gives better ventilation effectiveness than a model with complete mixing.

Measurements in test room: In the test room ventilation, ventilation effectiveness, air velocities and temperatures were measured. The tests showed that the steady state effectiveness was a function of the airflow and the heat load. Under summer conditions without direct solar radiation, the effectiveness varied from 4 to 40. This is a substantial improvement in air quality compared to complete mixing.

For winter conditions the steady state effectiveness is more variable, depending upon the convective air currents from cold windows and heating elements. But the tests showed that the effectiveness will always be better than complete mixing. Direct solar radiation at the floor also gives a reduction in the effectiveness but it still remains better than for complete mixing.

The transient effectiveness was always better than for complete mixing. This means that the room is emptied faster of contaminants than with complete mixing and that the air supply and exhaust are situated in two different zones.

The temperature effectiveness was not as good as the steady state effectiveness, 20 - 30% better than for complete mixing. This is probably due to the fact that the heat sources and the contaminant sources are not identical.

Air velocities and temperatures were also measured 3,5 and 10 cm above the floor in a distance of 60 cm from the air supply. The velocity was found to be a function of the Archimedean number of the supply air. A further treatment of this data showed that the velocity was mainly dependent on the heat load in the room.

The temperature in the same positions was dependent on the air velocity and the height of the supply opening. Besides the air quality, the thermal comfort is of essential interest for the ventilation engineers. The result above should be related to a thermal comfort index. At the present, there is no relevant index that completely covers the situation with low air velocity supply near the floor. If we compare with the results (draft criterium) obtained by Claus C. Pedersen, it can be seen that thermal comfort is obtained for a large range of heat loads and air flow rates.

The practical conclusion that can be drawn is that the fresh air supply could be reduced substantially compared to that of a system with complete mixing, without reducing the air quality. If the total supply of air to the room is reduced, the effectiveness is rapidly decreasing. This means that the total air flow rate should not be reduced too much. An increase in return air flow should probably be considered instead. The temperature of the supply air should be 19 - 21 ^{0}C in order to avoid thermal discomfort.

In the main part of the work, only filter clothing was used in the air supply opening. The effect of the width of the air supply opening was not examined in this work, which means that the supply opening used may not be the optimum solution with regard to the thermal comfort.

To improve the conditions in the zone of occupation the temperature difference between the floor and head level of the occupants should be smoothed out. More internal recirculation of air in the zone of occupation is one way to obtain this. Preliminary experiments have shown promising results.

3. DEFINING VENTILATION EFFECTIVENESS

Ventilation effectiveness may be defined in several ways. In App. V a more complete discussion of these problems is reviewed.

As a starting point, a ventilated room with air flow rate, $\dot{V}$, is used. Contaminants are supplied from sources in the room. Local elimination of contaminants is also introduced.

If we subtract the elimination rate from the production rate of contaminants, we may, for the steady state situation, with balance in generation and removal of contaminants in the room, write the mass balance equation as follows:

$$\dot{V} = \frac{Q - \sum_{j=1}^{m} \dot{E}_j}{C_{cm} - C_S} = \text{complete mixing flow rate.}$$

Q - contaminant production rate (mass/unit time)

E_j - Elimination source no. J

C_S- Concentration of contaminant in the supply air (mass/unit volume)

C_{cm}- complete mixing concentration

At this point we may state that ventilation air flow rates can be decreased by using different approaches.

i. Reduce contaminant generation.

ii. Increase local elimination.

iii. Increase concentration differences between supply and exhaust air.

The first two approaches are obvious and the practical implications are also obvious. The third one may seem to oppose air quality. The concept of ventilation effectiveness will show that this is not the case.

If we measure the concentration in the zone of occupation, C_0, an apparent air flow rate for the room can be calculated.

$$\dot{V}_a = \frac{\dot{Q} - \sum_{j=1}^{m} \dot{E}_j}{C_0 - C_S}$$

Now, we can define the ventilation effectiveness as the ratio between $\dot{V}_a$ and $\dot{V}_{cm}$.

$$\dot{\varepsilon}_{st} = \frac{\dot{V}_a}{V_{cm}} = \frac{C_{cm} - C_S}{C_0 - C_S} = 1 + \frac{C_{cm} - C_0}{C_0 - C_S}$$

C_{cm} is equal to the mean exhaust air concentration, C_E, and is no direct measure of the air quality. Through the ventilation effectiveness, the concentration in the zone of occupation can be determined. One can see that it is efficient to design systems having $C_{cm} > C_0$. This means that aiming at complete mixing is not the best guide to efficient ventilation.

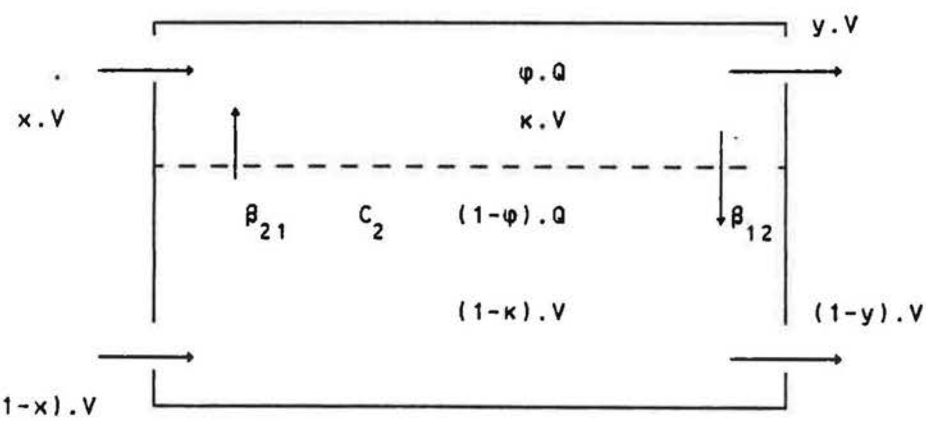


Fig. 1. Two zone mixing model. Each zone has good mixing.
ϕ-Relative production rate in zone 1
κ-Relative volume of zone 1
β_{12}-Interzonal mixing parameter
x - Part of air supplied from zone 1
y - Part of air exhausted from zone 1

To progress further in the analysis of ventilation effectiveness, a two zone single room mixing model is introduced, Figure 1.

From this model a transient method for effectiveness may be derived theoretically, see also App V. The room is filled with tracer gas. After the tracer gas supply is stopped at time, t = 0, the gas is diluted. The characteristics of the decay are shown in Figure 2.

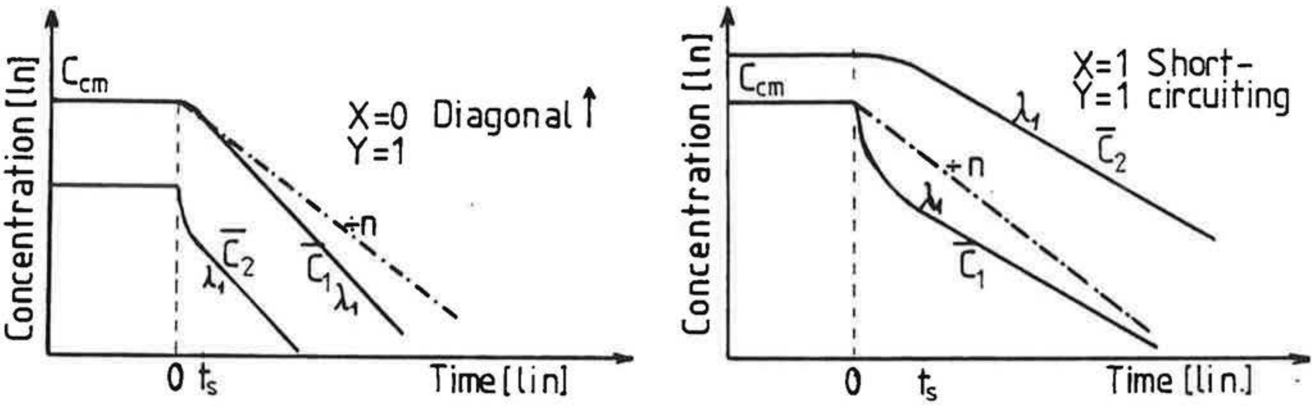


Fig. 2. Decay characteristics for a two-zone model.

Just after the tracer gas supply is stopped there will be a period of stabilization. After this period we will have a stabilized decay period. At complete mixing the slope is -n, which is the nominal air exchange rate, $\dot{V}/V$. The slope of the stabilized curves is λ_1, which is an apparent air exchange rate. In appendix V a more detailed analyse is done.

1. The whole system is controlled by the "time constant" $1/\lambda_1$, indicating that the decay curves become parallel after a certain time, t_s, using a semilog graph where the time scale is linear.

2. The slope of the decay curves during the stabilized decay period, $t>t_s$, is less than for a complete mixed decay if supply and exhaust are located in the same zone, (short circuiting). The slope is steeper if supply and exhaust are located in opposite zones (displacement).

3. The concentration is lowest in the supply zone.

The slope of the parallel curves, λ_1, omitting the sign, is an apparent air exchange rate, characterizing the whole room, not specifically the zone of occupation. The ratio - λ_1/n is a transient ventilation effectiveness, ε_{tr}. The information it gives is first of all, whether the ventilation system has a short-circuiting or not. The influence on air quality is very much dependent on whether the zone of occupation is in the supply zone or not. Since the index also is not dependent on the source, it is not a good air quality indicator. However, if the short circuiting takes place outside the zone of occupation, the index is proportional to the steady state ventilation effectiveness. Further, as a time constant indicator, it tells how "fast" the zone can be emptied of contaminants if the generation stops, compared to complete mixing.

The areas under the curves are a measure of the exposures. These can be used for assigning ventilation effectiveness, as shown by Sandberg. If the room is completely mixed at time zero, the ratio between the initial concentration and the area under the concentration curve measured at an arbitrary point is a local air exchange rate because, inverted, it gives the local age of the air at that point. Comparing this local air exchange rate with the nominal, it gives all information about the dilution process at this specific location. The local transient ventilation effectiveness is for this reason calculated as $\varepsilon^0 = n^0/n$. However, this index is not a complete air quality indicator either, because it does not depend on the contaminant source. The three efficiencies ε_{st}, ε_{tr}, and ε^0 together describe the ventilation system in a room very well.

Figures 3 and 4 show calculations of expected effectiveness as a function of the mixing parameter, for 4 different schemes of ventilation. The effectiveness decreases sharply when the mixing parameter falls below 4 for the ceiling type of short circuiting schemes. The most interesting feature to see is the very good performance of a vertical-up displacement type of system. The curve denoted ϕ = 1 indicates promising results. In practice this can be achieved if we are able to match the contaminant sources and ventilation system in a way that a plug flow type of contamination flow is obtained from zone 2 to zone 1.

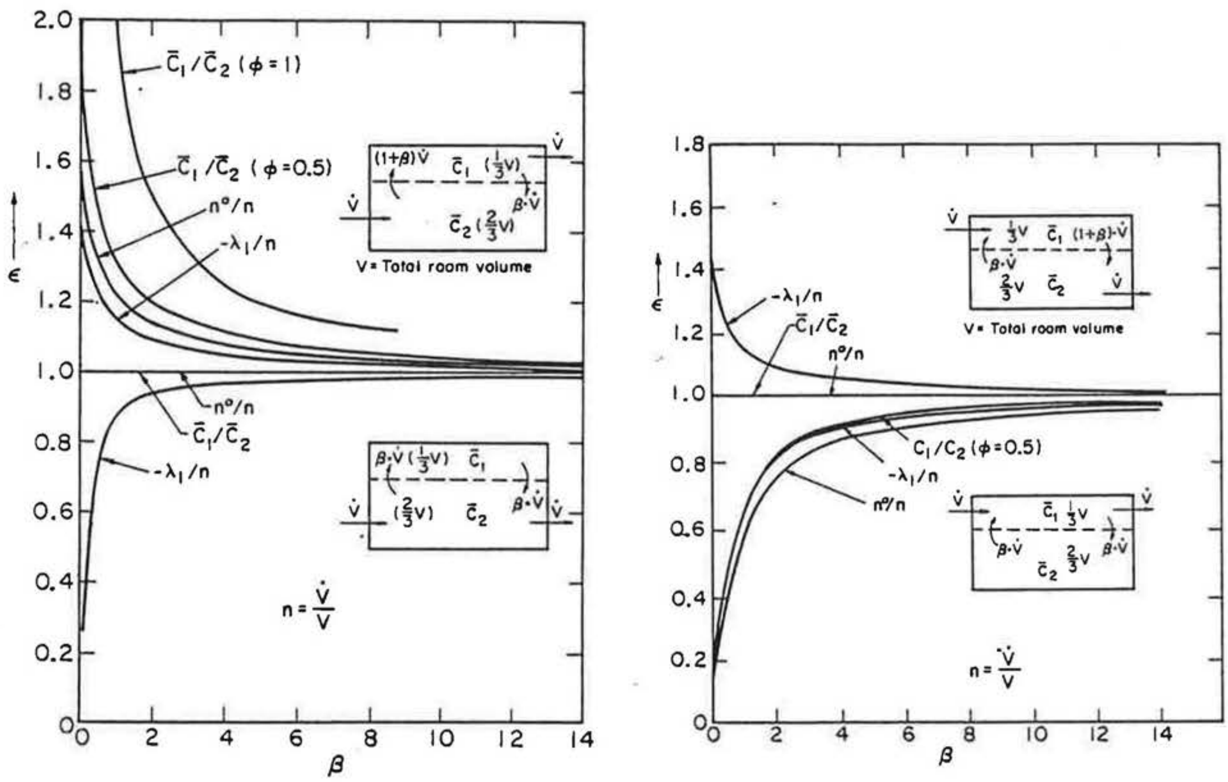


Fig. 3 and 4. Calculated ventilation effectiveness, using a two-zone mixing model. Short-circuiting + displacement vertical down.

4. ASPECTS OF MEASURING VENTILATION EFFECTIVENESS

Steady state ventilation effectiveness can be measured by measuring the concentrations of the actual contaminants at significant locations in the zone of occupation and in different exhaust ducts, together with measuring the different air flow rates. If the infiltration is not a significant part of the total ventilation, the complete mixed concentrations are determined by calculating an air-flow-weighted average concentration for the exhaust air flows.

The different sources can be simulated by using an appropriate tracer gas which disperses similarly to the actual contaminants.

An appropriate procedure for measuring transient effectiveness is to use a tracer gas. Local and overall effectiveness are determined from the same tests, by starting the dilution from a well mixed room. The nominal air exchange rate and the total air flow rate are determined from a well mixed decay.

5. DISPLACEMENT VENTILATION VERTICAL-UP FOR SMALL OFFICE ROOMS

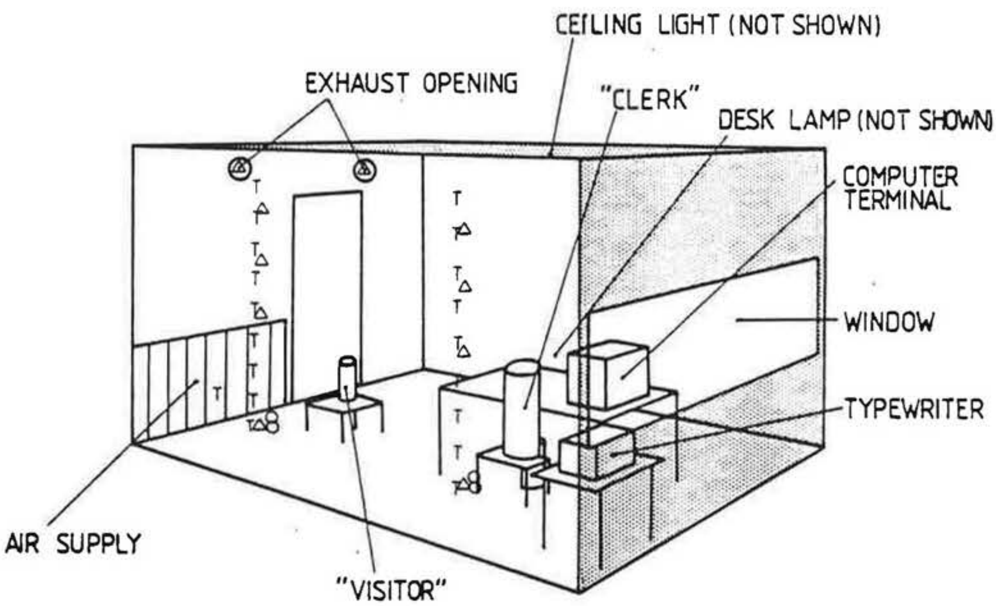


Fig. 5. Test room. Some of the measuring points:

o - air velocity

Δ - concentration

T - temperature

Tests have been carried out in a room of approximately 45 m^3 with a floor area of 16 m^2, Figure 5. Air was supplied through an adjustable opening in the short wall opposite the window wall. The opening consisted of a filter cloth. The height of the supply opening was varied, as well as the supply air flow rate and the temperature. The windows were simulated by water cooled or heated panels. Persons in the room were simulated by heated cylinders having a surface area and heat output equivalent to a sedentary average adult person. The contaminants from the persons were simulated bu using N_2O as a tracer gas. Data were collected and processed by a microprocessor-based, datalogging system. For summer conditions, two levels of heat load were used (test 7 to 17 in App. I):

Low heat load	High heat load	
Ceiling light 240 W	Ceiling light 240 W	1 desk lamp 75 W
Window ca. 120 W	Window 120 W	1 computer terminal 135 W
1 person 85 W	2 persons	1 typewriter 40 W

The test-room has some heat loss/accumulating ability, the real heat load with regard to the cooling ability of the air must be calculated from the temperature difference in/out of the room.

When the steady-state effectiveness was investigated, the tracer gas, N_2O, was supplied in the "persons". The concentration of the tracer gas and the temperature were measured several places in the room by means of a measuring column.

6. THE VENTILATION EFFECTIVENESS DEPENDS ON AIR FLOW AND HEAT LOAD

In a room with low velocity air supply placed near the floor, the air is transported from the floor to the ceiling by convective jets. A simplified illustration is shown in fig. 6. From this we can see that two circulation zones are established in the room, one lower, clean zone and an upper, polluted zone.

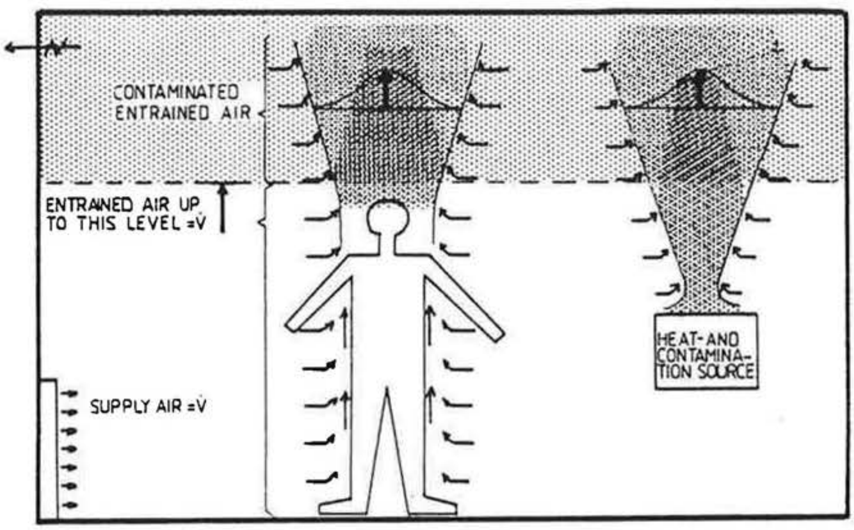


Figure 6. Simplified illustration of the air flow pattern in a ventilated room.

The supply air has some undertemperature related to the room air. Due to the undertemperature, the supply air disperses into the lower part of the room.

The convective jets decide the size of the zones. The air flow in the convective jets increases with the height, air is entrained along the jet. When the entrained air flow is equal to the supply air flow to the lower zone, the convective stream has to entrain air from the upper, contaminated part of the room. If much air is supplied to the lower zone, we will have a large lower zone.

The contaminants are usually supplied to the room from one or more of the heating sources in the room. In that way, the contaminants are transported to the upper part of the room and mixed with the air in the zone.

In reality, there will also be a transport of air from the upper to the lower zone, due to mixing between the zones ($\beta>0$). This means that we will also have some transport of contaminants to the lower zone. The recirculating air flow causes the lower zone to become larger than the supply air flow indicates.

In reality, we will not have a two zone model but a model with several stratas where the concentration is increasing with the height. In App. I, concentration profiles are shown as a function of the height above the floor.

6.1 The steady state effectiveness increases with increasing air flow rate.

From the results presented in fig. 7, one can see that the steady-state ventilation effectiveness depends on both supplied air flow and the heat load.

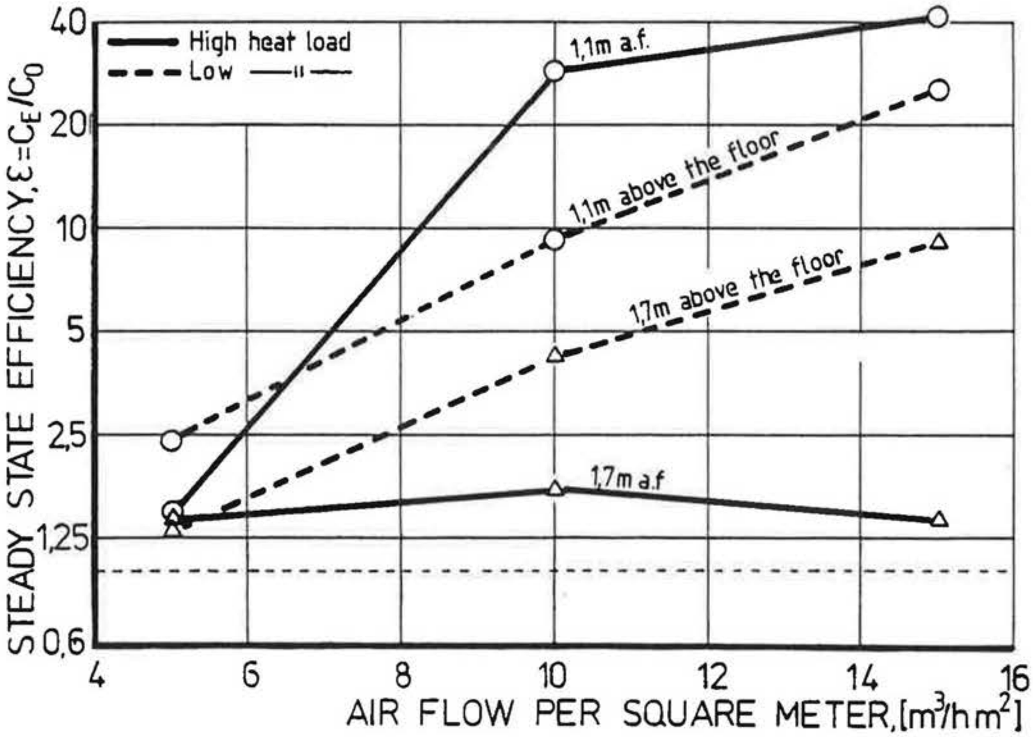


Fig. 7. Steady-state ventilation effectiveness vs air flow and heat load. The measurements are made near the writing desk, 1,1 m and 1,7 m above the floor.

From the measurement at level 1.1 one can see that the effectiveness rapidly increases with the air flow. At low air flows, the measuring point is partly situated in the upper contaminated zone. Increasing air flow caused the strata to rise and the measuring point gradually to be situated in clean air. It also seems to be a tendency for the high heat load to cause a clearer stratification.

For the measuring point 1,7 m above the floor and low heat load, the effectiveness is increasing with increasing air flow. This is due to the strata rising at increasing air flow. There is not enough supply air to raise the clean zone up to 1,7 m when the heat load is high.

6.2 The transient effectiveness indicates a two zone model

From the measurements shown in fig. 8, it is obvious that the transient effectiveness is always greater than 1. The best efficiencies were gained with the high heat load. This is owing to the fact that the supply and the exhaust openings are situated in different zones and that the stratification becomes clearer when the heat load and the temperature differences increase. At the highest air flow rate the effectiveness is decreasing. There are at least two possible reasons for this: the momentum flux of the supply air causes some mixing between the zones (larger β) or the volume of the upper zone is reduced. This will theoretically lead to a lower transient effectiveness.

There seems to be a coherence between the conclusions one can draw from the two ways of considering the effectiveness.

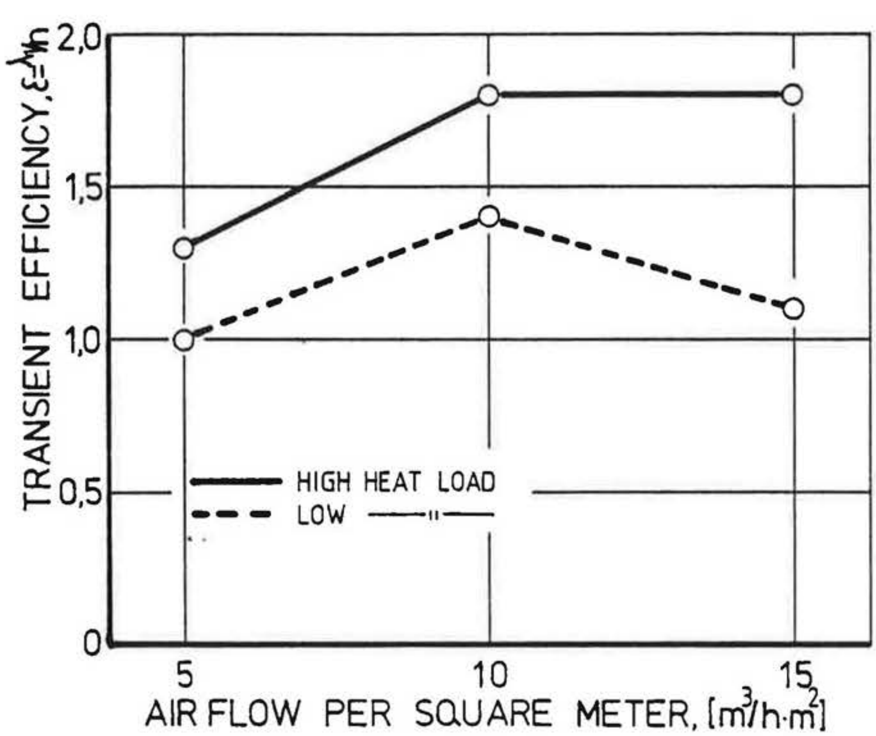


Fig. 8. The transient effectiveness versus heat load and supply air.

6.3 The influence from solar radiation on the ventilation effectiveness

In fig. 4.1. results are shown from an experiment with direct solar radiation at the floor level. The solar radiation was simulated by coating a part of the floor with heating film. We can see that the steady state effectiveness is reduced.

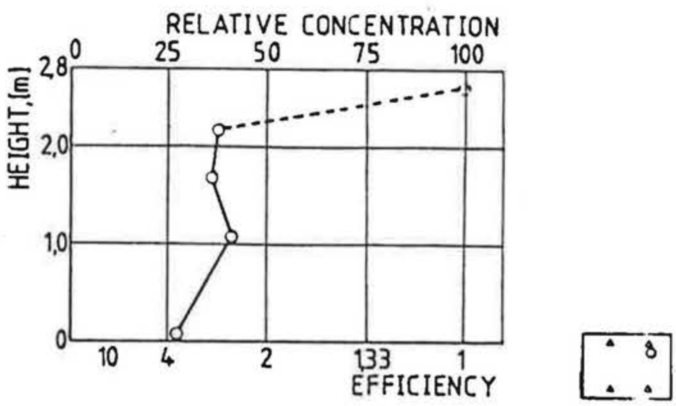


Fig. 9. Concentration plot from a test with direct solar radiation on the floor, air flow rate 10 m^2/hm^2, see also test 2 in App. I.

6.4 The influence from a cold window on the effectiveness

Experiments show that a cold window in the room with a convective heater below it causes some mixing in the room, see fig.10.

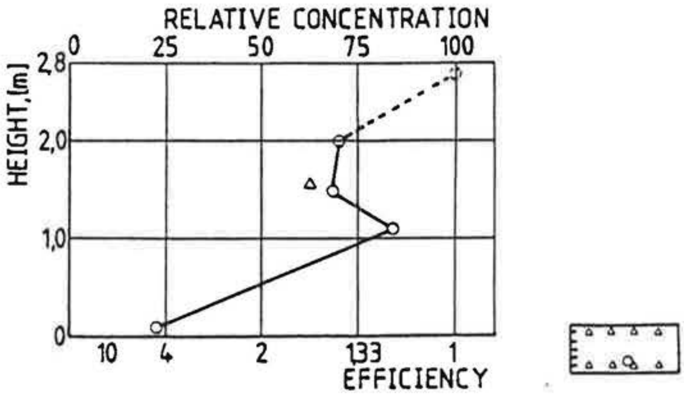


Fig.10. Concentration plot from a test with a cold window (11 ^{0}C) in the room. Heating panels below the window (600W). This test is from an earlier investigation, the room differs some form that shown in fig. 5. The air flow rate is 160 m^3/h. The internal loads are relatively small.

When the convective heater is not needed to cover the heat loss, the cold convective stream into the lower zone may cause the strata to rise to a higher level which might lead to an increased steady state effectiveness in the level 1,7 m above the floor, see fig.11.

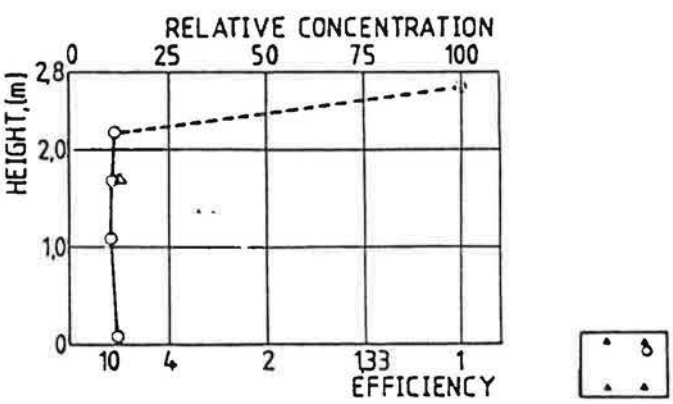


Fig.11. Concentration plot from a test with a window with a temperature of 8,5^{0}C where heat losses are covered by the internal losads, see also test 3 in App. I.

6.5 Temperature effectiveness

In the same way as we defined a steady-state ventilation effectiveness we can define a temperature effectiveness

$$\varepsilon_t = 1 + \frac{t_{cm} - t_o}{t_o - t_s} \qquad (4)$$

This effectiveness describes the ability of the system to remove surplus heat from the zone of occupation.

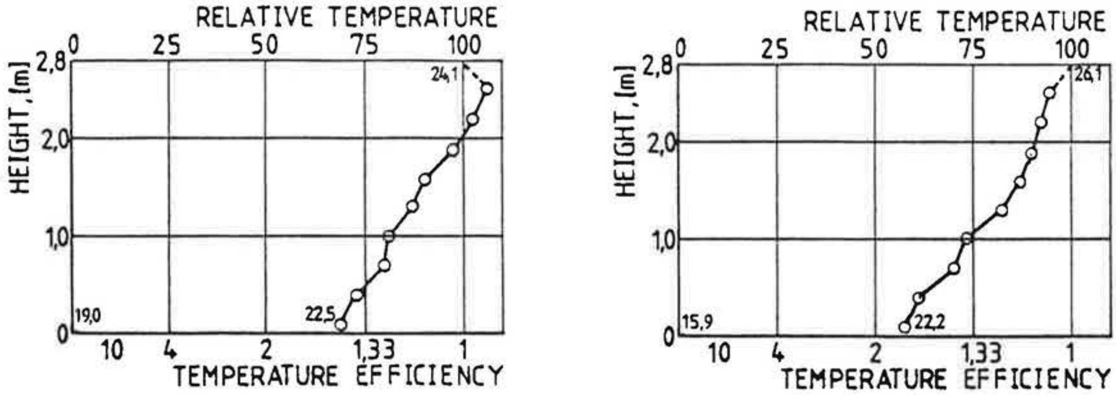


Fig. 12. Typical temperature plot adjacent to the desk, low heat load to the left, high heat load to the right. Air flow rate 160 m^3/h.

In contrast with the concentration profiles, the temperature is evenly increasing with the height, see fig. 12 and App. I. The cause of this is that the heat sources and the contaminant sources are not identical.

Only a few of the heat sources supply tracer gas to the room. The heat sources are also creating convective streams with different temperature levels. A plume with low temperature does not have the ability to entrain the warmest zones just beneath the ceiling. The contaminants are associated with the warmest convection streams.

In figure 13 the temperature effectiveness vs. heat load and air flow is shown. The effectiveness is always larger than unity but the best effectiveness is achieved just above the floor. The effectiveness is increasing with the air flow rate and the heat load.

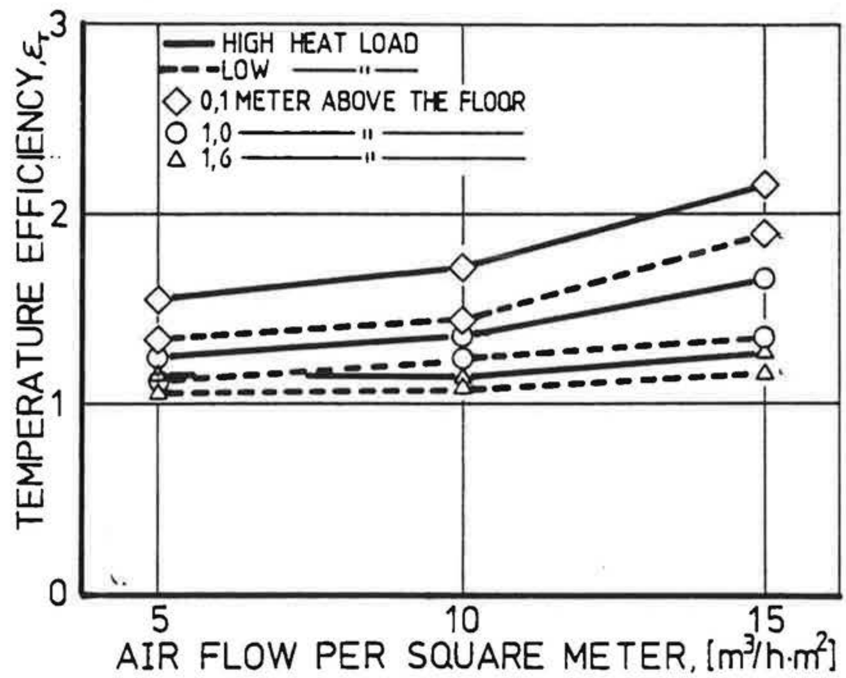


Fig. 13. Temperature effectiveness versus air flow rate and height above the floor.

6.6 Influence from real persons on the ventilation effectiveness

How will the system behave when a real person is introduced to the room? To answer this question one test with a real person in the room was carried out. A comparison of an investigation with a person in the room and another without a person in the room is shown in fig. 14.

From the figure one can see that the profiles and the effectiveness are somewhat influenced, but not so much that the mixing is essential. During the test, which lasted for half an hour, the test person opened the door a couple of times and walked around in the room.

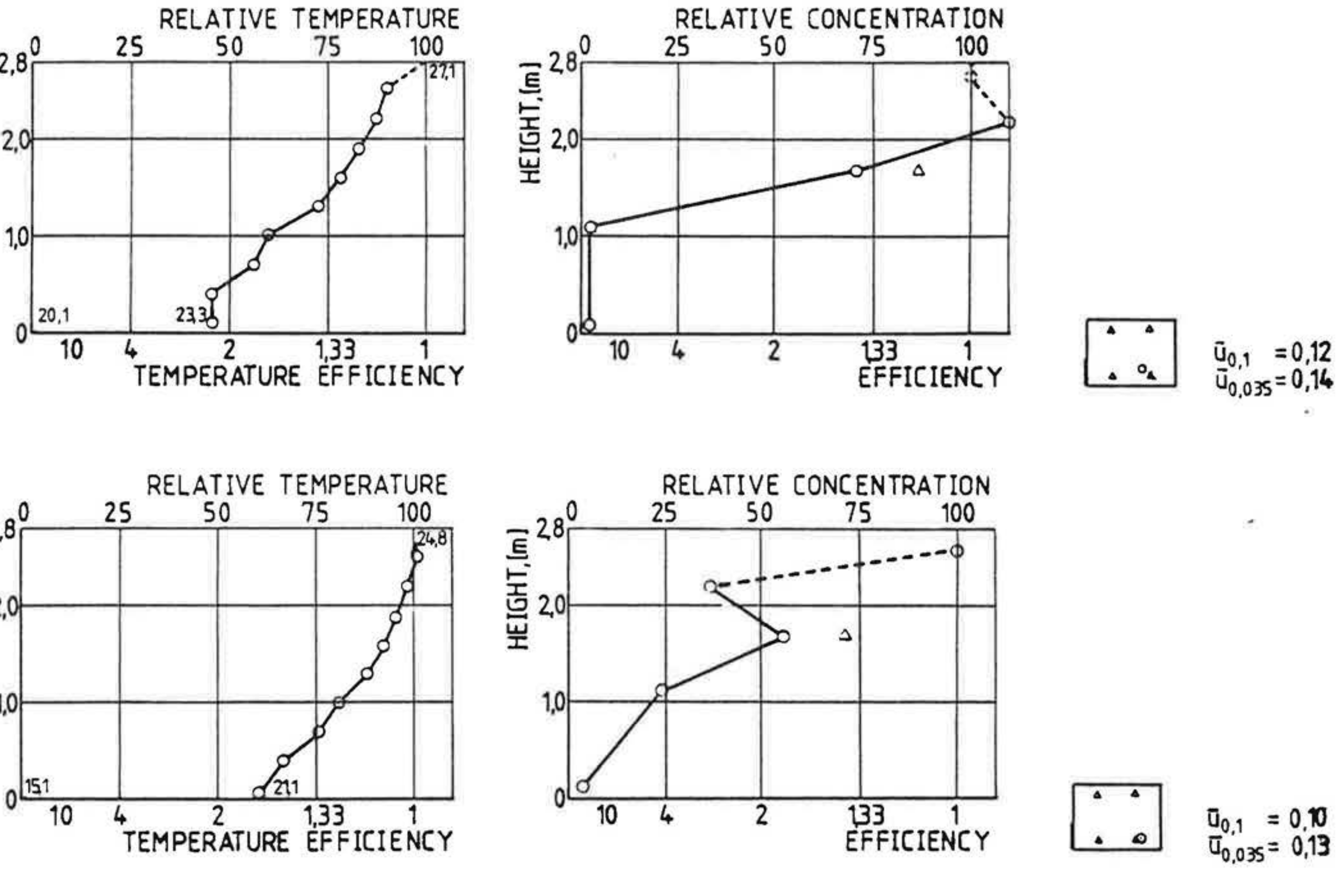


Fig. 14. Above temperature and concentration plots with a simulated person, below a test with a real person in the room.

6.7 Comparison with high wall air inlet

In fig. 15, a test with high wall air inlets is shown. From this we can see that the steady state effectiveness, at level 1.7 m, hardly becomes larger than one. When hot air is supplied, the steady state effectiveness decreases to 0,5 when a system with high wall exhaust opening is used. This can be explained by the short-circuit model: When hot air is supplied, a two zone model is established and the supply and the exhaust are situated in the same zone.

With the exhaust opening situated near the floor, there will not be a short circuiting system. When the overtemperature is very high, the steady state effectiveness at level 1,7 m increases when the exhaust opening is situated near the floor. The explanation is that the stratification sinks down to a lower level so that the measuring point becomes situated in the upper zone. The polluted plume does not have enough overtemperature to entrain the upper zone. Therefore, the upper zone will be "clean" and the lower zone polluted.

Compared with the low velocity system, we can see that high wall air inlets give poor results. However, one can always avoid short circuiting by proper selection of the location of the exhaust (in the other zone).

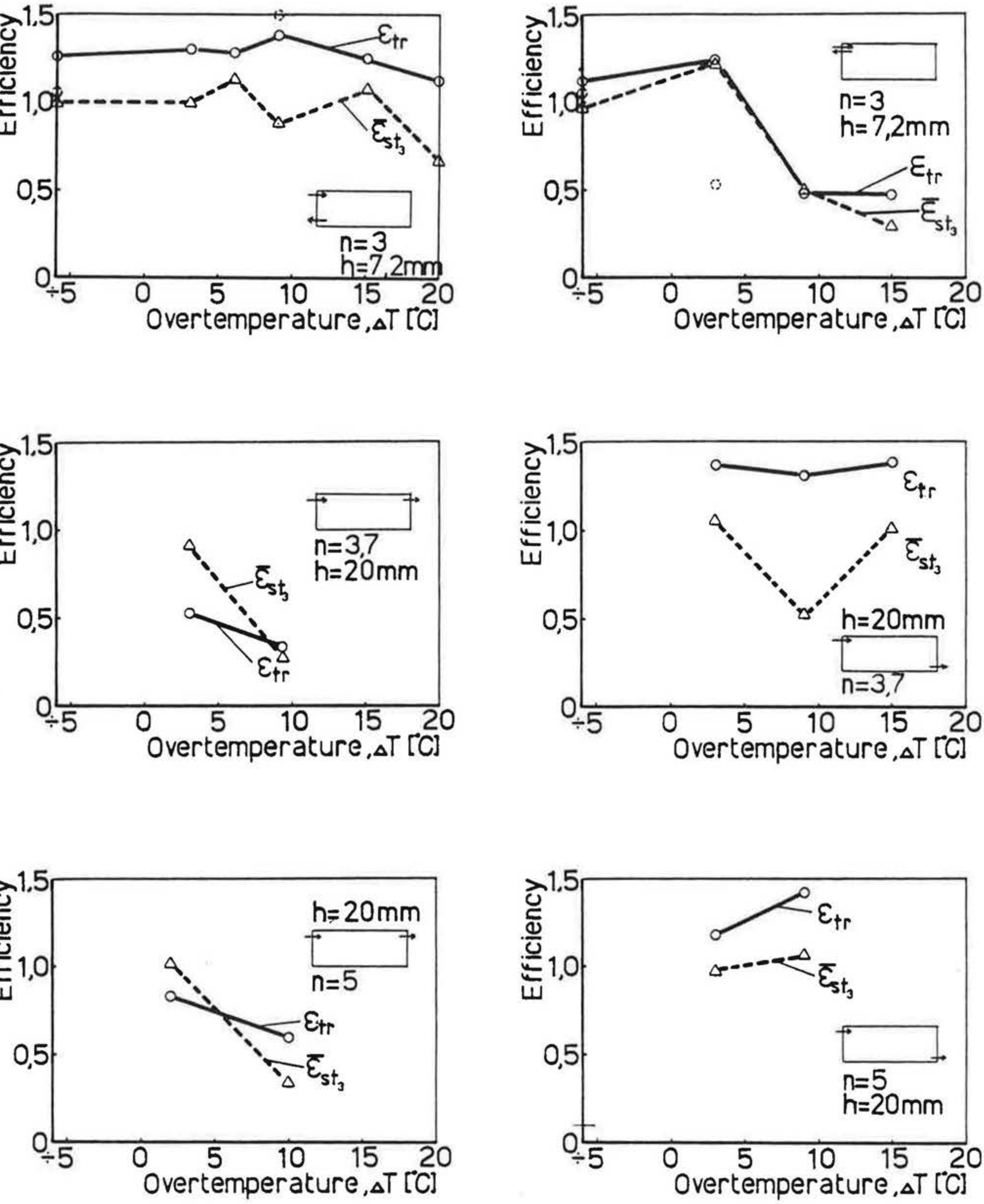


Fig. 15. Transient, ε_{tr}, and steady state, ε_{st}, effectiveness versus air flow rate (air exchange rate n), overtemperature of the supply air and the situation of the exhaust opening. Air supply through a slot adjacent to the ceiling, with a length of 3,45 m. Slot height h.

6.8 Comparison with theory

If the measured effectivenesses are compared with fig. 3 and 4,it is obvious that the expectations of the displacement principle are fulfilled. However, one should realize that the air circulation patterns are not so simple as the two-zone model. In reality, there will be several strata where temperature and concentration of tracer gas are increasing with the height. As explained earlier, the temperature and concentration effectiveness are not similar.

7. AIR VELOCITY AND TEMPERATURE IN THE TEST ROOM

Using displacement ventilation vertical up, it is reasonable to assume the highest air velocities just above the floor near the air supply. During our experiments we found the highest velocities in a distance of 0,4 - 0,8 m from the supply, 0,03 - 0,04 m above the floor. Velocities and temperatures were, among other measurements, recorded at a distance of 0,6 m from the opening 0,035 and 0,1 m above the floor. In fig. 14 the velocity ratio u/u0, (measured velocity/supply air velocity), is fitted against the Archimedean number of the supply air.

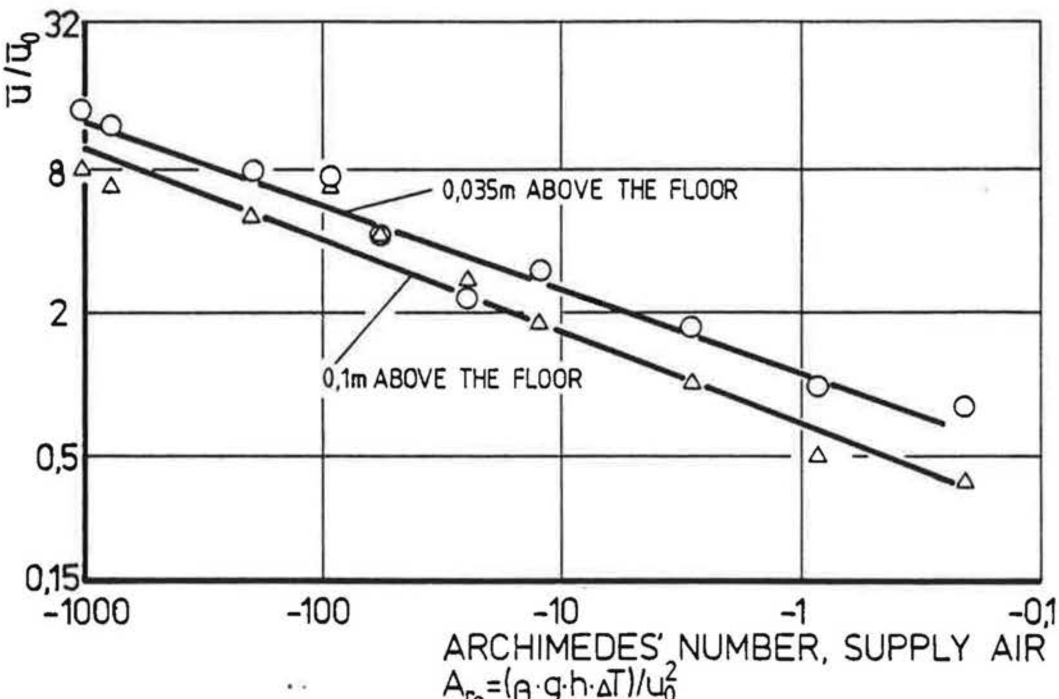


Fig. 16. Air velocity 0,1 and 0,035 m above the floor, 0,6 m from from the opening vs. supply air Archimedean number and supply air velocity. The width of the opening was 1,55 m. T = temperature difference in/out of the room.

The height, h, of the opening was varied from 0,045 m to 0,7 m. ΔT was varied from 3 to 14 K depending on the air flow and the heat load. The air velocity varied from 0,019 to 0,3 m/s depending on the air flow and the height. The fitted curves follow the expression:

For the level 0,035 m

$$\frac{U}{U_0} = 1,11.\ Ar_0^{0,367} \qquad (5)$$

For the level 0,1 m above the floor

$$\frac{U}{U_0} = 0,70.Ar_0^{0,39} \qquad (6)$$

The air velocity 0,6 m from the air supply opening as a function of the surplus heat removed from the room is determined from:

$$U = 0,019.\left(\frac{h}{\dot{V}_0}\right)^{0,07} . Q^{0,36} \qquad (7)$$

From the level 0,1 m above the floor

$$U = 0,014.\left(\frac{h}{\dot{V}_0}\right)^{0,16} . Q^{0,36} \qquad (8)$$

Several expressions might be developed from these formulaes.

In fig.15, the temperature smoothing-out vs height of the supply opening and the supply air velocity is shown.

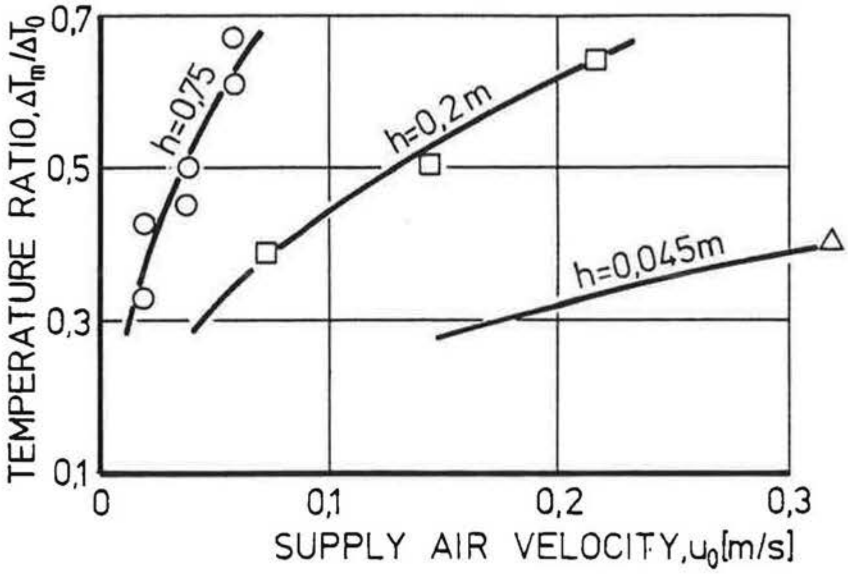


Fig. 17. Temperature ratio against supply air velocity and height of supply opening. ΔTm- temperature diff. between the exhaust air opening and a point in the supply air "jet" approx. 0,05 m above the floor.

The best empirical model fitted for the recorded values 0,6 m from the opening is

$$\frac{\Delta T_m}{\Delta T} = 2,7 . U_0^{0,47} . h^{0,44} \qquad (9)$$

The confidence intervals for the constants are quite wide, due to some correlation between the independant variables, which means that the model should not be used for extrapolation or other rooms.

In (9) U_0 could be substituted with $\dot{V}/h.b$ where b=1,55 m. If we do so we can see that the temperature ratio is nearly independent of the height h.

ΔTm is the temperature difference between the exhaust air opening and a point in the supply air stream approximately 0,05 m above the floor at a distance of 0,6 m from the opening. Assuming a straight temperature profile from the floor to the outlet, the temperature gradient in the zone of occupation could now be calculated.

8. DESIGN OF AIR SUPPLY OPENINGS

There are several ways of designing the supply opening. Filter clothing, perforated panels, porous materials, etc. on could be used. Filter clothing could be characterized as a porous material or as a perforated panel with a high rate of holes.

Some investigations have been carried out of perforated panels and filter clothings as supply opening, see App. VI. It seemed to be a tendency that the filter clothing resulted in lower air velocities than perforated panels. In the project reported here, filter clothing was used. A perforated panel with a high rate of holes would probably have done just as well. When using perforated panels, air is entrained into the small jets if the perforated areas are not too large.

This causes larger air flows and air velocities, but the temperature gradients are smoothed out faster.

9. COMFORT

The following quotation is from Fanger (1982) :

"The PMV and PPD indices express warm and cool discomfort for the body as a whole. But thermal dissatisfaction may also be caused by an unwanted heating of one particular part of the body (local discomfort). This can be caused by a too high vertical air temperature difference between head and ankles, by a too warm or cool floor, by a too high air velocity (draught), or by a too high radiant temperature asymmetry. Limits for these factors are listed for light, mainly sedentary activity in Sections A.1.1 and A.1.2. If these limits are met, less than 5% of the occupants are predicted to feel uncomfortable due to local heating or cooling of the body caused by each of the above mentioned factors. Note that the percentages may not be additive.

The experimental data base concerning local discomfort is less complete than for the PMV and PPD indices. Sufficient information is thus not available to establish local comfort limits for higher activities than sedentary. But, in general, man seems to be less sensitive at higher activities.

If the environmental conditions are inside the comfort limits recommended in this Appendix, more than 80% of the occupants are estimated to find the thermal conditions acceptable.

A.1.1 Light, mainly sedentary activity during winter conditions heating period)

- The operative temperature shall be between 20 and 24 ^{0}C.

- The vertical air temperature difference between 1.1 m and 0,1 m above floor (head and anke level) shall be less than 3^{0}C.

- The surface temperature of the floor shall normally be between 19 and 26 ^{0}C, but floor systems may be designed for 29 ^{0}C.

- The mean air velocity shall be less than 0.15 m/s.

- Radiant temperature asymmetry from windows or other cold vertical surfaces:

  The radiant temperature asymmetry shall be less than 10 ^{0}C (in relation to a small vertical plane 0.6 m above the floor).

- Radiant temperature assymetry from a warm (heated) ceiling:

  The radiant temperature asymmetry shall be less than 5 ^{0}C (in relation to a small horizontal plane 0.6 m above the floor).

A.1.2 <u>Light, mainly sedentary activity during summer conditions (cooling period).</u>

- The operative temperature shall be between 23 and 26 ^{0}C.

- The vertical air temperature difference between 1.1 m and 0.1 m above floor (head and ankle level) shall be less than 3 ^{0}C.

- The mean air velocity shall be less than 0.25 m/s."

Local discomfort has been closer investigated by Claus I.K. Pedersen (with expert advice by Fanger). Naturally, he has found the sensitivity to be highest where the skin is bare. The most sensitive point is the neck. He has also investigated local cooling of the ankles; the sensitivity to draught at the ankles is much less than at the neck.

In fig. 18, eq 5 and eq 9 are connected to Pedersen's equations. In the hatched area, more than 12,5% of the persons would feel discomfort. The air velocity frequency is supposed to be 0,2 Hz, and the ratio between the highest velocity and the mean air velocity was two.

The conclusion is that the quantity of surplus heat removed from the zone of occupation is limited by the thermal comfort.

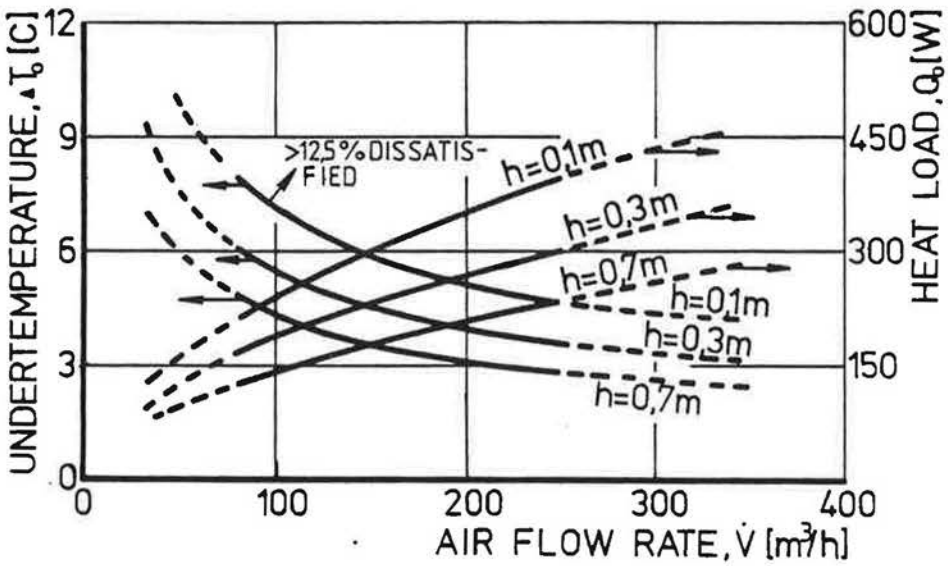


Fig. 18. Maximum heat load and undertemperature allowing 12,5% dissatisfied persons versus air flow rate and height of supply opening. The undertemperature is the temperature differense between a point 1,1 m above the floor and the supply air. The heat load is related to the same points.

10. SOME PRACTICAL CONSEQUENSES

In the previous part it has been shown that air supply direct to the zone of occupation and exhaust in the upper zone gives lower concentration of contaminants in the occupied space and higher exhaust air temperature. Higher exhaust air temperature means that more energy is removed from the room.

During the parts of the year with excess heat it is an advantage that the exhaust air is as warm as possible compared to the temperature in the occupied parts of the room. In the parts of the year with heating demand the opposite would be an advantage.

10.1 Removal of surplus heat

The gain in the parts of the year with excess heat in the room, the experimental system compared to complete mixing, is due to the temperature effectiveness. The air could be supplied with higher temperature, or if care is taken to avoid draught, the air flow rate could be reduced compared to complete mixing. To take advantage of free cooling no external recirculation of air should be used during this season.

Internal recirculation of air in the zone of occupation could increase the ability of removing surplus heat. Recirculation in this zone will smooth out the temperature difference between the floor and the head level of the occupants. This means that the air could be supplied with lower temperature or that the air flow rate could be decreased. Preliminary experiments indicates promising results. To keep a high effectiveness it is important that the recirculation between the upper poluted zone and the occupied space is kept at a low level.

10.2 Improved ventilation effectiveness means that fresh air supply can be reduced or increased recirculation of air should be used:

From the experimental results it could be computed an example for winter conditions:
Given a room with an air flow of 10 m^3/hm^2. The ventilation scheme is

complete mixing, and it is used 50% external recirculation. This could be compared to a system like that used in the experiments. We want to keep the same air quality as for complete mixing. Due to better effectiveness of the experimental system (fig.7) the:

- external recirculation could be increased to 64%

  or
- external recirculation could be dropped and the fresh air supply could be cept at 5 m^3/hm^2

Energy calculations of the system with 64% external recirculation shows that the energy consumtion is reduced with 10 - 20% compared to the complete mixing system in spite of the increased temperature of the exhaust air.

For the calculation the computer program "Royal DEBAC" was used for a one day calculation. Royal DEBAC is a energy demand calculation program adapted to the two zone model.

APPENDIX I

RESULTS: TEMPERATURES, CONCENTRATIONS, MEAN AIR VELOCITIES

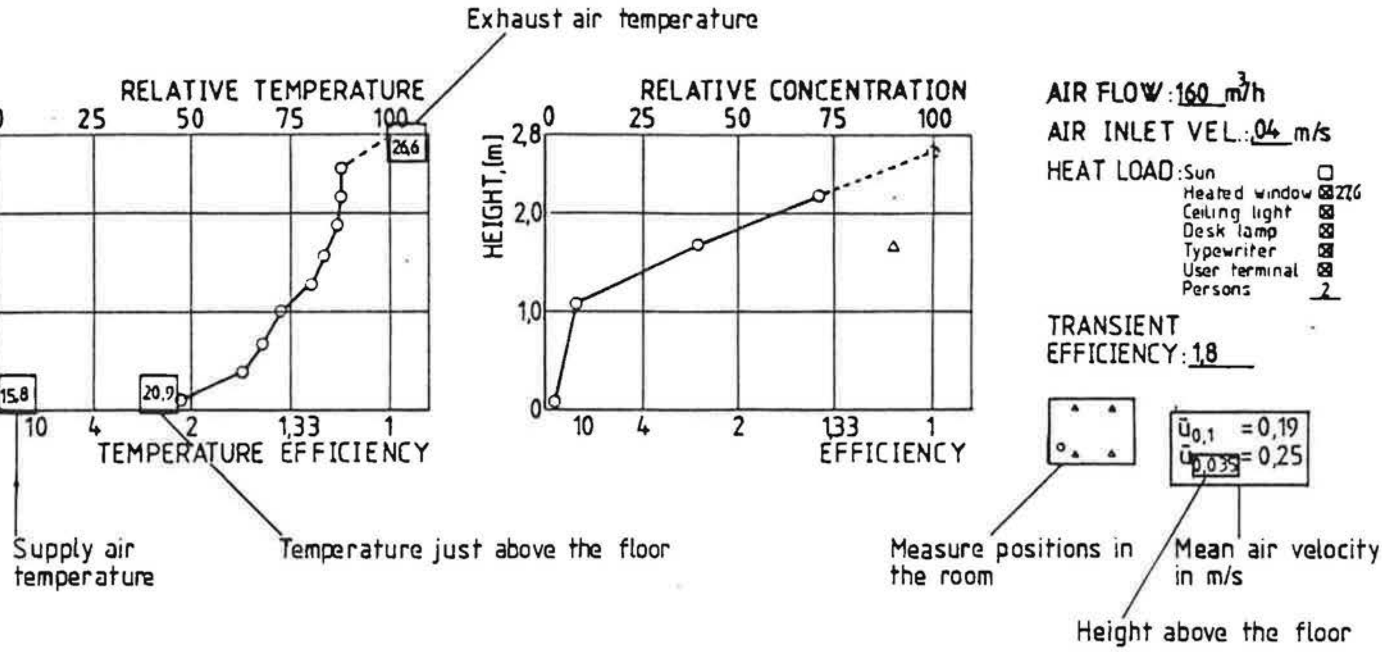


Fig. I.1. Explanation of what the numbers shown on the figures in App. I mean.

In the following pages, results are reviewed. The test conditions are shown on each figure.

Test 1 to 4 are reference tests and preliminary tests which were done to compare the test room to earlier test rooms (Skåret, Mathisen 1983). In these tests there were measured in 4 points in the room. In test 5 and 6 it was used perforated sheets instead of filter clothing. In test 7 to 17 the room was furnitured as a clerks room, see fig. II.2.

TESTS

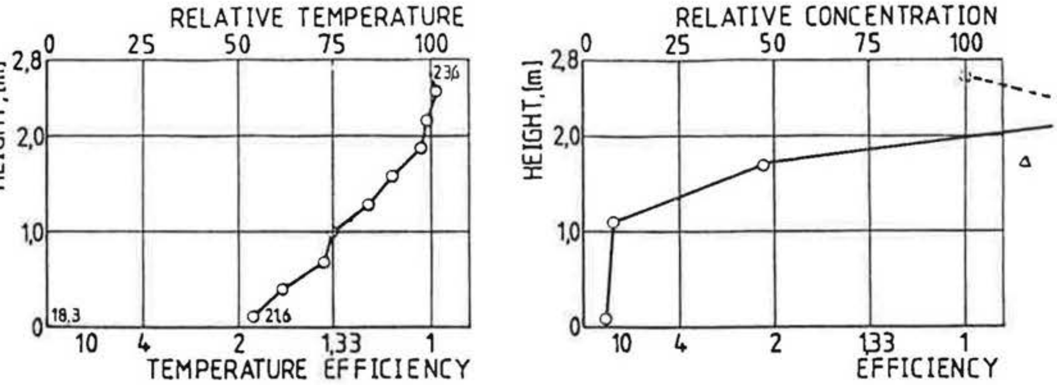
Test no.	Supply opening	Remarks
1	Filter clothing 1,55x0,3m^2	Reference test, summer without direct solar radiation, without furniture
2	Filter clothing 1,55x0,3m^2	Reference test, direct solar radiaton at the floor (830 W), Without furniture
3	Filter clothing 1,55x0,3m^2	Reference test, winter conditions, without furniture
4	Filter clothing 1,55x0,3m^2	Reference test, as no. 1, but with low air supply temperature, without furniture
5	Perforated sheet, 10% degree of perforation, two areas (0,7x0,7m^2) with 54 holes in each direction	Summer conditions without direct solar radiation, without furniture
6	As no. 5,but with one perforated area	
7	Filter clothing 1,55x0,75 m^2	With furniture in the test room
8	As no. 7	
9	As no. 7	
10	As no. 7	
11	As no. 7	
12	As no. 7	
13	Filter clothing 1,55x0,75 m^2	With a real person in the test room, with furniture
14	Filter clothing 1,55x0,2 m^2	
15	As no. 14	
16	As no. 14	
17	Filter clothing 1,55x0,045m^2	

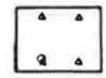
AIR FLOW: 160 m³/h

AIR INLET VEL.: 0,1 m/s

HEAT LOAD: Sun ☐
Heated window ☒ 2°C
Ceiling light ☒
Desk lamp ☐
Typewriter ☐
User terminal ☐
Persons 2

TRANSIENT EFFICIENCY: 1,9





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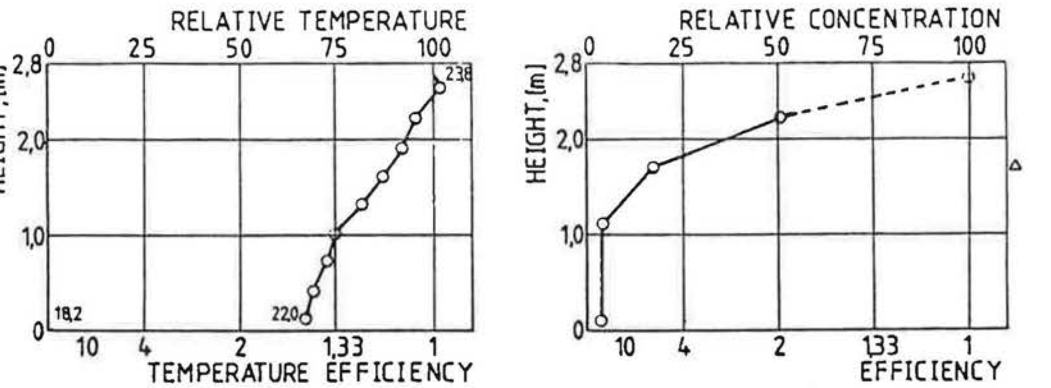
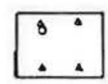
$\bar{u}_{0,1} = 0{,}14$





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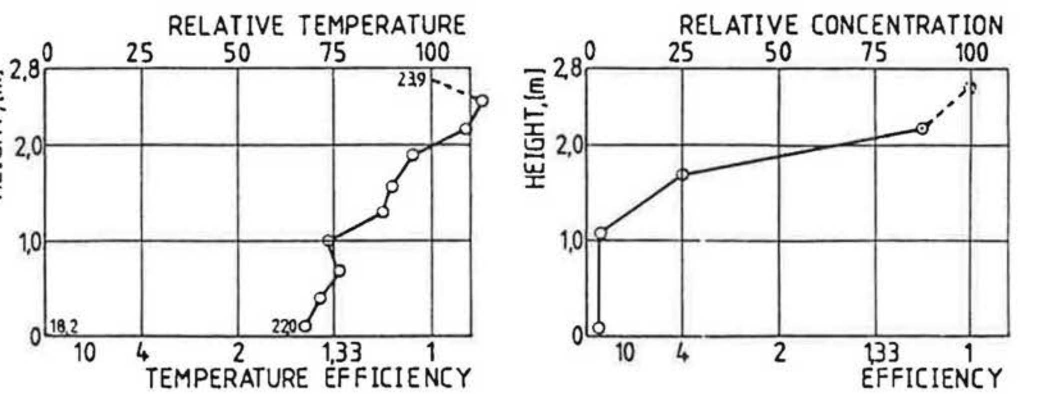
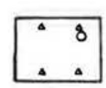
$\bar{u}_{0,1} = 0{,}08$





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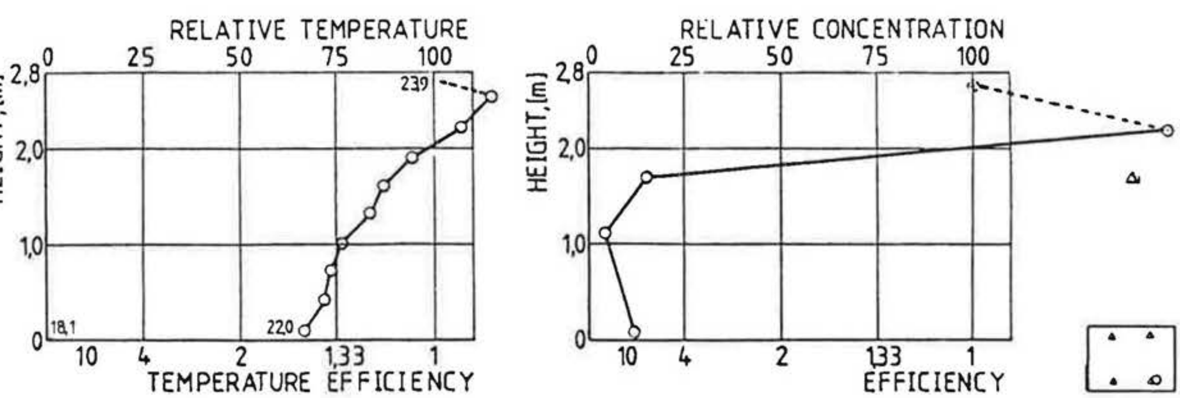
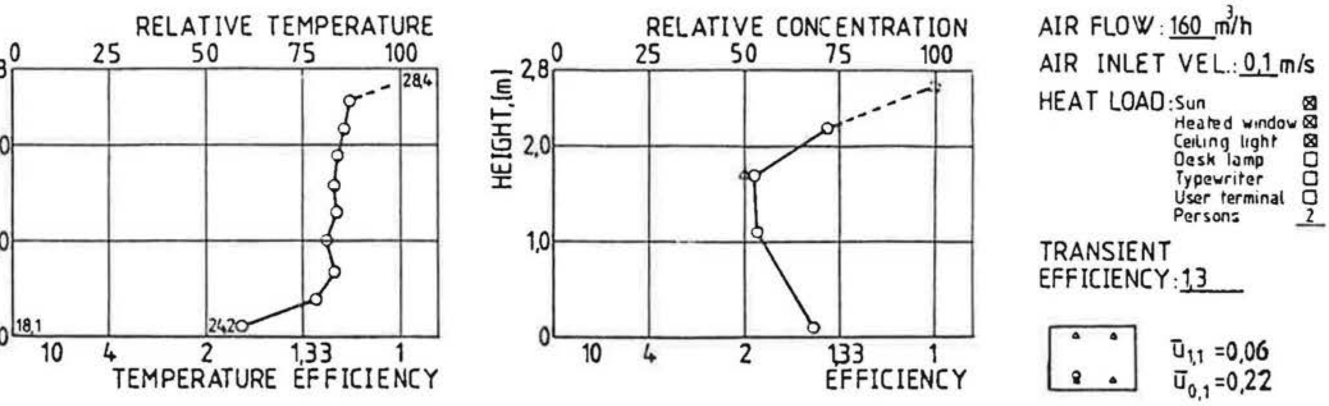
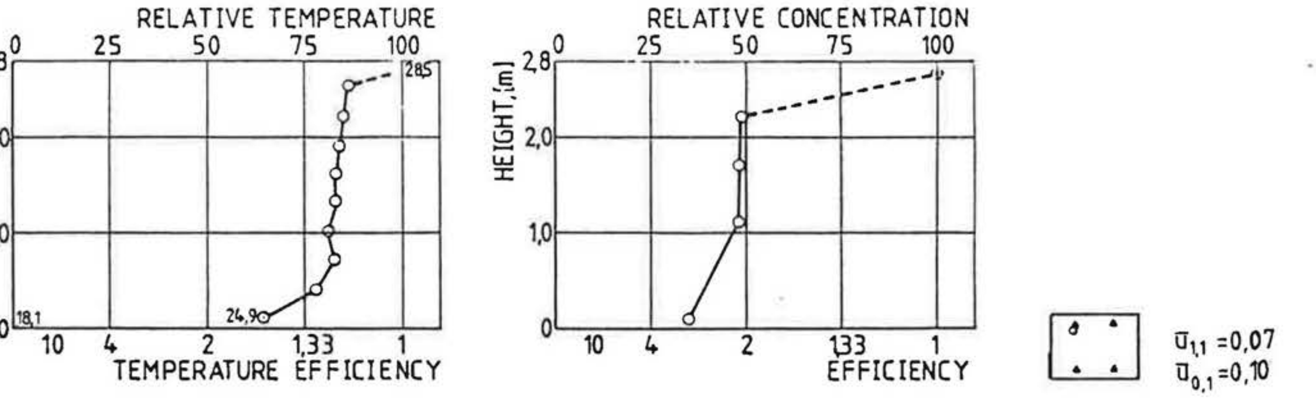
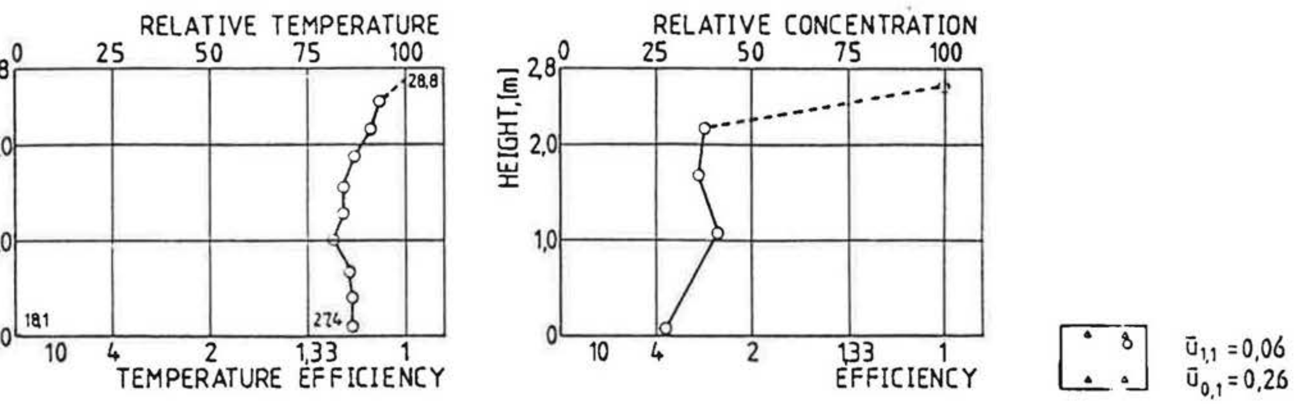


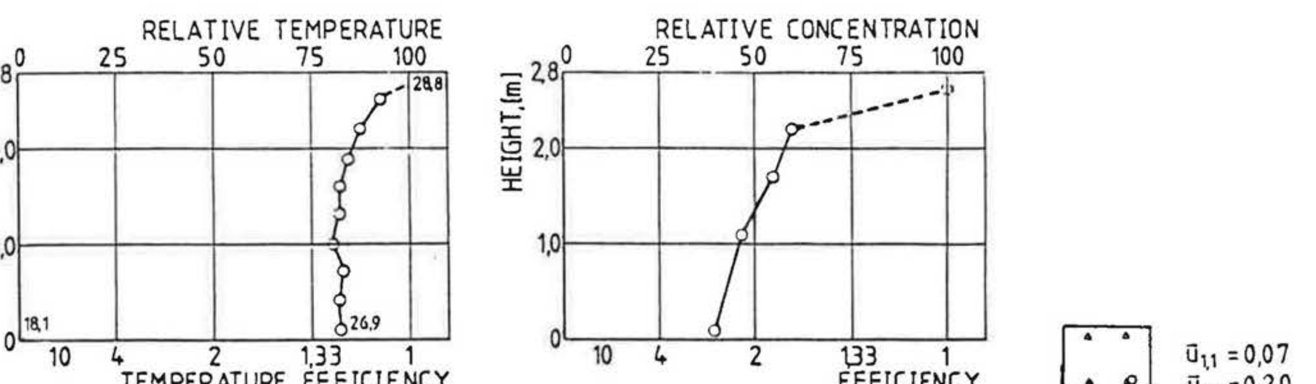
$\bar{u}_{1,1} = 0{,}06$
$\bar{u}_{0,1} = 0{,}12$

1	Filter clothing 1,55x0,3m²	Reference test, summer without direct solar radiation, without furniture

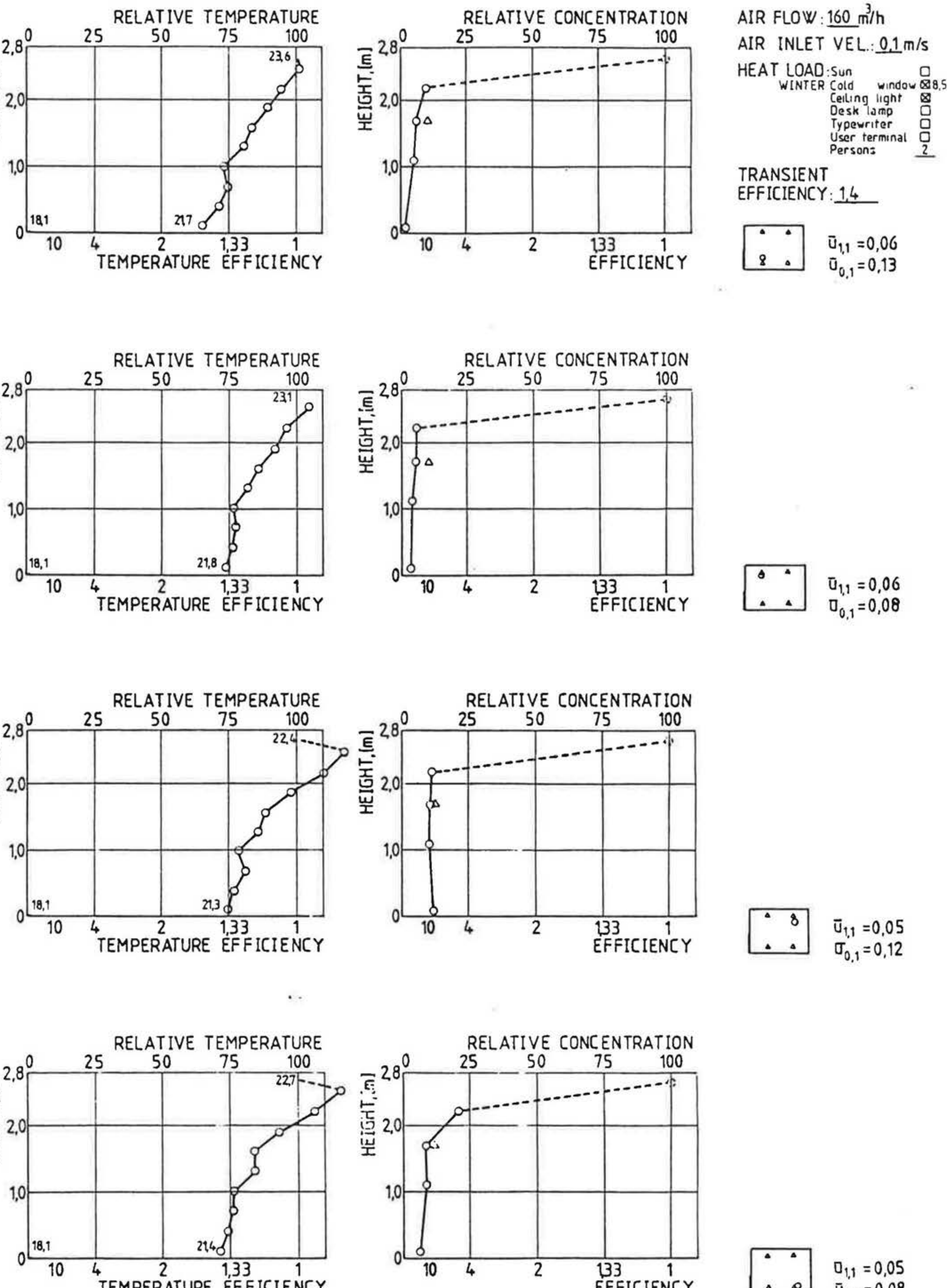








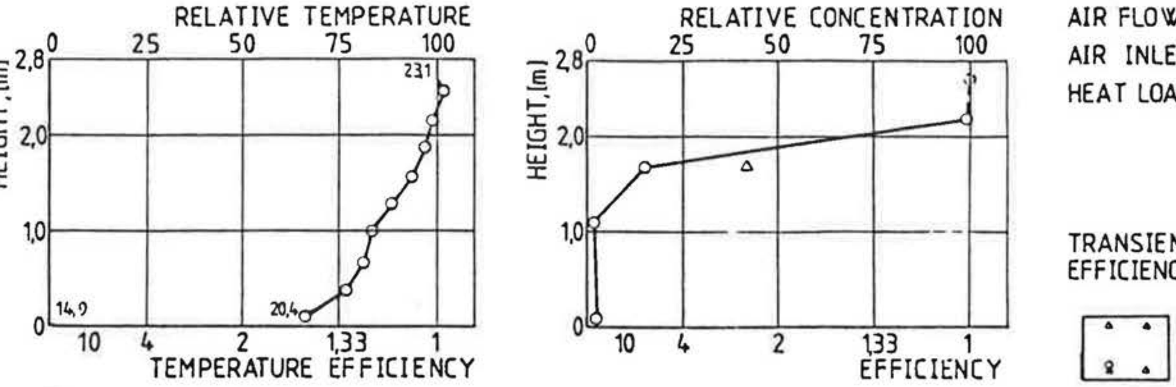
2	Filter clothing 1,55×0,3m^2	Reference test, direct solar radiaton at the floor (830 W), Whitout furniture



3	Filter clothing 1,55x0,3m^2	Reference test, winter conditions, without furniture

AIR FLOW: 160 m^3/h
AIR INLET VEL.: 0,1 m/s
HEAT LOAD: Sun ☐
Heated window ☒
Ceiling light ☒
Desk lamp ☐
Typewriter ☐
User terminal ☐
Persons 2

TRANSIENT EFFICIENCY: _____



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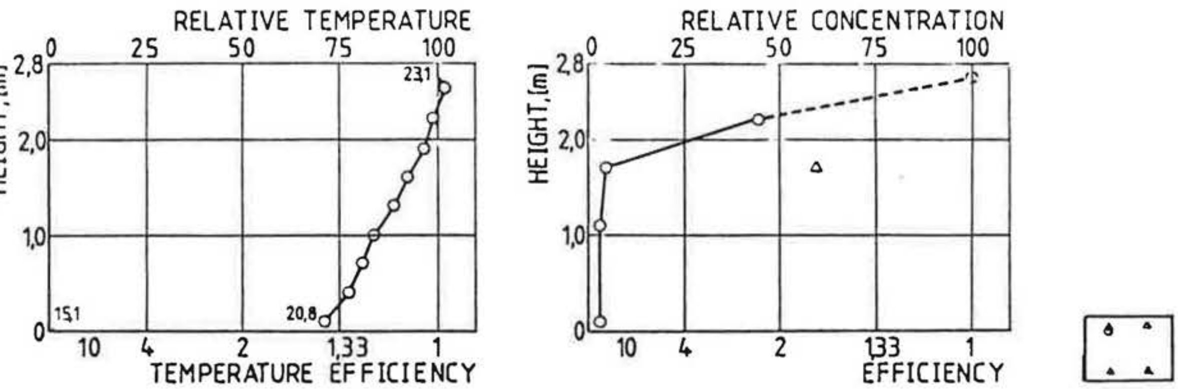
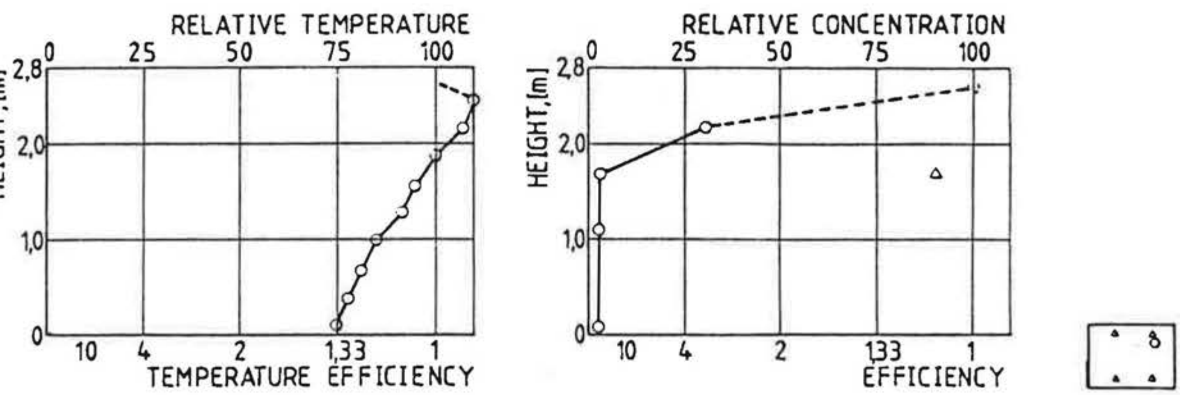
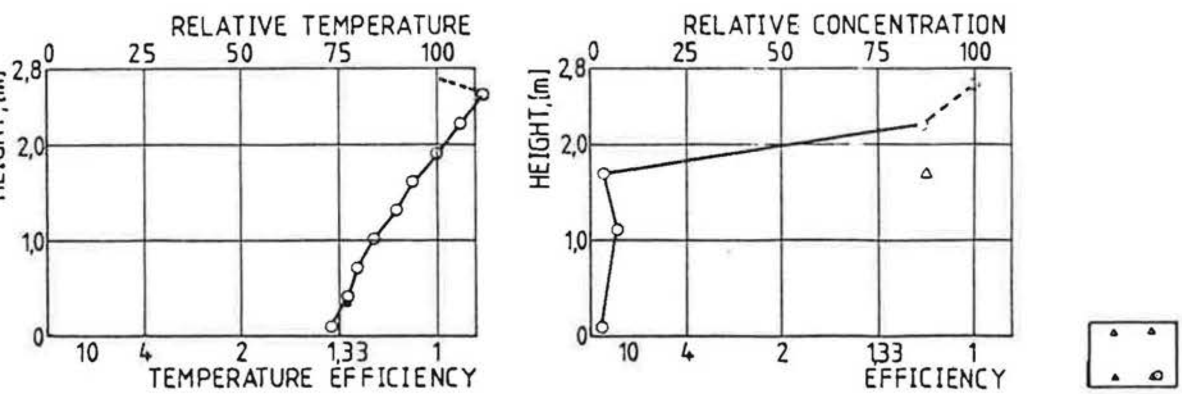
$\bar{u}_{0,1} = 0,13$



$\bar{u}_{1,1} = 0,06$
$\bar{u}_{0,1} = 0,08$

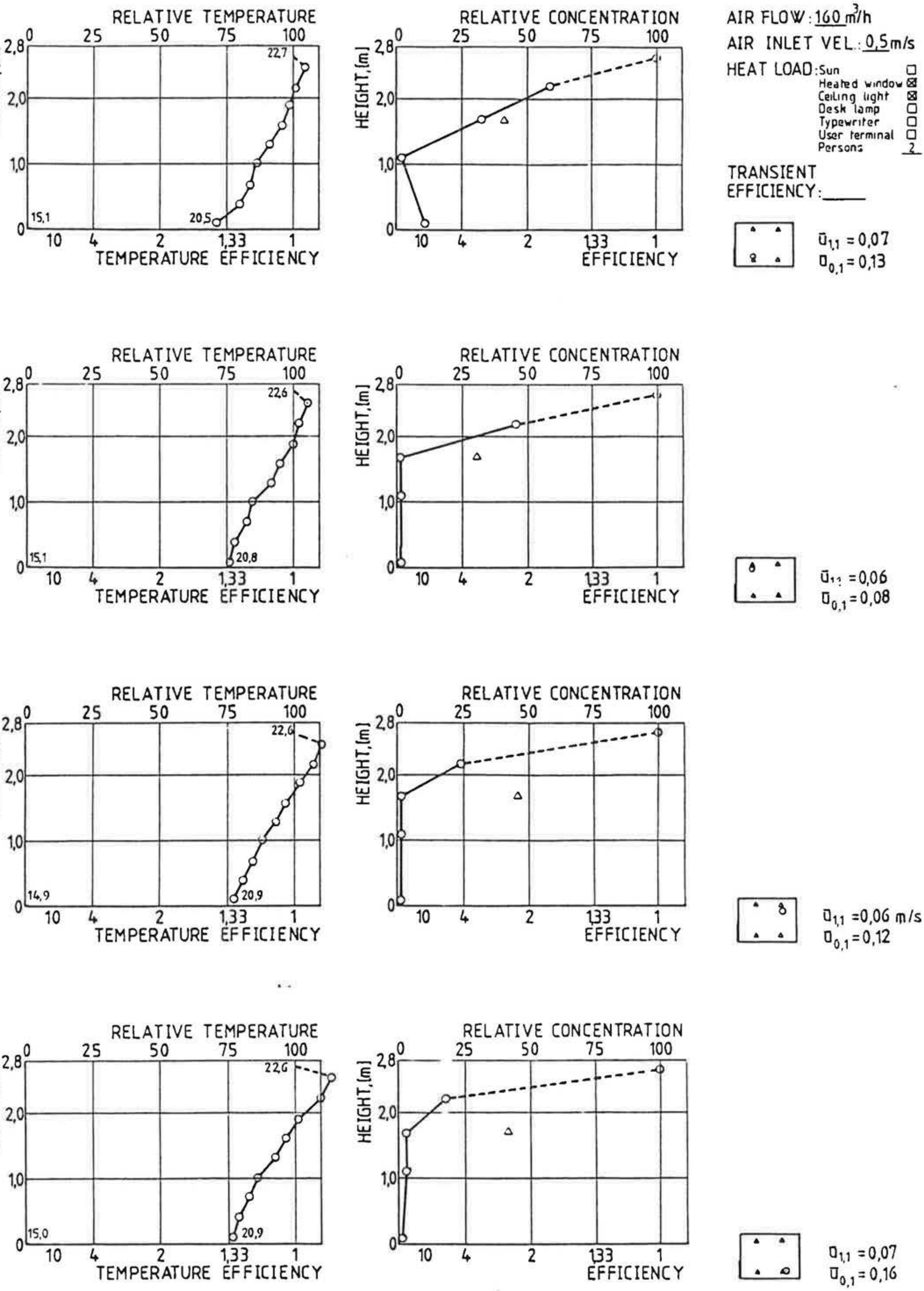


$\bar{u}_{1,1} = 0,05$
$\bar{u}_{0,1} = 0,09$



$\bar{u}_{1,1} = 0,06$
$\bar{u}_{0,1} = 0,12$

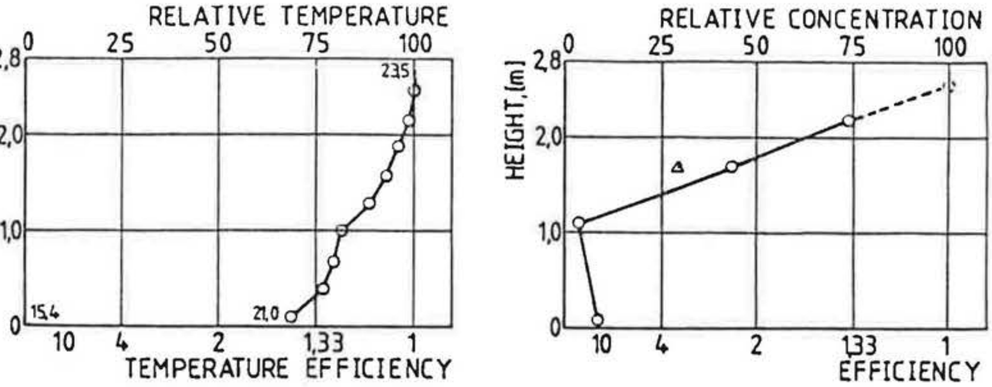
4	Filter clothing 1,55x0,3m^2	Reference test, as no 1, but with low air supply temperature, without furniture

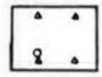


5	Perforated sheet, 10% degree of perforation, two areas ($0,7 \times 0,7 m^2$) with 54 holes in each direction	Summer conditions without direct solar radion, without furniture

AIR FLOW: 160 m³/h
AIR INLET VEL.: 1,0 m/s
HEAT LOAD: Sun ☐
Heated window ☒ 240
Ceiling light ☒
Desk lamp ☐
Typewriter ☐
User terminal ☐
Persons 2

TRANSIENT EFFICIENCY: 22





$\sigma_{1,1} = 0,08$
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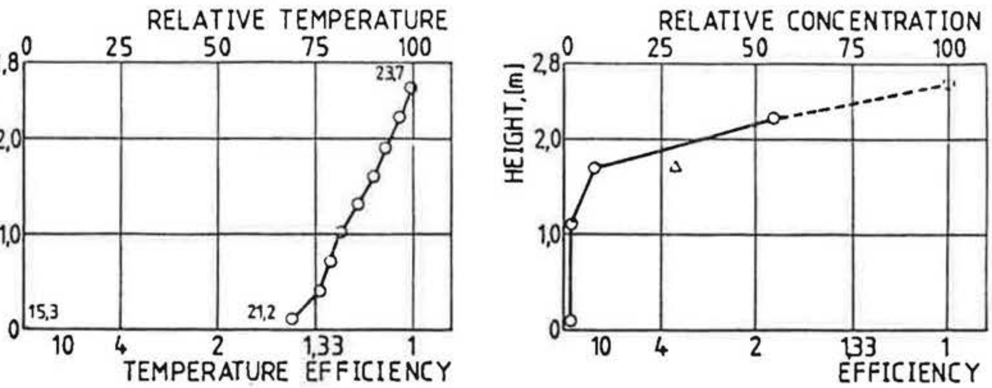

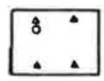

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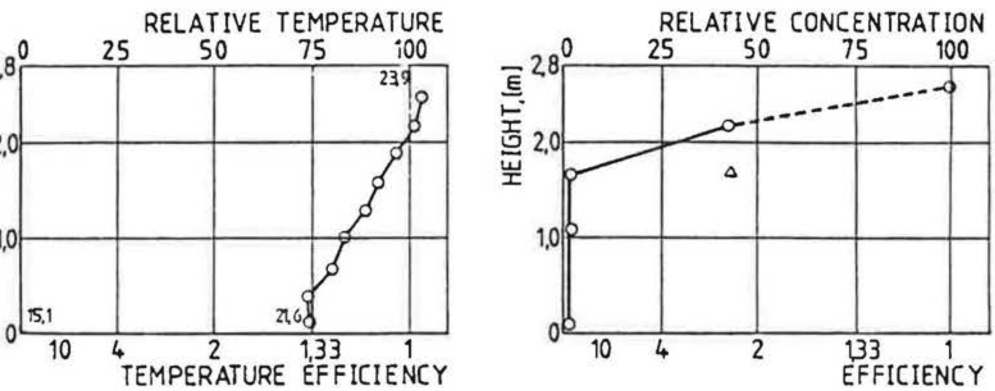
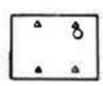
$\sigma_{0,1} = 0,11$





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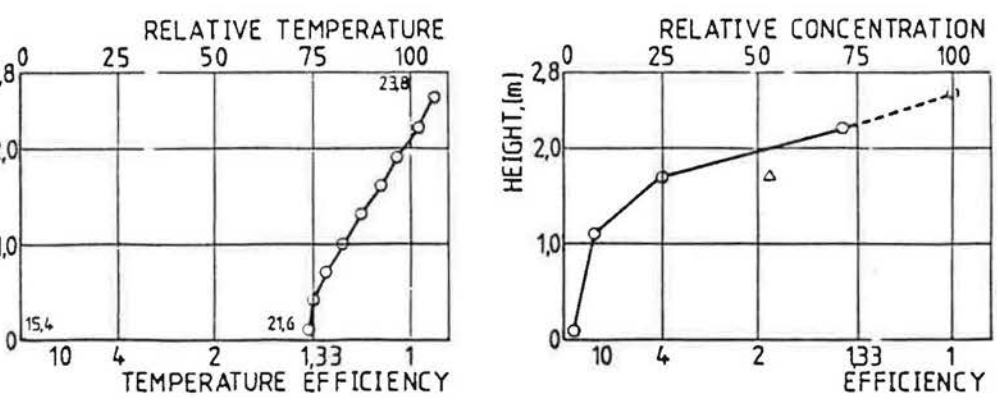
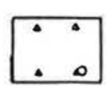
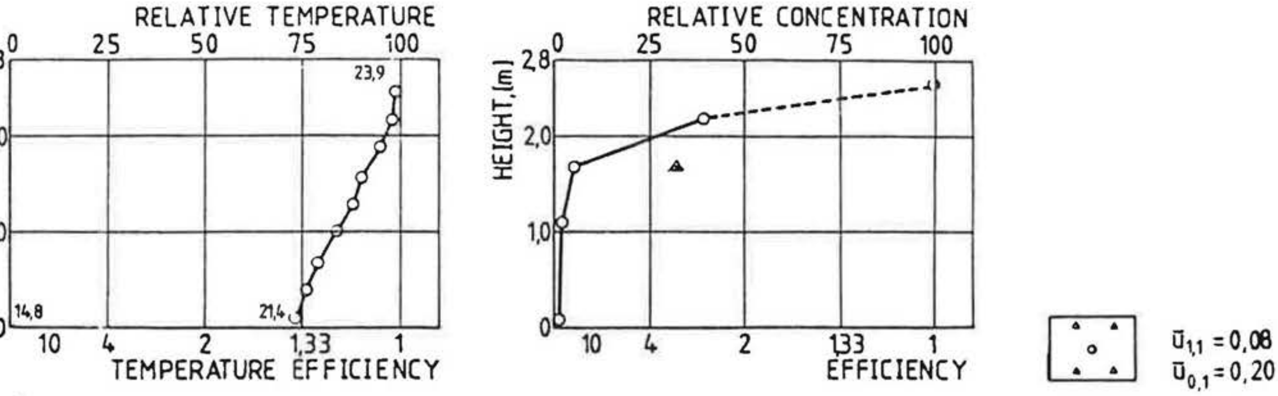


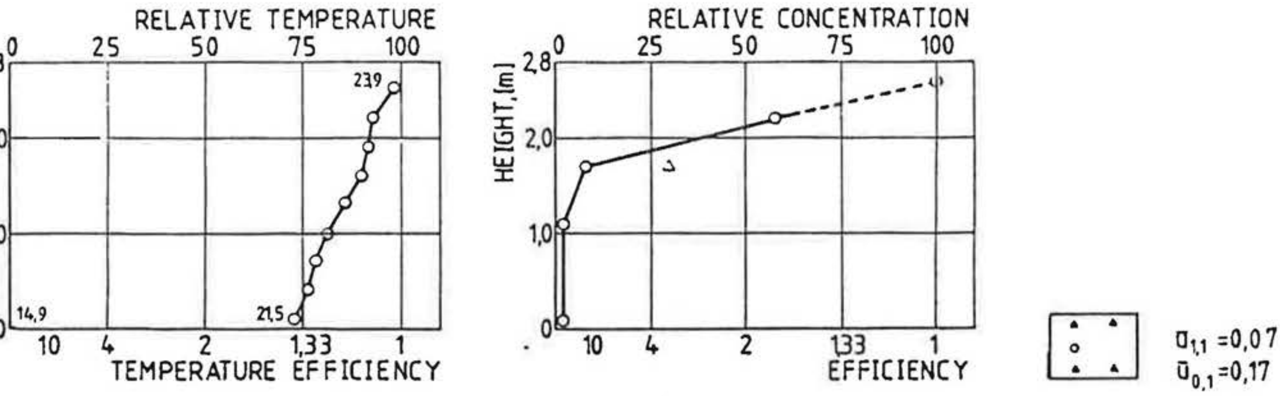


$\sigma_{1,1} = 0,08$
$\sigma_{0,1} = 0,20$

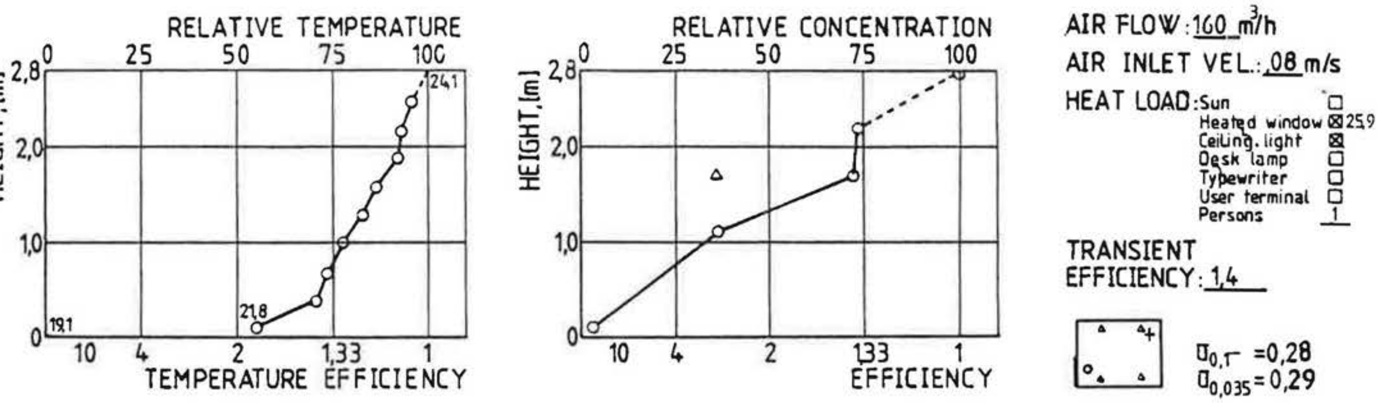
6	As no.5 but with one perforated area

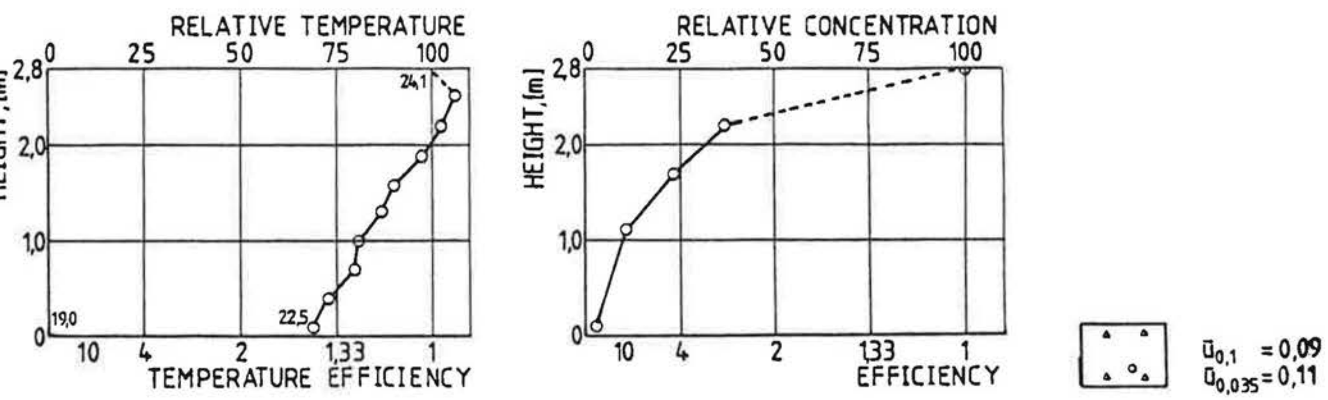


RELATIVE TEMPERATURE
0 25 50 75 100
23,9
14,8
21,4
10 4 2 1,33 1
TEMPERATURE EFFICIENCY
RELATIVE CONCENTRATION
0 25 50 75 100
HEIGHT, [m]
2,8
2,0
1,0
0
10 4 2 1,33 1
EFFICIENCY
$\bar{u}_{1,1} = 0,08$
$\bar{u}_{0,1} = 0,20$

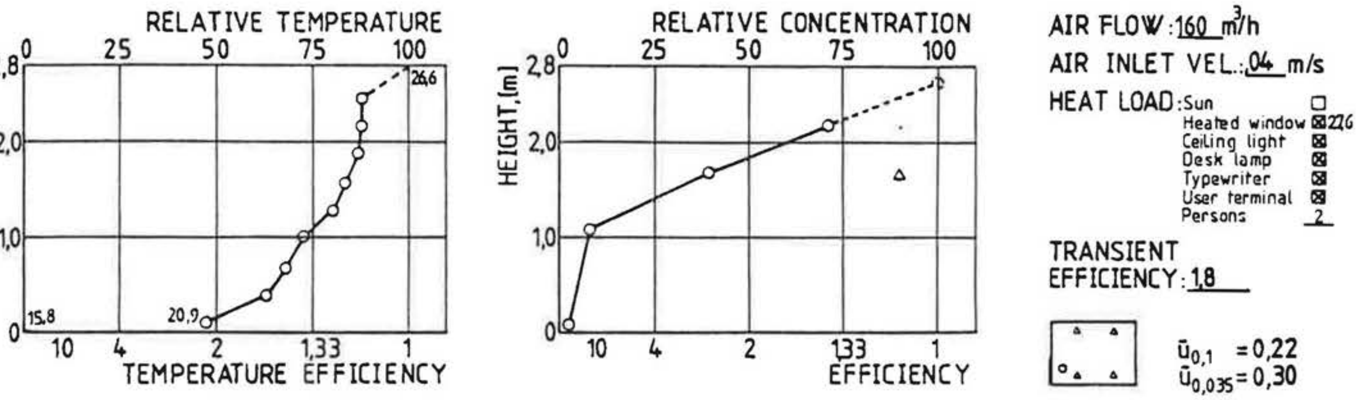


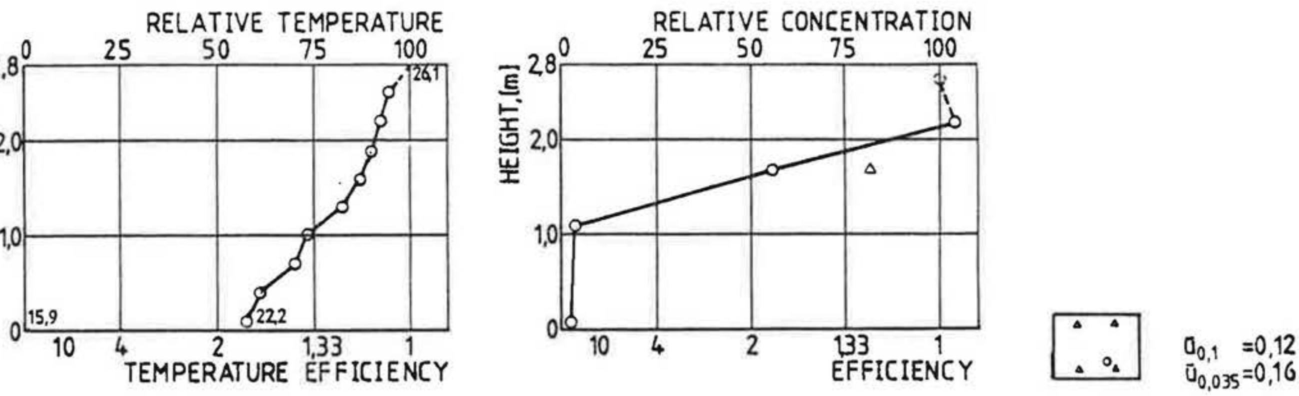
RELATIVE TEMPERATURE
0 25 50 75 100
23,9
14,9
21,5
10 4 2 1,33 1
TEMPERATURE EFFICIENCY
RELATIVE CONCENTRATION
0 25 50 75 100
HEIGHT, [m]
2,8
2,0
1,0
0
10 4 2 1,33 1
EFFICIENCY
$\bar{u}_{1,1} = 0,07$
$\bar{u}_{0,1} = 0,17$



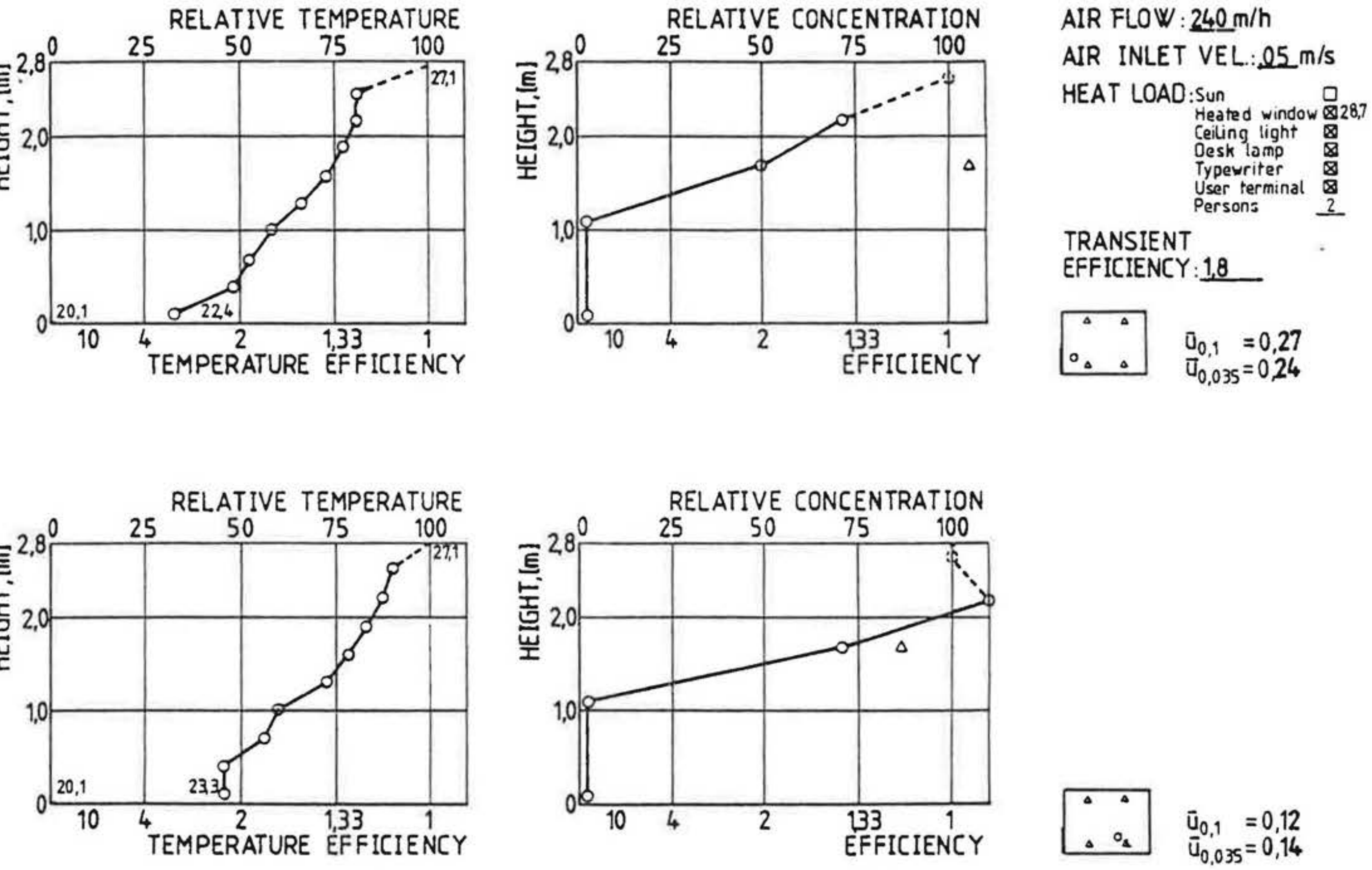


7	Filter clothing 1,55x0,75 m^2	With furniture in the test room

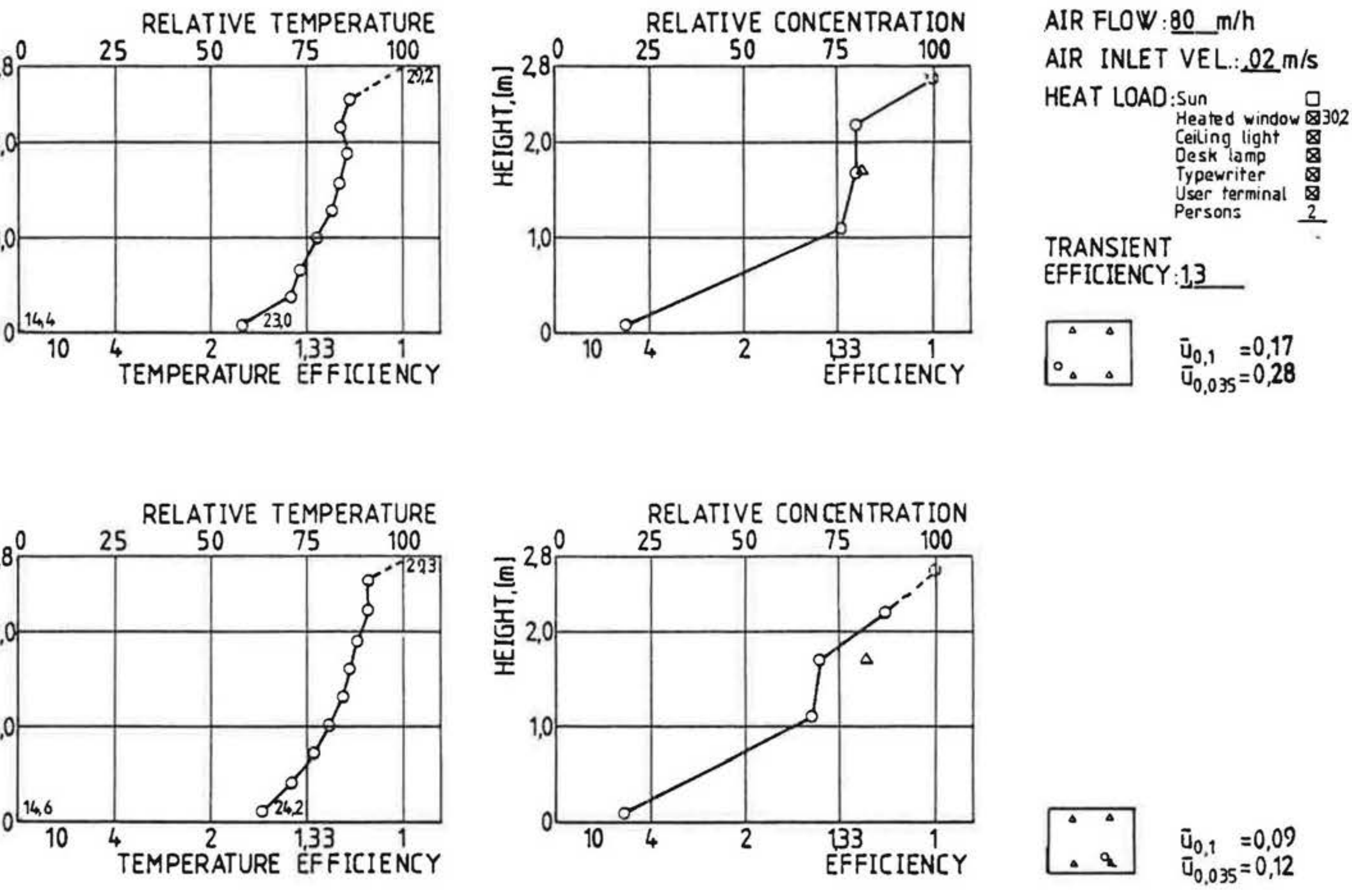




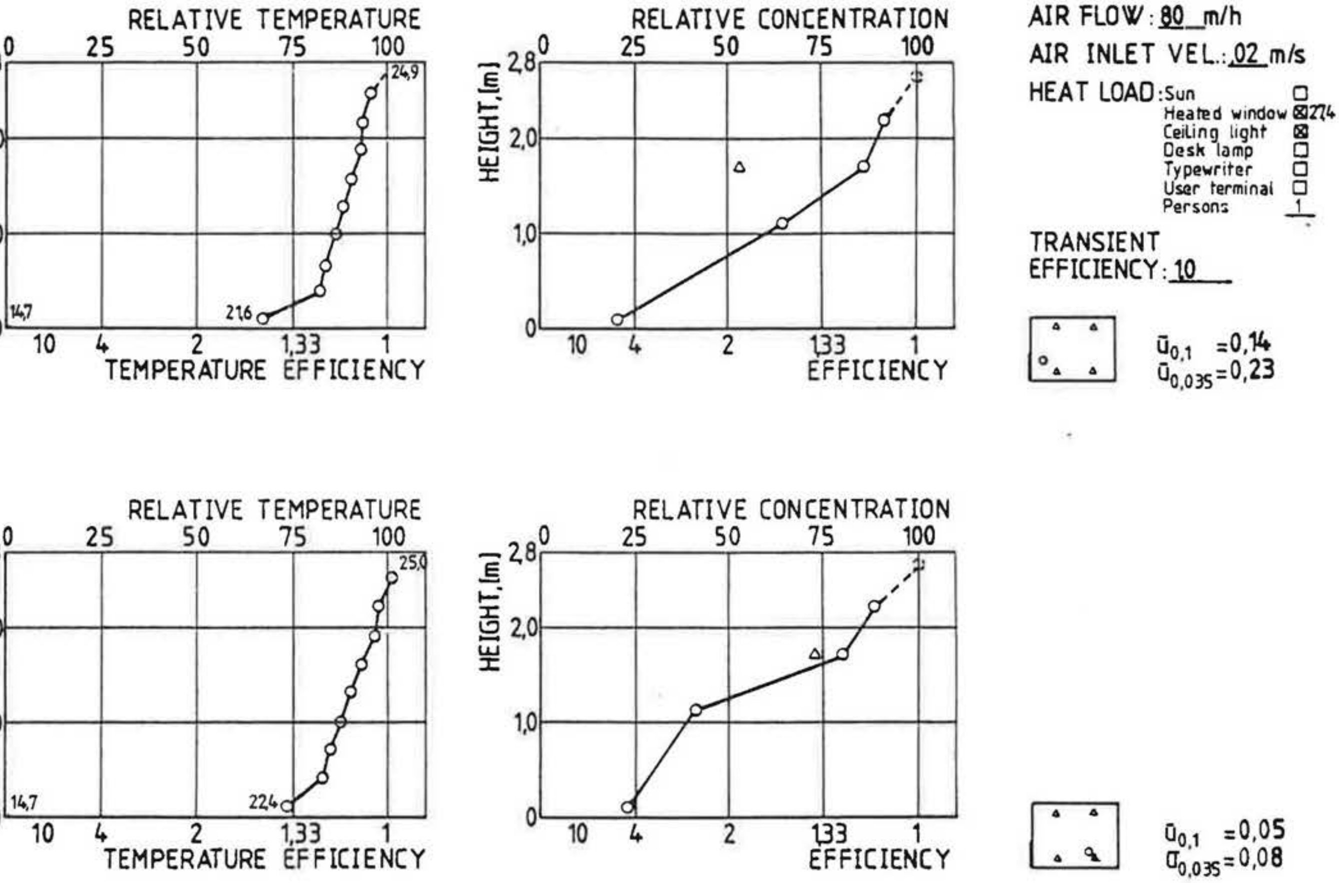
8	Filter clothing 1,55x0,75 m^2	With furniture in the test room



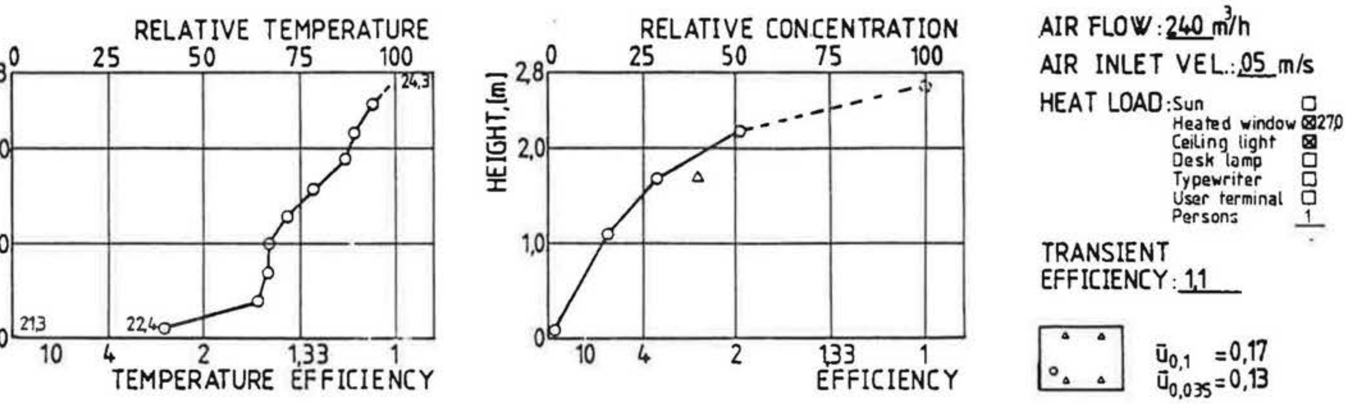
9	Filter clothing 1,55x0,75 m^2	With furniture in the test room

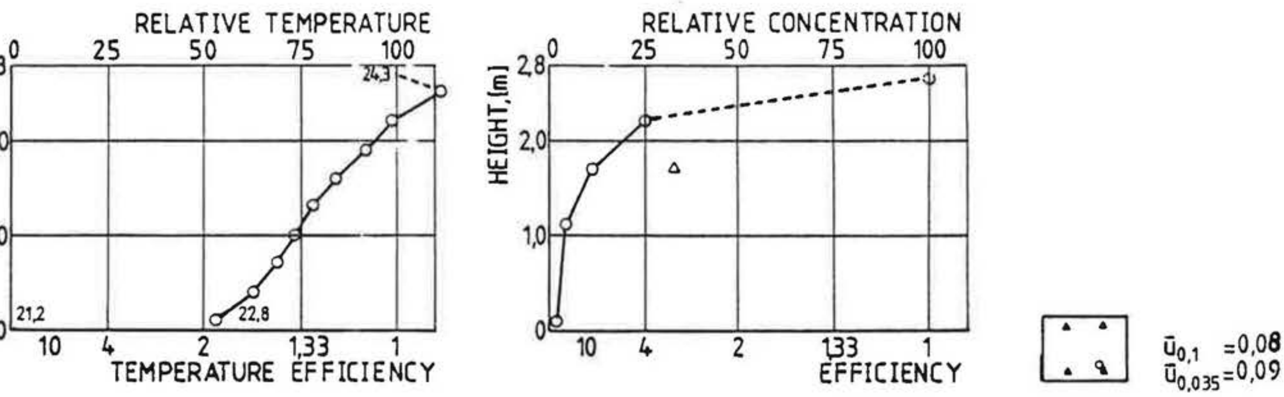


10	Filter clothing 1,55×0,75 m^2	With furniture in the test room

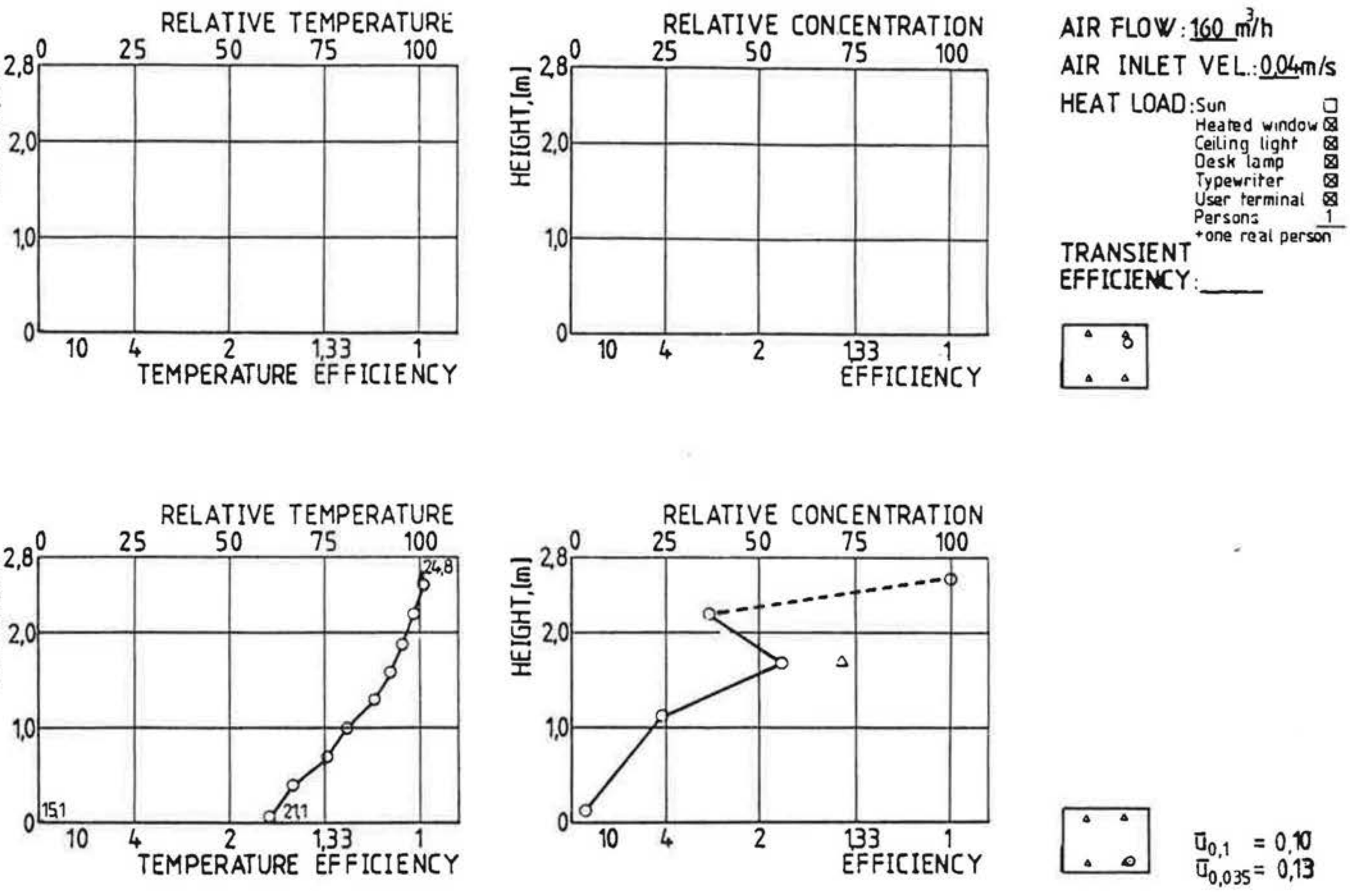


11	Filter clothing 1,55x0,75 m^2	With furniture in the test room

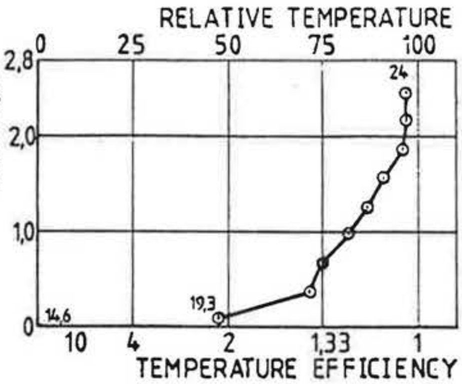


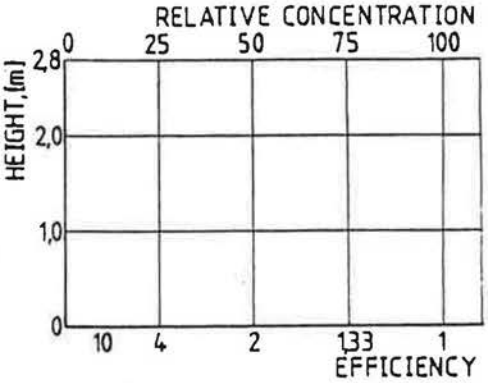


12	Filter clothing 1,55x0,75 m^2	With furniture in the test room



13	Filter clothing 1,55x0,75 m^2	With a real person in the test room, with furniture



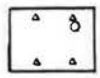


AIR FLOW: 160 m^3/h

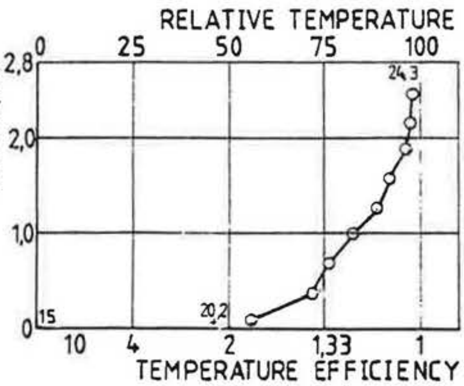
AIR INLET VEL.: 0,14 m/s

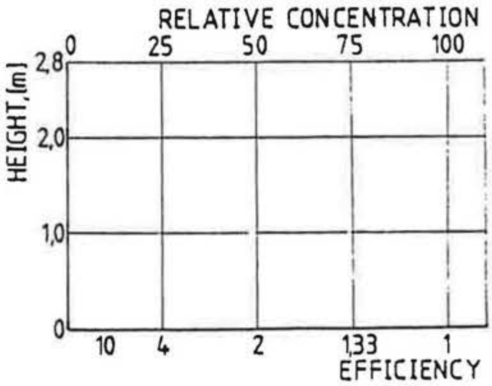
HEAT LOAD: Sun □
Heated window
Ceiling light
Desk lamp
Typewriter
User terminal
Persons 2

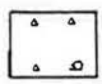
TRANSIENT EFFICIENCY: ____



$\overline{u}_{0,1}$ =0,15
$\overline{u}_{0,035}$=0,25

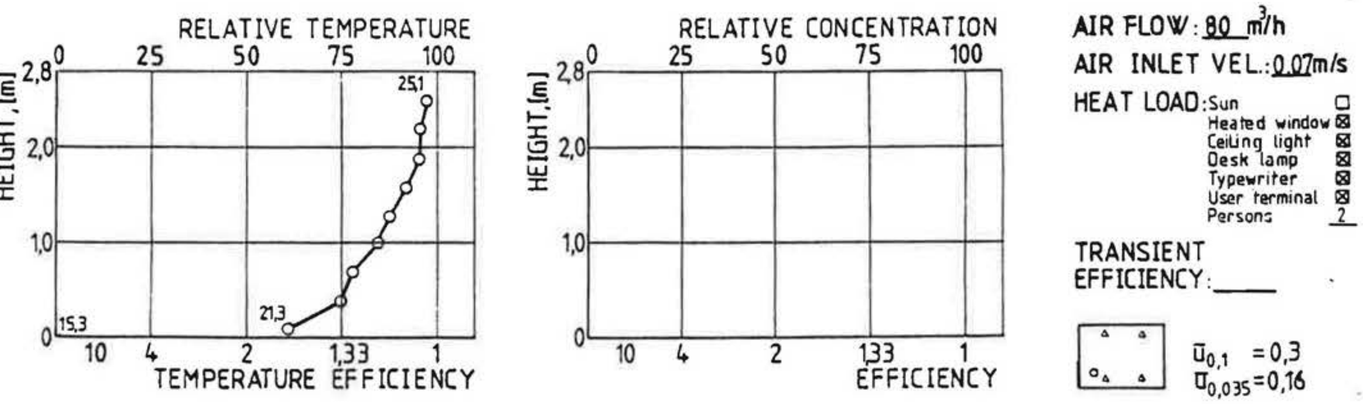


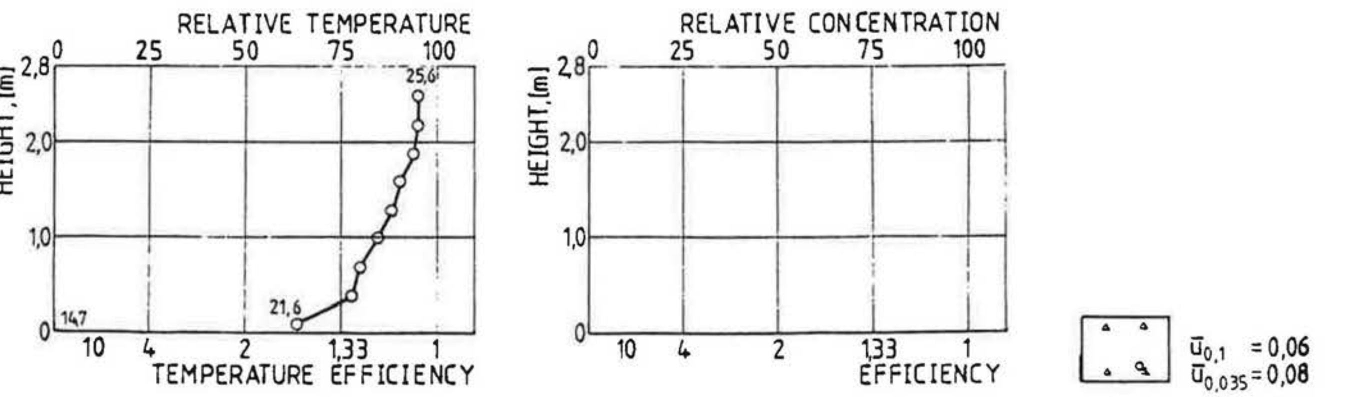




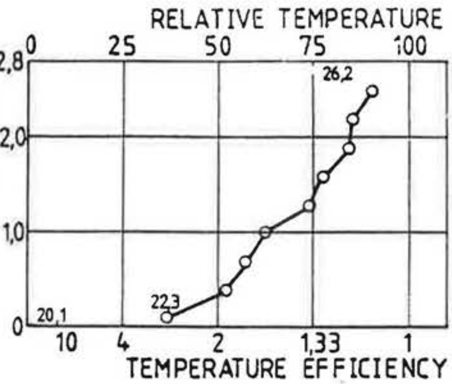
$\overline{u}_{0,1}$ =0,13
$\overline{u}_{0,035}$=0,16

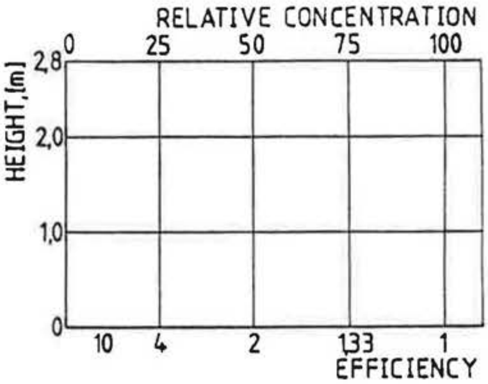
14 | Filter clothing 1,55x0,2 m^2 |





15 | Filter clothing 1,55x0,2 m^2 |



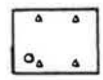


AIR FLOW: 240 m^3/h

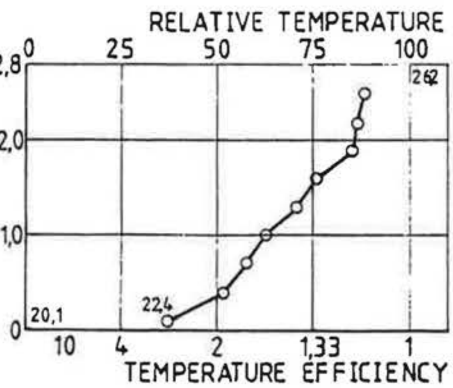
AIR INLET VEL.: 0,2 m/s

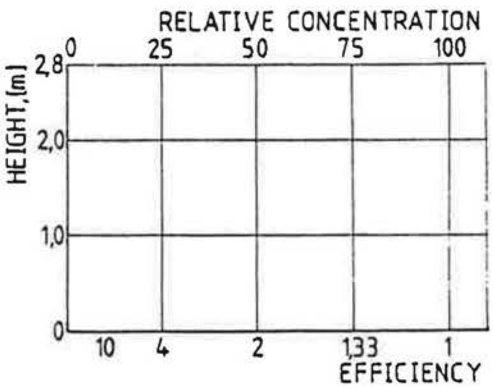
HEAT LOAD: Sun ☐
Heated window ☒
Ceiling light ☒
Desk lamp ☒
Typewriter ☒
User terminal ☒
Persons 2

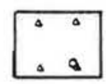
TRANSIENT EFFICIENCY: ______



$u_{0,1} = 0,11$
$u_{0,035} = 0,21$

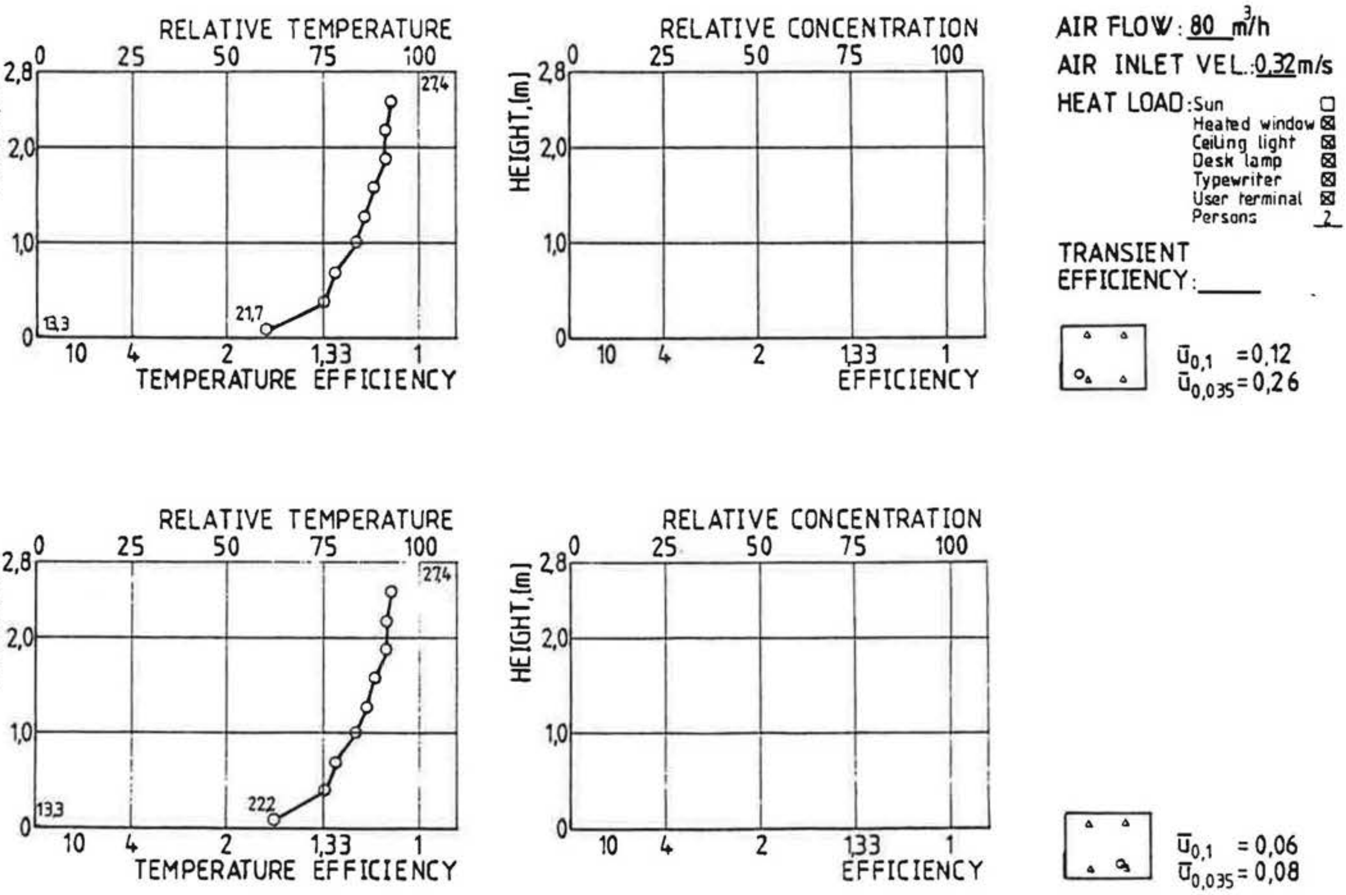






$u_{0,1} = 0,09$
$u_{0,035} = 0,15$

16 | Filter clothing 1,55x0,2 m^2 |



17 Filter clothing 1,55x0,045m^2

APPENDIX II

DESCRIPTION OF MODEL AND MODEL EQUIPMENT

The test room which was used for the laboratory investigations is shown in fig.II.1 and fig. II.2.

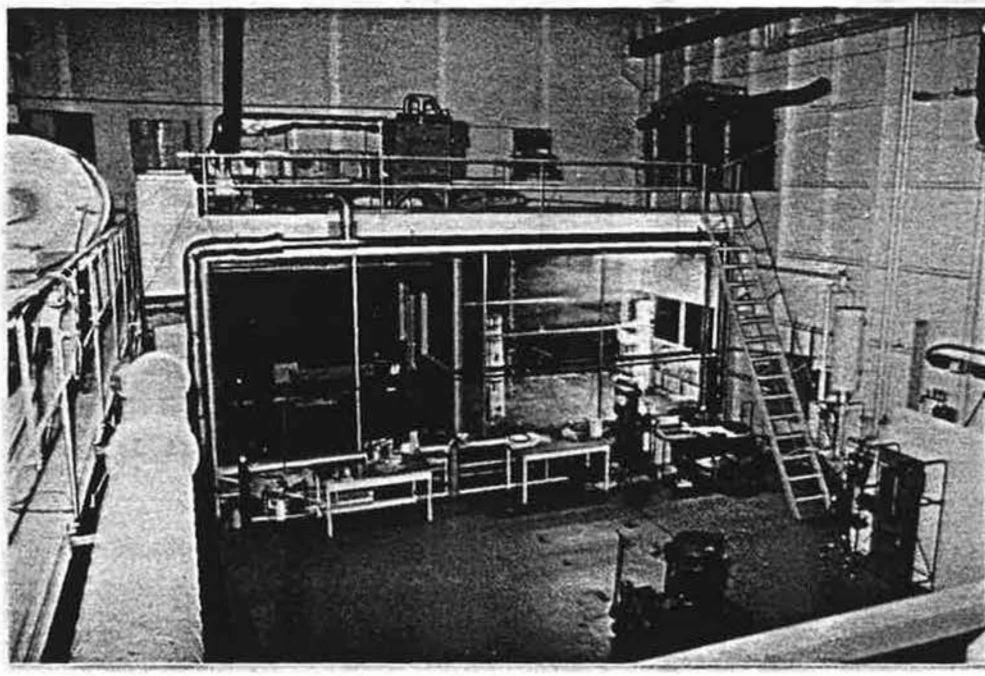


Fig.II.1 The test room which was used for the laboratory investigations.

The air inlet was an area 1,55 m long, with variable height. Warm radiators were used as "windows" for the experiments with summer conditions.

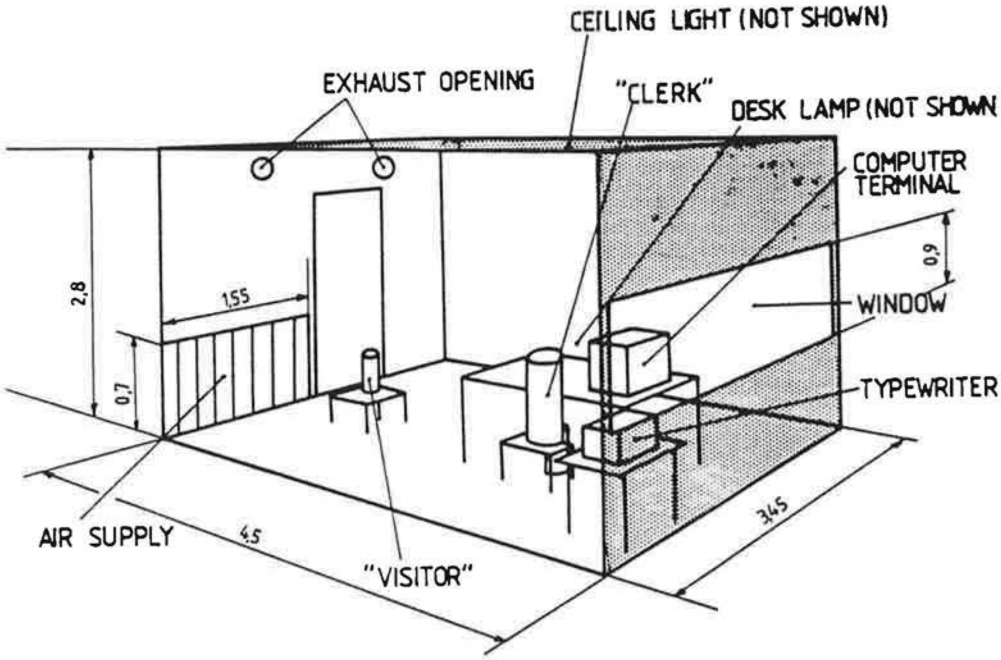


Fig.II.2 Test room with furniture.

The windows are shown in fig.II.3.

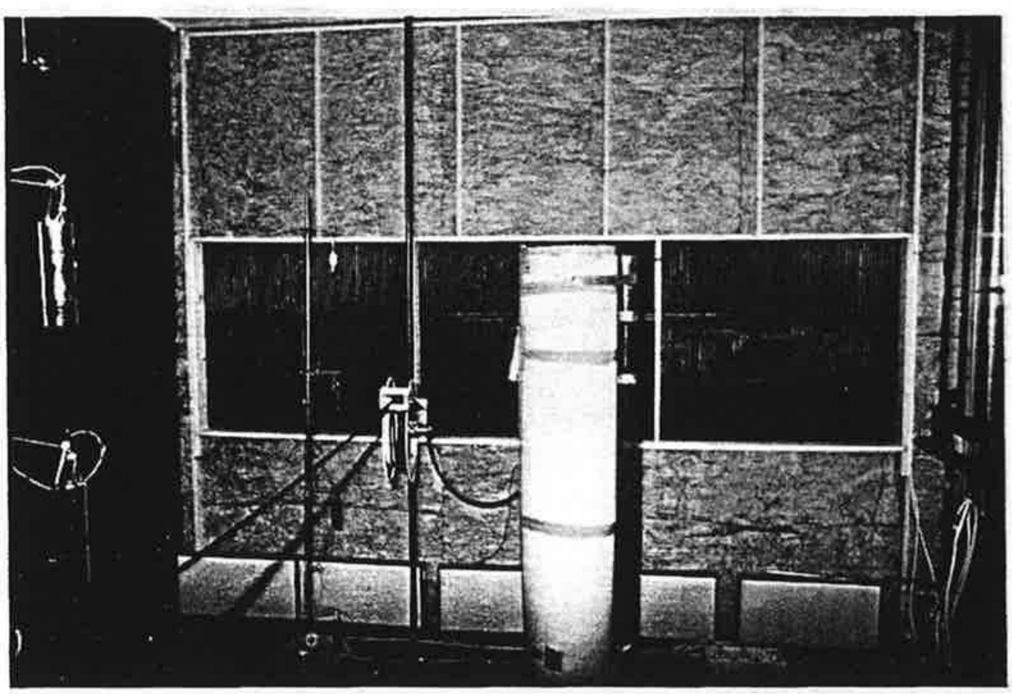


Fig.II.3. "Windows" with electric heaters beneath them. Window area is 3,2 x 0,9 m^2. The cylinder in front is a "person used in test 1 to 6.

II.1. Air supply.

The air supply is shown in fig.II.4.

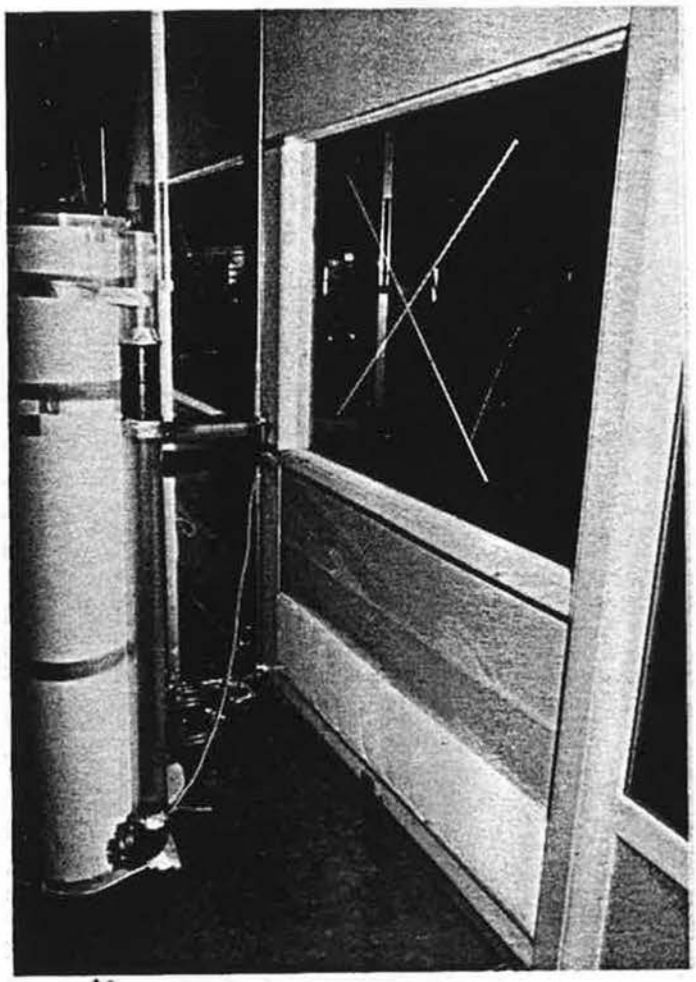


Fig.II.4. Air supply area.

Outdoor air was used in the experiments. The outdoor air was taken from a place far away from the discharge air area to avoid short circuit.

The air was filtered before it was heated in an electric heating element and blown into the room. The air volume was measured with an orifice plate and an inclined tube monometer. A damper was used to adjust the volume. To minimise the infiltration of air in the model, the evacuation volume was set by the pressure difference between the laboratory and the model. (This difference should be zero). The difference was measured with an electronic micromanometer. Tests have shown that the model has an air tightness about the same as that of a typical Norwegian private house.

The temperature of the inlet air was set by an electric heating element and a controller (proportioned, resit, derivative).

APPENDIX III

DESCRIPTION OF MEASURING EQUIPMENT

The ventilation effectiveness was measured by tracer gas. Some of the probes for tracer gas analysis, anemometers and temperature meters were located at a column.

Four of the probes were located at the column which could be moved through the room see fig.III.1.

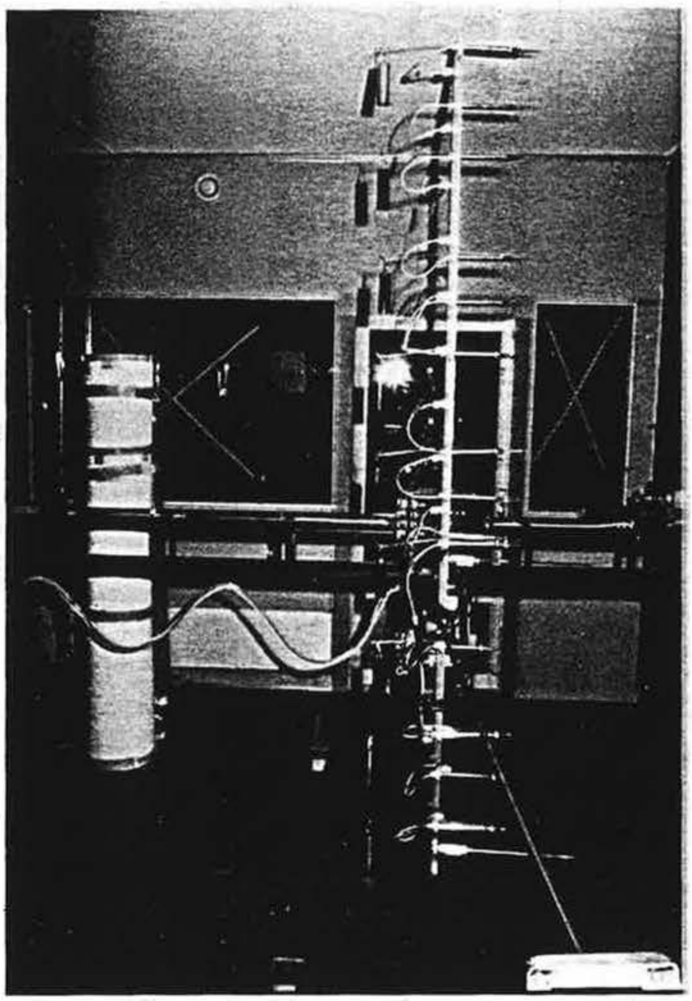


Fig.III.1. Column, equipped for temperature measurements, tracer gas sampling and air velocity measuring.

The column was moved by two electric motors with toothed wheels, controlled by the microprocessor. The position of the colum was monitored by two potentiometers located at the shaft of the motors.

Tracer gas techniques.

The measurements were done with an infrared analyser type URAS 1. Multipoint measurements with one analyser were conducted by means of an auxilliary pumping system as shown in fig. III.2.

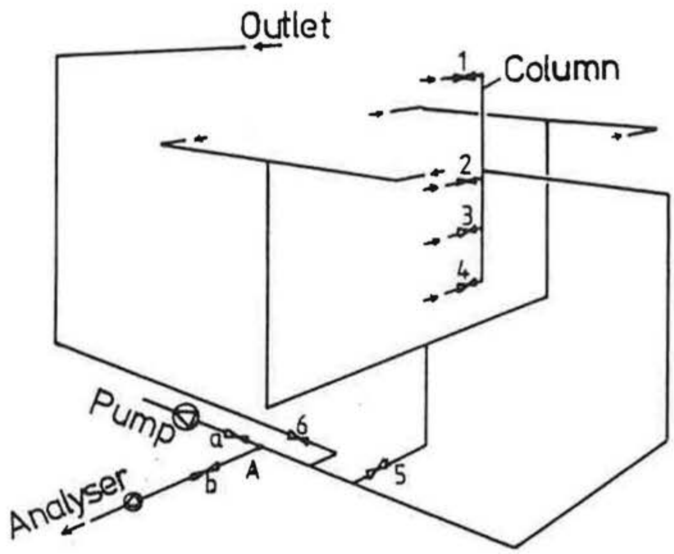


Fig. III.2. Tubing and valve arrangements for auxiliary suction from measuring points.

The vacuum pump was much more powerful than the pump of the analyser. For each sampling (opening of solenoid 1, 2, 3, 4, 5 or 6), the vacuum pump sucked for 18 seconds to bring a "fresh" sample to the solenoide marked A. The distance from this solenoid to the analyser was short. When the solenqid A closes port a, port b is opened and the analyser pump sucked for 70 seconds.

The measured values from the gas analyser were sampled by means of a microprocessor-based data-logger. This microprocessor was also used to control the solenoide valves and the motors moving the column.

The microprocessor was supplied by Ing. Paul Jorgensen in Trondheim and was based on Texas-components.

Air velocity.

In fig. III.3. one of the air velocity anemometers is shown. Two anemometers were used in tests 7 to 17 the anemometers were situated 0,1 and 0,035 m above the floor. In tests 1 to 6 the anemometers were situated 0,1 and 1,1 m above the floor.

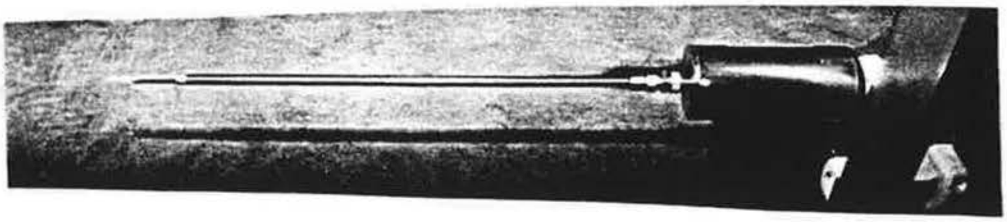


Fig.III.3. Hot film anemometer, type TSI.

The anemometers were calibrated in a TSI-calibration unit.

Air temperature

The air temperatures were measured with semiconductors of type AD 590. They were shaded from radiation with nickled cylinders, see fig. III.4.

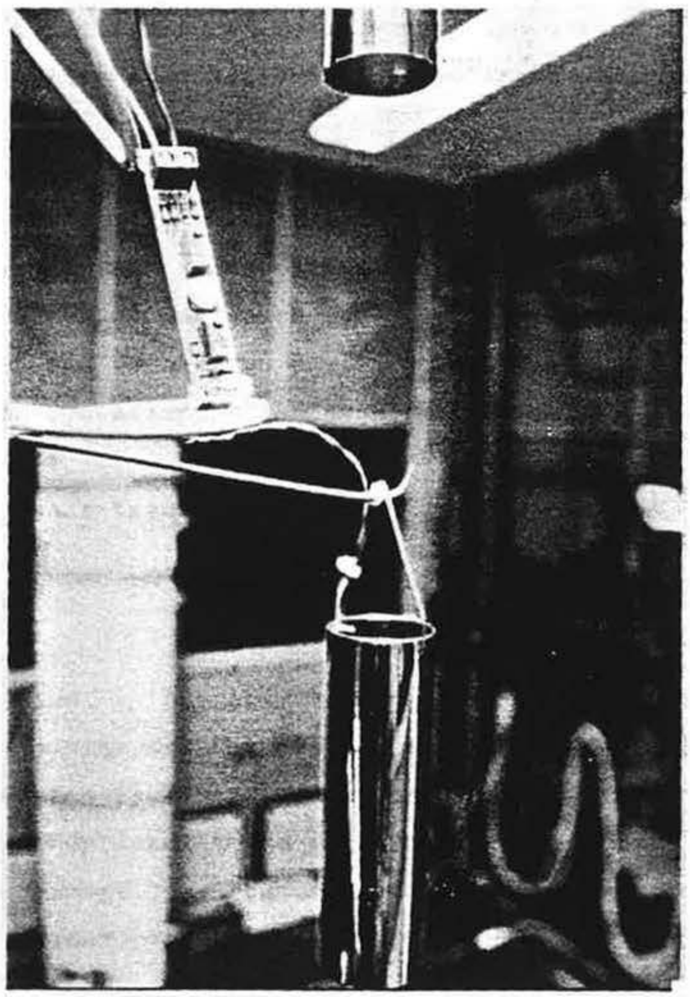


Fig. III.4. Cemiconductors for temperature measurements shaded by nickled cylinders.

APPENDIX IV

METHODS OF MEASURING

The sampling, analysing of measurements and moving of the column were done automatically as described in Appendix II.

The studies started with a "clean" room. The tracer gas supply and the sampling were started at the same time. The gas concentration increased until the steady-state was obtained. In the steady-state situation the column was moved to the other positions. Then it returned to the starting position, and the traser gas supply was turned off before starting the transient measurements.

N_2O was used as a tracer gas.

<u>Test results.</u>

Fig.IV.1. shows a typical test record. The ventilation scheme for this record is high wall supply and exhaust ("short circuiting scheme" from earlier tests). The concentrations are increasing gradually from zero up to steady level. The measured concentrations are fluctuating very much, except for the exhaust air measurements. This is due to the source characteristics which, together with the ventilation system, create much turbulence.

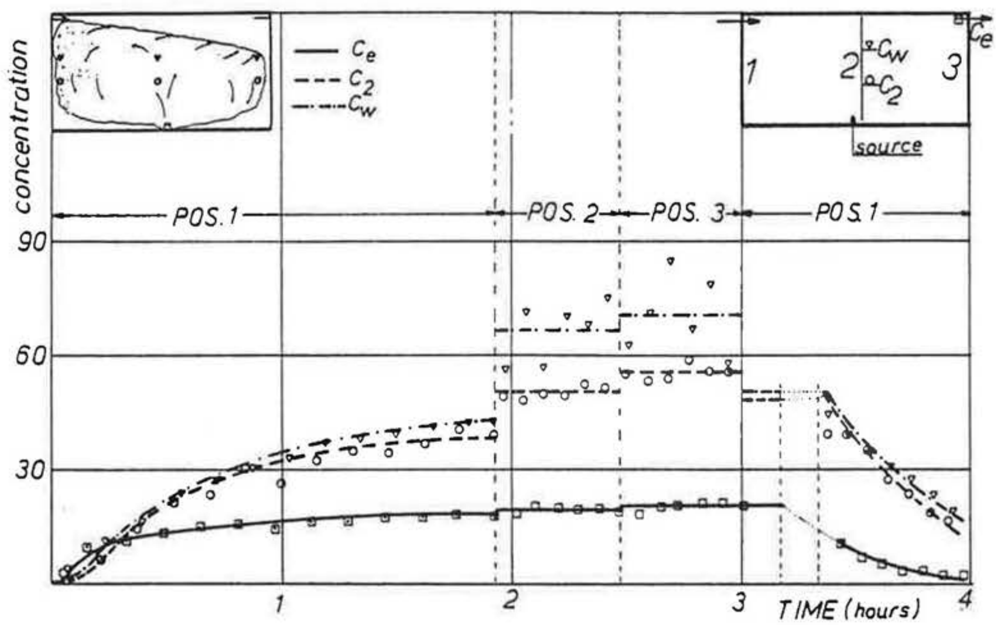


Fig. IV.1. Tracer gas test record. "Short-circuiting scheme". $T = +9^0C$. Earlier tests (Skåret, Mathisen 1982).

The fluctuations continue for the same reason through the steady state period. In the decay period (tracer gas supply is shut off), the concentrations decrease gradually to zero. During this period the measurements are not fluctuating. The reason for this is that the concentrations are distributed more evenly when the source is shut off. The curve-fitting, regression coefficients for all measurements are satisfactory in spite of the fluctuations. It should be mentioned that the heat sources are not switched off when the tracer gas supply is turned off.

In fig.IV.1 a lin.-log plot of the concentration is made for the decay period (after the tracer gas supply has been turned off).

IV.2. Statistical processing of data.

Ventilation effectiveness.

At each position of the column, five samples of tracer gas concentration at each level were taken. The mean value and sample standard deviation for each level and position were calculated. The results of these calculations are not shown in this report. The mean values were used in the following calculations of effectiveness.

The transient effectiveness was calculated from the decay of the tracer gas in the test room. The slope of the curves was fitted by the use of least squares method.

Air velocity.

The air velocity was measured for a period of 3 minutes. The mean velocity of this period was calculated and used in the further calculations.

APPENDIX V

VENTILATION EFFECTIVENESS, THEORY

The equation for conservation of mass for a single room can be written as follows:

$$V_R \frac{d\bar{C}_R}{dt} = \dot{Q} - \left[\sum_{i=1}^{n} (C_E - C_S) \cdot \dot{V}_i + \sum_{j=1}^{m} \dot{E}_j \right]$$

C_R = Spatial mean concentration (mass/unit volume).

C_E = Concentration of contaminant in the exhausted air.

C_S = Concentration of contaminant in the supply air.

E_j = Elimination source no. j.

Q = Contaminant production rate (mass/unit time).

V = Exhaust device no. i (equal amount of air is to be supplied).

V_R = Total room volume.

t = Time.

The left hand side of the equation represents the concentration change if contaminant generation and removal is not in balance. The conditions for the zone of occupation do not enter directly into this equation but are proportional to either C_R or C_E. However, the constant of proportionality differs between systems. The conditions are also influenced by the thermal conditions for the ventilation system and the room and,of course, also by the occupational characteristics and the characteristics of the pollution sources. As we can see, it is not possible to control the air quality by controlling the mass balance parameters, such as inlet/outlet conditions and ventilation air flow rate. In other words, it is not possible to design for air quality, by using the general mass balance equation.

A better approach is to develop relations between the conditions for the zone of occupation and the parameters in the mass balance equation.

It is appropriate to call these relations performance parameters or effectiveness. If we subtract the elimination rate from the production rate, we may, for the steady state situation, write the mass balance equation as follows:

$$\dot{V}_{cm} = \frac{\dot{Q} - \sum_{j=1}^{m} \dot{E}_j}{C_{cm} - C_S} = \text{complete mixing flow rate} \tag{2}$$

C_{cm} = complete mixing concentration.

Without going into details it is interesting, at this point, to state that ventilation air flow rates can be decreased by using three different approaches:

i. Reduce contaminant generation
ii. Increase local elimination
iii. Increase concentration differences between supply and exhaust air

The first two approaches are obvious and the practical implications are also obvious. The third one seems to oppose air quality. The concept of ventilation effectiveness will show that this is not the case.

If we measure the concentration in the zone of occupation, C_O, an apparent air flow rate for the room can be calculated.

$$\dot{V}_a = \frac{\dot{Q} - \sum_{j=1}^{m} \dot{E}_j}{C_O - C_S} \tag{3}$$

Now, we can define the ventilation effectiveness as the ratio between V_a and V_{cm}.

$$\varepsilon_{st} = \frac{\dot{V}_a}{\dot{V}_{cm}} = \frac{C_{cm} - C_S}{C_O - C_S} = 1 + \frac{C_{cm} - C_O}{C_O - C_S} \tag{4}$$

C_{cm} is equal to the mean exhaust air concentration, C_E, and is no direct measure of the air quality. Through the ventilation effectiveness, the concentration in the zone of occupatnon can be determined. Designing systems having $C_{cm} > C_o$ are efficient. This means that aiming at complete mixing is not the best guide to efficient ventilation. These conclusions are arrived at in a rather simple and obvious way without having any theoretical flow model for the room. To come further, a two zone single room mixing model is introduced, Figure V.1.

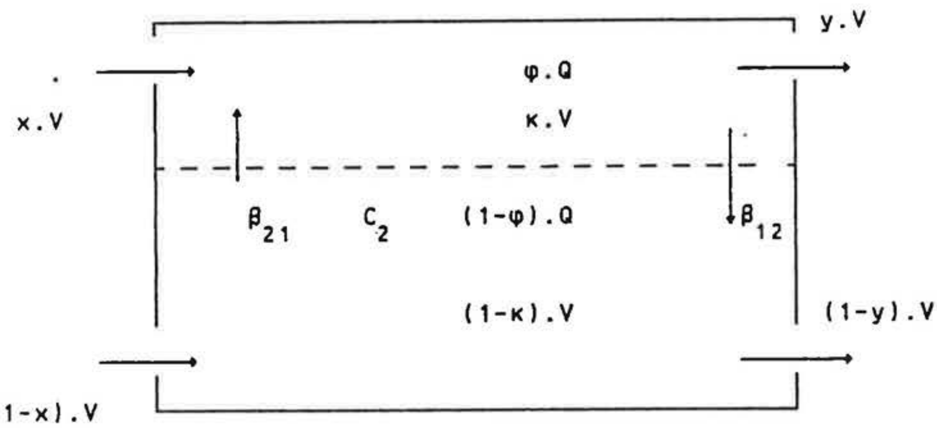


Fig. V.1. Two zone mixing model. Each zone has good mixing. ϕ-Relative production rate in zone 1. κ-Relative volume of zone 1. β_{12}-interzonal mixing parameter.

This model is chosen not only because it is simple but also because practical experience has indicated that ventilated rooms tend to behave like this. The two zones are the supply zone and the rest of the room. For simplicity, the supply air concentration is taken to be zero and the net production rate, Q_n, is introduced.

$$\dot{Q}_n = (\dot{Q} - \sum_{j=1}^{m} \dot{E}_j)$$

$$\frac{d\bar{C}_1}{dt} = \frac{\phi \dot{Q}_n}{k\ V} - \frac{Y + \beta_{12}}{k} n.\bar{C}_1 + \frac{(y - x) + \beta_{12}}{k} n.\bar{C}_2 \tag{5}$$

$$\frac{d\bar{C}_2}{dt} = \frac{(1 - \phi)}{(1 - k)} \frac{\dot{Q}_n}{V} + \frac{\beta_{12}}{1 - k} n.\bar{C} - \frac{1 - x + \beta_{12}}{1 - k} n.\bar{C}_2$$

C_1 = Mean concentration in zone 1.

C_2 = Mean concentration in zone 2.

κ = Relative volume of zone 1.

V = Total room volume.

ϕ = Relative production rate in zone 1.

Q_n = Total net production rate of contaminants.

β_{12} = Interzonal mixing parameter.

n = Nominal air exchange rate (V/V).

x = Relative supply air flow rate for zone 1.

y = Relative exhaust air flow rate for zone 1.

The mixing within each zone is assumed to be good. These equations can be rewritten:

$$\frac{d\bar{C}_1}{dt} = a_{11}\bar{C}_1 + a_{12}\bar{C}_2 + a_{10}$$

$$\frac{d\bar{C}_2}{dt} = a_{21}\bar{C}_1 + a_{22}\bar{C}_2 + a_{20} \qquad (6)$$

The steady state solution is obtained by letting the time derivatives become zero. The steady state solution is:

$$\bar{C}_1 = \frac{a_{10}a_{22} - a_{20}a_{12}}{a_{12}a_{21} - a_{11}a_{22}}$$

$$\bar{C}_2 = \frac{a_{20}a_{11} - a_{10}a_{21}}{a_{12}a_{21} - a_{11}a_{22}} \qquad (7)$$

The steady state ventilation effectiveness, letting zone 2 be the zone of occupation, is:

$$\varepsilon_{st} = \frac{C_{cm}}{C_2} = \frac{k}{n\phi} \cdot \frac{a_{12}a_{21} - a_{11}a_{22}}{\frac{a_{20}}{a_{10}} a_{11} - a_{21}} \tag{8}$$

New information is obtained by solving the equations, letting a_{10} = a_{20} = 0, i.e. solving for the step response.

$$\bar{C}_1 = K_1 e^{\lambda_1 t} + K_2 e^{\lambda_2 t}$$

$$\bar{C}_2 = K_1 \, {}^1k_2 e^{\lambda_1 t} + K_2 \, {}^2k_2 e^{\lambda_2 t} \tag{9}$$

$$\lambda_1 = \frac{1}{2} (a_{11} + a_{22}) + \frac{1}{2} \left[(a_{11} - a_{22})^2 + 4 \, a_{12} a_{21} \right]^{\frac{1}{2}}$$

$$\lambda_2 = \frac{1}{2} (a_{11} + a_{22}) - \frac{1}{2} \left[(a_{11} - a_{22})^2 + 4 \, a_{12} a_{21} \right]^{\frac{1}{2}}$$

$${}^1k_2 = \frac{\lambda_1 - a_{11}}{a_{12}}$$

$${}^2k_2 = \frac{\lambda_2 - a_{11}}{a_{12}}$$

APPENDIX VI

PERFORATED PANELS COMPARED TO FILTER CLOTHING AS AIR SUPPLY OPENINGS

Before the experiments started, it was done a simple test to find the best materials for the air supply opening.

From the results it can be seen that there is a tendency for perforated sheets to cause higher air velocities in the room than those caused by filther clothing. Only these degrees of perforation were tested, however,so it might be possible to obtain better results with other degrees of perforation and a different shaping of the supply opening.

In test 5 and 6, App.I, it was used perforated sheets. If test 5 is compared to test 3 (in App.I) one can see that both the air velocoity and the temperature efficiency are nearly equal. However, in test 3 the opening has less than half the height of the opening in test 6. Test 6 with half the area of that used in test 5 gives to high air velolcities.

Symbols:

u_0 - supply air velocity

u_m - maximum measured air velocity, measuring points shown at the figures

t_m - temperature measured in the same point as u_m

t_r - room air temperature, shown at the figure

t_0 - supply air temperature

$\Delta T_m = t_r - t_m$

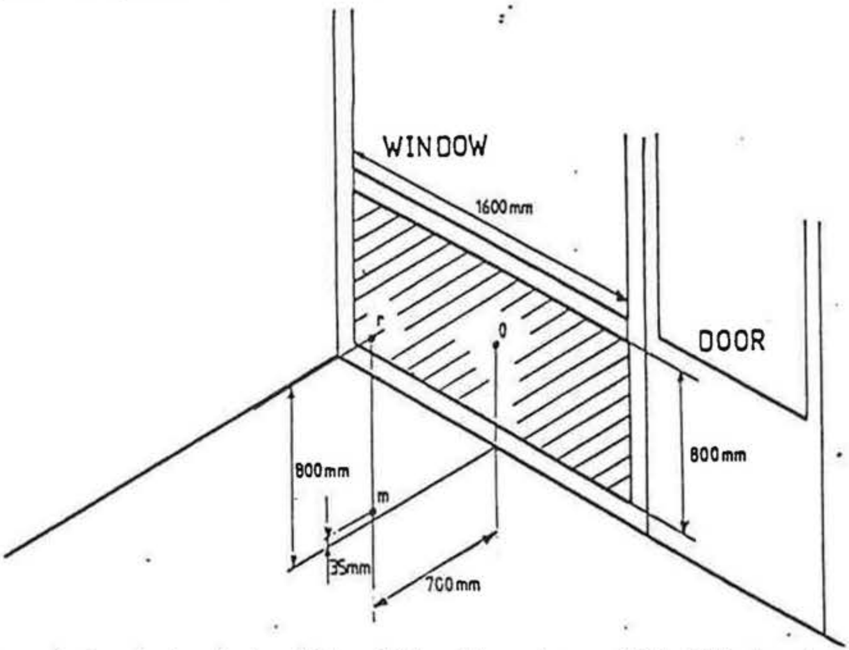
$\Delta T_0 = t_r - t_0$

$t_{0,1}$ - air temperature 0,1m above the floor

TEST NR.1

Air flow rate: 160m^3/h

Degree of perforation: 5,47%

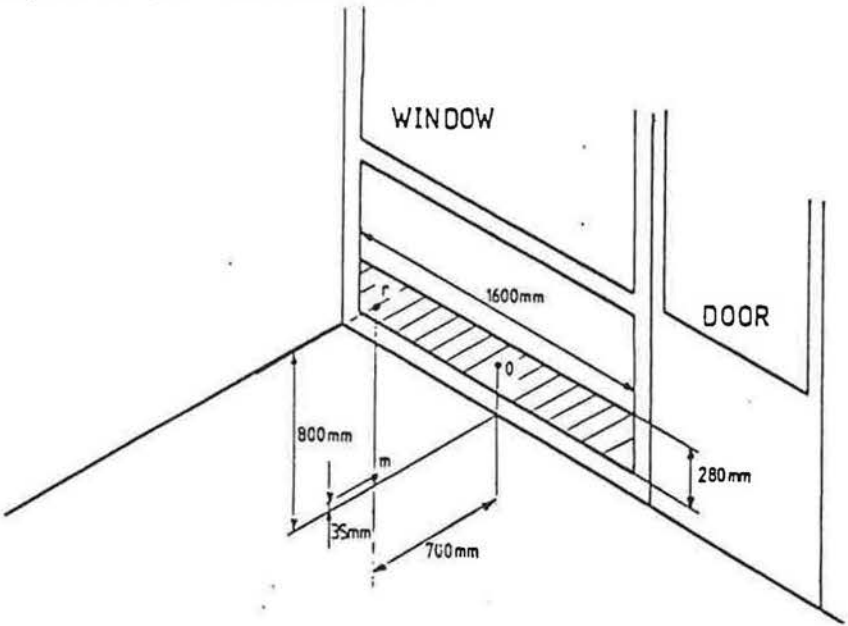


u_m	t_r	t_m	t_0	ΔT_m	ΔT_0	$t_{0.1}$	$u_{0.1}$	$\Delta T_0/\Delta T_m$	u_m/u_0		
0,26	21,3	17,5	15,6	3,8	5,7	18,8	0,15	1,5	7,4		

TEST NR.2

Air flow rate: $160 m^3/h$

Degree of perforation: 5,47%

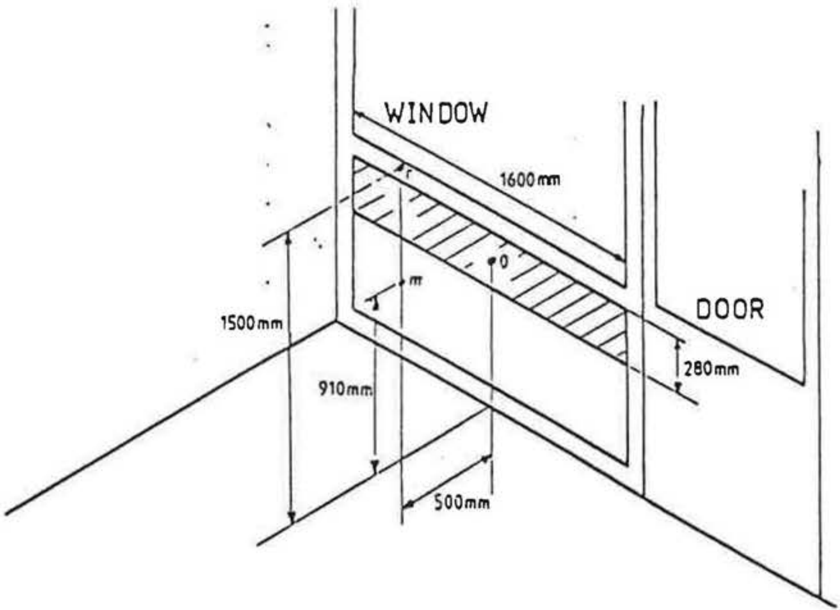


u_m	t_r	t_m	t_0	ΔT_m	ΔT_0	$t_{0.1}$	$u_{0.1}$	$\Delta T_0/\Delta T_m$	u_m/u_0		
0,45	21,6	18,1	15,4	3,5	6,2	18,7	0,1-0,2	1,77	4,54		
0,40	23,9	22,5	21,6	1,4	2,3	22,7	0,2	1,64	4,03		

TEST NR.3

Air flow rate: 160 m^3/h

Degree of perforation: 5,47%

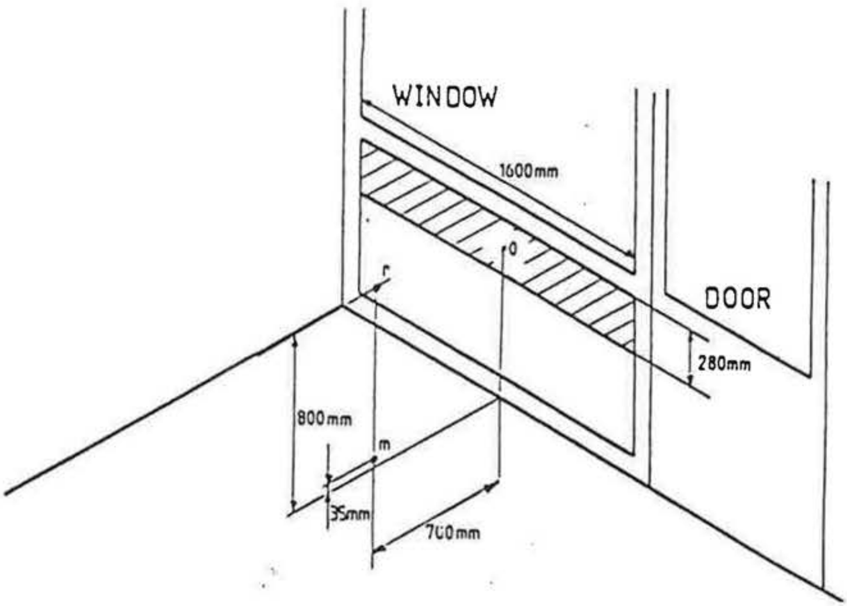


u_m	t_r	t_m	t_0	ΔT_m	ΔT_0	$t_{0.1}$	$u_{0.1}$	$\Delta T_0/\Delta T_m$	u_m/u_0		
0,48	23,7	22,6	20,9	1,1	2,8			2,55	4,83		
0,40	23,5	20,8	16,5	2,7	7,0			2,59	4,03		

TEST NR.4

Air flow rate: 160m^3/h

Degree of perforation: 100%, filter clothing

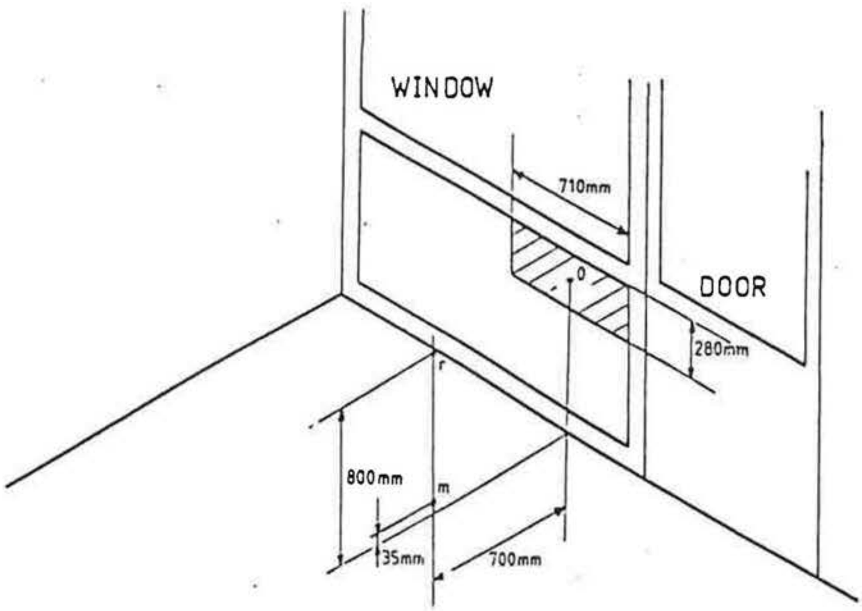


u_m	t_r	t_m	t_0	ΔT_m	ΔT_0	$t_{0.1}$	$u_{0.1}$	$\Delta T_0/\Delta T_m$	u_m/u_0		
0,26	23,3	19,7	15,6	2,6	7,7	20,4	0,10	2,96	2,62		
0,26	20,7	17,8	15,0	2,9	5,7	17,8	0,15	1,97	2,62		

TEST NR.5

Air flow rate: $160 m^3/h$

Degree of perforation: 100%, filter clothing

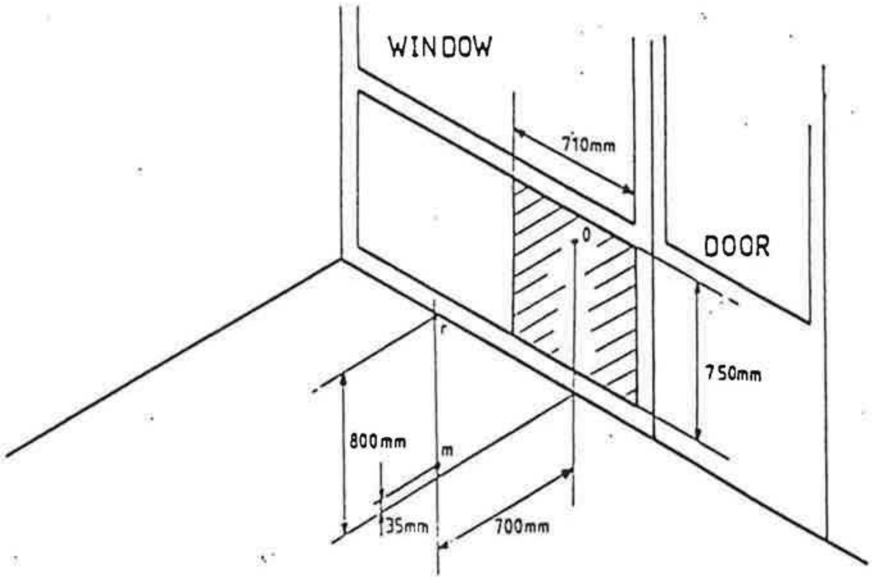


u_m	t_r	t_m	t_0	ΔT_m	ΔT_0	$t_{0.1}$	$u_{0.1}$	$\Delta T_0/\Delta T_m$	u_m/u_0		
0,40	21,5	17,5	15,4	4,0	6,1	18,4	0,20	1,53	1,78		

TEST NR.6

Air flow rate: $160 m^3/h$

Degree of perforation:100%, filter clothing



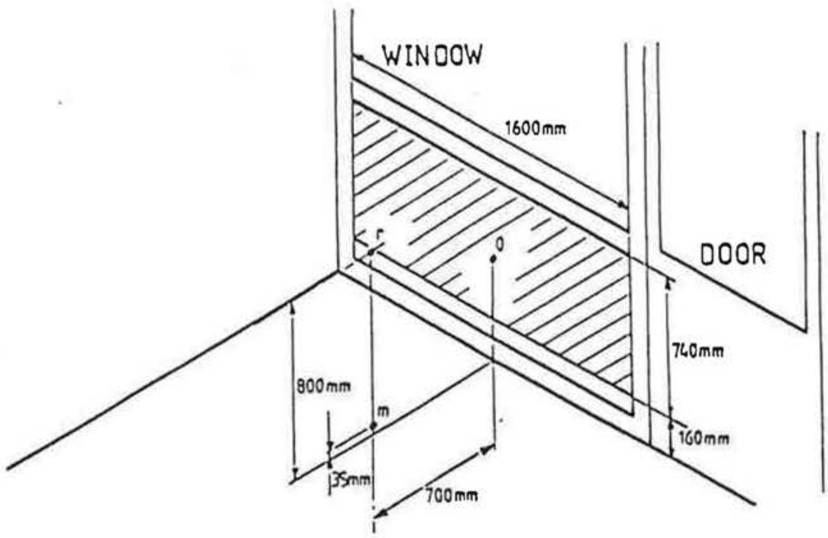
(6)

u_m	t_r	t_m	t_0	ΔT_m	ΔT_0	$t_{0.1}$	$u_{0.1}$	$\Delta T_0/\Delta T_m$	u_m/u_0		
0,30	21,2	17,6	15,4	3,8	5,8	17,8		1,53	3,60		

TEST NR.7

Air flow rate: $160m^3/h$

Degree of perforation: 100%, filter clothing

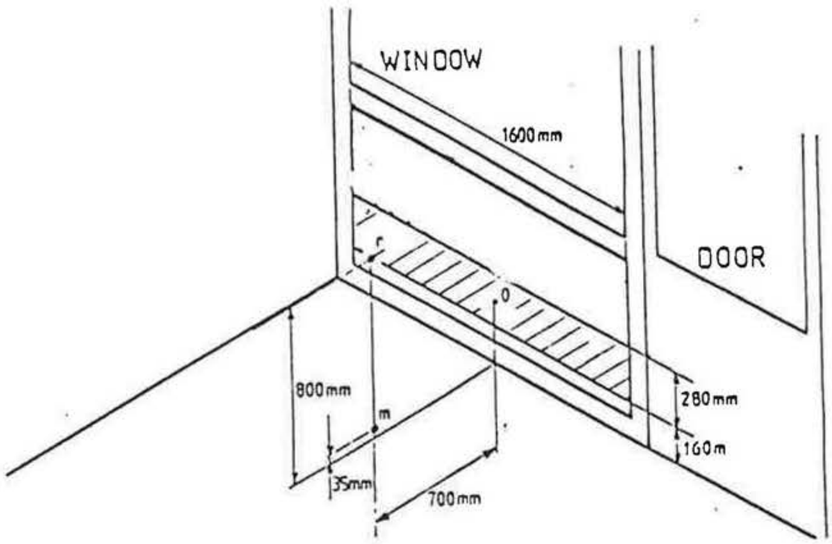


u_m	t_r	t_m	t_0	ΔT_m	ΔT_0	$t_{0.1}$	$u_{0.1}$	$\Delta T_0/\Delta T_m$	u_m/u_0		
0,15	18,5	16,0	14,7	2,5	3,8	16,3	0,1	1,52	4,00		

TEST NR.8

Air flow rate: $160m^3/h$

Degree of perforation: 100%, filter clothing

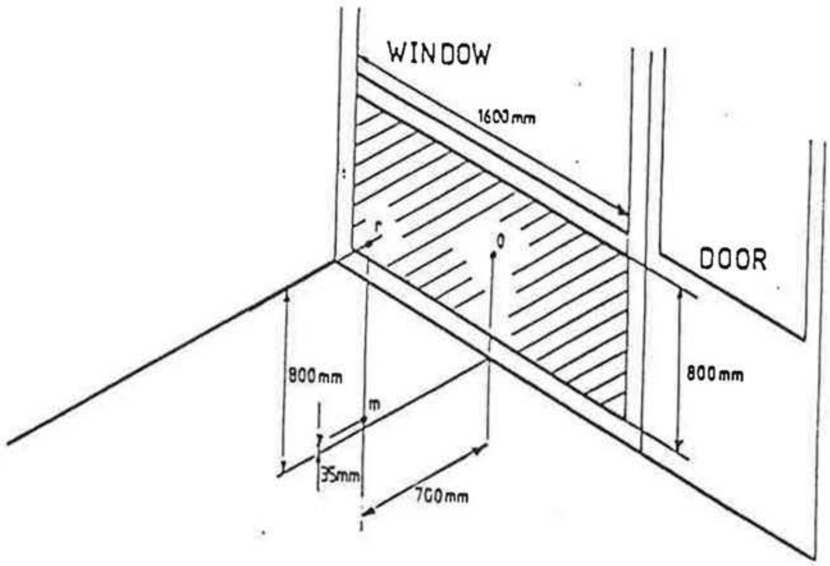


u_m	t_r	t_m	t_0	ΔT_m	ΔT_0	$t_{0.1}$	$u_{0.1}$	$\Delta T_0/\Delta T_m$	u_m/u_0		
0,17	19,1	15,8	14,9	3,3	4,2	16,2	0,12	1,27	1,71		

TEST NR.9

Air flow rate: 160 m^3/h

Degree of perforation: 28%



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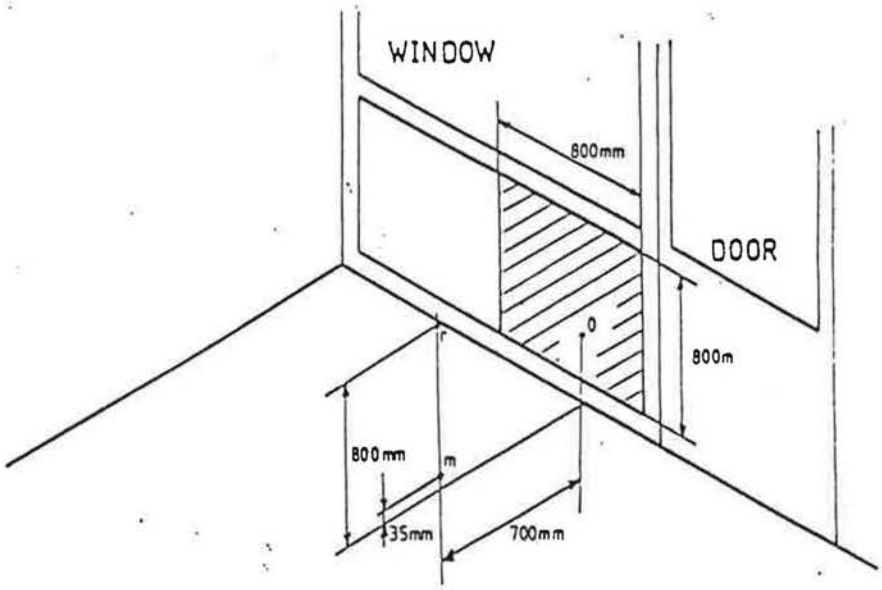
u_m	t_r	t_m	t_0	ΔT_m	ΔT_0	$t_{0.1}$	$u_{0.1}$	$\Delta T_0/\Delta T_m$	u_m/u_0		
0,22	20,0	16,5	15,7	3,5	4,3	17,0	0,1-0,15	1,23	6,34		
	19,8	16,9	15,6			17,5					

TEST NR.11

Air flow rate: 160m^3/h

Degree of perforation: 28%, chess pattern with 10 holes in each direction

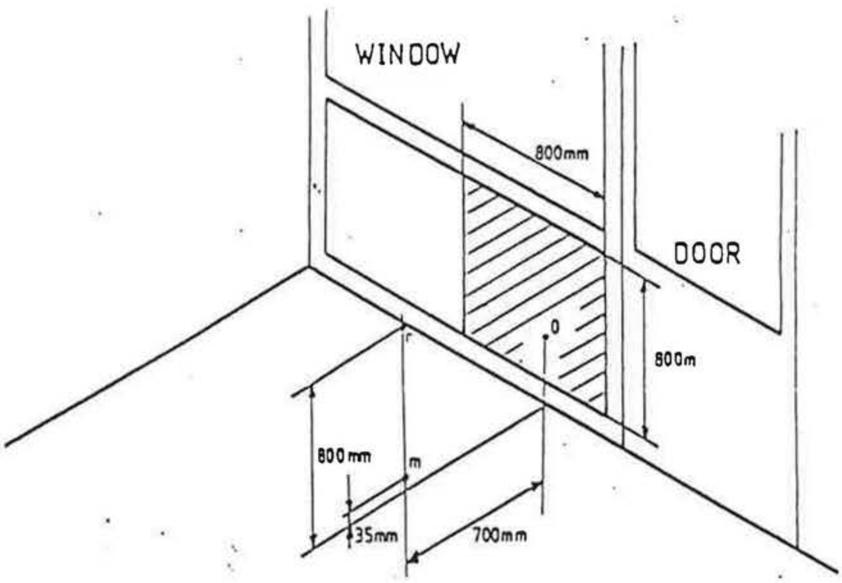


u_m	t_r	t_m	t_0	ΔT_m	ΔT_0	$t_{0.1}$	$u_{0.1}$	$\Delta T_0/\Delta T_m$	u_m/u_0		
0,22	20,5	17,8	15,7	2,7	4,8	18,1	0,22	1,78			

TEST NR.12

Air flow rate: $80m^3/h$

Degree of perforation: 28%, chess pattern with 10 holes in each direction

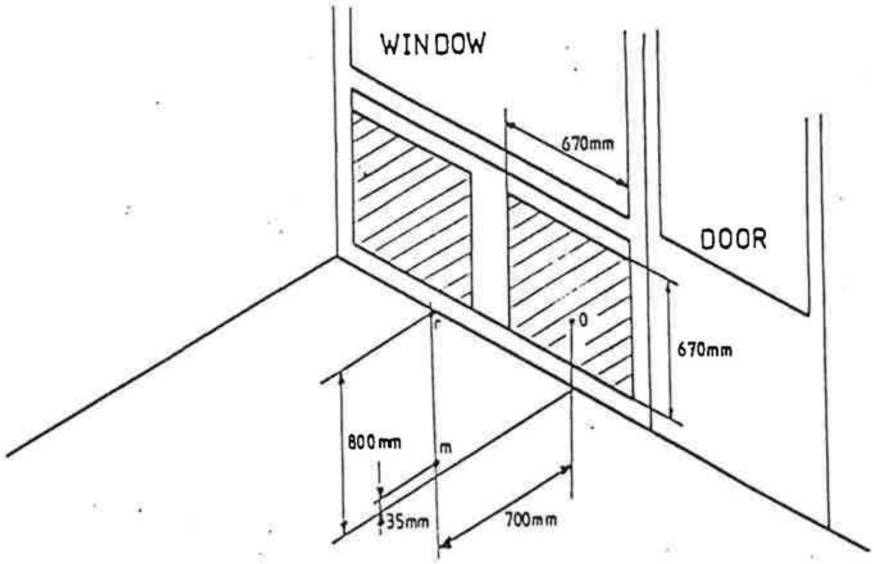


u_m	t_r	t_m	t_0	ΔT_m	ΔT_0	$t_{0.1}$	$u_{0.1}$	$\Delta T_0/\Delta T_m$	u_m/u_0		
0,20	20,9	18,3	16,0	2,60	4,9	18,7	0,1	1,88			

TEST NR. 13

Air flow rate: 160 m^3/h

Degree of perforation: 8,2-8,6%, hole diameter d_0=4,1-4,2 mm



u_m	t_r	t_m	t_0	ΔT_m	ΔT_0	$t_{0.1}$	$u_{0.1}$	$\Delta T_0/\Delta T_m$	u_m/u_0		
0,22	20,8	17,5	16,0	3,3	4,8	18,1	0,14	0,145	4,44		

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