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ROOM FLOW TESTS IN A REDUCED SCALE

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ABSTRACT

This study concerns the problems and prediction of room flow in airconditioning. By means a relatively simple example it is shown how difficult it is to form mathematical models, particular of the three-dimensional flow field occurring in practice.

After basic definitions, an explanation of the influence of different air flow syste on the structure of room flow is given. Measurements and observations found the microstru of the flow to be extremely complex so that a precise mathematical model formation seems be impossible. Measurements on isothermal and anisothermal models - reduced from the orig supply reliable dimensional results. Similarity-theoretical studies indicate that model t are principally a useful means of increasing the dimensional certainty, but in order to e ly reproduce the three-dimensional room flow, the test in the scale 1 : 1 remains essent

The author refers to the considerable lack of knowledge concerning the influence of microstructure of a room flow and the resulting influences for the comfort feeling.

1. INTRODUCTION

In the physics and here especially at dynamical and thermodynamical problems model tests a similarity-theoretical basis have been proved. Model tests are always required if no ce tain mathematical-physical prediction is possible for the system 'Original' and if the di mensional uncertainty is not given. Tests on reduced models and on simulators are execute onlyif they are more economical, i.e. cheaper than model tests in the scale 1 : 1. Fig. shows the relation of the model test in unit dimensions.

With respect to the motion intensities of the indoor air occurring in an airconditic or ventilated room, the thermodynamical system of the room flow seems to be extremely com plicated. The dimensional uncertainty should be shown on a simple anisothermal example of the trajectory calculation of a round free jet:

Various authors give the following empirical equation for the trajectory of a horizo tally blown-out round free jet, which is colder than the ambient temperature:

Regenscheit /2/ : $$\frac{y}{d} = 0,33 \cdot m \cdot Ar \cdot \left(\frac{x}{d}\right)^3 \qquad (1)$$

Koestel /3/ : $$\frac{y}{d} = Ar \left(\frac{a}{b} + 1,6 \cdot k\right) \left(\frac{x}{d}\right)^3 \qquad (2)$$

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Baturin /4/ : $\frac{y}{d} = Ar \left(\frac{x}{d}\right)^2 \cdot \left(0{,}72 \cdot \frac{x}{d} + 0{,}31\right)$ (3)

Müllejans /5/ : $y = \left(0{,}006 \cdot \frac{J\vartheta}{u_o^2}\right) \cdot x^2 \left(\frac{tg}{r} \cdot x + 1{,}5\right)$ (4)

where d = jet diameter
$= 2\,r$

y = vertical coordinate
m = mixed number

$= \frac{d}{x} \cdot 0.2$

x_o = core length of the jet

Ar = Archimedes characteristic

$= \frac{g \cdot d \cdot \Delta\vartheta}{u_o^2 \cdot T_u}$

g = gravitational acceleration

$\Delta\vartheta$ = ambient temperature-blown in temperature

u_o = outlet velocity of the air at the outlet

T_u = ambient temperature

x = horizontal jet coordinate

$\frac{a}{b} = \frac{(1 + 1/Pr)}{4}$

Pr = Prandtl characteristic

$= \frac{v}{a}$

v = cinematic viscosity

a = thermal conductivity

$k = \frac{x_o}{d} = 6.5$

γ = propagation of the jet
(tg = o.24)

The following data are given (for industrial hall air conditioning):

diameter of jet	$d = 0.05$ m
exit velocity of the supply air	$n_o = 10$ m/s
ambient temperature	$T_u = 295$ K
temperature difference	$\Delta\vartheta = 10$ K

Question: The vertical path coordinate y after a horizontal jet path of x = 10 m has to be determined. See Fig. 2

With the given data the following y-components can be calculated from 1 - 4.

Equation	y in m	average $\overline{Y}$ in m	deviation in % of $\overline{Y}$
(1)	17.56		11.66
(2)	2o2		192
(3)	191	1o5	181
(4)	8.6		8.19

The results of the above calculation explain all. The dimensional uncertainty is extremely great, even as for the example, the most simple form of jet - i.e., the round free jet - had been chosen. There are many reasons which are analysed in detail here.

In spite of the complexity of the indoor flows, a series of studies have been made which indicate the complicated methods of calculations. (procedure according to path 5 in Fig. 1). However, the limits are made to relatively simple two-dimensional forms of jet in rooms having a simple geometry. The starting point of all conclusions are the Navier-Stokes' differential equations, which resume the forces - as cause for the flow motion. Besides the continuity equation in recent studies the differential equation system will be enlarged by a turbulence model - turbulent energy-transportation equation and kinetic energy dissipation. 6, 7, 8.

A further condition for the validity of the mathematical models is to be seen in the fact that the total room is part of a direct flow caused by the supply air momentum. The result of the calculation is the curve of equal velocity as shown in Fig. 3.

For the dimensioning of air flow systems the mathematical models have not succeeded until now, as

1. the results are frequently different from the reality in the original,

2. the chosen simple geometries cannot be found actually, and

3. the methods of calculation can only be executed by giant computers.

Therefore, the room flow model test is still today the usual means of gaining a dimensional certainty for a draft-free airconditioning of a room. Therefore the path 2 or 3 in Fig. 1 has to be used. So the problems are not getting simpler, because complex diffe ential equation systems supply complicated dimensionless number groups and so the question of transmissibility of model test results is interesting. The following should show if and under what conditions model tests can be executed - and this is presently the only way to make some more precise statements on the original.

2. PHENOMENOLOGICAL DESCRIPTION OF A ROOM FLOW

The motion intensities of the indoor air in the occupied zone have a decisive importance f the comfort of the human being. So the ratios of the forces are also in the flow, which cause the motion in defined ranges. The sense and interpretation of results of model tests as well as of similarity-theoretical and of analogous basis, depend on the room situation and the chosen air flow system. Therefore in the following the air flow system should be defined:

2.1 Definition of Air Flow Systems

Principally, according to Ref. 10, a distinction is made between the following air flo systems:

1. tangential air flow system
2. diffuse air flow system
3. displacement air flow system
4. micro-climate air flow system
5. jet air flow system.

In Fig. 4 the principal mode of operation of systems 1 - 4 is shown. The definitions are the following:

To 1.: Tangential Air Flow System, Fig. 4a

"The supply air which enters in any form of jet coming from one or more air outlets of the room, flows in the course of its flow trajectory tangently to one or several room enclosing surfaces."

A distinction is made between three typical systems:

1. air flow by a high-pressure inductive unit, mounted at the window. Compare Fig.4a/1.
2. from the linear ceiling-slot-outlets the supply air is blown into the room in the form of even wall jets. Compare Fig. 4a/2.
3. from disk valves, anemostats etc. the supply air flows out even-to-ceiling in form of a radial wall jet.

To 2.: Diffuse air flow system, Fig. 4b

"Above an occupied zone to be determined, air outlets are installed in a room, through which the supply air enters the room in high-inductive jet forms and at an appropriate momentum so that the air jets have reduced their velocity when reaching the occupied zone (by means of room air admixing) so that no drafts occur. In the occupied zone, a very uniform room air motion fluctuating according to magnitude and direction may be stated. It can be called "diffuse"."

To 3.: Displacement air flow system, Fig. 4c

"Displacement air flow systems are understood to be vertical or horizontal displacement flows convection free and poor in turbulences, the forces of inertia of the flow are considerably higher than those of thermal forces or other disturbances."

To 4.: Micro-climate air flow system, Fig. 4d

"Primary air from a pressure plenum generally is supplied to the hollowly formed chair or table legs. By means of an inductive system in the table distribution duct or at the back of the chair, the room air (secondary air) is mixed with the primary air. The induction ratio as $\dot{V}$/primary to $\dot{V}$/secondary is approximately 2 : 1. The supply air generally is discharged at a flow angle to the vertical axis of 0-20 deg. The blowing direction has to be chosen so that the head of the person seated should not be in the direct range of the jet, but in its inductive field. The jet momentum has to be chosen so that the jet direction in the occupied zone is stable at all load situations. So the

1. air flow in the occupied zone is stable and the great whirls of air circulation occurring in such rooms do not influence the occupied zone,
2. prepared supply air will be directed to the breathing zone of the human being (micro-climate zone), and
3. sufficient convection in the field of the seated person can be obtained by means of the secondary air mixing."

To 5.: Jet air flow system

This system is used exclusively for hall situations (ventilating and airconditioning of large rooms, such as theatres, cinemas, concert halls, etc.). It can be defined as follows:

"Jet air flow is said to be a system whose supply air is blown into the room twisted or not-limited from nozzles or similar air outlets, so that high volume rates per outlet are discharged, the volume rate per m^2 ground area is much higher than 30 m^3/h supply air and the total hall area is used for room air mixing."

Now the phenomenology of the total room flow can be explained and the particulars of the various air flow systems can be treated in detail.

2.2 The topology of the indoor air motion

Independently of the definition of various air flow systems there are common characteristics of a room flow. Besides the measurement technique of the motion intensities, the visual study of a room flow has extreme importance. That is why the room flow has to be made visible by appropriate means. Owing to intensive observations with measurement techniques, the following main characteristic of a room flow can be defined:

1. There is no two-dimensional or nearly-two-dimensional room flow. Each room flow is extremely three-dimensional.

Reason:

1. A room as shown in Fig. 5 is airconditioned by a high-pressure inductive unit. If a light section is made through the plane I, it seems that the flow is two-dimensional. Only by a light section vertical to it (e.g. through plane II) it can be stated that the flow is three-dimensional.

2. Room flows are not stable and are characterized by vortex formations. These swirls have the following characteristics:

   - different sizes (macro- and micro-swirls)
   - different forms (circuit, ellipse etc.)
   - different durabilities of life
   - in most cases: no fixed site.

3. Stable and instable room flow zones are mostly unsteady.

   Already the observations 1 - 3 show why a mathematical treatment of such flow systems is very difficult. But other facts have been realized:

4. Curves of equal velocities cannot be measured unless you are in the direct surrounding of the air outlet and if it is measured in the primary range of the jet.

Only ranges of equal velocities can be considered, as shown in Fig. 6.12.

In addition, the following physical characteristic of room flow are known: 11

5. At a measurement site the room air velocities vary continuously depending on intensity and direction. The limit frequency of this macro turbulence will be at 1 c/s.

6. The measure of dispersion can only be sensibly interpreted in the statistics. It is a normal distribution so that by means of the probability net a useful form of illustration of the motion intensity can be found."

If the structure of a room flow is analysed more precisely further specific characteristics can be realized:

7. Air flow systems have specific characteristics concerning the measure of dispersion (frequency/amplitude) of the room air motion.

By means of Fig. 6 and 7 these facts should be explained:

In a testing room, room flow measurements have been made using at the same room geometry three air outlet variants. See Fig. 6. Comparison tests have been made with the same size of momentum regarding the volume flow rate and the outlet velocity of the supply air. The temperature differences between room air and supply air varied between 0 K and 8 K. In a certain height above the floor and always at the same site of measurement, and by means of a high-sensitive anemometer, the room air motion intensity had been registered. If the motion intensities at various cooling loads for the variants 1 - 3 are compared, the following can be stated:

8. If the cooling load of a room and the temperature difference between room air and supply air have increased, the arithmetical mean of the motion intensity has hardly changed. However, the amplitude and frequency will increase with higher cooling load.

9. The amplitude and frequency are characteristically different between the tested variants 1 - diffuse air flow system - and variant 2 - tangential air flow system. The diffuse system indicates a greater amplitude and also a greater frequency.

10. The tested mixed system, variant 2, is hardly different with respect to amplitude and frequency between isothermal situation and maximum cooling load.

Very probably these different motion intensities also influence differently the comfort feeling. It cannot yet be stated which is better or worse. On the other hand, the points 1 - 10 and in particular the points 8 - 10 explain how difficult a mathematical treatment of a room flow will be, and it can be stated:

1. A model test is also problematical owing to the mentioned phenomena.
2. Tests in a reduced scale have to be interpreted extremely critically.

Reason: The aim of a room flow model test will be to make statements and predictions concerning the comfort situation in a ventilated or airconditioned room.

This complex, complicated phenomenology of a room flow can be understood much better, if a similarity-theoretical test is executed. This is explained in the following:

3. Physical analysis of the room flow

As can be realised, the mathematical treatment of the room flow via the possibility 3/5 according to Fig. 1 is not very promising. One should proceed by means of the possibility 2 - dimension analysis - dimensionless number group - model test. The detailed procedure of a dimension analysis is described in Ref. 1 and 12. Here only the result of a possible dimensionless number group should be mentioned. Accordingly, a room flow can be described as follows:

$$\underbrace{Ne,\ Ar,\ Re,\ Re^{*},\ \frac{u_o}{n\ l}}_{\text{velocity field}},\ \underbrace{Pe,\ Pe^{*},\ \bar{Nu}}_{\text{temperature field}},\ \underbrace{(\Pi_n \ldots .)}_{\text{geometry parameter for both fields}} = 0 \qquad (5)$$

Symbols:

$$Ne = \frac{F_N}{\varrho \cdot v^2 \cdot l^2} = \frac{\text{normal force}}{\text{inertia force}}$$

= Newton characteristic

$$Ar = \frac{g \cdot \beta \cdot \Delta\vartheta \cdot l}{v^2} = \frac{\text{therm. buoyancy force}}{\text{Inertia force}}$$

= Archimedes-number

$$Re = \frac{v \cdot l}{\nu} = \frac{\text{inertia force}}{\text{viscosity force}} = \text{Reynold number}$$

$$Re^* = \frac{v \cdot l}{A_\tau / \varrho} = \frac{\text{inertia force}}{\text{turbulent viscosity}} = \text{turbulent influence}$$

$$\frac{u_o}{n \cdot l_o} = \frac{\text{outlet momentum}}{\text{momentum in the room}} = \text{momentum changing number}$$

$$Pe = \frac{v \cdot l}{a} = \frac{\text{convection}}{\text{heat conductivity}} = \text{Peclet number}$$

$$Pe^* = \frac{v \cdot l}{A_q / \varrho} = \frac{\text{convection}}{\text{turbul. thermal conductivity}} = \text{turbulent influence}$$

$$Nu = \frac{\alpha \cdot l}{\lambda} = \frac{\text{thermal conductivity in boundary layer}}{\text{thermal conductivity in non-moving fluid}} = \text{Nusselt characteristic}$$

Also at simple room geometries the number of geometry parameters is considerable. Owing to Equation 5, weight consequences for the execution of model tests have to be respected:

If several dimensionless numbers are important and at the same time characterize a physical procedure, model tests can be executed only

a) by application of the partial similarity

b) by variation of the material values, 13.

In the procedure according to a) - partial similarity - the dimensionless number is chosen from the dimensionless number group known to have an important influence on the physical process. So, for instance, Eq. 5 could be simplified for processes being exclusively or mainly under the influence of thermal buoyancy and inertia force, as per

$$(Ar, \Pi_n) = 0 \qquad (6)$$

It is known that this simplification is allowed for vertical displacement flows (laminar flow system from bottom to top and top to bottom). From Eq. 6 clear model conditions may be derived and so tests can be executed in the same fluid and also in the model under variation of the material values and/or of the corresponding basis numbers.

Furthermore, Eq. 5 indicates that for isothermal flows the equation can be considerably simplified:

$$(Ne, Re, Re^*, \frac{u_o}{n \cdot 1}, \Pi_n \ldots) = 0 \qquad (7)$$

Here again the partial similarity offers

$$(Re, Re^*, \frac{u_o}{n \cdot 1}, \Pi_n) = 0 \qquad (8)$$

clear model conditions, so that model tests in a reduced scale and also by means of other fluids are possible, 13.

For all other room flows, i.e., for most of the problems, no characteristic can be neglected in the indicated dimensionless number group (5), because according to the experiences to date, all dimensionless numbers have the same importance, 12. Therefore it has to be stated:

that room flow tests - except the two mentioned cases - in a reduced model cannot lead to any transmissible result regarding the magnitude, temperature and motion intensities.

Therefore the 'improper models' in the scale 1 : 1 are absolutely required.

In ordinary rooms up to a height of 3.00 to 4.00 m, it causes no problems and it is even cheaper to execute model tests in the scale 1 : 1. For larger areas (industrial halls, theaters, concert halls etc.) it is in general not possible. However, in the following it is reported that nevertheless model tests in reduced scale could be reasonable under the mentioned conditions, and what other possibilities exist for analogous procedures.

4. Model technique and room flow tests

Before some reasonable testing methods at reduced models can be studied, some indications should be given to the test in the scale 1 : 1. Today such tests are executed too expensively, e.g. by completely furnishing the testing room, decorating with curtains and flowers etc. For the experienced testing engineer - and without experience it is even more difficult at the scale 1 : 1 - a test in an empty room will be sufficient to be able to produce results. However, it is better to say that owing to these tests he may give the instructions to the project engineers for the realization of the air flow system. Here generally it is only important that the room flow tests are made rather early - the best is the preplanning time to avoid expensive replanning.

4.1 Isothermal model test at reduced models

Isothermal tests concerning room flows or partial problems of the room flow (e.g. free jet behaviour) are executed. Then it is required that in case of Re = Konst. at the same fluid the model velocity has to be chosen too high, as

$$u_M = u_H \cdot \frac{l_H}{l_M} \qquad \text{for Re = const.} \tag{9}$$

U = flow velocity

H = original

M = model.

Therefore the procedure of the variation of the material values is better used shifting from air in the original to water as model fluid. For the determination of model velocity the following is given:

$$\text{Re} = \text{Konst. } U_M = u_H \cdot \frac{l_H}{l_M} \qquad \frac{v_M}{v_H} \tag{10}$$

with

$$\left.\begin{array}{lllll} \text{for } H_2O & v_M & = & 1.004 & \cdot\ 10^{-6} \\ \text{and for air} & v_H & = & 15.11 & \cdot\ 10^{-6} \end{array}\right\} \frac{m^2}{s} \begin{array}{l} \text{at } 20^\circ\text{ C} \\ \text{" } 20^\circ\text{ C} \end{array}$$

So a large range for choosing the length scale and the model velocity results:

$$u_M = 0.0665 \cdot u_H \cdot \frac{l_H}{l_M} \tag{11}$$

Preferably water models serve as visual studies and as demonstration models and are also easier to produce. Knowledge and experiences can be gained via Ref. 14.

- penetration depth of free jets
- penetration depth of swirls
- turbulent mixing possibilities
- Coanda effects
- influence of constructive subjects on the formation of a flow.

The results and knowledge can be transferred in an isothermal flow case to the original with air as fluid. Such tests are reasonable for basic studies as well as in the teaching field, but practically the application is restricted.

If mainly two-dimensional orientated potential flows are considered, this is partly the case at the depression orientated flows (exhaust air hood flow, laminar flow, natural hall ventilation and the like), so the utilization of the analogy between electric circuit and the potential flow should be taken into account. It is known that for both potential fields, analogue connections are opposed as in Table 1. Two examples should be mentioned to clarify the application of this procedure.

Table 1 Comparison of Analogous Numbers and Equations for a Potential Flow and Electric Circuit

	mass circuit	electric circuit
analogous numbers	pressure difference ΔP mass circuit $\dot{m}$ flow resistance W	$\hat{=}$ voltage U $\hat{=}$ amperage J $\hat{=}$ resistance R
Proportionalities	$\Delta p \sim \dot{m}$ $\Delta p = W \cdot \dot{m}$	$U \sim J$ $U = R \cdot J$
equations	$\frac{\partial \dot{m}}{\partial V} = \frac{\partial^2 \emptyset_S}{\partial x^2} + \frac{\partial^2 \emptyset_S}{\partial y^2} + \frac{\partial^2 \emptyset_S}{\partial z^2}$	$\frac{\partial J}{\partial V} = \frac{\partial^2 \emptyset_e}{\partial x^2} + \frac{\partial^2 \emptyset_e}{\partial y^2} + \frac{\partial^2 \emptyset_e}{\partial z^2}$

V = volume element
$\emptyset$ = potential
s = circuit field
e = electric field

Example 1, Fig. 9 shows in a principal section a manufacturing hall where Kalite electrolysis is conducted. The hall has a width about 41.00 m, lateral 13.00 m and in the middle a height of 19.00 m. In a height of 5.00 m above hall ground area the electrolyte tanks are located. Up to now the hall was naturally ventilated. Therefore at the sides of the hall, above the tank, ventilating slots can be found. Via these slots the air enters and flows out via the openings in the roof ridge. As at these electrolyte procedures mercury vapors occur, the natural ventilation of the hall will not be sufficient. It should be ventilated now as it is illustrated in Fig. 9.

For constructural reasons the exhaust air can only be discharged asymmetrically to one side of the hall. It must be learned how air penetration is possible below the electrolyte tanks, i.e. the dangerous field.

Solution of the problem: The hall section will be imitated similar to the geometry in a certain scale by conductive paper. Fig. 10 shows both circuits for the pressure lines and flow line investigation.

The circuit required for measuring the velocities corresponds to that at the measurement of the pressure lines. Fig. 11 shows the flow lines referred to in Fig. 9. It can be seen clearly that the right side of the hall is insufficiently ventilated. Therefore several models with additional roof openings and guide plates have been tested, one result of these tests is illustrated in Fig. 12 as an example. It can be seen that in this system a better penetration of the hall is guaranteed, 15.

Further examples and the basis of these procedures are described for instance in Ref.15. By means of these simple and less expensive methods, surprising results can be obtained if the requirements as to the exactitude of the results are not too high. The exactitude, referring to the velocity values at the mentioned example, would be at $\pm$ 30 % referring to the value investigated.

4.2 Anisothermal Model Tests at Reduced Models

For numerous hall situations model tests have been made. In a rather new study, 8, the function of a complicated mathematical model has been described which in the case of real validity model tests would no longer be required. It should be considered in future that owing to the mentioned physical connections, anisothermal model tests at reduced objects cannot surrender transmissible results with respect to motion intensity and temperature at site. Nevertheless, anisothermal tests with a constant Ar-number can be of help, as knowledge of the reduced model concerning stability of the air flow systems concerned can be obtained. Such a procedure is described in detail in Ref. 12 where, owing to a qualitative judgment of various model situations, one can construct a variant without any possibility of adjustment of the air outlets after mounting. In Fig. 13a the section of the hall may be seen.

The principle of the air flow can be seen in Fig. 13b. At a reduced model in the scale 1 : 6.62 under the condition Ar = Konst., qualitative tests on the stability of the room flow have been made. Fig. 13c shows a result and evaluation. If such results are transferred to the original, most of the errors can be avoided. Therefore, also in such cases model tests are needed. The model technique cannot supply complete dimensional certainty. Only air flow systems from bottom to top can also be predimensioned exactly for hall systems.

5. FINAL REMARKS

Room flows present extremely complicated physical systems. The possibility of obtaining by means of mathematical models a sufficient dimensional certainty is very restricted except in some simple cases. The model test is a useful means to increase the dimensional certainty.

Anisothermal model tests give only qualitative indications which, however, are very important for the planning and execution of installations. With regard to the influence of the microstructure of a room flow and the resulting influences on the comfort feeling, many factors are still unknown. This lack of knowledge should be supplied.

Nomenclature

A_q = turbulent change quantity (thermal conductivity)

A_τ = turbulent viscosity

a = temperature conductivity coefficient

d = jet diameter = 2r

F_N = normal force

g = gravitational acceleratior

k = $\frac{x_0}{d}$ = 6.5

H = Original

l = characteristic length

M = model

m = mixing number = $\frac{d}{x_o}$ = 0.2

n = air changing number in the room

T_u = ambient temperature

u = flow velocity

u_o = outlet velocity of the air at the outlet

x = horizontal jet trajectory

x_o = core length of the jet

y = vertical coordinate

α = coefficient of heat transfer

β = cubic expansion coefficient

∂V = volume element

$\Delta\vartheta$ = characteristic temperature difference

λ = thermal conductivity

ν = kinetic viscosity

Π_n = various geometry parameter

ρ = air density

$\emptyset$ = potential

Symbols: e = electric field s = flow field

Dimensionless numbers:

$$A_r = \frac{g \cdot d \cdot \Delta\vartheta}{u_o^2 \cdot T_u} = \frac{g \cdot \beta \cdot \Delta\vartheta \cdot 1}{u_o^2} = \frac{\text{therm. driving force}}{\text{inertia force}} = \text{Archimedes number}$$

$$Ne = \frac{F_N}{\rho \cdot u_o^2 \cdot l^2} = \frac{\text{normal force}}{\text{inertia force}}$$

= Newton characteristic

$$Nu = \frac{\alpha \cdot l}{\lambda} = \frac{\text{thermal conductivity in boundary lay}}{\text{thermal conductivity in non-moving fluid}}$$

= Nusselt characteristic

$$Pe = \frac{u_o \cdot l}{a} = \frac{\text{convection}}{\text{heat conductivity}}$$

= Péclet-number

$$Pe^* = \frac{u_o \cdot l}{A_q / \rho} = \frac{\text{convection}}{\text{turbul. thermal conductivity}}$$

= turbulent influence

$$Pr = \frac{\nu}{a} = \frac{\text{cinematic viscosity}}{\text{thermal conductivity}}$$

= Prandtl characteristic

$$Re = \frac{v \cdot l}{\nu} = \frac{\text{inertia force}}{\text{viscosity force}}$$

= Reynolds number

$$Re^* = \frac{v \cdot l}{A_\tau / \rho} = \frac{\text{inertia force}}{\text{turbul. viscosity}}$$

= turbulent influence

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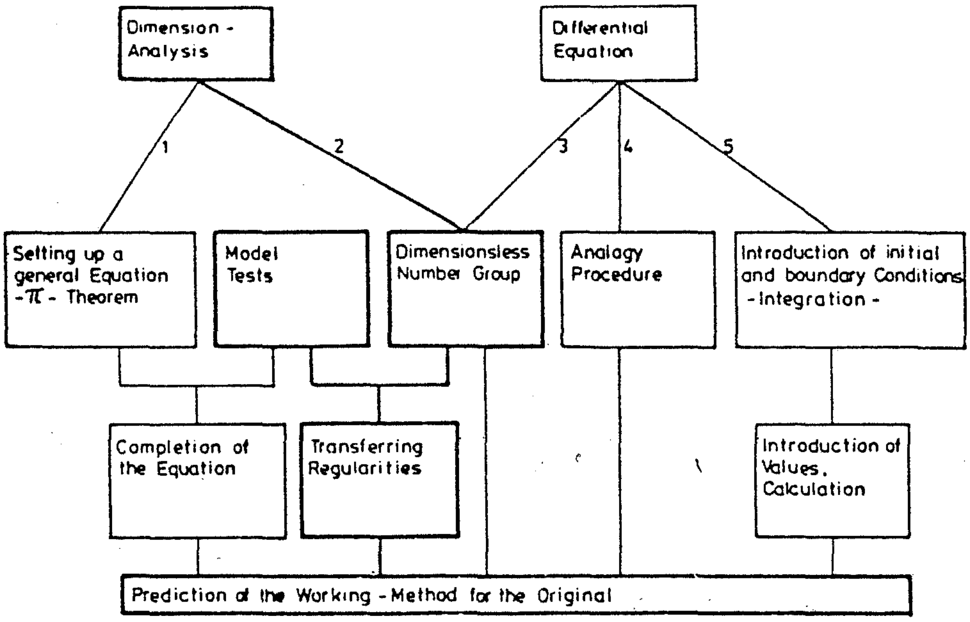


Fig. 1 Explanation of the working method of an original

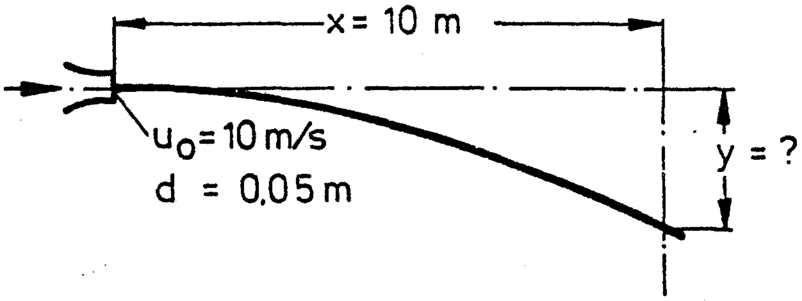


Fig. 2 Explanation of the calculation - example

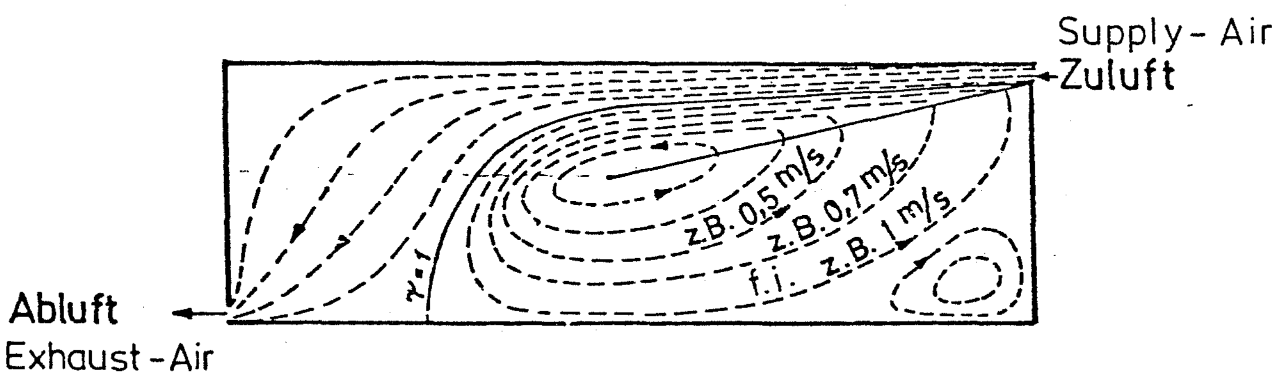
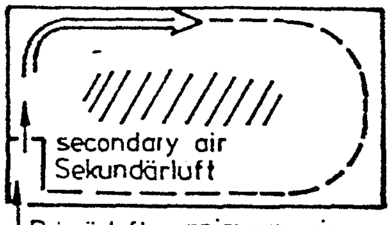


Fig. 3 Flow line diagram of a tangential air flow

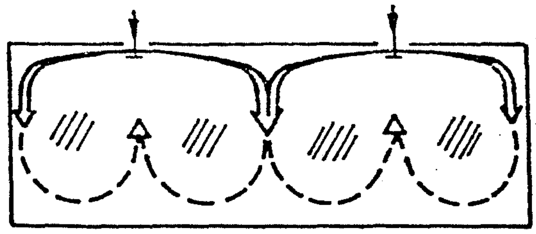


Primärluft primary air

/// Tranquillitas - Zone

Tangentiales LFS bei Hochdruck-Induktionsanlage

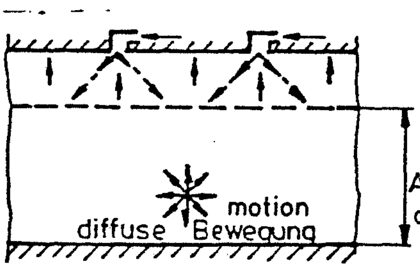
a/1 Tangential Air Flow System at high Pressure Unit



Tangentiales LFS bei linearen Decken-Schlitzauslässen

Tangential Air Flow System at linear Ceiling Slot Air Outlets

a/2

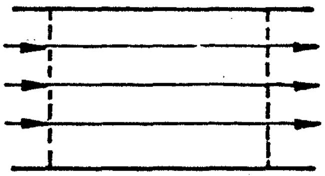


Aufenthaltszone
occupied zone

z. B.: wechselseitig unter 30 - 45° ausblasende Einzelstrahlen (lineare Anordnung)

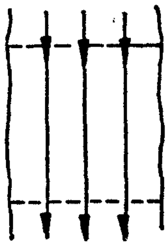
f.i.: alternatingly discharge individual jets at 30-45° (linear arrangement)

b



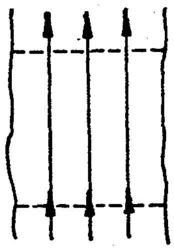
horizontales System

c/1 horizontal system



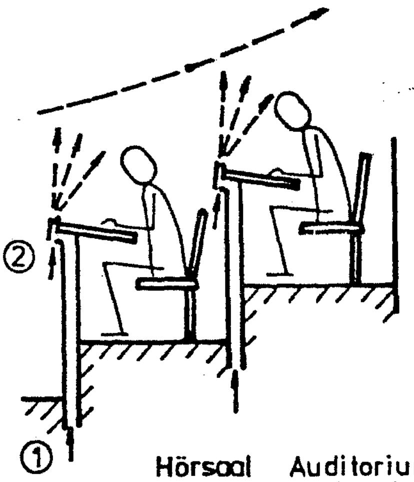
von oben nach unten

c/2 from top to bottom

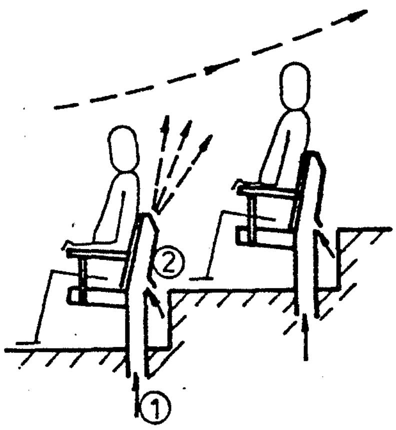


von unten nach oben

c/3 from bottom to top



primary air
① Primärluft
② Induktionsluft aus dem Raum
induction air from the room



Hörsaal Auditorium

Theater / Konzert theatre/concert

Micro-climate-air distribution system for auditoria and theatres (system Krantz)

Mikroklima-Luftführungssystem für Hörsäle und Theater (System Krantz)

d/1

d/2

a = tangential Air Flow System
b = diffuse Air Flow System
c = displacement Air Flow System
d = micro-climate Air Flow System

Fig. 4 Explanation of air flow systems

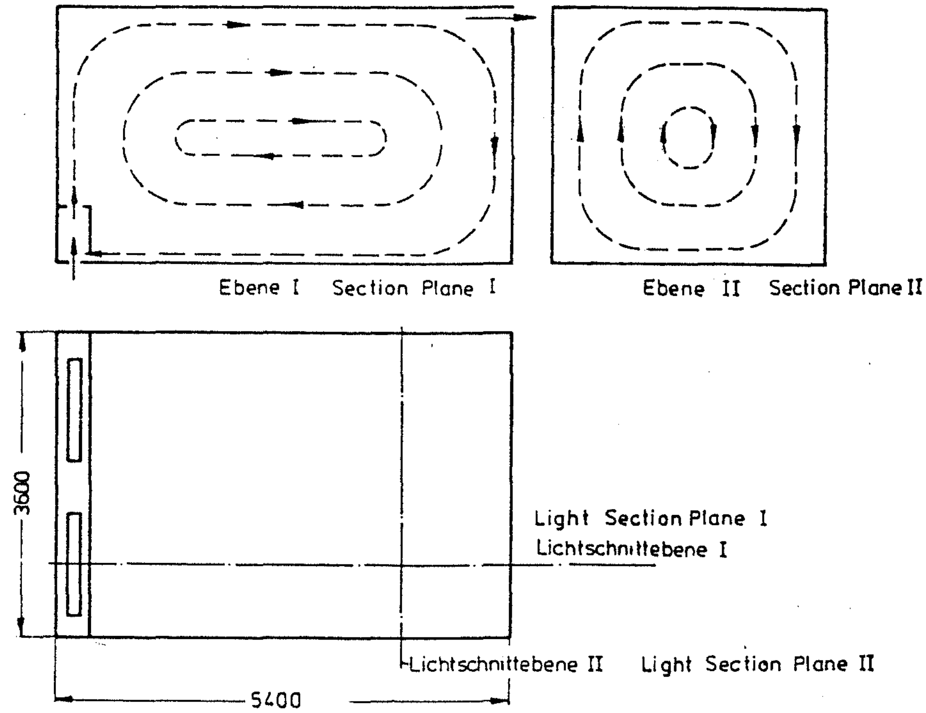


Fig. 5 Explanation of the 3-dimensional character of a room flow

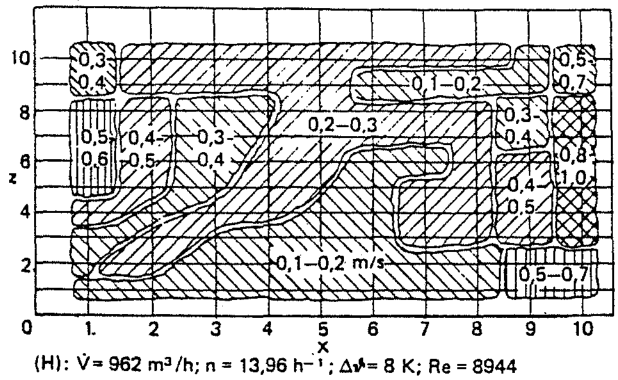


Fig. 6 Topography of an indoor air motion, e.g. tangential air flow system (Re concerned to outlet Flow)

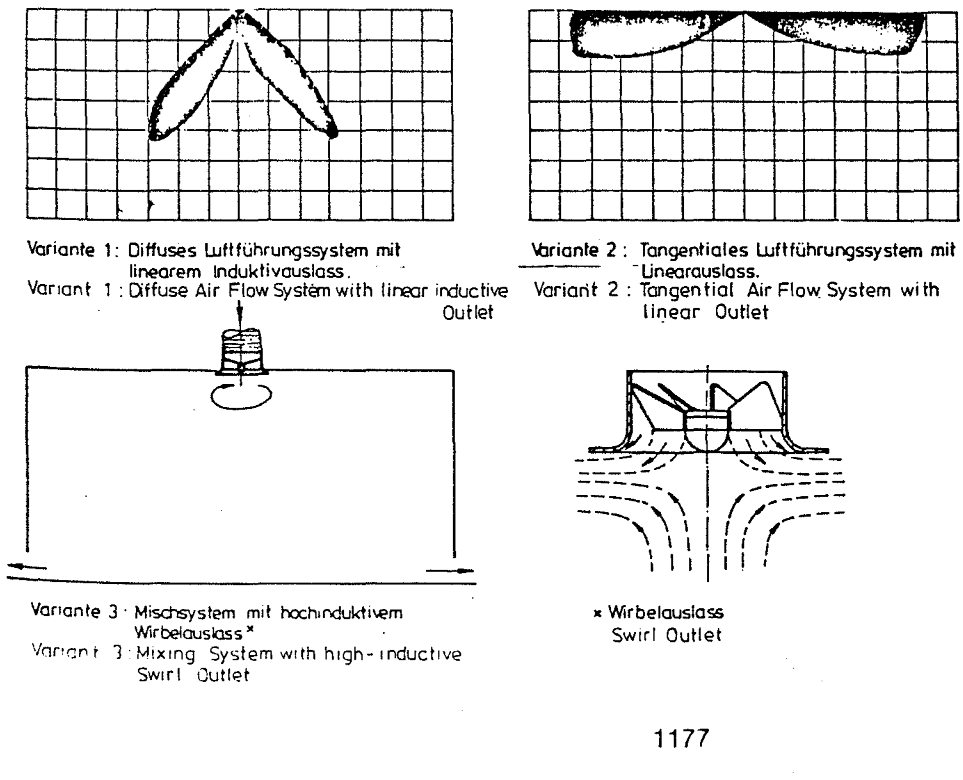


Fig. 7 Explanation of testing variants

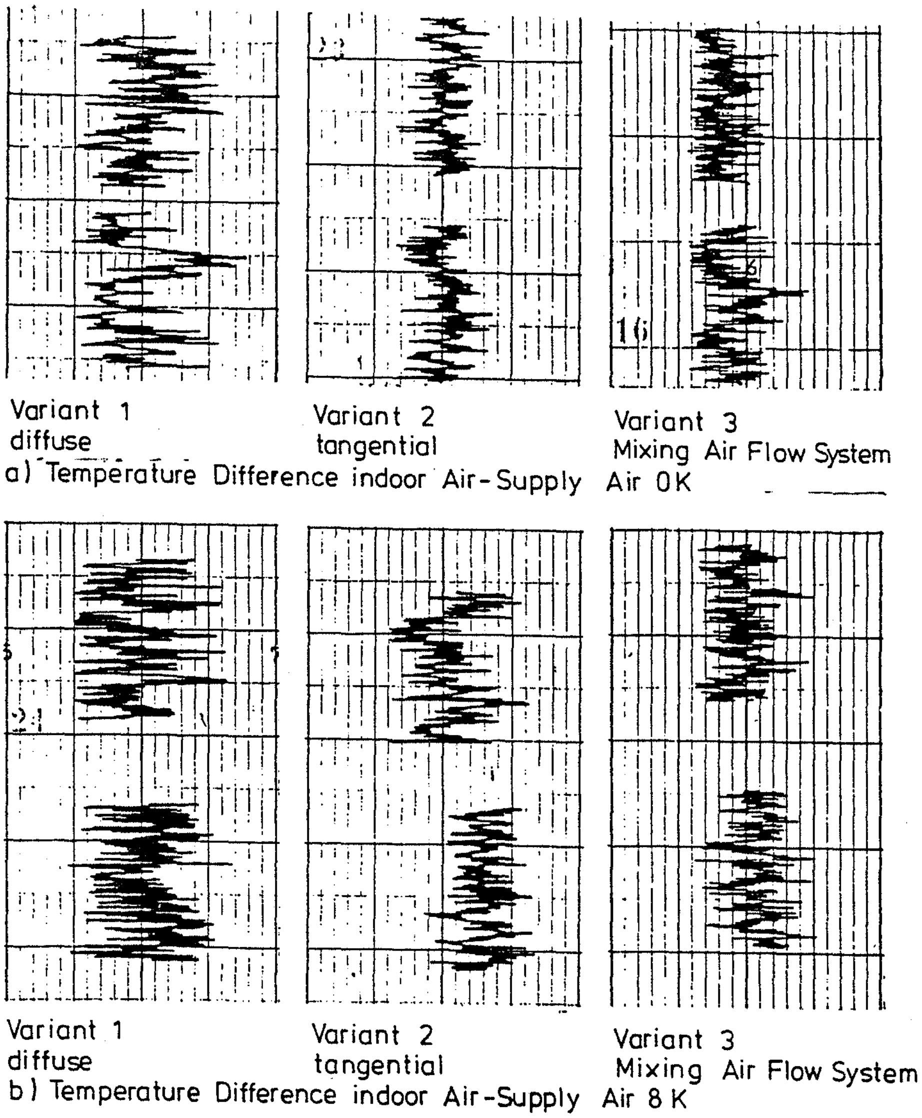


Fig. 8 Listed motion intensities of two representive measuring sites in the occupied zone

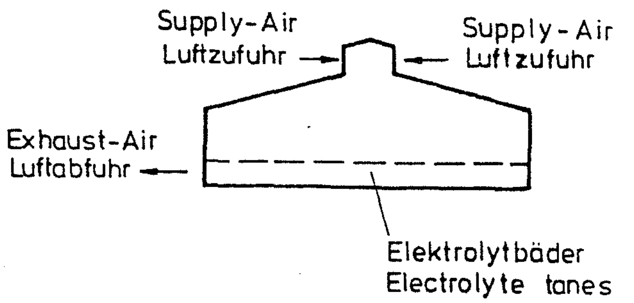


Fig. 9 Schematic section of the hall

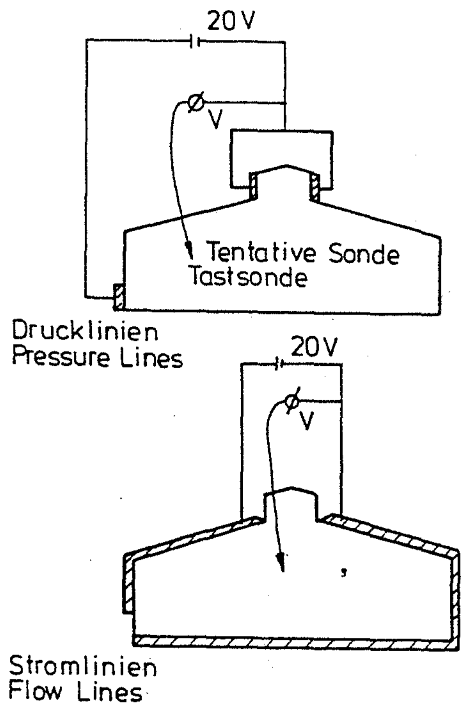


Fig. 10 Circuit diagram for pressure and flow line investigation

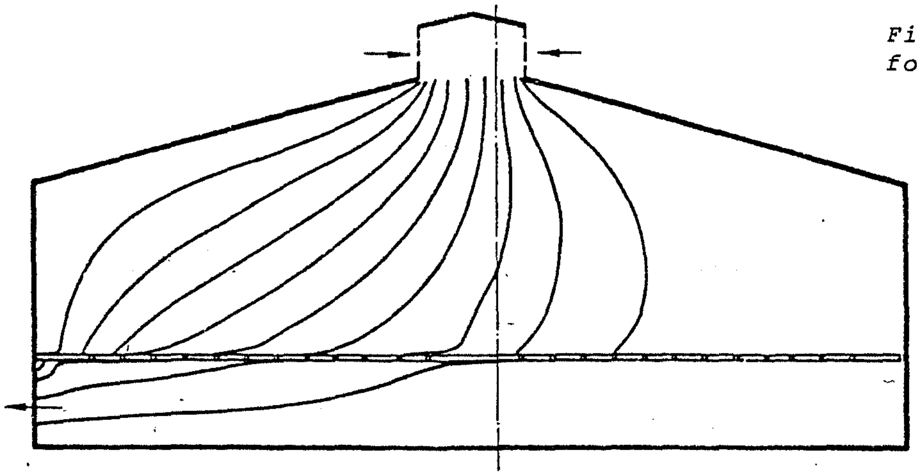
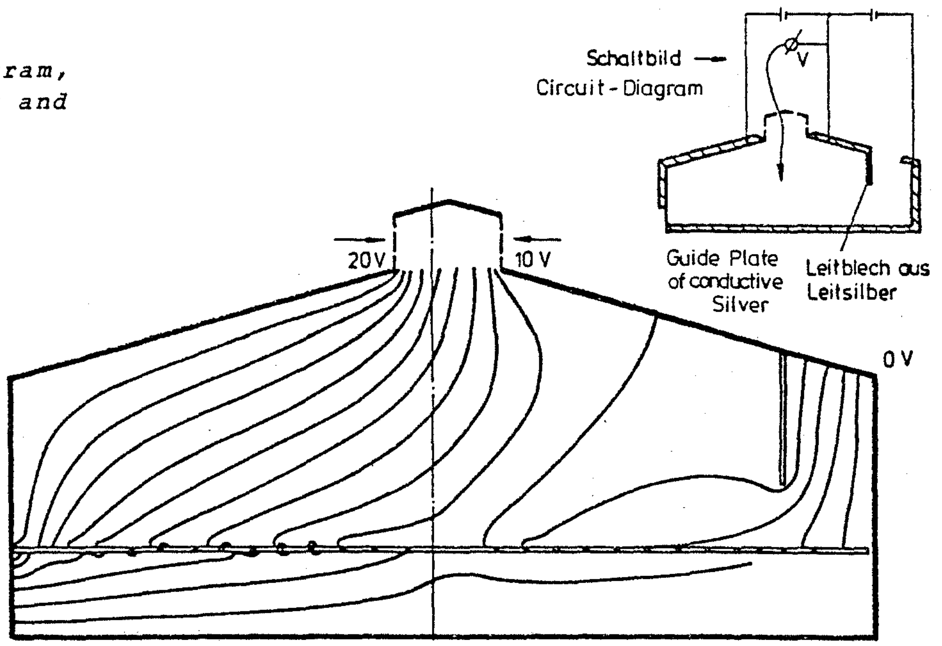


Fig. 11 Flow line diagram for situation in Fig. 9

Fig. 12 Flow line diagram, additional roof-opening and guide plate



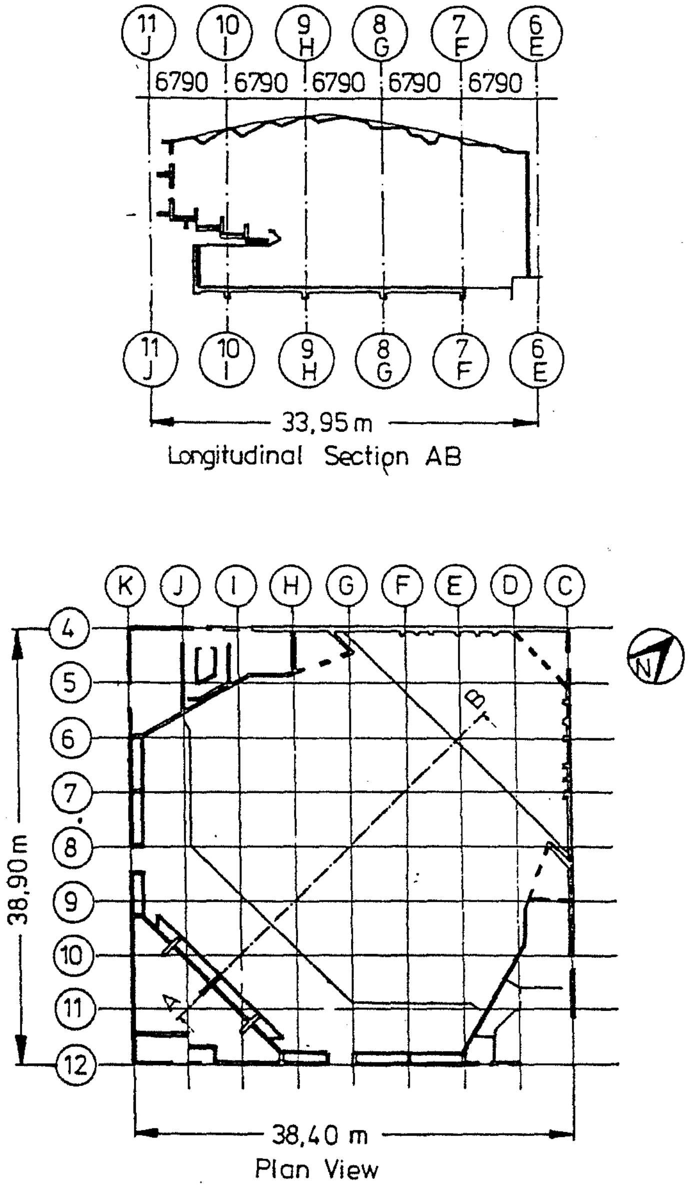
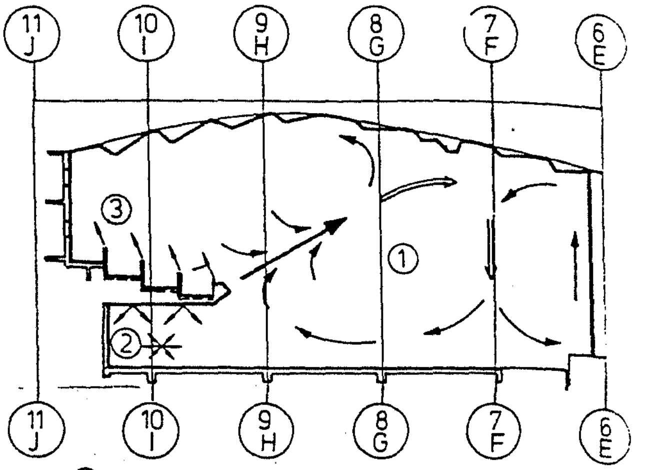
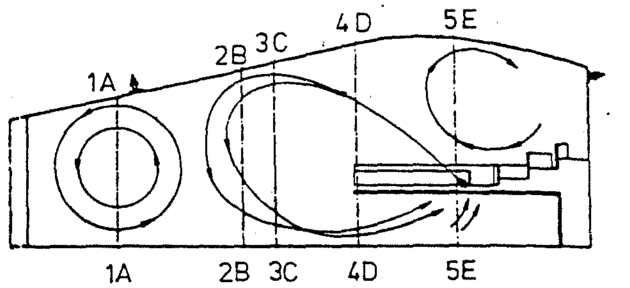


Fig. 13a City Hall, Aachen



Zone ① Jet Air Flow with Formation of Roll
Zone ② Diffuse Air Flow System
Zone ③ Micro- Climate- Air Flow System

Fig. 13b Combined air flow systems for Aachen-City Hall



Remarks to the individual Sections :
1A : stable Roll-Motion
2B, 3C, 4D : Back Flow in Direction to lateral Galleries
5E : Strate upward Motion

Position of the Jets :
Main Gallery : Parallel to Room Axes, blown out Angle 60°
Lateral Gallery : Parallel to Room Axes, blown out Angle 60°

Fig. 13c Flow line diagram of a testing variant

Fig. 13 Model Tests, City Hall Aachen

DISCUSSION

E. RODAHL, Prof., UNIT/NTM, Trondheim, Norway:

COMMENTS

a. The Reynolds number and the Archimedes number are obviously in conflict with each other. I would, however, like to underline that the flow is independent of the Reynolds number over a certain value, such as mentioned by Professor Steimle in an indirect way.

b. A main objective when applying models (reduced scale) is to estimate indoor climate parameters on the design stage. At present, the modelling technique is not developed enough so that it is possible to transfer (for instance) model velocities to prototype velocities with sufficient accuracy. However, the flow picture in the model is of great qualitative value for the designer.

It is also possible to study air flow in confined spaces by water models. Water models are particularly well suited for qualitative studies owing to the use of various colors.

QUESTIONS

The use of electric models seems very interesting.

a. Are the model results verified by other means?

b. Is the characteristic "Thermal Ceiling" in industrial halls as good in electric models as in water or air models?

Editor's Note: The author's response was not received in time for publication.