TEKNISKA MEDDELANDEN

63

INST. för UPPVÄRMNINGS-
och VENTILATIONSTEKNIK
KTH, STOCKHOLM.

1975:2 (vol. 3)

TEKNISKA MEDDELANDEN

rekvireras från sekr. A-M. Hoel, Inst. för Uppvärmnings- och ventilationsteknik, Kungl. Tekniska Högskolan, 100 44 Stockholm el. per tel. 08/23 65 20 ankn. 1680. Lösa häften kr 15:-. Pren. 50:- per volym omfattande 4 häften. Likvid insättes på postgirokonto nr 15653-9 under adress Tekniska Högskolan, 100 44 Stockholm. På talongen skall skrivas "Gottskrives kontonr 711-2, 523-1".

Förord

Återigen innehåller TEKNISKA MEDDELANDEN en monografi. Det handlar denna gång om ventilation av bostäder - ett område som blivit aktuellt inte minst till följd av energikrisen. Som bekant kan endast vid s.k. FT-system värmeåtervinning appliceras på ett enkelt sätt, och man kan förvänta sig att en strävan mot ett minskat energibehov för ventilation kan komma att inriktas mot en ökad användning av sådana system även inom bostadssektorn.

För att en systematisk installation av FT-system skall bli framgångsrik med hänsyn till energibesparingen krävs emellertid att den ofrivilliga ventilationen nedbringas till ett minimum. För att möjliggöra ett tillfredsställande gott underlag för en bedömning av värmebehovet för en sådan ofrivillig ventilation krävs bl.a. kännedom om fönstrens täthet samt om de lagar som gäller för strömningen genom spalter, t.ex. vid fönster. Arbetet i detta nr av TEKNISKA MEDDELANDEN syftar till att ge en sådan kunskap, och i avsnitt 2 ges bl.a. en teoretisk och experimentell redovisning av infiltrationen vid fönster. Vidare görs en jämförelse mellan olika länders normer m.m.

Ventilationen av en byggnad påverkas av många faktorer, bl.a. uteklimatet (t.ex. utetemperaturen och vind). Redovisning av variationer av storleksordningen ± 15 % i ventilationsflöde till följd av variationer i vindstyrka och vindriktning har rapporterats, och likaså kan man visa att flödesvariationer till följd av svängningar i utomhustemperaturen lätt når upp till ± 10 %. Speciellt är detta fallet vid höga byggnader, där både vindens och temperaturens inverkan blir accentuerad. I avsnitt 1 av monografien diskuteras dessa variationer, och dessutom behandlas vissa andra störningar av intresse för ventilationsteknikern.

En stor del av arbetet behandlar mätningar av flöden mellan rummen i en byggnad. Spårgasteknik - se avsnitt 3 - har använts för mätning av flödena i ett antal höga bostadshus. Redovisning sker inte bara av tekniken vid mätningen utan även av de intressanta resultat som nåtts, och vilka torde vara av intresse för ventilationstekniker. Arbetet avslutas med några synpunkter på fortsatta undersökningar.

En nyhet är att meddelandet denna gång utges helt på engelska - något som föreslagits från vissa av våra prenumeranter. Synpunkter på detta emotses - skall TEKNISKA MEDDELANDEN fortsättningsvis utges - helt eller delvis - på engelska?

INNEHÅLLSFÖRTECKNING

Sammanfattning

Den ökade höjden och tätheten hos bostadsbebyggelsen har förorsakat en ökad efterfrågan på ett behagligt inomhusklimat. Samtidigt föreligger också starka krav på en effektivare energianvändning i bostäder. En följd har blivit önskemål om förbättrad kontroll av ventilationsförloppet i byggnaden.

Då ventilationen i en byggnad störs av olika yttre inflytanden, sker påverkan till stor del via sprickorna i fasaden. Det har därför funnits ett behov av noggrannare kännedom om luftströmningsförloppet genom dylika sprickor. I föreliggande arbete har förloppet studerats för olika tryckdifferenser, och det visas att läckflödet kan beräknas med uttrycket

$$v = a \cdot l(\Delta p)^{1/\beta}$$

om β bestäms ur följande uttryck

$$\beta = 2{,}0 - e^{-5{,}0\, a\Delta p}$$

där Δp = tryckdifferensen över sprickan (Pa)
a = sprickans proportionalitetskonstant ($m^3/m \cdot s \cdot Pa^{1/\beta}$)
l = sprickans längd (m)

Tidigare har β schablonmässigt satts till 1,5. Här visas att β inte överstiger 1,2 för en spricka med stort strömningsmotstånd vid tryckdifferenser upp till 500 Pa. Vid 500 Pa innebär detta att det beräknade flödet blir 182 % större än det som erhålls med β = 1,5.

Mätning av luftflöden mellan rum är ett viktigt medel för analys av ventilationsförloppen i en byggnad. För detta ändamål har en spårgasmetod utvecklats, som förutsätter utvärdering med dator. Ekvationer som beskriver variationen hos spårgaskoncentrationen i ett rum, då gas sprides i alla rum i byggnaden, har uppställts, varigenom flödet mellan de olika rummen kan beräknas ur mätdata. Datorprogrammet innefattar också en statistisk behandling, som eliminerar inverkan av tillfälliga mätfel. Luftflödena kan bestämmas med ett fel mindre än

3 % i datorprogrammet; mätfelen tillkommer dock.

Fem apparater för kontrollerad förångning av koldioxid från torris har framställts. Kontrollen sker medelst styrning av tillförseln av elenergi. Luftflödesmätningar har skett i lägenheter i två 16-våningshus. Koldioxidspridarna har ställts en i varje rum i lägenheten och gas har producerats i de olika rummen omväxlande.

Luftflödesmätningen i fastigheterna skedde både sommar- och vintertid, och inverkan av ändringen i utetemperatur på funktionen hos ventilationen undersöktes. Totala frånluftsflödet, uppmätt med lägenhetens alla fönster stängda, var i ett hus med enbart mekanisk frånluftsventilation vintertid 35 % större på 15:e våningen och 6 % mindre på första våningen jämfört med flödet sommartid. I det andra huset, som hade FT-system, var motsvarande värden 19 % ökning på 14:e våningen och 27 % minskning på första våningen. Infiltrationen från byggnadernas vertikala schakt till lägenheterna tycks starkt påverka dessa värden.

Ventilationsförloppet i byggnaderna studerades också med en nätverksmetod. Vintertid visade det sig vara fullt möjligt att en lägenhet i byggnadens övre del genomströmmas av luft som redan förorenats i byggnadens lägre del. Flödesändringen mellan sommar och vinter kan minskas genom att vintertid effekten hos tilluftsfläkten ökas och effekten hos frånluftsfläkten minskas.

VENTILATION OF DWELLINGS AND ITS DISTURBANCES

Hiroshi Honma

CONTENTS

Ventilation of Dwellings and its Disturbances

Introduction

The concentration of population into cities and rationalisation of natural resources are making it necessary for buildings to be built at greater densities and constructed to greater heights. This gives rise to some unfavourable effects on the ventilation of buildings. Doctors have pointed out that there is an unexpectedly broad and fast spread of pathogenic bacteria in unsuitably ventilated buildings. High rise buildings, in particular, are influenced by natural disturbances such as wind or thermal currents, to a far greater extent than lower buildings. A highly impermeable exterior wall construction and an improved technique of ventilation system design are required in modern buildings. Almost all cities suffer from the pollution of their atmosphere. It is becoming increasingly difficult to use outside air directly for the ventilation of buildings in large cities. Under such circumstances, the factors which in conventional design techniques have been ignored due to their slight influence, must be subjected to re-examination, in parallel with the employment of newly developed equipments or control techniques.

This report deals firstly with factors which have a bearing on the leakage of air through exterior walls. The shape of an air passage through a crack in an exterior wall is too irregular, and its depth relatively small, so that the phenomenon of air leakage through a crack cannot be treated either as a laminar or turbulent flow, or as a combination of the two. This affects the exponent of the pressure difference in the air infiltration equation. As the range of the working pressure difference at exterior walls becomes larger, there is an increasing need for a more precise grasp of the exponential term. Air flow through simply shaped gaps was measured over a wide range of pressure differences. By using the results of measurements and experimental work by many researchers concerning air leakage through window cracks of various kinds, an equation is proposed for the evaluation of the exponent from the character of an air passage and a working pressure differential.

Secondly, the tracer gas technique of air change measurement is developed so as to be capable of application for the measurement of air transfer between the rooms in a dwelling. From the points of view of hygiene and heating engineering, it is not only the total air change but also the manner of air flow which must be known in a building. This technique can also be used for the analysis of air flow in an entire building, by aggregating the results of air flow measurements relating to each part of the building. Owing to the difficulty in regulating tracer gas production and complications in the theory relating to variations in concentration, current ventilation measurement by the tracer gas technique has remained in a comparatively primitive state. The controlled production of carbon dioxide gas is accomplished by regulation of the supply of electric heat to dry ice in a thermally insulated vessel. An equation for the variation in concentration in every room of a building was derived from the differential equation relating to the transportation of a tracer gas. The air transfers are calculated from the resultant variations in concentration in the rooms by using a numerical analysis technique.

The measurement method was employed in two 16-storey blocks of flats in a suburb of Stockholm. The stack effect on the ventilation systems was examined by measuring the air flows at several points in the buildings.

1 Ventilation of Dwellings

1.1 The requirements on ventilation

The following discomforts may be intimately felt in our everyday life. A shortage of fresh air causes psychological stuffiness, while the physical results are an accumulation of sultriness, body odours and cigarette smoke. On the other hand, excessive ventilation causes drafts and gives rise to a restlessness in the atmosphere of a room. The principal factors concerned with regard to the comfort of a room are temperature, humidity and air movement, and they are very closely interrelated. Ventilation forms an inescapable part of the three factors of room comfort. Furthermore, it is also involved in the transmission of odour and airborne bacteria between the rooms of a house.

The scientific study of ventilation began in Germany and England in the middle of the nineteenth century. The reason for this study was the need to maintain room air in a hygienic condition.The index of condition first applied was the concentration of carbon dioxide gas.

The medical requirements to be satisfied by fresh air can be calculated by different methods based on different points of view. The primary requirement on air as a source of oxygen for normal activity by an adult is calculated as follows:

respiration rate	24 times per minute
respiration volume	0.8 litre
concentration of oxygen	
in fresh air	21% by volume
in expired air	16% by volume

Consequently, the need for oxygen is

$$0.8 \frac{l}{\text{time}} \times 24 \frac{\text{times}}{\text{min.}} \times (0.21 - 0.16) \frac{l}{l} = 0.96 \frac{l}{\text{min.}} = 0.016 \frac{l}{\text{sec.}}$$

When the allowable lower limit of oxygen in room air is 19%, the ventilation requirement for continuous use of a room is

$$\frac{0.96\ l/\text{min.}}{(0.21 - 0.19)l/l} = 48\ l/\text{min.} = 0.8\ l/\text{sec.}$$

In the 1930s, ventilation was studied from the point of view of comfort in a room by Yaglou and his associates at the Harvard School of Public Health (1935, 36, 37 and 55). Next to the thermal character of the air, body odour was found to be a primary index of the quality of room air in relatively comfortable conditions. According to their findings, body odour decays much faster in the first few minutes than may be expected on the basis of the physical theory of dilution. This is probably the reason why per capita air space governs the ventilation requirement from the point of view of body odour. At a ventilation rate of 0.024 m^3/sec or less for a normally hygienic adult, the level of body odour cannot be maintained below the olfactory threshold. For an air space with a volume of 13.3 m^3 per person, which is almost the same as the condition in dwellings, the minimum fresh air requirement is 0.003 m^3/sec per person in order that a person entering from relatively clean air may get the impression of allowable odour intensity in the room, while it is 0.002 m^3/sec per person in order that the occupants of the room may maintain an air quality of fair to good. When the per capita air space is only 2.8 m^3, the requirement rises to 0.012 m^3/sec per person in order that the impression upon entering the room may be the same.

The ventilation in a room containing smokers was also studied by Yaglou and his associates. Their conclusions were as follows. The odour from cigarette smoking takes a long time to disperse; in other words, the intensity of the odour rather increases during the first three hours. The principle to be applied to this type of odour is therefore ventilation of a relatively small space at high efficiency. The odour from cigarette smoking is so strong that it masks body odour in normal everyday conditions, and in a room where the occupants include smokers, the ventilation must be based on the number of smokers. In order that the odour may be kept below an acceptable level for the non-smoking occupants, at least 0.012 m^3/sec of ventilation is required for each smoker. In order that the impression on entering the room may be kept below the acceptable level, the requirement rises to 0.017-0.019 m^3/sec.

In calculating summer air conditioning requirements, the emission of heat and humidity from a human body exerts a great

influence on the ventilation load. In a space of a very enclosed character, such as a shelter, heat and humidity emission is the dominant load on a ventilation system. On hot days, as much as 0.028 m^3/sec per person of outside air may be required in order that temperature and humidity in the shelter may be kept at tolerable levels. See Baschiere, 1965.

In addition to the above fundamental requirements there are many sources of air aggravation which must be eliminated or diluted by means of ventilation. The dominant sources of aggravation in dwellings are dust emission due to activity of the occupants, odour and steam given off by cooking, combustion gases and pathogenic bacteria. The ventilation regulations in building codes or technical recommendations laid down by authorities in different countries are based on the above factors, but the climate and the living habits of the citizens of the country also form the basis of decision. For instance, the Swedish Building Code SBN 67 lays down in the ventilation regulations for dwellings that :

the rate of air flow q_a in a flat situated in a block with a mechanical ventilation system shall not be lower than the values given by the expression

$$q_a = 0.000612 - 0.00000111\ F$$

where q_a is the normal value of rate of air flow in $m^3/sec/m^2$ of floor space, and F is the total floor area of a flat in m^2. In addition, the rates of air change mechanically extracted from kitchens, bathrooms etc shall not be less than

kitchens and kitchenettes	0.0222 (0.0111) m^3/sec
bathrooms, shower rooms and WCs	0.0167 (0.0083) m^3/sec

The values in brackets can be used according to local codes. The extraction rate in kitchens where cooking is in progress must be increased by effective handling to 0.055 - 0.083 m^3/sec.

The rate of air change in a room and the per capita ventilation rate are other ways of expressing the ventilation requirement in dwellings. The ventilation of a detached house is very often regulated only by the minimum size of opening in an exterior wall.

1.2 The effect of ventilation on the heating load

Owing to the development of industrialised buildings and building components, it is becoming increasingly evident that many aspects of heating load estimation must be improved. There is a tendency for heat gains due to solar radiation and internal heat production in a building to be calculated accurately and to be separately subtracted from the estimated heating load. See Raiss (1966) and Horie (1969).

The changes in heating load calculation specifications in the German Industrial Standard DIN 4701, from the beginning of this century to the present time, are very instructive for all engaged in heating engineering. See Kollmar (1949), Krischer (1959) and Esdorn (1972). Even at present, estimation of a heating load includes many uncertain factors. These uncertainties are usually compensated for by paying increased attention to certain other factors. The transmission heat loss in exterior walls was usually increased by the addition of a certain percentage, for walls of different orientation, to the loss calculated on the basis of an overall heat transfer coefficient and temperature difference. In the DIN of 1909, the addition for a northerly wall was +20% and that for a southerly wall was 0%. In the DIN of 1959 these rates were changed to +5% and -5% respectively. In the beginning, a ventilation load was calculated as a term subsidiary to the transmission heat loss, and it was only in 1959 that the ventilation heat loss was to be calculated separately. As a consequence, the share of total heat loss due to the transmission heat loss has been decreasing and that due to the ventilation heat loss, increasing.

Let us examine the behaviour of ventilation heat losses in the heating loads of dwellings, which are estimated according to the Swedish Building Code SBN. For a detached house, the ventilation load constitutes about 20% of the total heating load. As regards flats, the ventilation load is the same for the same type of flat as long as the required rate of ventilation is maintained, but the transmission heat loss varies according to the position of the flat. For an end flat on a top floor, the share of the ventilation load is between 30 and 40 per cent,

while that for a flat enclosed on all four sides by other flats is between 50 and 60 per cent. The load for a whole block approaches the latter figure as the scale of the block increases. In the above example the allowable minimum rate of ventilation is employed in the calculation. In practice, in order to allow for various disturbances which will arise in service and to give the system a certain overload capacity, a ventilation system is run at a much higher rate.

The transmission heat loss follows the change in outdoor temperature with a certain time lag, but the ventilation heat loss changes as a result of changes in temperature difference and in ventilation rate. Jackman (1974) introduced the concept of the "wind-temp number" into the calculation of the infiltration heat loss. The wind-temp number is expressed as a function of outdoor temperature, wind direction and wind velocity, obtained from a statistical treatment of meteorological data. He clearly pointed out that the maximum heating load does not always coincide with the lowest outdoor temperature.

In air conditioning calculations, full consideration is given to the chronological difference between solar irradiation load and ventilation load. In adapting the outdoor temperatures for purposes of heating load design, the daily fluctuations in outdoor temperature are considered, in view of the thermal mass of the walls, to have a lesser effect on the heating load. However, the same design temperature is applied in estimating ventilation heat losses. The error in employing the same design temperature may have been compensated for when the transmission heat loss formed the major proportion of the heating load. When the ventilation system is run at all times to satisfy the minimum requirement, the ventilation heat loss directly follows the variations in diurnal temperature. In a district where the diurnal change in outdoor temperature is relatively large, a lower temperature is to be applied for the ventilation load calculation than for the transmission heat loss calculation. And the occasional coincidence of the two loads must be considered dynamically in estimating the maximum heating load.

Excessive or undesirable ventilation has been reduced by means such as high-precision window construction and the application of weather strips. But every building must conform to the minimum ventilation requirement as outlined in 1.1. In the next stage in the improvement of ventilation techniques, the over capacity of ventilation systems must be reduced. To achieve this, the effectiveness of ventilation must be improved by more reasonable design of air transportation in a building, better control of a ventilation system, and better distribution of air flow in a room.

1.3 The composition of ventilation

1.3.1 The mechanism of ventilation

When all houses were small, ventilation requirements were satisfied merely by opening a ventilator in an exterior wall. When buildings were constructed with several storeys, ventilation stacks were incorporated in the structure. But even then ventilation could be performed without mechanical aids. The natural forces of wind and heat movement provide the motive power for the ventilation in such buildings.

In a detached or low-rise apartment house, the ventilation requirement can be satisfied by means of natural ventilation, with the exception of partially and temporary mechanical ventilation in kitchens and toilets, owing to their relatively light occupation and relatively large exterior surface. In large blocks of flats, however, centralised forced ventilation systems comprising a fan and a duct system become necessary. A ventilation system is very often combined with an air conditioning or heating installation. The performance of a ventilation system depends on many factors such as climate, the scale of the building, the allowable tolerance of control, etc. The components of a ventilation system can be classified into motive power which causes air flow, and flow resistances in each element of the air transportation system. The unavoidable air passages in a building structure, such as window cracks and staircases, come into interaction with these. When a ventilation system is run by mechanical force, the natural forces which play an important part in the ventilation of small buildings act rather as disturbing factors.

In planning the ventilation of a room, it is not only the quantity of the air change but also its quality which is an important consideration. When the air flow is unevenly distributed in a room, it may not only give rise to complaints by the occupants on the grounds of draughts or stagnation, but may also adversely affect the efficiency and economy of the system. The rate of air flow and the positions and design of the registers for supply and return air have a direct effect on the distribution of air in a room. At the same time, the air flow is also influenced by the furniture in the room, by the convection which may be caused by a temperature difference at wall surfaces, by a human body, a heat source or even solar radiation from a window. The air flow in a room is also disturbed by inward and outward movement of air through a crack around a window or through other air passages.

1.3.2 Natural ventilation of a room

The following two methods are generally used for calculating the natural air change in a room:

A. the crack length method
B. the air change method.

The first method is based on the effective air passage and the difference in working pressure over the passage. These passages are usually cracks in windows, window frames and other joints in a building structure, and capillaries through building materials can also be regarded as air passages. The distribution of air passages in a room is also dependent on the relative position of the room in the building. From the point of view of ventilation, a building must be regarded as an organic composition of its air passages. Moreover, the balance of air flow in a building is very sensitive to changes in wind direction, for instance, and a reversal of the direction of flow through an air passage is not a rare occurrence. This gives ventilation in a room a very complicated character and makes prediction of ventilation very difficult.

The second method is based on data obtained by statistical treatment of empirical results. Bahnfelth and his associates (1957) conducted a number of air change measurements in single-storey brick and veneer structure residential buildings with warm air heating system . The results were treated statistically, and it was found that the relationship between the air change number and the indoor-outdoor temperature difference was 0.012/^{o}C, and that between the air change number and the wind velocity was 0.0053/m/sec. The wind velocity was measured at the building site at a height of 6.9 m. Many other measurements of air change in

detached houses, performed in the USA, also give the same trends regarding temperature difference and wind velocity, but the figures vary over a wide range due to conditions such as the structure of the building and the climate.

The rate of air change in a room for the estimation of the heating load is based on such experimental data. But even the pressure balance of a complicated building is difficult to decide. In estimating the ventilation requirement by the air change method, only the absolute value of the air change in a room is given, and there is no information on where this air comes from. In ventilation calculations there remain many details which must be decided on the basis of the designer's experience.

1.4 Uncertain factors in ventilation

1.4.1 The effect of a temperature difference between the inside and outside of a building

The climate undergoes a seasonal alteration, and the temperature also changes. The difference between the maximum and minimum monthly average outdoor temperature is about 20°C in Stockholm, and a seasonal temperature difference of this or greater magnitude occurs in almost all cities. On the other hand, the room temperature is usually maintained at about 20°C. The difference in density between the inside and outside air, due to this temperature difference, causes a pressure difference over the external shell of a building.

In the field of ventilation engineering, air may be treated as a perfect gas, and the relationship between density and temperature may then be expressed by the following equation:

$$\rho_2 = \frac{273 + t_1}{273 + t_2}\,\rho_1 \qquad 1.1$$

where ρ_1 and ρ_2 are the densities of air at temperatures t_1 and t_2 respectively.

The change in atmospheric pressure with height is expressed by the following equation:

$$P_h = \frac{P_0}{10^{\frac{h}{674\,(273+\,t\,)}}} \qquad 1.2$$

where P_h is the pressure at a height h above a reference elevation, P_0 is the standard pressure at the standard elevation, and t is the average temperature of the column of air between the two elevations. If the difference in height is small, the equation can be simplified to read

$$P_h = P_0 - g\,\rho\,h \qquad 1.3$$

where ρ is the density of air.

Consider a cylinder of 60 m height which stands vertically on the ground. There is no air leakage through the cylinder wall, only through a small hole at the bottom of the wall. If the atmospheric temperature is -10°C and the temperature of the air inside the cylinder is maintained at +20°C, then the wall at the top of the cylinder is pressed outwards by the following pressure difference:

$$\Delta P = P_{i,60} - P_{o,60} = \frac{101\,300}{10^{\frac{60}{674(273+20)}}} - \frac{101\,300}{10^{\frac{60}{674(273-10)}}} = 80.2 \text{ Pa}$$

where $P_{i,60}$ and $P_{o,60}$ are the pressures inside and outside the cylinder respectively at a height of 60 m.

The pressure distribution along the cylinder wall is shown in Figure 1.1a. Furthermore, if the cylinder also has a small hole at the top, there is an upward flow of air through the cylinder. The rate of change in pressure difference with height is reduced when the intensity of air flow increases, as shown by dashed lines in Figure 1.1b. The level at which the pressure difference is zero is called the neutral zone. As will be seen from Figure 1.1c, the level of the neutral zone changes according to the distribution of the flow resistances of the openings in the wall.

A practical building has many points around entrance doors, windows and ventilation openings in the external shell where there is a lack of fit. In addition to these, there are a large number of cracks scattered irregularly all over the external shell. Tamura and his associates (1966) carried out a number of measurements of the pressure distribution over the exterior walls of tall buildings. An example of the vertical distribution of the pressure difference over the exterior wall of a building is reproduced from their report in Figure 1.2a. The object of measurement was a nine-storey building with an air conditioning system. During the measurements, the air conditioning system was shut down, and the dampers at air intakes and outlets were closed. The neutral zone of the pressure difference, over a range of outside temperatures, is situated at a height of 72% of the height of the building, measured from the bottom. Because there were many openings at the top of the building for the air conditioning system, the neutral zone was situated above the mid-height of the building. Several other measurements were made, with the many openings at the top and the registers inside the building being sealed with plastics sheets. The neutral zone descended to 62%, as shown in Figure 1.2b. The actual pressure difference to height ratio was 82% of the theoretical value calculated on the assumption of no air leakage across the external shell.

1.4.2 The effect of wind on ventilation

Both the direction and velocity of wind change at random over a relatively short period, and because of this it is difficult to carry out a field study of wind effect.

The wind at a certain site is affected by the topographical configuration of the country around the site, and the effect of wind is influenced by conditions above the level of the ground around the building. For this reason it is almost impossible to construct a general equation which will relate wind data at a meteorological station to that at a building site situated some distance from the station.

The average wind velocity near the ground surface changes with height, and it is also affected by the roughness of the ground surface. The wind velocity at a certain height is approximated by the following equation

$$u_h = \left(\frac{h}{h_0}\right)^{\alpha} \cdot u_0 \qquad 1.4$$

where u_h is the wind velocity at height h, and u_0 the velocity at a reference height h_0. The value of the exponent α depends on conditions above ground. Davenport (1963) derived the following figures for the power law index α of respective areas from observation data of profiles of average wind speed in sufficiently high ranges by treating various topographical conditions as fluid dynamic roughness:

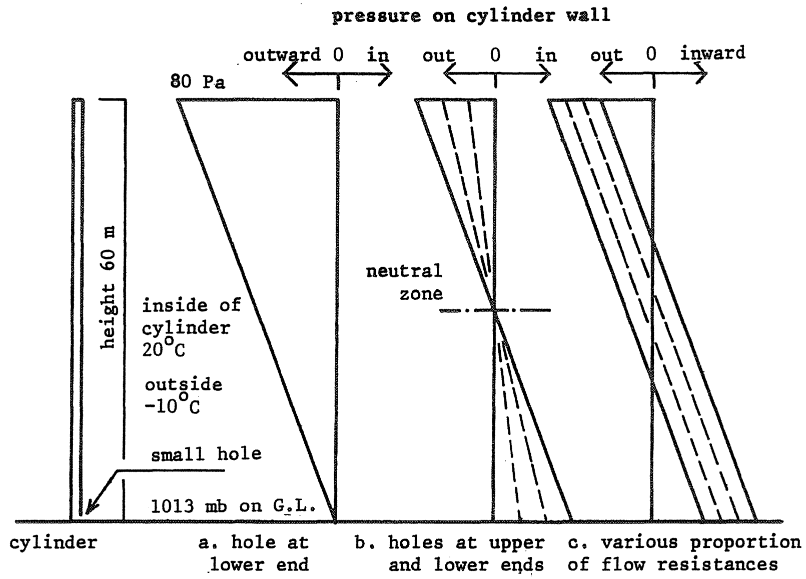
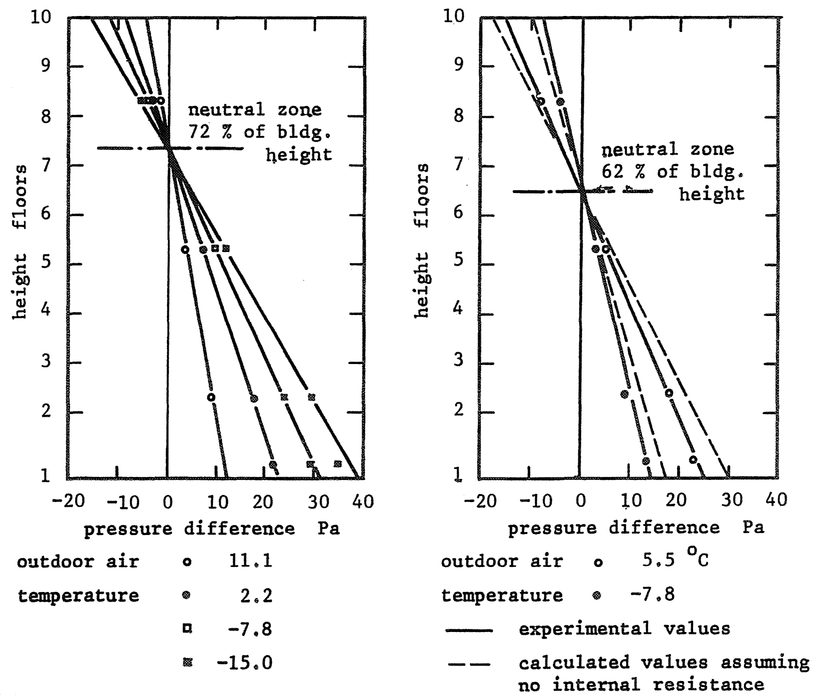


Fig. 1.1 Pressure distribution along cylinder wall caused by temperature difference



a. ventilation system off

b. ventilation system off vertical air shafts sealed

Fig. 1.2 Pressure difference on exterior wall caused by temperature difference (Tamura et al. 1966)

exponent	condition of ground
0.10	open sea
0.16	flat open country
0.28	woodland forest
0.40	urban area

Besides the average speed, the component of gusts, which are caused by turbulences in the air flow, is included in the wind. The proportion of gusts to corresponding average wind speed becomes greater, where the surface roughness of a ground is greater. By Shellard (1963), the proportion, which were met in the range of building height, and which have durations of three seconds or more, was 1.5 in the western coast of England, and was 1.8 in extensively build-up areas.

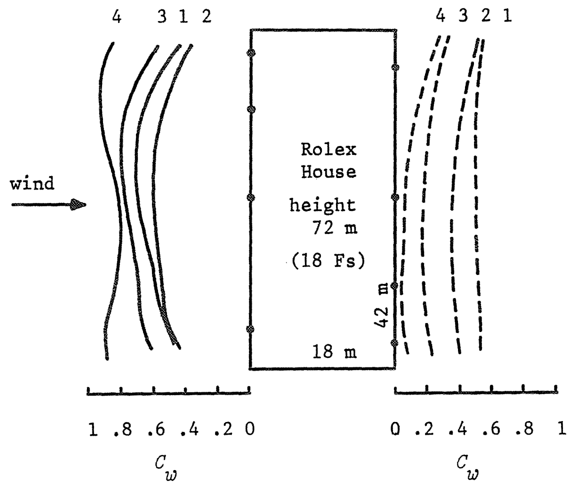
When wind strikes a small flat surface at right angles, the energy contained in the air movement is converted and acts on the surface as a pressure. This is called a velocity pressure. The relationship between wind velocity and velocity pressure is expressed by the following equation :

$$P_v = \frac{1}{2} g \rho u^2 \qquad 1.5$$

where P_v is the velocity pressure at wind velocity v. The wind pressure acting on part of the external wall of a building varies according to the direction of the wind relative to the building, the shape of the building and the relative position of the wall. The ratio of a working pressure P_w at a certain point on a wall to a velocity pressure is known to be stable over a range of wind velocities. This ratio is called the wind pressure coefficient C_w, and its functional meaning is expressed as

$$C_w = \frac{P_w}{P_v} \qquad 1.6$$

Wind conditions and building conditions vary over a considerable range, and for this reason no general expression is as yet available concerning the relationship between coefficient and wall.



wind pressure coefficient at 13th floor

1 : wind tunnel - model in isolation
2 : " " - model in situ (impermeable walls)
3 : " " " " " (permeable walls)
4 : full scale test

Fig. 1.3 Wind pressure distribution on exterior walls
Comparison between full scale and model tests
(Newberry et al., 1968)

The distribution of wind pressure over the external shell of a building is influenced not only by the shape of the building but also by the air flow through the building. Dick (1949) found in his experiments on the ventilation in tenement buildings that the wind pressure coefficient of a windward wall is considerably higher than the commonly recognized values obtained from wind tunnel tests, while, to the contrary, it is unreasonably low in the glancing and leeward walls. He concluded that this phenomenon is due to the difference in the resistances offered to the inflow and outflow of the air through a house. This was supported by the results of research into wind pressure on a rectangular 18-storey building (aspect ratio 2.3), by Newburry and his associates (1968). It is stated that "in the wind tunnel tests, the model made from permeable planes simulates the pressure distribution of the actual building better than the model made from solid planes". The comparison of wind pressure coefficients by Newburry is shown in Figure 1.3.

1.4.3 Other disturbances

The performance of a ventilation system is influenced by the way the occupants use the building, for instance by the opening and closing of entrance doors and windows. In a high-rise building the air flow at the ground floor entrance may cause a strong air flow in a staircase or lift shaft when there is a large difference between indoor and outdoor temperature. Considerable attention has been devoted to the prevention of such an air flow in multi-storey buildings, the measures taken ranging from double doors to pressurization or depressurization of the ground floor to prevent the air flow through the vertical shaft.

As regards the infiltration of air through a crack around a window, seasonal changes in humidity have a considerable effect on air infiltration through wooden windows. Wind pressure also affects infiltration of air through a window crack by displacing the window in the frame.

Tamura and his associates (1973) measured air leakage through the external shell of tall buildings by pressurising them on the inside. By performing the air leakage measurement under different outdoor temperature conditions, the air leakages through the building proper, bottom separation and top separation were obtained separately. Their results showed that the rate of air leakage through the exterior wall varies from 0.0013 to 0.0024 m^3/sec per m^2 of outside wall at a pressure difference of 75 Pa. This air leakage is equivalent to a rate of flow through an orifice with an area of 1.9 - 3.7 cm^2/m^2 of exterior wall.

1.5 Ventilation design by the network method

As mentioned above, the ventilation system of a building consists of mechanical elements such as a fan and a duct system, and air passages in exterior walls and partitions in the building. The latter very often remain outside the control of ventilation design. The performance of a ventilation system is also influenced by disturbances such as wind, thermal effects and the activity of the occupants. Furthermore, owing to the fact that the effect of a pressure difference on air movement is not linear, analysis of the performance of a ventilation system is complicated.

The relationship between the air flow v in a duct and the pressure difference ΔP causing this flow is expressed by

$$\Delta P \simeq \text{const. } v^2 \qquad 1.7$$

The equation generally used for air leakage through a small crack is

$$\Delta P = \text{const. } v^\beta \qquad 1.8$$

The exponent β is dependent on the fluid dynamics character of the passage and the active pressure difference, and has a value between 1 and 2. This term will be discussed in detail in Chapter 2.

The equation expressing the balance between air flows entering and leaving a certain space is a function of the pressure in the space and the adjacent spaces, and of the flow resistances in every air passage.

Generally, it is impossible to give a theoretical solution for this equation because of the non-linearity of each component. In addition, the flow balance in a building containing several rooms is expressed by a series of flow balance equations of the above type for every room. When the building has a ventilation system, the flow balance of the system must also be treated simultaneously.

Rydberg (1959) constructed a large number of equations for the rational design of various kinds of ventilation systems and for the examination of the effects of various disturbances on these, by treating all components comprised in a ventilation system in a highly conceptual manner. Two equations are quoted below from his work.

In a tall building with an extract ventilation system, the resistance of an exhaust register to counteract the change in the ventilation rate due to a window being opened, is calculated as follows:

$$P_R = \frac{P_C - P_D\,(\eta_1)^2 \sum_{i=1}^{n} \left(\frac{1}{i}\right)^2 - 2P_D\,\eta_1 \sum_{i=1}^{n} \left(\frac{1}{i}\right)}{2\eta_1 + (\eta_1)^2} \qquad 1.9$$

where η_1 is the increase in the rate of ventilation in the room in which the window is opened, P_C, P_R and P_D are the flow resistances at the window cracks, exhaust register and one floor height of vertical duct respectively, and n is the number of rooms connected to the vertical duct

Further, the change η_2 in the rate of ventilation in the flat above the one where the window has been opened is expressed by the following equation

$$\eta_2 = \eta_1 \left\{ \frac{P_R + P_D}{P_C + P_R + P_D} \right\}^{\frac{1}{2}} \qquad 1.10$$

Den Ouden (1967) applied an electric analogue method to solve the flow balance in a building. According to Ohm's law, the electric current in a resistance is usually linearly proportional to the active electric potential. Den Ouden combined the resistance of a special lamp and one or several solid resistances to replace the relationship between air flow and pressure difference by the relationship between an electric current and the potential.
The air flow system in a building can be replaced by an electrical network by employing the special combination of resistances, and the performance of the system under various conditions examined by electrical network calculations.

The other method of solving the flow balance problem is to seek the balanced condition by solving the series of flow balance equations in every part of the building by an iterative method using a digital computer. When an imaginary network is composed of all the components involved in the ventilation of a building, and a computer program is compiled for this network, the pressures and the flow rates in every part of the building can be calculated in a very short time with a high degree of precision. The performance of the system under various conditions can be examined by relatively easy treatment of the input data, and very intimate comparisons of the performance under these conditions can be achieved with great accuracy. This method has gained wide acceptance for the analysis of ventilation systems in high-rise buildings.

Provided that the pressures in adjacent spaces are known, the flow balance equation for the room shown in Figure 1.4 is expressed as follows:

$$\frac{1}{R_c}\left(|P_o - P_i|\right)^{\frac{1}{\beta_w}} \pm \frac{1}{R_d}\left(|P_c - P_i|\right)^{\frac{1}{\beta_d}} \pm \frac{1}{R_s}\left(|P_s - P_i|\right)^{\frac{1}{2}} \pm \frac{1}{R_r}\left(|P_r - P_i|\right)^{\frac{1}{2}} = 0 \qquad 1.11$$

where R_c, R_d, R_s and R_r are the flow resistances at the window, door, supply register and exhaust register respectively, P_o, P_i, P_c, P_s and P_r the pressures outside and inside the room, corridor, and the branch points in the supply and return ducts to the room

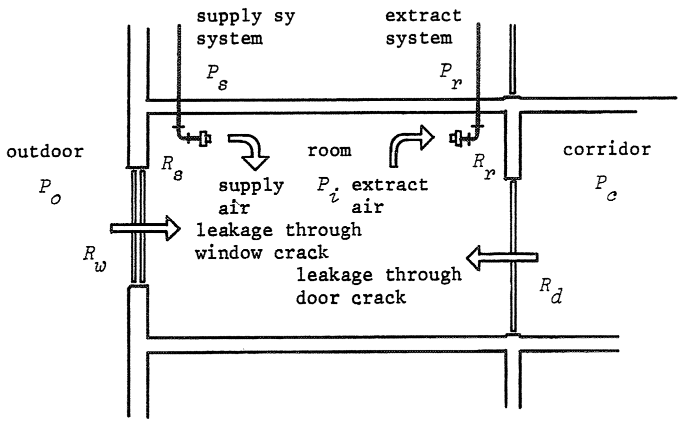


Fig. 1.4 Flow balance in a room

respectively. The positive sign is applied to every term when the corresponding air flow is inwards, and vice versa.

The iterative calculation to obtain the balanced pressure P_i in the room in Equation 1.11 is carried out as follows. When the pressures P_o, P_c, P_s and P_r are known, the equation is expressed as $f(P_i)$, a function of the pressure in the room. Suppose that the pressure P_i is made up of an assumed value $\mathring{P}_i$ and a correction term $\ddot{P}_i$, so that

$$P_i = \mathring{P}_i + \ddot{P}_i \qquad 1.12$$

The correction term $\ddot{P}_i$ can be obtained by the Newton method as follows:

$$\ddot{P}_i = \frac{f(P_i) - f(\mathring{P}_i)}{f'(\mathring{P}_i)} = -\frac{f(\mathring{P}_i)}{f'(\mathring{P}_i)} \qquad 1.13$$

Here $f'(P_i)$ is the first differential coefficient of equation 1.11 with respect to P_i.

The value of $\mathring{P}_i$ first assumed can be modified by the correction term $\ddot{p}_i$ according to Equation 1.12. In the next step of the iteration process, the newly modified pressure P_i is substituted for the assumed pressure $\mathring{P}_i$. The balanced condition is obtained by repetition of this process until the correction term attains a value below a tolerable limit.

When a building consists of a large number of rooms, the balance equations for every space must be processed simultaneously. The above iteration process may result in divergence since in attaining the balanced pressure in a space it is assumed that the pressures in adjacent spaces are stable, but, naturally, where there are a large number of spaces the pressures in adjacent spaces must also be modified simultaneously with the process applied for the space in question. Another cause of divergence may be that the pressure first assumed in each space does not approximate very well to the balanced condition. To avoid such divergence in the course of iteration, the rate of modification

must be decreased by using a correction term δ. Equation 1.12 is then altered to read

$$P_i = \mathring{P}_i + \delta\, \ddot{P}_i \qquad 1.14$$

The recommended value of the correction factor in this case is about 0.5.

As an example, the ventilation in a 16-storey block of flats with an extract ventilation system is treated by the network method. The design air flow rate is 0.05×10^{-3} m^3/s for each flat. The pressure loss at window cracks, the exhaust register and one floor height of vertical duct are 25, 10 and 2 Pa respectively at the design air flow rate at the first floor. The pressure difference by height and the effect of the temperature difference are treated by equation 1.2. The absolute pressure (mb) outdoors are shown in Figure 1.5 for each floor.

The performance of a fan in a ventilation system is influenced by changes in the conditions of the building and in the climate. The following equation is used to express the fan characteristics:

$$P_e = \kappa_1 - \kappa_2 q_f \qquad 1.15$$

where P_e is the pressure difference caused by the fan, κ_1 and κ_2 are constants to be determined from the characteristic curve applicable to the operating range, and q_f is the rate of air flow delivered by the fan.

By opening the window on the first floor, the volume extracted from the first floor is increased by 56 %, and that extracted from the second floor decreased by 13 %, as shown in Figure 1.5a. In the example illustrated in Figure 1.5b, the flow resistance at the exhaust register of the first floor is changed to 50 Pa at the design flow rate, and the effect of the fan is correspondingly increased. The increase in volume of air extracted from the first floor is reduced to 21 %, and the volume of air extracted from the second floor is slightly changed by the window being opened on the first floor.

Legend

a height in floors
b absolute pressure of outdoor air at each elevation in mb
c pressure caused by supply fan in Pa
c´ " " exhaust fan in Pa
d volume of flow at supply fan in $\times 10^{-3}$ m^3/s
d´ " " at exhaust fan in $\times 10^{-3}$ m^3/s
e air leakage at top of vertical shaft in $\times 10^{-3}$ m^3/s
f pressure difference between verticla shaft and outdoor of the same elevation in Pa

each floor:
g pressure difference between room and outdoors of the same elevation in Pa
h air leakage through window crack in $\times 10^{-3}$ m^3/s
i " " crack around entrance door in $\times 10^{-3}$ m^3/s
j volume of air supply in $\times 10^{-3}$ m^3/s
j´ " " extraction in $\times 10^{-3}$ m^3/s

k air leakage at bottom of vertical shaft in $\times 10^{-3}$ m^3/s

Conditions of case a

Design ventilation rate 0.05 m^3/s (each floor)
Air temperature outside 20.0 °C, indoor 20.0

Air flow coefficients of cracks

	a $m^3/m\ s\ Pa^{1/\beta}$		l m
window	0.0731	10^{-3}	80.0
entrance door	-		-
top of shaft	-		-
bottom of shaft	-		-

Pressure drop (at design flow rate)

	supply	return
register (1st. floor)	- Pa	10 Pa
vertical duct (between each connection)	-	2

Charactor of fans

	κ_1 Pa	κ_2 Pa/(m^3/s)
supply	-	-
exhaust	451	50

Conditions for case b

Design ventilation rate 0.05 m^3/s (each floor)
Air temperature outside 20.0 °C, indoor 20.0

Air flow coefficients of cracks

	a $m^3/m\ s\ Pa^{1/\beta}$		l m
window	0.0731	10^{-3}	80.0
entrance door	-		-
top of shaft	-		-
bottom of shaft	-		-

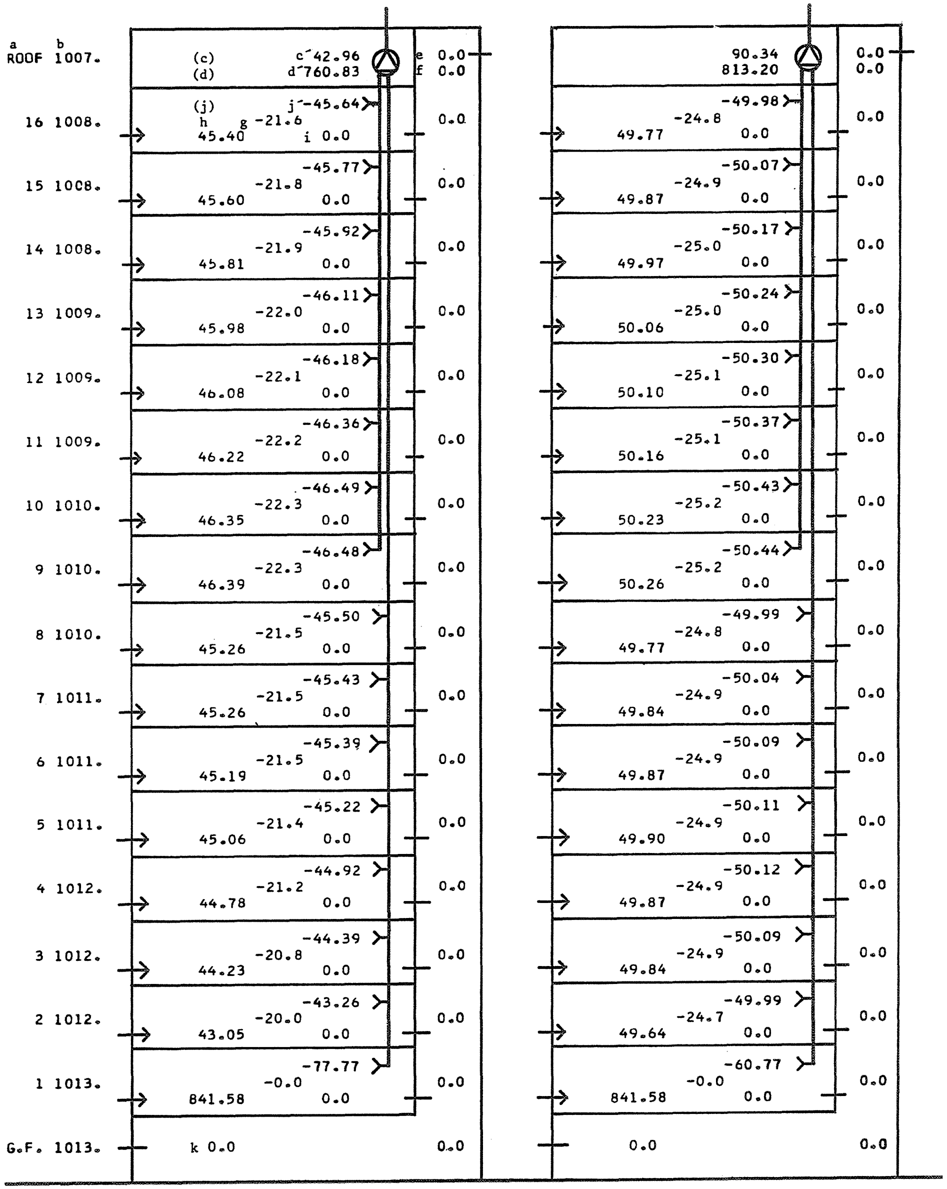
Pressure drop (at design flow rate)

	supply	return
register (1st. floor)	- Pa	50 Pa
vertical duct (between each connection)	-	2

Charactor of fans

	κ_1 Pa	κ_2 Pa/(m^3/s)
supply	-	-
exhaust	491	50

Fig. 1.5 Continued



a. small resistance of flow at extract registers

b. large resistance of flow at extract registers

Fig. 1.5 Comparison of changes in flow balance by opening window of 1st floor

2 The exponent in the air infiltration equation

2.1 The air infiltration equation

The ventilation of a detached house is usually satisfied by natural means. The main routes of air flow are the gaps at window cracks and the joints in wall panels. In a tall building with a mechanical ventilation system, the disturbance due to infiltration of air through such cracks plays an important part in relation to the performance of the ventilation system. The following equation is conveniently applied for the air flow through a gap of length l:

$$v = a \cdot l \cdot (\Delta P)^{\frac{1}{\beta}} \qquad 2.1$$

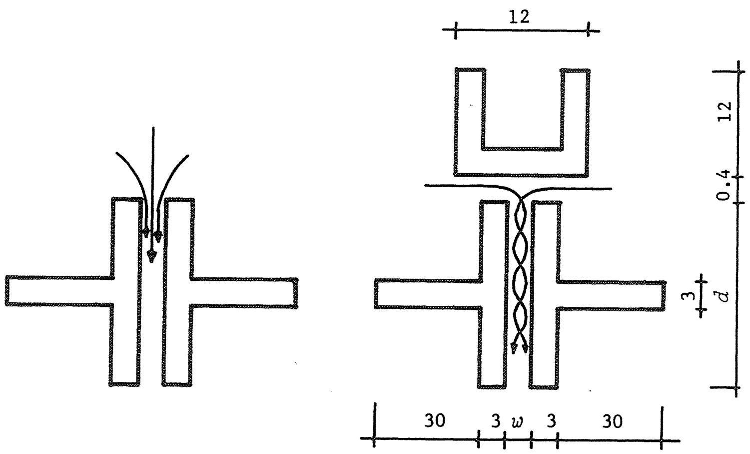
where a is a proportionality constant of a gap and ΔP the pressure difference across the gap. In the following part of this chapter, the reciprocal of the exponent of the pressure difference is simply termed "exponent". It has been known that the exponent β of the pressure difference varies depending on the conditions of the air passage and the acting pressure difference itself. But experiments on various types of cracks, performed by different researchers, show that the exponent varies in an irregular manner. The primary cause of this irregularity may be the difference in the shape of the air passage, but may also be due to the fact that the ranges of pressure employed in the different measurements are different.

As a result of the proliferation of multistorey blocks of flats equipped with mechanical ventilation systems, there has been a considerable rise in the range of pressure differences which act across the exterior walls of such buildings. In view of the fact that newly constructed buildings tend to be higher, the structure of their walls must have a higher degree of impermeability.

A very accurate assessment of the performance of a ventilation system is possible by the application of the network method. A more intimate understanding of every factor involved in ventilation is necessary in order that more realistic results may be obtained by the method. Knowledge of the value of the exponent in the infiltration equation is one of the most important elements

in a ventilation calculation. When the exponent is expressed by a simple equation over the range of pressure differences likely to be encountered in tall buildings, this must be conveniently used in examining the influence of infiltration on ventilation and consequently in more realistic design of ventilation.

In this chapter, the character of the exponent is examined for air flows through various sizes of simply shaped gaps over a wide range of pressure differences. The results are compared with those of different researchers relating to air leakage measurements through various kinds of window cracks and gaps in building structures.



a. plain slit b. slit with obstacle

unit : mm

Fig. 2.1 Sections of air passages

2.2 Air flow through plain gaps

2.2.1 Measuring equipment

Owing to its simple shape and precise geometry, a space between two plain surfaces was chosen for examination of air leakage. The simple shape made possible convenient examination of the results by the theory of fluid dynamics. Air flow through an actual window crack is extremely complicated and is almost impossible to treat by the theory of fluid dynamics. The values chosen for the widths w and depths d of the gaps were as follows. See Figure 2.1.

Width w, mm	0.075	0.15	0.35	0.79	1.05
Depth d, mm	30.0	19.7	7.9		

The gaps were constructed between the top faces of two aluminium T bars. The two bars of about 1 m length were cut from long runs of bar in order that there may be even contact between the two faces over the whole length. The two faces were also polished by rubbing them against each other, a slip of emery paper being placed between them. The surfaces therefore have fine polish scratches of emery paper No 280 at right angles to the direction of air flow. It was not possible to measure the fit between the two polished faces but, when a source of light was observed through the gap, the intensity of illumination appeared uniform over the entire length. The two bars were fastened to each other by screws at each end and at two points near their centres,there being sheet metal spacers between them. The spacers were cut from copper or brass sheets. The above gap widths are the average thicknesses of every set of four spacers. As a result, the gap was divided into three lengths of 317 mm each. This length was considered to be long enough for the end effects to be negligible. The case was made from galvanized iron plates which were soldered at each corner to prevent air leakage. The aluminium bars were mounted on the case with bolts. Considerable care was taken to prevent distortion which may be caused as a result of mounting the bars on the case. Airtightness was further ensured by coating with grease every contact surface on the bars, the case and bolt holes. The viscosity of the grease used was sufficient to withstand a

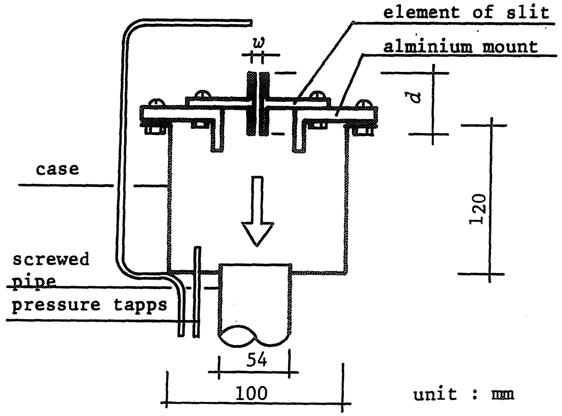


Fig. 2.2 Air leakage measurement equippment

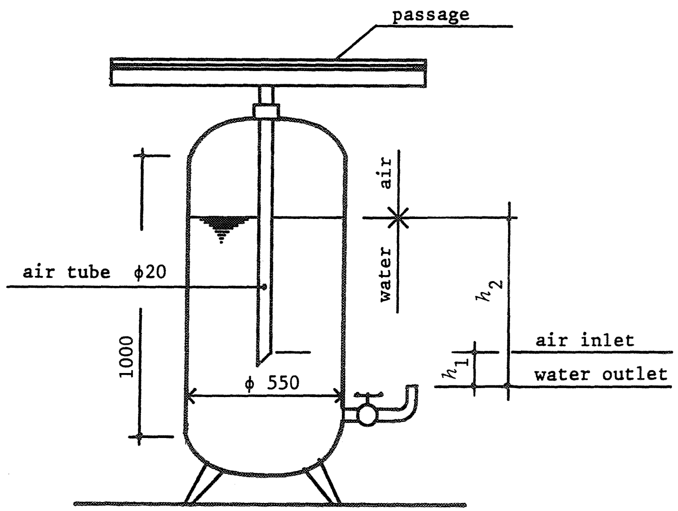


Fig. 2.3 Water vessel for measurement of low flow rate

pressure difference of up to 1000 Pa. A section of the equipment is shown in Figure 2.2. The case was connected to the flow rate measurement and air suction devices by means of screwed connectors in order to minimise air leakage between the case and the flow rate measuring equipment.

The pressure difference across the gaps was measured by a micro-manometer. The minimum sensitivity of the instrument was 0.1 Pa. The pressure difference applied across the gaps ranged from 1 Pa to over 500 Pa. The upper limit was varied depending on the flow resistance of the gap and the capacity of the air suction apparatus. The pressure difference of 500 Pa is equivalent to the velocity head due to a wind velocity of 28 m/sec.

For low rates of flow, a water displacement method was employed for flow rate measurement. The air passing through the gap was lead into the top of an airtight vessel, and the volume of the displaced water was measured. This installation is shown in Figure 2.3. Nearly constant flow was obtained irrespective of the water level in the vessel by placing the air inlet below the water surface. A smooth air flow was ensured by making the air outlet at the bottom end of the air tube in the form of a group of small perforations. Due to the difference h_2 between the water level inside the vessel and the level of the water outlet, the pressure in the air space is lower than atmospheric pressure, and the volume of air which displaces the water must be corrected by this pressure difference when the air flow through the gap is calculated from the water volume. Two orifices of 8.95 and 32 mm diameter were used for measurement of larger flow rates. These measurements were carried out in the laboratory of the Department of Heating and Ventilation, Royal Institute of Technology Throughout the measurement period, the room temperature was between 18.5 and 20.5°C. Ordinary room air was passed through the gaps.

2.2.2 Measurement results

The flow rate was not measured at previously determined steps of pressure difference, but when the air flow reached a steady condition, the pressure difference and the flow rate were read

simultaneously. There was no objection to irregular values of the pressure difference, since this was quite convenient for effective and precise treatment of the many readings when these were processed in the computer. The readings are the mean values of about three minutes of steady conditions. Both the flow rates and pressure differences cover wide ranges, and the results have therefore been plotted on graphs to logarithmic scale, as shown in Figures 2.4 a - f. The air flow rates were converted into the flow rate for a gap of one metre length, so that the unit of flow rate is cubic metres per second per metre.

The slopes corresponding to values of 1.0 and 2.0 of the exponent β of the pressure difference, in log-log scale, are drawn in Figure 2.4a for reference. The figures show that the slope of the flow rate-pressure difference curve is different for each gap. Generally speaking, the slopes of gaps with larger flow resistances, which means a greater depth and/or a lesser width, are steeper than those of gaps with smaller flow resistances. Judging from the slopes, the flow through a gap with a large resistance is almost directly proportional, over the range of the pressure differences employed, to the pressure difference. The slopes decrease and the exponent β approaches 2.0 as the resistances of the gaps become smaller. It seems that the value of the exponent β never be smaller than unity.

Further, the slopes of the flow rate-pressure difference curves are not constant for each gap. The slopes decrease as the active pressure difference increases, and the transition from a steeper to a flatter slope takes place at a lower pressure difference in the flow through gaps of lower flow resistance than in the flow through gaps with higher flow resistance.

In order to examine the effect of turbulence, measurements were made with and without an obstacle positioned 0.4 mm from the leading edge of the gap, as shown in Figure 2.1. However, no difference could be noted in the changes of the slopes between the gaps without and with the obstacle.

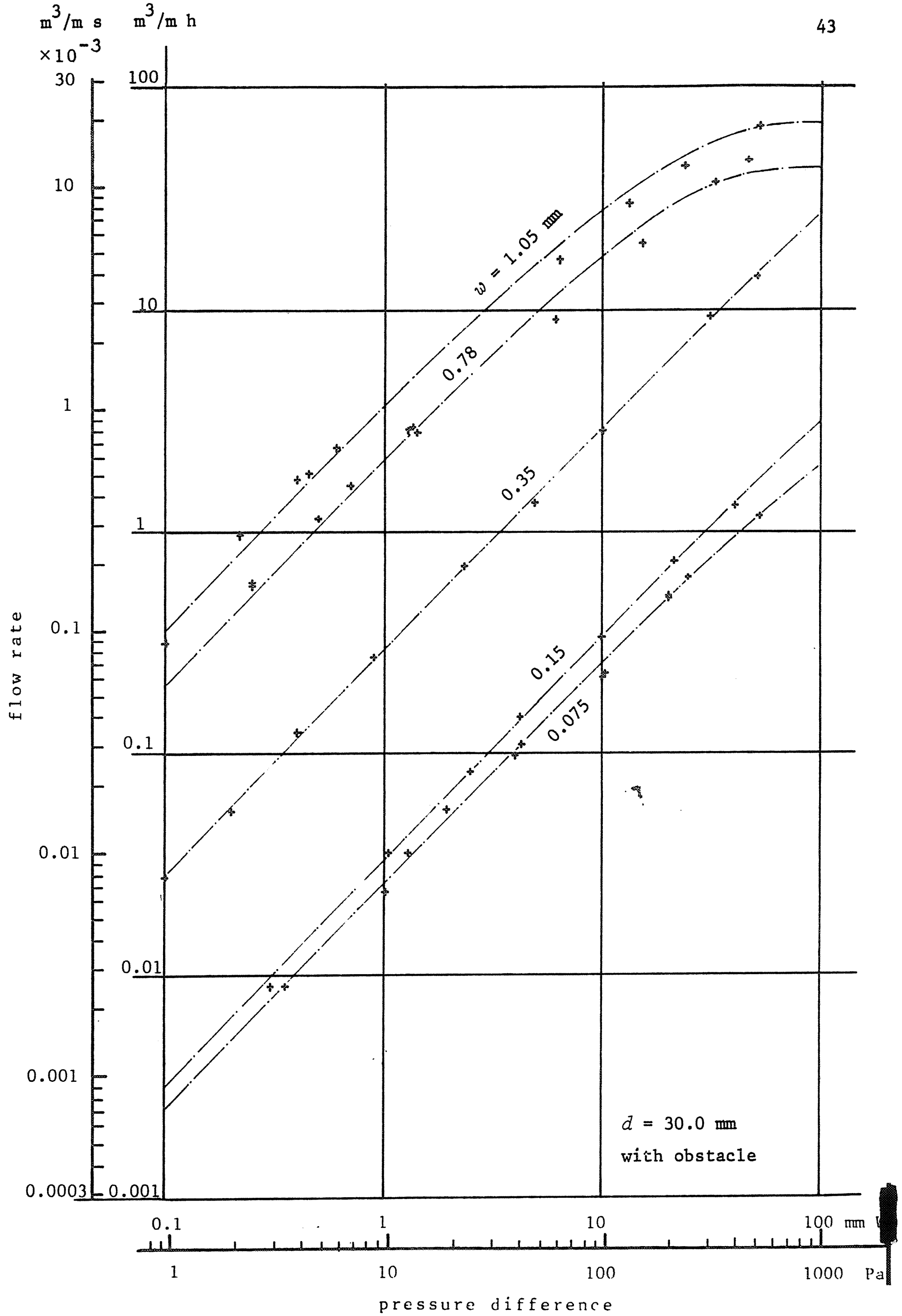


Fig. 2.4 b Relations between volume of flow and pressure difference aluminium gaps with obstacle, depth 30.0 mm

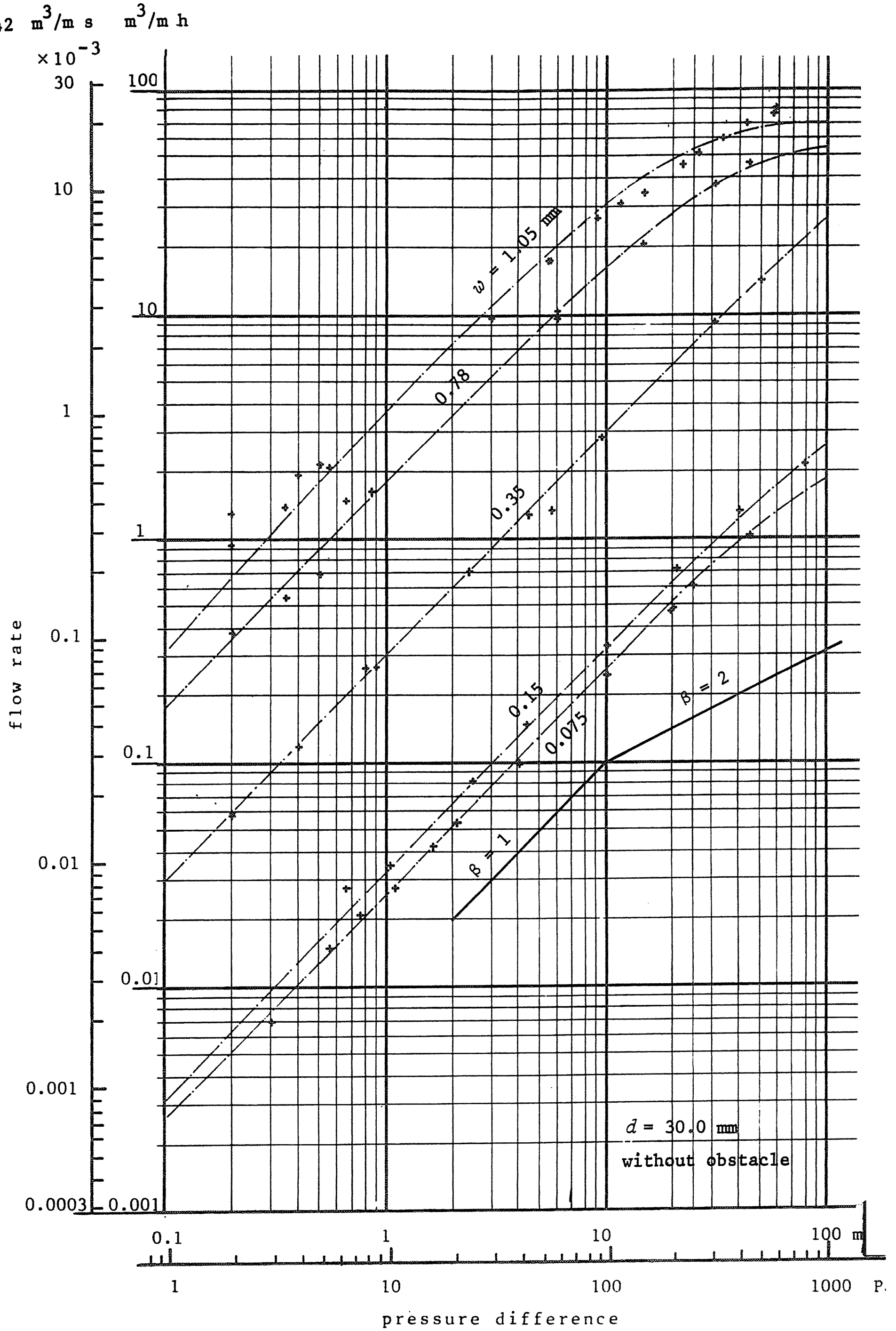


Fig. 2.4 a Relations between volume of flow and pressure difference aluminium gaps, depth 30.0 mm

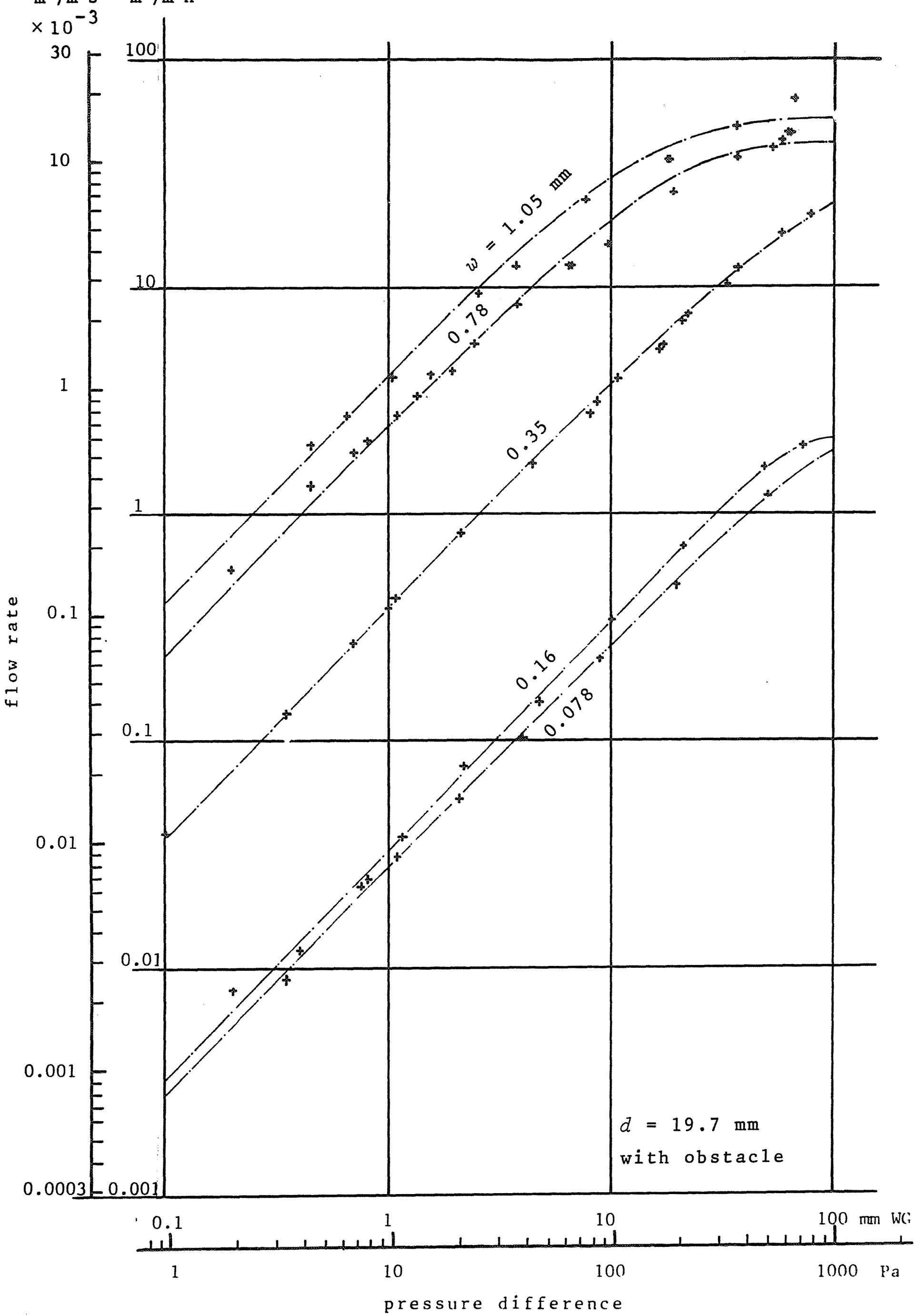


Fig. 2.4 d Relations between volume of flow and pressure difference aluminium gaps with obstacle, depth 19.7 mm

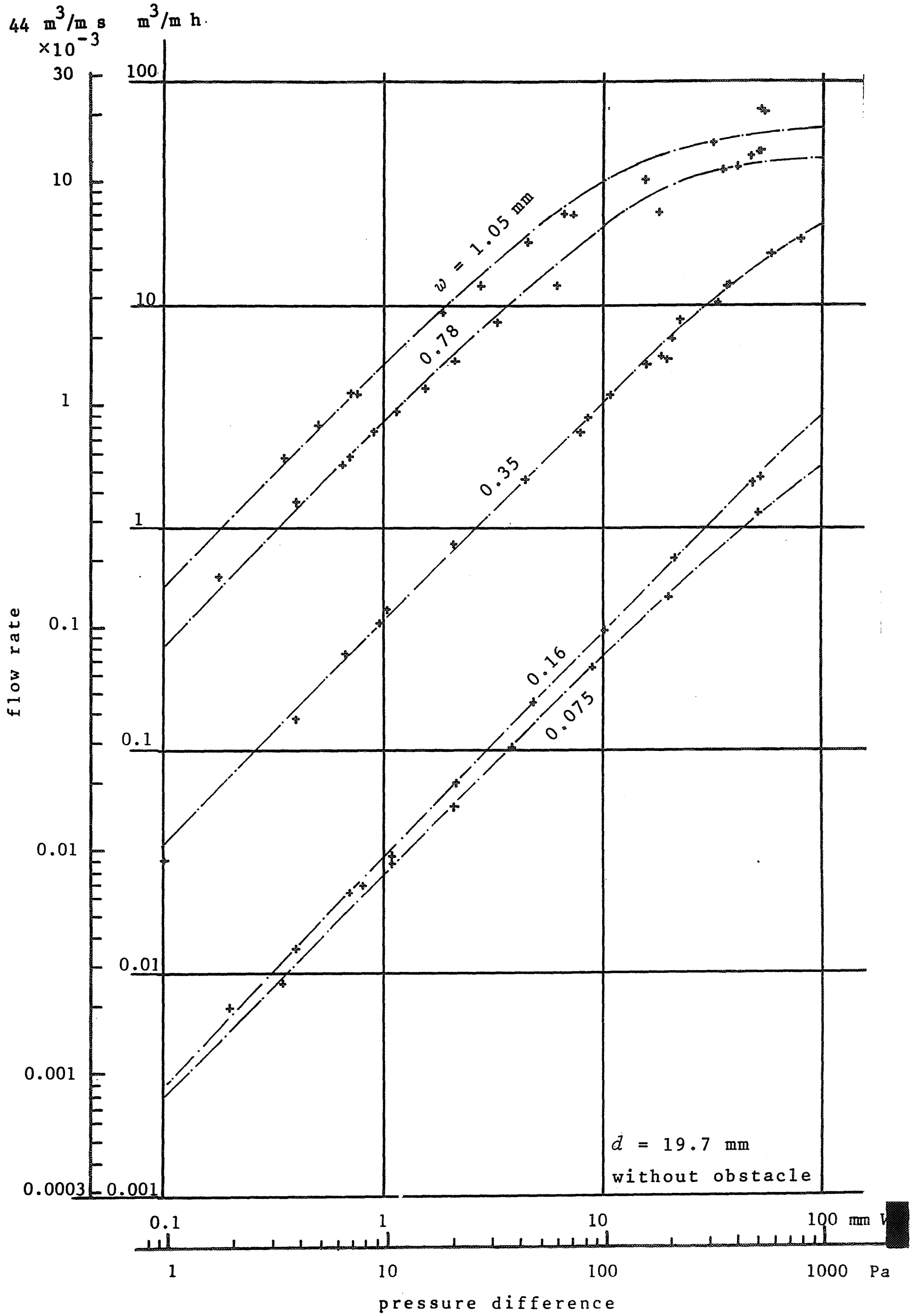


Fig. 2.4 c Relations between volume of flow and pressure difference aluminium gaps, depth 19.7 mm

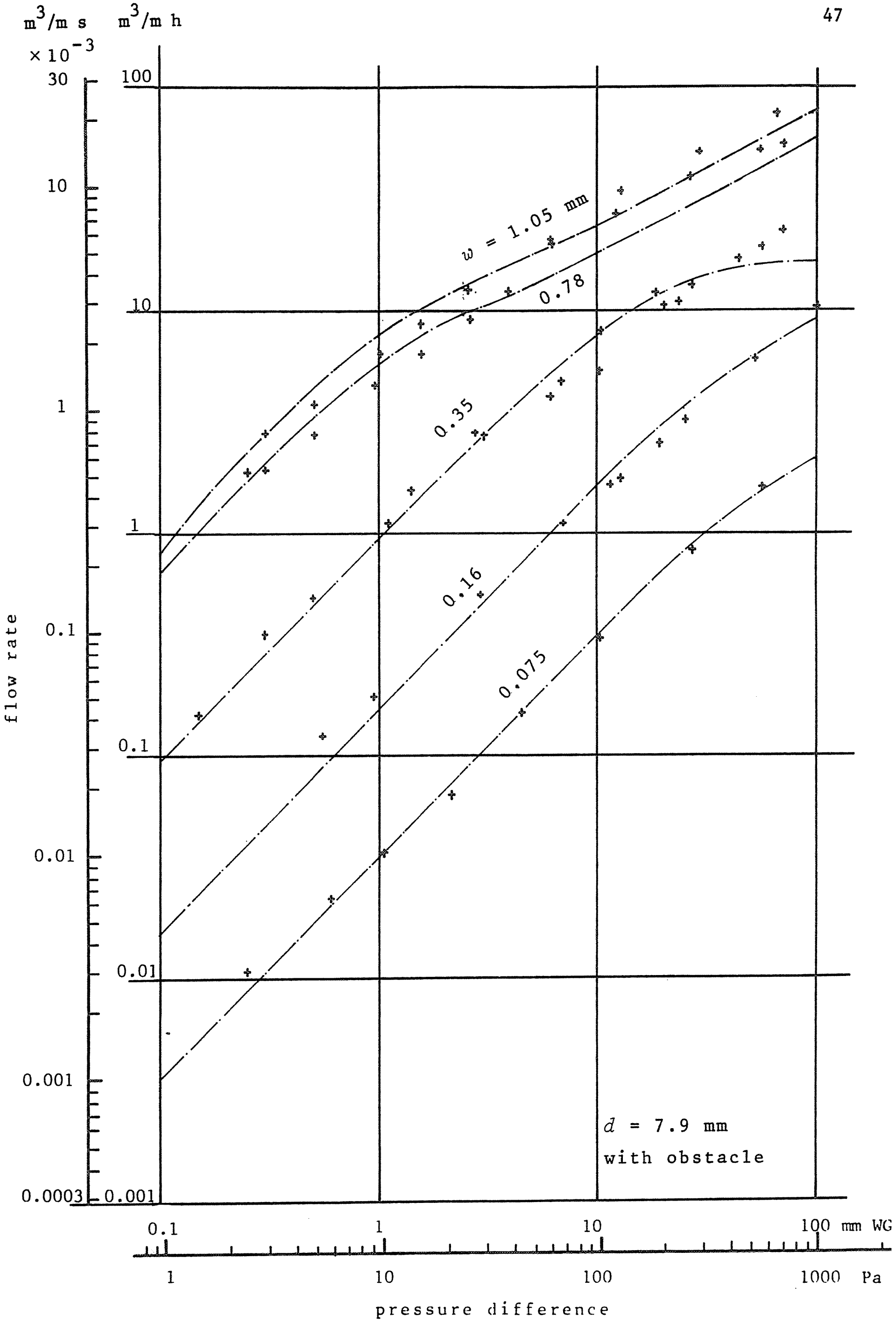


Fig. 2.4 f Relations between volume of flow and pressure difference aluminium gaps with obstacle, depth 7.9 mm

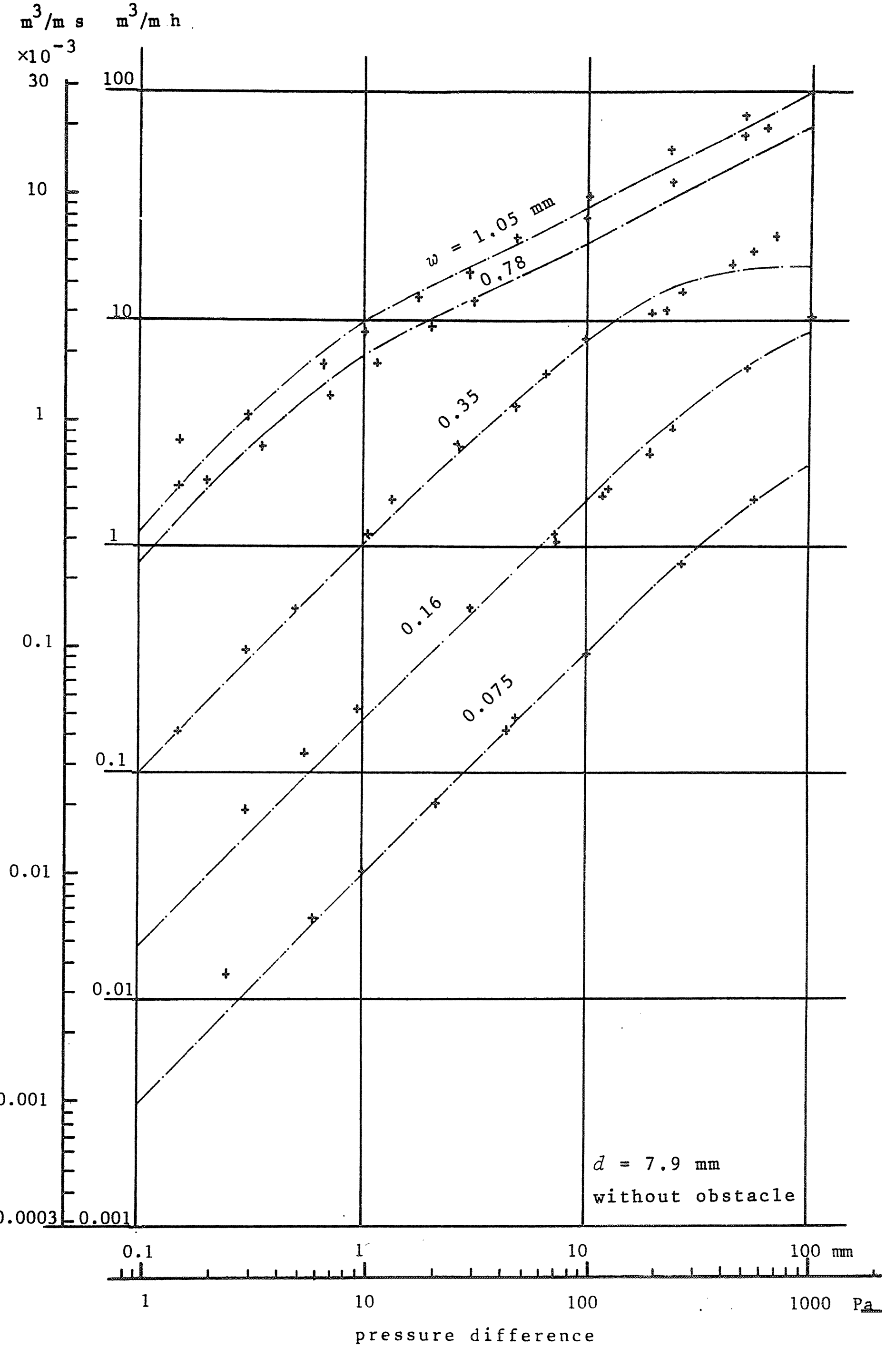


Fig. 2.4 e Relations between volume of flow and pressure difference aluminium gaps, depth 7.9 mm

2.2.3 Statistical treatment of the results

When the general theory of fluid flow is applied to the flow through a gap, the energy equation is made up of the active pressure difference ΔP, the friction pressure loss P_f and the acceleration pressure loss P_a, as follows:

$$\Delta P = P_f + P_a \qquad 2.2$$

When the average velocity of air through a gap is $\bar{u}$, the acceleration pressure is expressed by the equation

$$P_a = \frac{1}{2} g \rho \bar{u}^2 \qquad 2.3$$

where ρ is the density of the air.

The friction pressure loss P_f is also approximated as a function of the average velocity. The friction pressure loss is linearly proportional to the average velocity in the laminar flow range, as follows

$$P_f = \text{const.}\ \bar{u} \qquad 2.4$$

And in the turbulent flow range, the friction pressure loss is proportional to the square of the average velocity, as follows

$$P_f = \text{const.}\ \bar{u}^2 \qquad 2.5$$

The flow rate through a gap is directly expressed by the product of the average velocity and the area of the air passage. Consequently, the energy equation 2.2 can be rewritten as a quadratic equation for the flow rate, with a term to the first power for the pressure difference. In the actual phenomenon of air flow through a fine crack, it may be supposed that there is a considerable range over which laminar and turbulent flow exist simultaneously.

It may be unreasonable to treat the flow as only turbulent flow, even though turbulence takes place in the flow through the gap. The reason is that the depth of passage through a crack is not

large enough for the theory of turbulent flow to be applied, which assumes sufficiently developed turbulence. In such a case, the first power term for the flow rate has a significant effect in the energy equation. The following expression can then yield the energy balance:

$$\Delta P = \frac{1}{2} g \rho v^2 + \{ \text{const.}\ \xi v + \text{const.}\ (1 - \xi) v^2 \} \qquad 2.6$$

where ξ is the part of the friction loss due to laminar flow.

When the exponent in Equation 2.1 is expressed by a simple equation, this may be helpful in effectively performing a network calculation for a ventilation system. The exponent is governed by the relationship between the first and second power terms for the flow rate. From the equation for the energy balance, Equation 2.6, it is easily seen that the exponent of the pressure difference is in the range 1.0 - 2.0. But this relationship cannot be solved theoretically.

Among others, the following function was chosen to express the exponent β, because it includes only one constant A:

$$\beta = 2.0 - e^{-A \cdot \Delta P} \qquad 2.7$$

The value of β remains approximately unity over the range where the resulting pressure difference is small, and approaches two when the pressure difference increases. The intensity of this approach is governed by the constant A.

The combinations of the proportionality constant α and the constant A in the equation for the exponent were calculated, using the principle of least squares, from the experimental results for air flow through gaps. The flow rates through each gap covered a wide range, and the least squares method was employed to minimise the rates of deviations from the flow rates, instead of minimising the deviations themselves. The statistical method of calculating a mean value from weighted data was applied for this calculation, the reciprocal of the flow rate being adopted as the weight of each measurement.

The relationships between the proportionality constant a and the width w of gaps are shown in Figure 2.5. The figure shows that the proportionality constants a for gaps of the same width differ clearly by depth. On average, the standard deviation of the experimental values from the approximate equation was 11%. The deviation was only a few per cent for flows through gaps of large flow resistance, but it reached 20% for flows through gaps of small flow resistance.

The air flow was calculated by substituting the results for a and A from the least squares method into Equation 2.7 and Equation 2.1. The calculated values are shown in Figures 2.4a - f by dashed lines. From the character of Equation 2.7, the change in air flow through a gap of small flow resistance shows singularity over the range where the exponent changes rapidly.

In the following examination of the results, the air flow through each gap is represented by the statistically calculated value. The average velocity of air flow can be obtained by simply dividing the flow rate by the area of the gap in the range of laminar flow. In the range of turbulent flow, a contraction of flow takes place at the leading edge of a gap, and the average velocity consequently increases. The acceleration pressure loss was calculated by Equation 2.3 for the several pressure difference steps for each gap, both with the contraction being ignored and the contraction coefficient assumed at 0.8. The acceleration pressure loss part of the total pressure difference required to cause the flow was also calculated for each gap and each pressure difference. This is shown in tables 2.1 a-f.

In the flow through the gap of 0.075 mm width and 30 mm depth, the share due to acceleration is only 3 % at a pressure difference of 500 Pa. This means that 97 % of the pressure is used in overcoming friction, see table 2.1 a. In the case of the gap of 1.05 mm width and 7.9 mm depth, the acceleration share is as much as 93 %, see table 2.1 e. If contraction takes place at the leading edge, then this share exceeds 100 %. In the flow through a gap of a small flow resistance, the acceleration share is calculated to be smaller in the higher range of pressure difference than in the middle range . This may be influenced by the character of the applied equation 2.7.

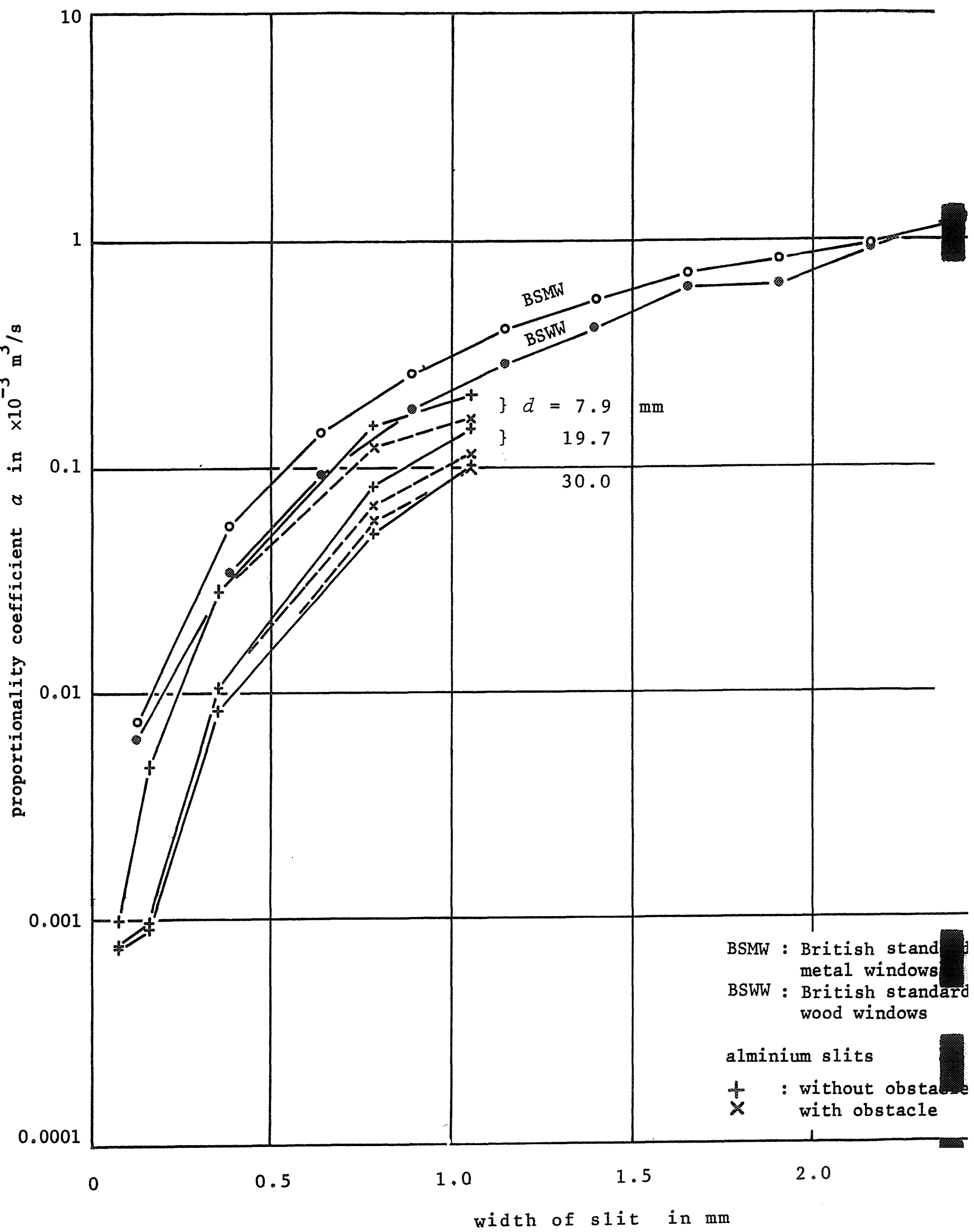


Fig. 2.5 Variation in proportionality coefficients by widths of slits

In fluid dynamics, the critical Reynolds number for a flow along a flat surface is around 500. Here, the reference length of the flow is the depth of the boundary layer. Reynolds number was calculated for the flow through a gap by substituting half the width of a gap for the reference length, as follows:

$$Re = \frac{\bar{u}}{\nu} \cdot \frac{w}{2} \qquad 2.8$$

Here, ν is the kinematic viscosity of the air.

Table 2.1 a - f show that when Reynolds number is about 500, the share of the pressure for acceleration of the air , calculated without consideration of contraction, is between 16 and 35 % of the total pressure difference.

The depth of a gap plays no part in the calculation of Reynolds number, provided that the depth affects the velocity. But the part played by acceleration is smaller in the case of a deeper gap than in the case of a shallower gap, when the Reynolds numbers of both are around 500. It is seen from this that friction exerts considerable influence on flow through gaps.

The theoretical solution given for flow through a space between two flat plates in the laminar flow range, assuming a parabolic velocity distribution, is as follows:

$$\bar{u} = \frac{w}{12\,\rho\,\nu} \frac{d\,P}{d\,d} \qquad 2.9$$

Where $\frac{d\,P}{d\,d}$ is the pressure gradient.
In the range of laminar flow the roughness of the wall has no effect, only the viscosity of the fluid. This equation applies for a steady condition of laminar flow, but the average velocities in the gaps were estimated by this equation. The acting pressure difference and the depth of the gap concerned were employed in calculating the pressure gradient. No consideration was given to transient effects at the entrance and exit of the gap.

The theoretical and experimental velocities are compared in tables 2.1 a-f. The experimental velocities coincide with the

theoretical velocities in the case of gaps whose flow resistances are in the middle of the range covered by the experiment series.

In the case of small flow resistances the experimental results gradually deviate as the pressure difference is increased. It is considered that this deviation is due to the development of turbulence in the flow. The deviation begins in the Reynolds number range below 500, and the starting point is also affected by the depth of the gap.

The experimental flows through gaps of higher flow resistances exhibited far larger velocities than the theoretical values. The deviations were almost the same for each gap all over the range of the pressure differences employed. In the case of gaps of the same width, the deviation is greater for the gap of larger depth. To a limited extent, this may be due to edge effects.

The values of the acceleration pressure loss and Reynolds number may be regarded as an indication of the value of the exponent. When one or both of these are large, the exponent approaches 2.0, while the exponent is very nearly unity when they are small.

The following experimental equation is given for air flow through a window crack by Thomas and his associate (1953). The constants are based on various window cracks examined in the UK, Germany and Canada.

$$\Delta P = \frac{0.022 \times 10^{-3}}{(w_m)^2} (v + 236\, v^2) \qquad 2.10$$

where w_m is mean face clearance in mm.

The flow rates shown in the lines "experimental equation" in tables 2.1 a-f were calculated by applying the width w of the gaps for the mean face clearance w_m in Equation 2.10. The calculated flow rate predicts satisfactorily the experimental results for gaps with small flow resistances.Equation 2.10 was found to result in overestimation of the flow through a gap in the case of higher flow resistances and lower pressure differences. This is discussed further in 2.3.1.

Table 2.1 c Characters of flow through plain slit, width 0.35 mm, depth 19.7 mm

PRESSURE IN PA	1.00	2.50	5.00	10.00	25.00	50.00	100.00	250.00	500.00
AIR VELOCITY IN M/SEC.	0.031	0.076	0.153	0.305	0.759	1.506	2.953	6.902	12.231
FLOW RATE IN X10^{-3} M^{3}/M S	0.011	0.027	0.053	0.107	0.266	0.527	1.033	2.416	4.281
THEORY OF LAMINAR FLOW	0.010	0.025	0.049	0.098	0.246	0.492	0.984	2.459	4.918
EXPERIMENTAL EQUATION	0.062	0.147	0.271	0.478	0.945	1.509	2.333	3.996	5.887
EXPONENT OF PRESSURE	1.000	1.000	1.000	1.001	1.002	1.004	1.008	1.019	1.037
REYNOLDS NUMBER	1.	4.	7.	14.	35.	70.	138.	322.	570.
ACCELERATION IN PA	0.001	0.004	0.014	0.056	0.347	1.366	5.252	28.701	90.131
IN PERCENT	0.1	0.1	0.3	0.6	1.4	2.7	5.3	11.5	18.0
CONTRACT COE. 0.8	0.1	0.2	0.4	0.9	2.2	4.3	8.2	17.9	28.2

Table 2.1 d Characters of flow through plain slit, width 0.35 mm, depth 7.9 mm

PRESSURE IN PA	1.00	2.50	5.00	10.00	25.00	50.00	100.00	250.00	500.00
AIR VELOCITY IN M/SEC.	0.080	0.201	0.401	0.796	1.940	3.689	6.606	11.738	14.006
FLOW RATE IN X10^{-3} M^{3}/M S	0.028	0.070	0.140	0.279	0.679	1.291	2.312	4.108	4.902
THEORY OF LAMINAR FLOW	0.025	0.061	0.123	0.245	0.613	1.226	2.453	6.132	12.264
EXPERIMENTAL EQUATION	0.062	0.147	0.271	0.478	0.945	1.509	2.333	3.996	5.887
EXPONENT OF PRESSURE	1.000	1.001	1.002	1.005	1.011	1.023	1.045	1.108	1.204
REYNOLDS NUMBER	4.	9.	19.	37.	90.	172.	308.	547.	653.
ACCELERATION IN PA	0.004	0.024	0.097	0.382	2.267	8.200	26.291	83.010	118.192
IN PERCENT	0.4	1.0	1.9	3.8	9.1	16.4	26.3	33.2	23.6
CONTRACT COE. 0.8	0.6	1.5	3.0	6.0	14.2	25.6	41.1	51.9	36.9

Table 2.1 a Characters of flow through plain slit, width 0.075 mm, depth 30.0 mm

PRESSURE IN PA	1.00	2.50	5.00	10.00	25.00	50.00	100.00	250.00	500.00
AIR VELOCITY IN M/SEC.	0.010	0.025	0.049	0.098	0.245	0.487	0.960	2.287	4.191
FLOW RATE IN X10^{-3} M^{3}/M S	0.001	0.002	0.004	0.007	0.018	0.037	0.072	0.172	0.314
THEORY OF LAMINAR FLOW	0.000	0.000	0.000	0.001	0.002	0.003	0.006	0.016	0.032
EXPERIMENTAL EQUATION	0.003	0.007	0.015	0.029	0.071	0.136	0.252	0.532	0.890
EXPONENT OF PRESSURE	1.000	1.000	1.000	1.001	1.001	1.003	1.005	1.013	1.026
REYNOLDS NUMBER	0.	0.	0.	1.	2.	5.	10.	23.	42.
ACCELERATION IN PA	0.000	0.000	0.001	0.006	0.036	0.143	0.555	3.151	10.581
IN PERCENT	0.0	0.0	0.0	0.1	0.1	0.3	0.6	1.3	2.1
CONTRACT COE. 0.8	0.0	0.0	0.0	0.1	0.2	0.4	0.9	2.0	3.3

Table 2.1 b Characters of flow through plain slit, width 0.35 mm, depth 30.0 mm

PRESSURE IN PA	1.00	2.50	5.00	10.00	25.00	50.00	100.00	250.00	500.00
AIR VELOCITY IN M/SEC.	0.024	0.061	0.122	0.244	0.609	1.216	2.422	5.977	11.666
FLOW RATE IN X10^{-3} M^{3}/M S	0.009	0.021	0.043	0.085	0.213	0.425	0.848	2.092	4.083
THEORY OF LAMINAR FLOW	0.006	0.016	0.032	0.065	0.161	0.323	0.646	1.615	3.230
EXPERIMENTAL EQUATION	0.062	0.147	0.271	0.478	0.945	1.509	2.333	3.996	5.887
EXPONENT OF PRESSURE	1.000	1.000	1.000	1.000	1.000	1.001	1.001	1.004	1.007
REYNOLDS NUMBER	1.	3.	6.	11.	28.	57.	113.	279.	544.
ACCELERATION IN PA	0.000	0.002	0.009	0.036	0.223	0.890	3.535	21.521	82.000
IN PERCENT	0.0	0.1	0.2	0.4	0.9	1.8	3.5	8.6	16.4
CONTRACT COE. 0.8	0.1	0.1	0.3	0.6	1.4	2.8	5.5	13.5	25.6

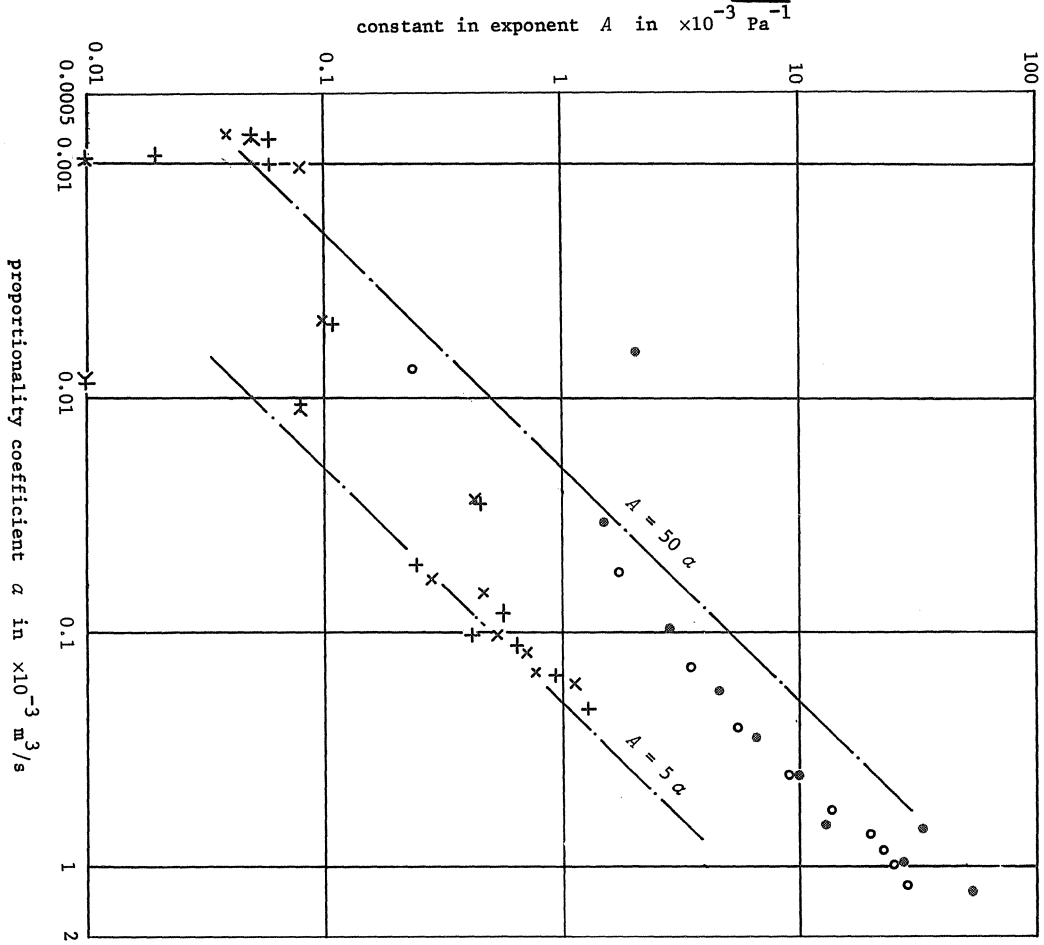


Fig. 2.6 Relation between proportionality coefficients and the constants in exponent

- O British standard metal window
- ● British standard wooden window
- + alminium slit without obstacle
- × alminium slit with obstacle

Table 2.1 e Characters of flow through plain slit, width 1.05 mm, depth 7.9 mm

PRFSSURE IN PA	1.00	2.50	5.00	10.00	25.00	50.00	100.00	250.00	500.00
AIR VELOCITY IN M/SEC.	0.188	0.469	0.933	1.838	4.341	7.791	12.437	16.810	15.652
FLOW RATE IN X10 $^{-3}$ M ^{3}/M S	0.198	0.493	0.980	1.930	4.558	8.181	13.059	17.650	16.435
THEORY OF LAMINAR FLOW	0.662	1.656	3.311	6.623	16.556	33.113	66.226	165.564	331.127
EXPERIMENTAL EQUATION	0.440	0.877	1.409	2.187	3.763	5.555	8.100	13.159	18.866
EXPCNENT OF PRESSURE	1.001	1.003	1.005	1.010	1.026	1.051	1.099	1.229	1.406
REYNOLCS NUMBER	26.	66.	131.	257.	607.	1090.	1740.	2352.	2190.
ACCELERATION IN PA	0.021	0.133	0.525	2.036	11.352	36.573	93.190	170.251	147.612
IN PERCENT	2.1	5.3	10.5	20.4	45.4	73.1	93.2	68.1	29.5
CONTRACT CCE. 0.8	3.3	8.3	16.4	31.8	71.0	114.3	145.6	106.4	46.1

Table 2.1 f Characters of flow through plain slit with obstacle, width 1.05 mm, depth 7.9 mm

PRESSURE IN PA	1.00	2.50	5.00	10.00	25.00	50.00	100.00	250.00	500.00
AIR VELOCITY IN M/SEC.	0.141	0.351	0.699	1.381	3.301	6.060	10.101	15.000	14.964
FLOW RATE IN X10 $^{-3}$ M ^{3}/M S	0.148	0.368	0.733	1.450	3.466	6.363	10.606	15.750	15.712
THEORY OF LAMINAR FLOW	0.662	1.656	3.311	6.623	16.556	33.113	66.226	165.564	331.127
EXPERIMENTAL EQUATION	0.440	0.877	1.409	2.187	3.763	5.555	8.100	13.159	18.866
EXPCNENT CF PRESSURE	1.001	1.002	1.004	1.008	1.020	1.039	1.077	1.182	1.332
REYNOLDS NUMBER	20.	49.	98.	193.	462.	848.	1413.	2099.	2094.
ACCELERATION IN PA	0.012	0.074	0.294	1.148	6.565	22.127	61.475	135.557	134.906
IN PERCENT	1.2	3.0	5.9	11.5	26.3	44.3	61.5	54.2	27.0
CONTRACT CCE. 0.8	1.9	4.6	9.2	17.9	41.0	69.1	96.1	84.7	42.2

2.3 Flow of air through various window cracks

2.3.1 Air flow measurement on British Standard Windows

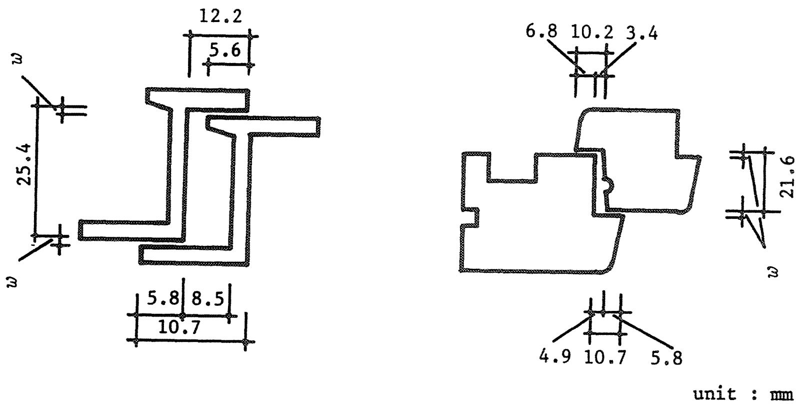
The air infiltration measurements on British standard metal and wooden windows were performed by Thomas and his associates (1953). In one part of these measurements, the members of the standard window were set at various gap sizes, and the flow of air through these gaps was measured over a range of pressure differences. Detailed results are given in the tables in their report. Sections through metal and wooden windows are shown in Figure 2.7. The gap widths ranged from 0.127 to 2.41 mm, and the applied pressure differences from 1.3 to 125 Pa. Figures 2.8 a and b show the flow rate-pressure difference diagram which is reproduced from the tables. The difference in the slopes of the curves for various sizes of gaps can be clearly seen in Figures 2.8 a and b. Also, the pressure at which the change in slope takes place gradually decreases as the gap size increases.

As will be seen from Figure 2.7, the shapes of the air passages are not straight, and the widths also vary along the passage. The total depths of the passages are 36.8 and 34.3 mm respectively for the metal and wooden windows. The total depths at the points at which the passages are contracted to the dimensions marked as widths (or face clearances in the original paper) are 11.4 and 12.7 mm respectively for the standard metal and wooden windows. The proportionality constants a and the constants A in the exponent equation were calculated for the British standard windows by the same method as that used for the aluminium gaps. The resultant proportionality constants are compared with those for the aluminium gaps in Figure 2.5. Here, it is the size of the face clearance and not the average clearance all over the passage which is applied for the width. When the face clearances are referred to the widths w of the aluminium gaps, it is found that the proportionality constants of the British windows are nearer to those of the aluminium gaps of depth 7.9 mm than to those of depth 30 mm.

The ideal method of calculating the frictional flow resistance of a deformed passage is to treat the passage as a sequence of fluid dynamic resistances, in the same way as is done in obtaining the k-value in heat transmission calculations. But this is

2.2.4 Approximate equation for the exponent

As explained above, the slope of the flow rate-pressure difference curve is affected not only by the difference in working pressure but also by the fluid dynamics character of the passage. The character of the passage is influenced by many factors such as width, depth and shape. The proportionality constant a is adopted here to represent all these factors. Each combination of the proportionality constant a and the constant A in the exponent is plotted on logarithmic paper, as shown in Figure 2.6. It will be seen in this figure that the constant A varies linearly with the proportionality constant a, except in the range where the proportionality constants have extremely small values.



a. metal window

b. wood window

Fig. 2.7 Sections of British standard windows

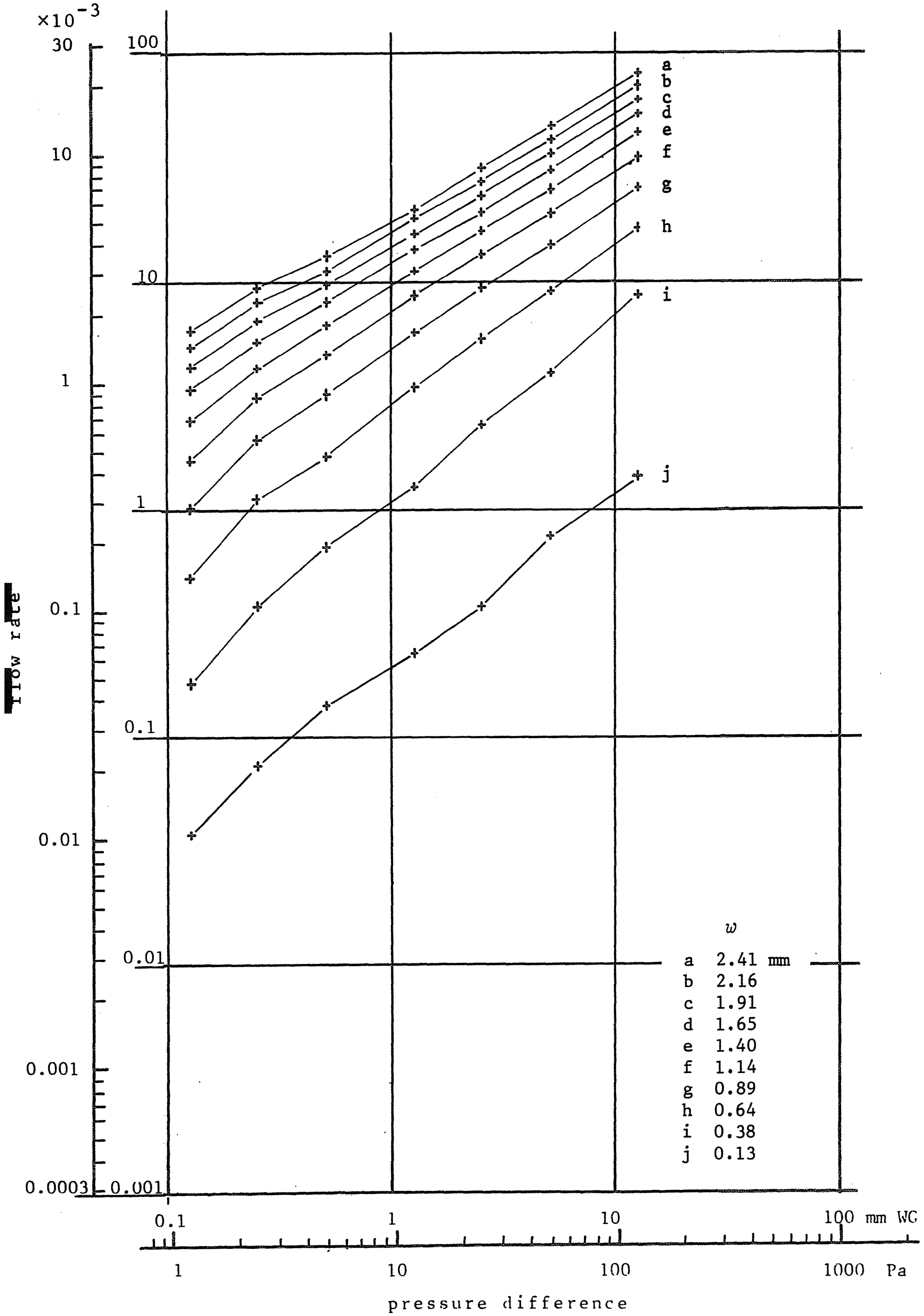


Fig. 2.8 b Relations between volume of flow and pressure difference
British standard wood windows

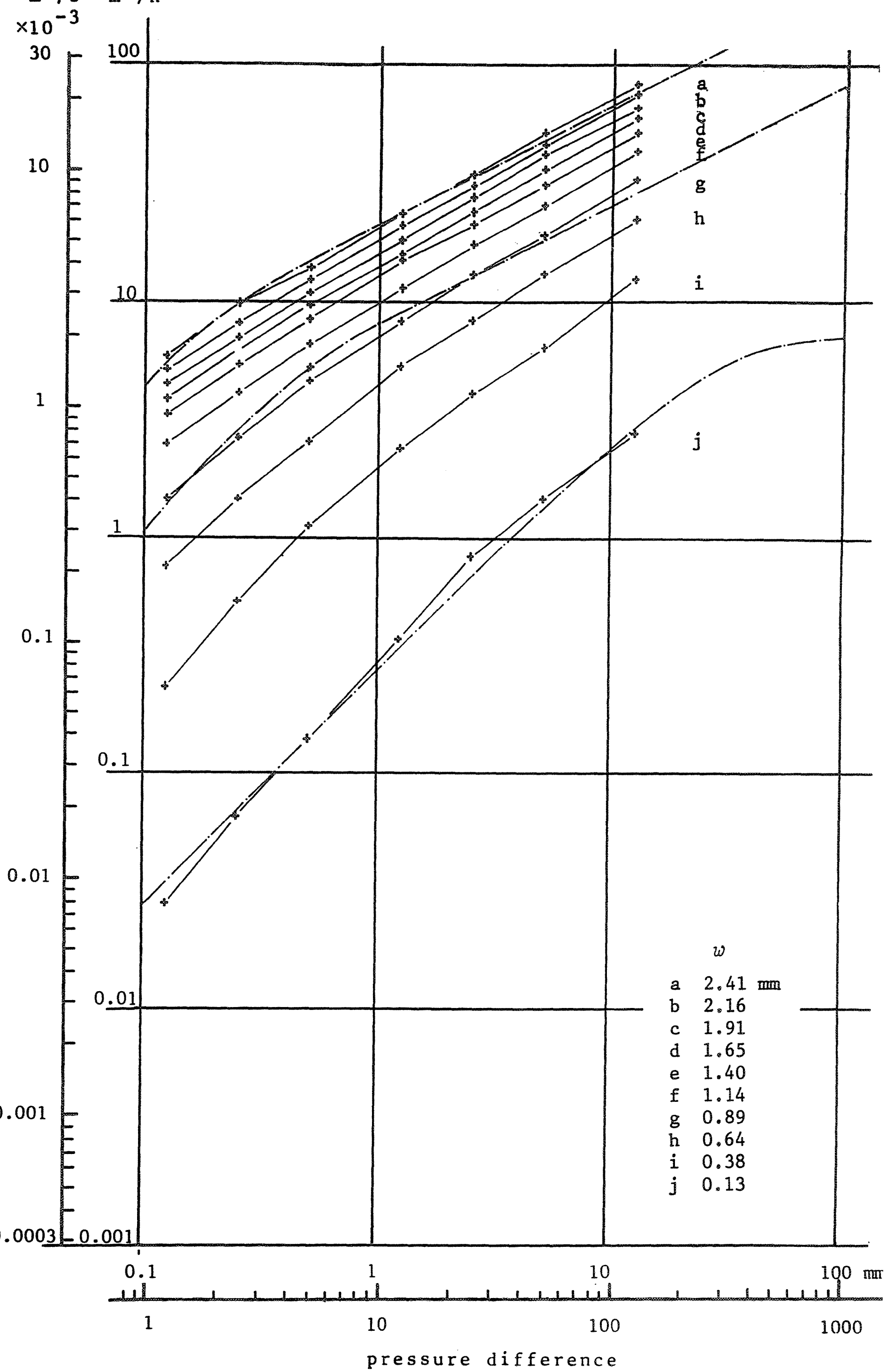


Fig. 2.8 a Relations between volume of flow and pressure difference
British standard metal windows

impossible, because, for instance, the friction factor changes when the flow is transformed from the laminar to the turbulent range. The average face clearance of a deformed air passage has only slight significance from the point of view of fluid dynamics. The width and depth of the most contracted part of a passage exert a dominant influence on the flow of air through the passage. When contractions and expansions are repeated over the cross sectional area of a passage, the acceleration term in Equation 2.6 must be multiplied by the number of times n that the contractions are repeated,

$$\Delta P = \frac{1}{2} (n) \, g \, \rho \, v^2 + \{ \text{const.} \; \xi \, v + \text{const.}' \, (1 - \xi) \, v^2 \} \qquad 2.11$$

The average velocities were estimated in Equation 2.9 by adopting the face clearances and the total length of the contracted parts of the sections of the British standard windows. In the central range of face clearances, the estimated velocities were in satisfactory agreement with the experimental velocities. In the case of small face clearances, there was very great deviation between the experimental and estimated velocities. The same was found in conjunction with flow through the aluminium gaps. The transient state at the edges and the deformed parts in a passage may have an effect on the deviation.

The relationship between the proportionality constants a and the constants A in the exponent for the British standard windows is also plotted in Figure 2.6. The points relating to the same types of sections show, in this case also, a strong tendency for the constants A to vary in proportion to the proportionality constants a. The disposition of the points between the British standard windows and the aluminium gaps is probably due to the difference in the pressure differences employed for the two groups. The depths and the shapes of the passages also seemd to affect the relationship.

2.3.2 Flow of air through some other types of windows

A series of air leakage measurements were performed by Birkeland and his associates (1955, 1958) on a number of wooden windows up to a pressure difference of 700 Pa. Most of the window types employed were Norwegian double windows in common use. Measurements were made on each type with one of the double windows closed, both of the windows closed, and a weather strip attached to one of the windows. Their report gives the diagram of air leakage-pressure difference curves for each window type for the various cases. For each case, the air flow rates were read from the figures at every 100th Pa, and the proportionality constants a and the constants A were calculated by the least squares method. The results obtained are shown in Figure 2.9. The proportionality constants of the cases where weather strips had been applied were in the range of 0.01 and 0.02×10^{-3} $m^3/m \; s \; Pa^{1/\beta}$, and those of the cases without weather strips in the range 0.05 - 0.10×10^{-3} $m^3/m \; s \; Pa^{1/\beta}$ for both single and double windows.

Various types of wooden and metal windows were examined in the USA and published in ASHRAE Transactions and some other bulletins. Several cases, with flow rates given under a wide range of pressure differences, were selected, and the proportionality constants and the constants in the exponent equation calculated from these by the method of least squares. The results are shown in Fig. 2.9. The source references and the appropriate conditions are listed in Table 2.2.

The reference infiltration of double-hung wooden windows and light steel casements, and their average infiltration ratio at pressure difference between 12.5 and 171 Pa are given in the report from the National Bureau of Standards, USA. The constant A of the exponent equation was calculated from the average infiltration ratio, and the proportionality constants from the minimum average and maximum reference infiltrations for various crack conditions in double-hung wooden windows and light metal casements. The results are shown in Figure 2.9 in the form of a horizontal span for each of the double-hung wooden windows and light metal casements.

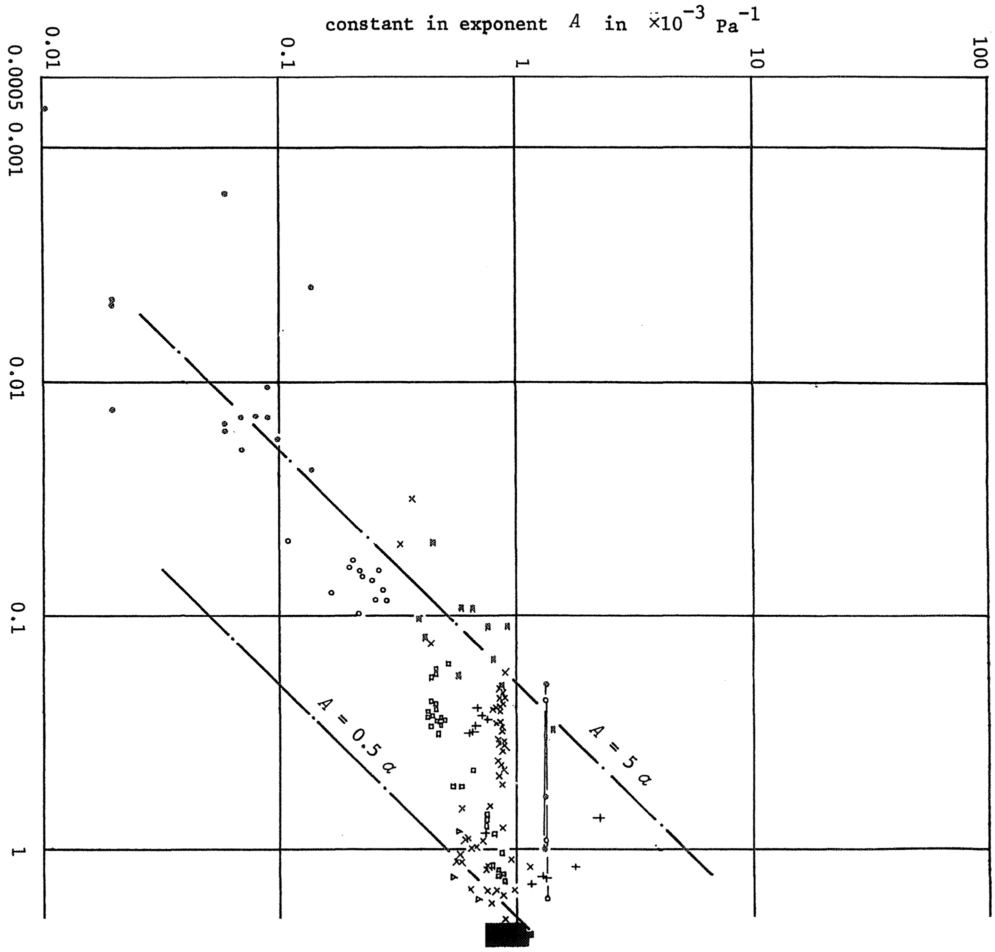


Fig. 2.9 Relation between proportionality coefficients and the constants in expon[redacted]

- ● Norwegian BRI, weather stripped
- o " " non-WS
- ⌘ University of Minnesota
- ◻ ASHRAE T. A. 686 & 704
- × " 779
- + " 817
- △ " 1760, revolving doors
- —o—NBS, double hung wood window
- —●— " , light steel casement

Table 2.2 Conditions of air leakage measurements

kind of crack	weather strip	ranges of experiments width mm	depth mm	pressure Pa	experimentarist	year	source
aluminium gaps		0.075 to 1.05	7.9 to 30.0	1 to 750	own	1974	
British standard metal and wood windows	with and without	0.127 to 2.413	36.8(11.4) and 34.3(12.7)	1.24 to 124	D.A. Thomas, et al	1953	JIHVE
Norwegian wood windows	with and without			to 686	R. Wigen	1958	report No. 28 Norwegian BRI
Joints of exterior wall panel of wood	fibrous filler			to 686	Ø. Birkeland, et al.	1958	report No. 15 Norwegian BRI
double hung wood windows	with and without	to 6.35		to 374	F.C. Houghten et al.	1924	T.A. No. 686 ASHRAE
double hung wood sash	ditto	to 6.35		to 374	C.C. Schrader	1924	T.A. No. 704 ASHRAE
metal and wood windows	ditto	to 6.35		to 374	A.C. Armstrong	1927	T.A. No. 779 ASHRAE
revolving doors	with seals			to 249	L.F. Schutrum et al.	1961	T.A. No. 1760 ASHRAE
wood and metal casement	with and without			to 249	C.E. Lund et al.	1952	Bulletin No. 35 Univ. of Minnesota
double hung wood windows light steel casement	ditto			to 174	E.F. Coleman et al.	1940	report BMS45 NBS

Infiltration of air through revolving doors was measured by Schutrum and his associates. Although the clearance itself is larger than in the case of ordinary windows, the measurement was made with both new and worn seals. As will be seen in Figure 2.9, the proportionality constant for revolving doors is in the group of large constants.

Most of the proportionality constants obtained in the American measurements are in the range 0.1 - 3.0 $\times 10^{-3}$ m^3/m s $Pa^{1/\beta}$, including windows both with and without weather strips. The constant *A* is mostly situated between 0.1 -1.5 $\times 10^{-3}$ Pa^{-1}. These figures are very large compared with the Norwegian results in which *A* was mostly 0.1 $\times 10^{-3}$ Pa^{-1} for windows with weather strips and 0.3 $\times 10^{-3}$ Pa^{-1} for windows without weather strips.

The constants *A* calculated from the Norwegian results are very low compared with the results relating to other windows. The combinations of the proportionality constants a and the constants *A* were recalculated by the same method, but with the range of pressure differences limited to below 400 Pa. As a result of this change, the proportionality constants showed increases of several per cent and the exponent constants increases of up to 10-40%. But their relative positions were even lower than those of the other windows.

Air infiltration through grooves in wooden wall structures, which were filled with various fibrous materials, was also measured in Norway. The constants calculated from the results relating to the grooves are plotted in Figure 2.9. The results show that both the proportionality constants a and the constants *A* are very much smaller than those for window cracks. There has recently been an increasing demand for more impermeable exterior wall structures, and sealing materials are extensively applied. This will result in a reduction in the sizes of cracks, and the flow of air through such cracks will have to be treated in the same way as air flow through capillary or porous materials.But even in such cases the theory shows that the flow is linearly proportional to the pressure gradient through such a material, see Carman (1956), and the value of the exponent approaches unity very closely.

When a high pressure difference acts on a window, this may be displaced in its frame. In some of the measurements, the flow rate - pressure differential curves to logarithmic scale showed steeper slopes in the higher pressure difference ranges than in the lower ranges. This may be due to the cracks being opened up by the displacement. But the opposite case, that of the cracks being reduced, could not be discerned from observation of the flow rate - pressure difference curves. All the results have therefore been plotted in Figure 2.9 without distinction.

2.4 Proposal of a function for the exponent

As seen in Figure 2.6, the change in the constant β for different cracks shows a great dependence on the proportionality constant a as expressed in the following equation :

$$A = \text{const.}\ a \qquad 2.12$$

These lines are plotted in Figures 2.6 and 2.9 with the values of the constant equal to 0.5, 5.0 and 50.0. Most of the points are contained within the area bounded by the lines 0.5 a and 50.0 a, except for the fact that the points relating to very fine cracks tend to extend to the left of this area. The line 5.0 a is situated in the centre of the area. In view of this, the following equation is used to cover a considerable range of variations in the the exponent in the air infiltration equation :

$$\beta = 2.0 - e^{-5.0\, a\, \Delta P} \qquad 2.13$$

The values of the exponent β and the rates of air flow are listed in ranges of pressure difference and proportionality constant in Table 2.3. It is to be noted that a considerably lower value must be used for the exponent when the flow resistance of a crack is high.

The value of the exponent does not exceed 1.2 for air leakage through a crack of large flow resistance, such as weather stripped windows, under pressure differences up to 500 Pa. When the pressure difference is 500 Pa, the air leakage calculated by applying 1.2 to the exponent is 182 % larger than that calculated by conventionally putting the exponent equal to 1.5.

In order to illustrate the effect of the exponent, the ventilation system of the tall block referred to in 1.5 will again be examined by the network method. When there is a stack effect on the block, the pressure difference at the exterior wall is changed by the elevation, and consequently the level of air leakage at the the window cracks also change. The change is first calculated using the conventional value of 1.5 for the exponent in the air leakage equation. The results are shown in Figure 2.10 a. The proposed equation 2.13 was then applied in the air leakage calculation by the network method. The effective pressure of the fan was assumed to be constant independent to the flow rate for the convenience of the comparison. The results are shown in Figure 2.10 b.

The ventilation of the flat on the first floor is calculated to be 0.066 m^3/s in case a, but the ventilation of the same flat becomes 0.073 m^3/s, when the proposed equation is applied to the exponent of the pressure term of the infiltration equation. Besides, when a smaller value is applied to the exponent, the apparent resistance of air flow becomes smaller than when a larger value is used in the pressure range of the examination. Consequently, the pressure difference in the flat on the first floor is only -13 Pa to the outside in case b, when the difference is 38 Pa in case a.

Table 2.3 Exponent and volume of flow through various proportionality coefficients and pressure differences

PROP. COEF.	PRESSURE DIFFERENCE IN PA																				
	2.	4.	6.	8.	10.	20.	30.	40.	50.	60.	70.	80.	90.	100.	150.	200.	250.	300.	350.	400.	500.
0.001	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.1	0.1	0.1	0.1	0.1	0.1	0.2	0.2	0.3	0.3	0.4	0.5
0.002	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.01
	0.0	0.0	0.0	0.0	0.0	0.0	0.1	0.1	0.1	0.1	0.2	0.2	0.2	0.2	0.4	0.5	0.6	0.7	0.9	1.0	1.2
0.005	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.01	1.01	1.01	1.01	1.01
	0.0	0.0	0.0	0.0	0.0	0.1	0.1	0.2	0.2	0.3	0.3	0.4	0.4	0.5	0.7	1.0	1.2	1.4	1.7	1.9	2.3
0.007	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.01	1.01	1.01	1.01	1.01	1.01	1.02
	0.0	0.0	0.0	0.1	0.1	0.1	0.2	0.3	0.4	0.4	0.5	0.6	0.7	0.7	1.1	1.4	1.8	2.1	2.4	2.7	3.3
0.010	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.01	1.01	1.01	1.01	1.02	1.02	1.02
	0.0	0.0	0.1	0.1	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0	1.4	1.9	2.3	2.8	3.2	3.6	4.3
0.025	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.01	1.01	1.01	1.01	1.01	1.01	1.02	1.02	1.03	1.04	1.04	1.05	1.06
	0.0	0.1	0.1	0.2	0.2	0.5	0.7	1.0	1.2	1.5	1.7	1.9	2.1	2.4	3.4	4.4	5.3	6.1	6.9	7.6	8.8
0.050	1.00	1.00	1.00	1.00	1.00	1.00	1.01	1.01	1.01	1.01	1.02	1.02	1.02	1.02	1.04	1.05	1.06	1.07	1.08	1.10	1.12
	0.1	0.2	0.3	0.4	0.5	1.0	1.5	1.9	2.4	2.8	3.3	3.7	4.1	4.5	6.3	7.8	9.1	10.2	11.1	11.9	13.0
0.075	1.00	1.00	1.00	1.00	1.00	1.01	1.01	1.01	1.02	1.02	1.03	1.03	1.03	1.04	1.05	1.07	1.09	1.11	1.12	1.14	1.17
	0.1	0.3	0.4	0.6	0.7	1.5	2.2	2.8	3.5	4.1	4.7	5.3	5.8	6.4	8.7	10.5	11.9	13.0	13.8	14.4	15.1
0.100	1.00	1.00	1.00	1.00	1.00	1.01	1.01	1.02	1.02	1.03	1.03	1.04	1.04	1.05	1.07	1.10	1.12	1.14	1.16	1.18	1.22
	0.2	0.4	0.6	0.8	1.0	1.9	2.9	3.7	4.6	5.3	6.1	6.8	7.4	8.1	10.7	12.6	14.0	14.9	15.6	16.0	16.2
0.250	1.00	1.00	1.01	1.01	1.01	1.02	1.04	1.05	1.06	1.07	1.08	1.10	1.11	1.12	1.17	1.22	1.27	1.31	1.35	1.39	1.46
	0.5	1.0	1.5	2.0	2.4	4.7	6.6	8.4	10.0	11.4	12.6	13.7	14.6	15.4	18.0	19.2	19.4	19.3	18.9	18.4	17.4
0.500	1.00	1.01	1.01	1.02	1.02	1.05	1.07	1.10	1.12	1.14	1.16	1.18	1.20	1.22	1.31	1.39	1.46	1.53	1.58	1.63	1.71
	1.0	2.0	2.9	3.8	4.7	8.7	11.9	14.5	16.6	18.2	19.4	20.4	21.2	21.7	22.7	22.4	21.7	20.9	20.2	19.6	18.8
0.750	1.01	1.01	1.02	1.03	1.04	1.07	1.11	1.14	1.17	1.20	1.23	1.26	1.29	1.31	1.43	1.53	1.61	1.68	1.73	1.78	1.85
	1.5	2.9	4.3	5.7	6.9	12.3	16.2	19.1	21.2	22.6	23.7	24.3	24.8	25.0	24.9	24.1	23.2	22.6	22.1	21.9	21.7
1.000	1.01	1.02	1.03	1.04	1.05	1.10	1.14	1.18	1.22	1.26	1.30	1.33	1.36	1.39	1.53	1.63	1.71	1.78	1.83	1.86	1.92
	2.0	3.9	5.7	7.4	9.0	15.4	19.8	22.7	24.6	25.8	26.6	27.0	27.2	27.2	26.6	25.7	25.1	24.8	24.7	24.9	25.5

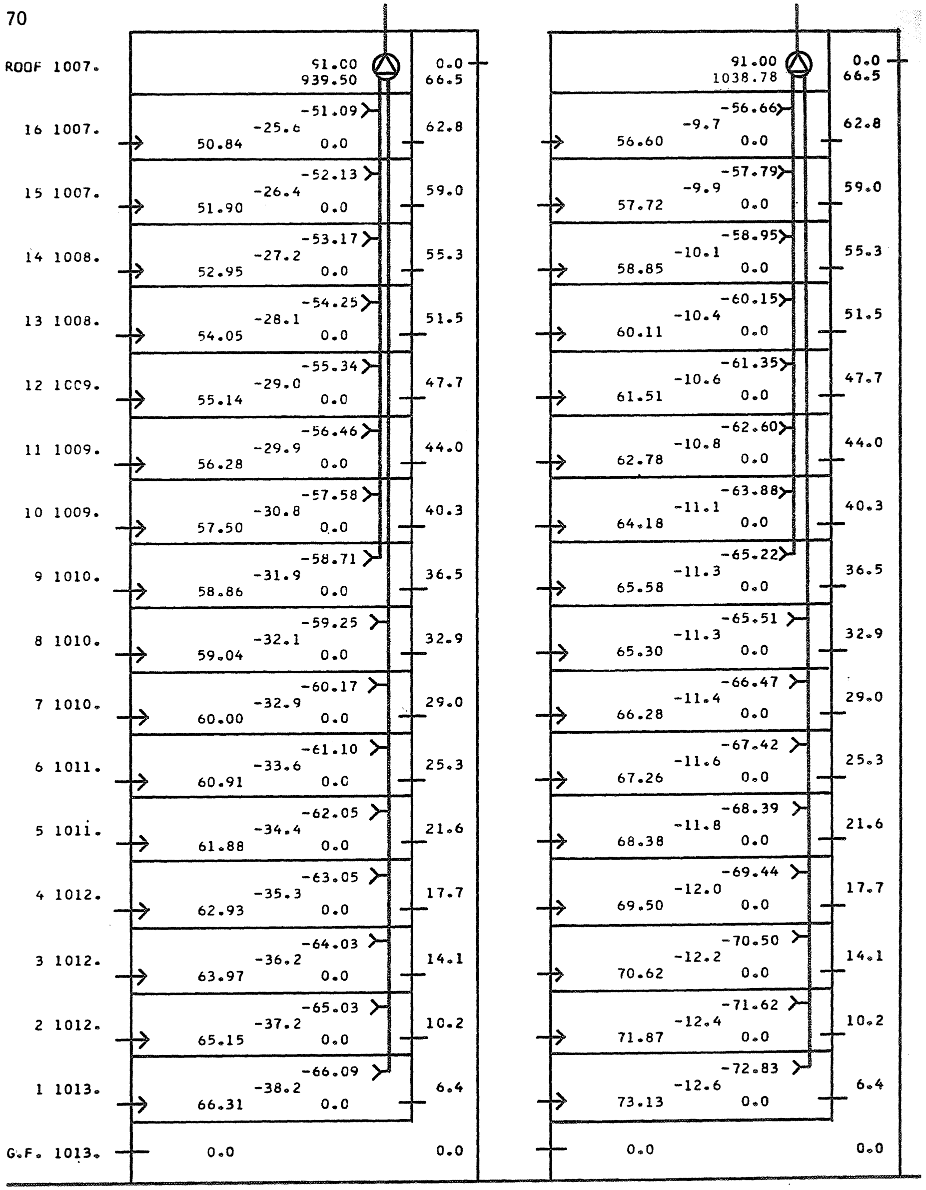
PROP. COEFFICIENT IN X10^{-3} M^3/M S AT 1 PA

upper row : exponent
lower row : volume of flow in x10^{-3} m^3/m s

3 A method of measuring air movement in a dwelling

The importance of the pattern of air movement in a building divided into a number of rooms will be recognized by all who are engaged on ventilation. The reasons why ordinary theories of flow measurement cannot be applied in this case are that the pressure differences between rooms are relatively small, and the air passages are irregular in shape. The natural conditions of flow may easily change when equipment for the measurement of flow rates or pressure differences is installed in the building. A number of air flow measurements have been carried out by various methods with the intention of clarifying the flow patterns between rooms. This problem has received stringent treatment, particularly in conjunction with the ventilation of hospitals.

The tracer gas technique is considered to be the most convenient one for use in such measurements without disturbance to the natural conditions of flow. The method of deciding the air changes in a room from the decay curve of a tracer gas has been applied for ventilation measurements in spaces of different sizes. But, in the field of ventilation engineering, application of the method has remained at a relatively undeveloped stage. The pattern of air flow in a house can be obtained when the simple treatment is repeated room by room. But this measure requires considerable exertion and a long measurement period. And it is possible for the conditions of flow to change over such a long period. In this chapter, the theory relating to the variation in concentration of a tracer gas is extended to a series of rooms, and an analytical method for constant air flows is introduced by employing newly developed techniques of numerical analysis and computer application.



a. exponent $\beta = 1.5$ b. exponent $\beta = 2.0 - e^{-5\,a\,\Delta P}$

conditions : same to Fig. 1.5 b except outdoor temperature -10 °C

legends : see Fig. 1.5 on page 35

Fig. 2.10 Comparison of calculated air leakage by conventional and presented exponents

3.1 The tracer gas technique

3.1.1 Tracer gases and detectors

Tracer gases must have the following characteristics:

a. The content of the gas in ordinary air must be relatively small, and there must be no source of the same gas in the building concerned.

b. It must be possible for the low concentration to be accurately assessed and detected.

c. The density of the gas must be as near as possible to that of air.

d. The gas must not react with the constituents of air.

e. The gas must not react with, or be adsorbed onto, the surfaces of walls, furniture, clothes, etc.

f. The gas must not be harmful for the human body.

g. The gas must not be flammable.

h. The gas must be easy to handle, easily available and inexpensive.

Many kinds of gas have been employed for tracer purposes in various ventilation engineering field measurements. The following gases are the major tracers in use at present:

Helium (0.555), water vapour (0.62), hydrogen (0.695), propane (1.56), carbon dioxide (1.529), nitrous oxide (1.530), sulphe dioxide (2.180), freon 12 (3.931)

The figures in brackets are the ratios of the specific densities of the gases to that of air at $0^{o}C$ and atmospheric pressure. Radioactive tracer gases such as Algon 41, Krypton 85 and Xenon 133 may be included in the group of tracer gases, but they are considered unsuitable for ventilation measurements in dwellings because of radiation hazards.

The methods for the detection of tracer gas concentrations are divided into chemical and physical methods. Continuous or rapidly repeating measurement of the variation in concentration is required in ventilation measurement; in addition, a concentration meter used

in the field must be compact. For these reasons, either the thermal conductivity or thermal absorptivity of a gas is usually chosen as the principle of a concentration meter.

Each gas has its own thermal conductance characteristic. Therefore, provided that the temperature and pressure of the mixture are kept constant, the conductivity of a mixture of gases varies with the change in the composition of the mixture. In a thermal conductivity meter or katharometer, the change in the thermal conductivity of a mixture is detected by a hot wire or a thermistor as a change in the heat loss or temperature. See Coblentz (1957). In consequence, the change in concentration is indicated as an electrical signal.

An infrared gas analyser detects the composition of a mixture of gases by the difference in the absorptivity of an infrared beam. This technique is gaining increasing popularity. The sensing element of this analyser is not in direct contact with the sample gas, and this may be a favourable characteristic for long-term application of the instrument.

3.1.2 Development of tracer gas application for ventilation measurements

The most intuitive application of a tracer gas technique is literally to observe the movement of the tracer which is transported by the movement of air. In most cases, the direction or pattern of air flow can be deduced from this method, but a flow velocity cannot. Visible smoke tracers can be classified in this category. An example of flow pattern measurement by application of a radioactive tracer gas is found in the report on ventilation measurements by Howland and his associate (1960).

The first application of the tracer gas technique for quantitative measurement of ventilation was employed by Max von Pettenkofer. The decay in the concentration of a tracer gas in a room after production of the gas had ceased is expressed by an exponential function as

$$C_{\tau} = C_0 \cdot e^{-n \cdot \tau} \qquad 3.1$$

where C_0 and C_τ are the concentrations of a tracer gas at time 0 and time τ respectively, and n is the air change number of the room.

If the concentrations at times 0 and τ are known, then the air change number of the room is calculated as follows:

$$n = \frac{1}{\tau} \ln \frac{C_0}{C_\tau} \qquad 3.2$$

The equation means that the logarithm of time decay in concentration of a tracer gas is proportional to the ventilation number. The tracer gas production need not be known and the concentrations in Equation 3.2 may be relative values. When homogeneous distribution of a tracer gas is assured in a room, a steady result can be obtained by plotting the concentration change on semi-logarithmic paper so that it is applicable for ventilation measurement in spaces with a range of volumes.

The measurements of air flows between connected rooms by tracer gas techniques have been investigated by many researchers. Several major works are introduced in the following.

When a tracer gas is blown off for an instant in one of two coupled rooms, and if air flows from the producing room to the secondary room, then there is a time lag in the variation in concentration in the secondary room in relation to that in the producing room, and the curve of concentration in the secondary room exhibits a peak value. Dick (1949) settled the equation for calculation of the air flow rate from the primary room to the secondary room by using the time lag and the ratio of the peak concentration $C_{max,2}$ in the secondary room to the initial concentration C_{01} in the primary room as

$$v_{21} = V_2 \frac{C_{max,2}}{C_{01}} \frac{n_2 - n_1}{\left(\frac{n_1}{n_2}\right)^{\frac{n_1}{n_2-n_1}} - \left(\frac{n_1}{n_2}\right)^{\frac{n_2}{n_2-n_1}}} \qquad 3.3$$

where v_{21} is the air flow rate from the primary to the secondary room, V_2 the volume of the secondary room, n_1 and n_2 the air change numbers in the two rooms : the air change numbers can be decided by independent measurements in the respective rooms.

Jenning and his associate (1971) performed a mathematical analysis of the concentration of contamination in a room, in which the rates of air movements, production of contamination, concentration of supplied air could all be treated as changeable in time. The mathematical method of expressing the variation in concentration can be applied identically to a tracer gas by replacing contamination source by tracer gas production. Some examples of contamination were numerically analysed by reference to a function of the ventilation condition. In their paper, a mixing factor of contamination is mentioned in relation to the efficiency of ventilation.

The method of concentration analysis in a room which is subject to harmful gaseous pollution is also introduced into the equation of mass balance by Muller (1961, 64). Intermittent gas production and intermittent ventilation were analysed in his work.

Several measuring techniques by constant production of a tracer gas were carried out. Baird (1969) measured air leakages between operating theatre units in a hospital from the ratio of the steady concentrations in the source room and receiving room, by producing tracer gas constantly for several hours. Foord and his associates (1973) erected a monitoring system for the spread of contaminated air in a hospital.

In a long period of ventilation analysis in a room, the automatic control of production of a tracer gas to keep a constant concentration has been carried out in the field research station of Holzkirchen near Munich.

In ventilation measurements in a series of rooms, there are reports that simultaneous production of several kinds of gases is under investigation by several research institutes.

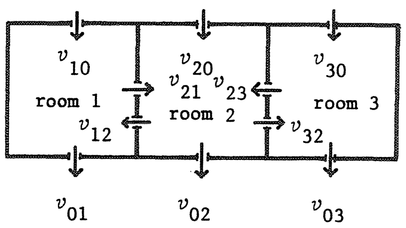


Fig. 3.1 Air flows between three rooms

3.1.3 Extension of the tracer gas technique

The tracer gas technique of ventilation measurement has been developed in the ways outlined in the previous section. The technique will be further developed by considering the theory on the variation in concentration of a tracer gas.

The finite differential equation for the mass balance of a tracer gas in a series of rooms is constructed as follows. Let us consider three rooms with air flowing between them, as shown in Figure 3.1. The air flows between the rooms which are assumed to be constant during measurement are designated v_{ij} where the first subscript indicates the room in which the air is received, and the second one the room from which the air flows. Let the concentrations of the gas in rooms 1, 2 and 3, at time τ_0, be C'_{01} , C'_{02} and C'_{03} respectively. The concentrations at time τ_1 will then be C'_{11} , C'_{12} and C'_{13} , the duration intensity of tracer gas production in room 2 being G_2 . The equilibrium equation during the time interval for the gas in room 2 is

$$C'_{12} V_2 = C'_{02} V_2 + G_2 \Delta\tau + v_{21} C'_{01} \Delta\tau + v_{23} C'_{03} \Delta\tau + v_{20} C'_0 \Delta\tau - \left(v_{12} + v_{32} + v_{02} + \frac{G_2}{\rho} \right) C'_{02} \Delta\tau \qquad 3.4$$

where room number 0 designates outdoors, and the concentration of the tracer gas in the outside air is expressed by C'_0. $\Delta\tau$ is the time interval between τ_0 and τ_1, V_2 the volume of room 2, and ρ the density of the tracer gas. If, compared with the ventilation rate of the room, the volume of the released gas is small, the term $\frac{G_2}{\rho}$ can be ignored. Equation 3.4 is the linear equation for air flows v_{ij}. The same type of equation can be drawn up for every room. If concentration measurement is repeated in the three rooms several times at certain intervals, then the same type of equation can be drawn up at every time interval, and the air flows can be found by solving these as a series of linear equations.

In solving the above serial equations for air flows, there are two methods which make it easier to distinguish the constant terms in the equations and consequently help in obtaining reliable solutions for the air flows. The first method is to have a periodical change in gas production in each room. By virtue of the character of an exponential function, the change in time in

gas concentration becomes very slight after a long period of constant production, or suspension of production, of the gas. The periodical change in the rate of gas production will enable clear separation of the constant terms in the serial equations to be effected. The second method is to have time differences in the phases of variation in concentration between the rooms. When air flows into a room from several rooms, the rooms from which the incoming air flows emanate can be distinguished by this means.

Technically, it appears very difficult to obtain reliable results of air flows by the finite differential equations, particularly when the time interval between concentration measurements is not short and the concentration measurement is not very accurate, since there are many obstructions in the process of gas production and gas distribution and in concentration measurement. The accuracy of air flow calculation can be enhanced when the variation in concentration is expressed by a transient equation instead of a finite differential equation. The equation describing the variation in concentration in a room in a series of adjacent rooms is rather a complicated one, but solutions with a precision sufficient for engineering purposes can be obtained by treating the equations with the recently developed numerical analysis technique and with the aid of a high speed computer. Also, the influences of the above obstructions will be minimised when a sufficient number of the concentration measurement data are treated by employing a statistical technique in the calculations.

If tracer gas concentrations in every room are measured in terms of the difference from the concentration of the gas in the outside air, the absolute concentration $C'_{\tau i}$ must be expressed as the sum of the measured concentration $C_{\tau i}$ and the concentration in the outside air, C'_0, as

$$C'_{\tau i} = C_{\tau i} + C'_0 \qquad 3.5$$

where the first subscript of the concentration indicates the time and the second one the room.

3.2 Equation of tracer gas production

Tracer gas must be produced in every room in order to distinguish the variation in concentration in every room as clearly as possible. The intensity and period of gas production should be altered by some simple means with regard to the conditions in the building under study.

Carbon dioxide gas is evaporated by electric heat from a piece of dry ice which is placed in a thermally insulated vessel. The apparatus is described in 4.1.1. The rate of production is controlled by the supply of electric heat.

When the intensity of gas production at a steady unit heat supply is U, the steady gas production G_h due to an arbitrary steady heat supply H will be

$$G_h = U \cdot H \qquad 3.7$$

When constant heat supply begins, there is a delay in the intensity of gas production due to the heat capacity of the electric heater and the vessel, as shown in Figure 3.2. It is supposed that when heat is not supplied before time τ_0 and unit heat supply begins at this time, the change in gas production G_h at an arbitrary time τ is expressed by the following equation:

$$\begin{array}{lll} \tau \leqq \tau_0 & H = 0 & G_h = 0 \\ \tau > \tau_0 & H = 1 & G_h = 1 \cdot U\,(1 - e^{-B(\tau-\tau_0)}) \end{array} \qquad 3.8$$

When an arbitrary heat H is supplied instead of the unit heat supply, the intensity of gas production after time τ_0 will be as follows:

$$G_h = H \cdot U\,(1 - e^{-B(\tau-\tau_0)}) \qquad \tau > \tau_0 \qquad 3.9$$

The constant B is the time constant of the gas producer and has the dimension sec^{-1}.

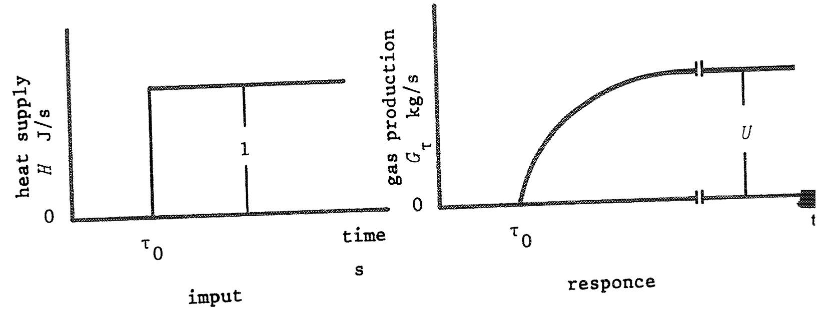
Substitution of this relationship into Equation 3.4 gives

$$\begin{aligned} C_{12} V_2 = {} & C_{02} V_2 + G_2 \Delta\tau + v_{21} C_{01} \Delta\tau + v_{23} C_{03} \Delta\tau \\ & - (v_{12} + v_{32} + v_{02} + \frac{G_2}{\rho}) C_{02} \Delta\tau \\ & + (v_{21} + v_{23} + v_{20} - v_{12} - v_{32} - v_{02}) C_{\bar{0}} \Delta\tau \end{aligned} \qquad 3.6$$

The last term of Equation 3.6 can be eliminated as long as the total amount of incoming air is equal to the total amount of outgoing air, provided that the volume of released tracer gas is negligibly small in comparison and the concentration in the outside air remains constant during a measurement. This means that the equation is independent of the concentration of tracer gas in the outside air. In the following, all equations are dealt with under this condition.

The part of the fifth term in brackets is the total rate of outward air flow in the room. In the following, the total rate of outward air flow in the room is expressed by $v_{\bar{0}2}$ as

$$v_{\bar{0}2} = v_{12} + v_{32} + v_{02} + \frac{G_2}{\rho}$$



$$\begin{array}{ll} \tau \leqq \tau_0 \quad H = 0 & G_h = 0 \\ \tau > \tau_0 \quad H = 1 & G_h = U(1-e^{-B(\tau-\tau_0)}) \end{array}$$

Fig. 3.2 The response of production of CO_2 gas to unit variation of heat supply

When the response of gas production to a sudden change in heat supply is expressed by Equation 3.9, the change in the intensity of gas production due to a series of sudden changes in heat suppl; is expressed as a summation of the responses which are the produc of all past changes in heat supply and the respective responses t a unit change in heat supply. For instance, when heat supply has changed as shown in Figure 3.3, the intensity of gas production at time τ, which is between τ_4 and τ_5, is expressed by the following equation:

$$G_\tau = G_p + (H_0 - 0)\,U(1 - e^{-B(\tau-\tau_0)}) + (H_1 - H_0)\,U(1 - e^{-B(\tau-\tau_1)}) + (H_2 - H_1)\,U(1 - e^{-B(\tau-\tau_2)}) + (H_3 - H_2)\,U(1 - e^{-B(\tau-\tau_3)}) + (H_4 - H_3)\,U(1 - e^{-B(\tau-\tau_4)}) \qquad 3.10$$

The freezing point of dry ice is -78.5°C, and the inside of the vessel is assumed to be kept at about this temperature. Heat will therefore penetrate through the insulated wall of the vessel and gas production will be caused by this. When an apparatus has been kept with dry ice inside it for a long time, the temperature and consequently gas production will reach a steady state. Let the intensity of heat penetration be H_p and the gas production due to this, G_p. It may be assumed that supply of heat began an infinitely long time before this, and the following equation is given for the gas production (the broken line in Figure 3-3).

$$G_p = H_p \cdot U\,(1 - e^{-(\infty)}) = H_p \cdot U \qquad 3.11$$

Let us also suppose that no heat had been supplied until time τ_1 when H_0 is zero, and the equation for gas production is generalised as follows:

$$G_\tau = G_p + G_h = G_p + \sum_{i=1}^{n} (H_i - H_{i-1})\,U(1 - e^{-B(\tau-\tau_i)})$$

$$\tau_{n+1} > \tau \geqq \tau_n \qquad 3.12$$

When heat supply is kept constant and gas is distributed alternately to every room at a constant rate from an apparatus by means of an exchange valve, the change in intensity of heat supply can be treated

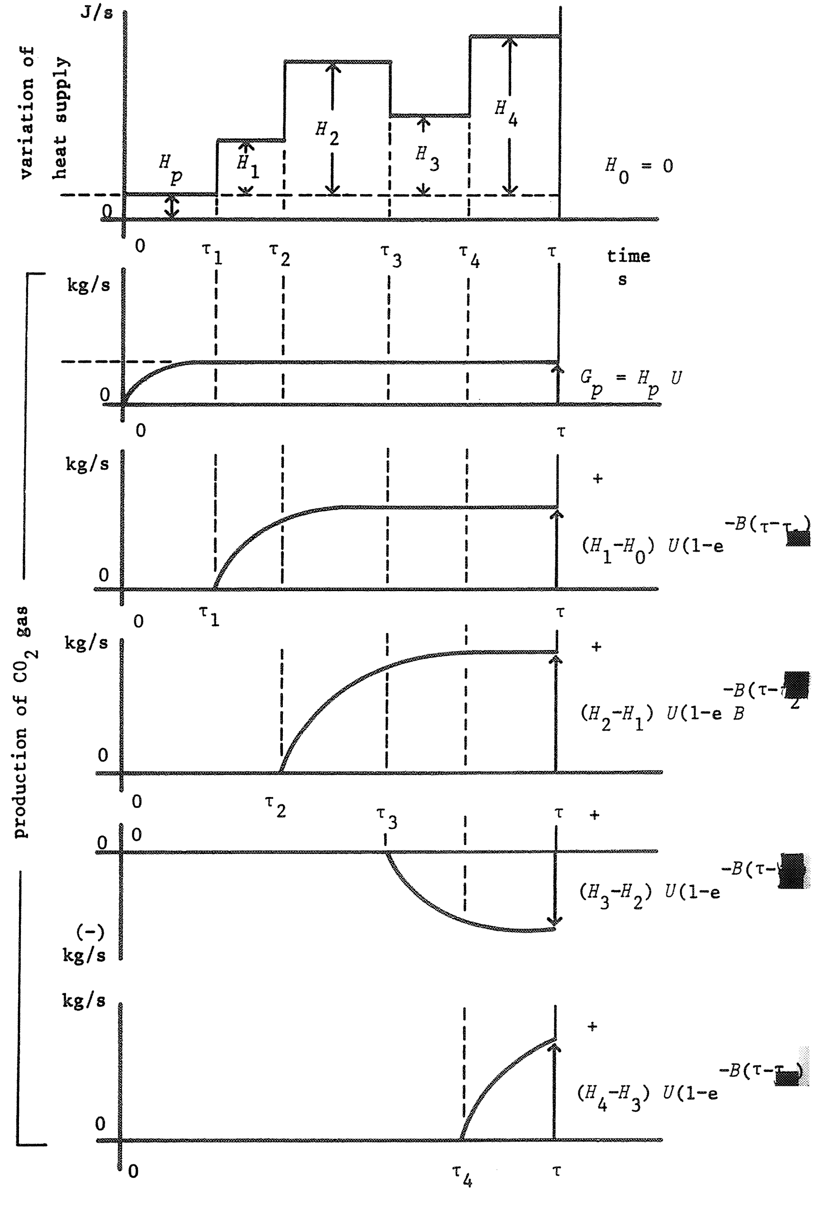


Fig. 3.3 Arbitrary square-wave heat supply and the production of CO_2 gas

as a term H_i which is the summation of the electric heat supply and the heat penetration, as

$$H_i = + (H_p + H_e) \quad \text{when the valve is open}$$

$$H_i = - (H_p + H_e) \quad \text{when the valve is closed} \qquad 3.13$$

At the same time, the response of gas production is considered to be very fast, so that the time constant B can be assumed to be infinite. Eventually, the following equation is applied for gas production by this type of control:

$$G_\tau = \sum_{i=1}^{\tau} (H_i - H_{i-1}) \, U \, (1 - e^{-\infty})$$

$$= (H_p + H_e) \cdot U \quad \text{when opning the valve} \qquad 3.14$$

$$= 0 \quad \text{when closing the valve}$$

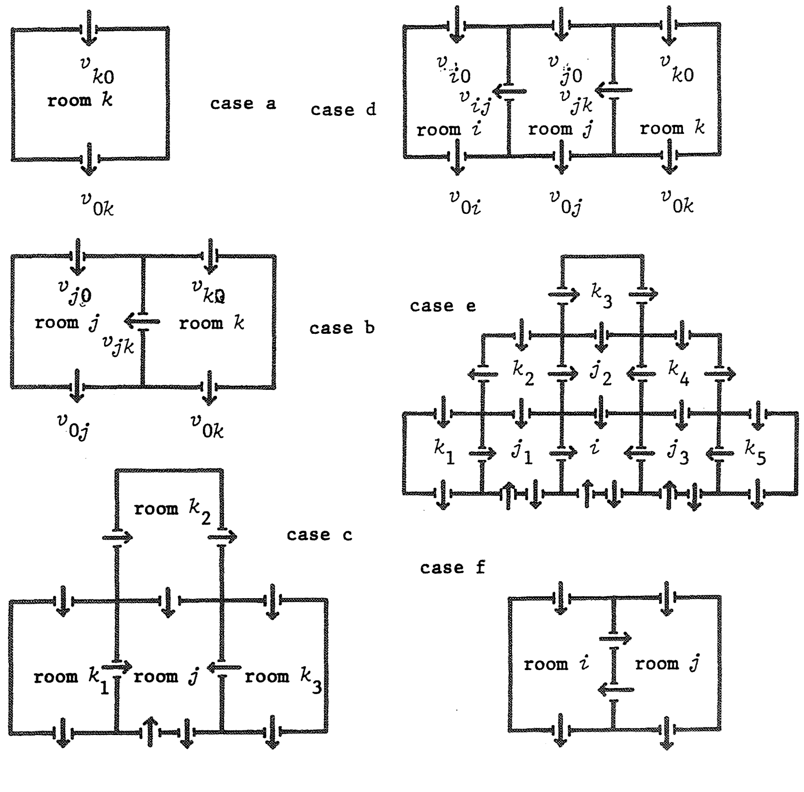


Fig. 3.4 Various cases of air flows

3.3 Theory of variation in concentration in a series of rooms

3.3.1 Single room

Suppose a room k which has a constant air change with the outside, as shown in Figure 3.4 a. The pressure and temperature of the system are kept constant for a sufficiently long time before and during a measurement. Let us suppose that a tracer gas is produced at intensity $G_{\tau k}$ in the room. Let the rate of air change and volume of the room be v_{0k} and V_k respectively. The flow balance equation 3.4 of the tracer gas in the room is modified as follows:

$$V_k \, \Delta C_k = G_{\tau k} \, \Delta\tau - v_{0k} \, C_k \, \Delta\tau \qquad 3.15$$

The air flowing into the room is here supposed to be instantly mixed with the room air, and the concentration in the room is homogeneous. The change in concentration of the gas in the room over a time interval $\Delta\tau$ is ΔC_k. This equation is converted into a differential equation of concentration as

$$V_k \frac{dC_k}{d\tau} + v_{0k} \, C_k = G_{\tau k} \qquad 3.16$$

The general solution of this equation is effected with the aid of a differential operator D as

$$\left(D + \frac{v_{0k}}{V_k} \right) C_k = \frac{G_{\tau k}}{V_k} \qquad 3.17$$

$$C_{\tau k} = e^{-\frac{v_{0k}}{V_k}\tau} \int e^{\frac{v_{0k}}{V_k}\tau} \, \frac{G_{\tau k}}{V_k} \, d\tau$$

If the intensity of tracer gas production in room k is expressed by equation 3.12, the intensity of gas production between time τ_0 and τ_1 is constant and equal to G_{pk}, and the concentration in room k between time τ_0 and τ_1 is then solved from equation 3.17 by substituting the concentration C_{0k} in room k at time τ_0 as the initial condition:

$$C_{\tau k} = C_{0k}\, e^{-\frac{v_{\bar{0}k}}{V_k}\tau} + \frac{G_{pk}}{v_{\bar{0}k}}\left(1 - e^{-\frac{v_{\bar{0}k}}{V_k}\tau} \right) \tag{3.18}$$

$$\tau_1 > \tau > 0$$

The concentration between time τ_1 and τ_2 is solved by substituting the following equation for the production of the gas $G_{\tau k}$:

$$G_{\tau k} = G_{pk} + (H_1 - H_0)\cdot U_k\left(1 - e^{-B_k(\tau-\tau_0)} \right) \tag{3.19}$$

and also by substituting the concentration at time τ_1 which is obtained from Equation 3.18 as

$$C_{\tau k} = C_{0k}\, e^{-\frac{v_{\bar{0}k}}{V_k}\tau} + \frac{G_{pk}}{v_{\bar{0}k}}\left(1 - e^{-\frac{v_{\bar{0}k}}{V_k}\tau} \right) + (H_1 - H_0)\,\frac{U_k}{v_{\bar{0}k}(v_{\bar{0}k} - B_k\cdot V_k)}\left\{ v_{\bar{0}k}\left(1 - e^{-B_k(\tau-\tau_1)}\right) - B_k\cdot V_k\left(1 - e^{-\frac{v_{\bar{0}k}}{V_k}(\tau-\tau_1)}\right)\right\} \tag{3.20}$$

$$\tau_2 > \tau \geqq \tau_1$$

Accordingly, the concentration between time τ_n and τ_{n+1} is generally solved as the following equation by substituting equation 3.12 for gas production and deciding the initial condition at each boundary of change in heat supply successively as

$$C_{\tau k} = C_{0k}\, e^{-\frac{v_{\bar{0}k}}{V_k}\tau} + \frac{G_{pk}}{v_{\bar{0}k}}\left(1 - e^{-\frac{v_{\bar{0}k}}{V_k}\tau} \right) + \sum_{m=1}^{n} (H_m - H_{m-1})\,\frac{U_k}{v_{\bar{0}k}(v_{\bar{0}k} - B_k\cdot V_k)}\left\{ v_{\bar{0}k}\left(1 - e^{-B_k(\tau-\tau_m)}\right) - B_k\cdot V_k\left(1 - e^{-\frac{v_{\bar{0}k}}{V_k}(\tau-\tau_m)}\right)\right\} \tag{3.21}$$

$$\tau_{n+1} > \tau \geqq \tau_n$$

This equation is expressed in the following by $f_1(G_{\tau k})$ as the result of gas production in the own room .

3.3.2 Influence of directly adjacent rooms

Assume two coupled rooms k and j as shown in Figure 3.4 b. In room k the concentration varies as expressed by Equation 3.21, and a constant rate of air v_{jk} flows from room k to room j. The tracer gas is produced in room j at a different pattern from that in room k, for instance as in rooms 1 and 2 in Figure 3.6. The production of tracer gas in room j is expressed by $G_{\tau j}$. The conservation equation of the tracer gas in room j is deduced in the same way as in Equation 3.4, and is changed into a differential equation as follows:

$$V_j \frac{dC_j}{d\tau} = G_{\tau j} + v_{jk}\, C_{\tau k} - v_{\bar{0}j}\, C_j \tag{3.22}$$

where $v_{jk}\, C_{\tau k}$ is the rate of flow of a tracer gas from room k to room j.

The concentration in room j at time τ is solved by the same process as that in room k. The solution is expressed as the **effects of the** gas production in room j and the adjacent room k as

$$C_{\tau j} = f_1(G_{\tau j}) + v_{jk}\cdot f_2(G_{\tau k}) \tag{3.23}$$

The term $f_1(G_{\tau j})$ is given here by substituting the constants of gas production in room j into Equation 3.21. The effect of gas production in room k on room j is

equation 3.24 : see page 86 3.24

When room j receives air from different rooms, as shown in Figure 3.4 c, the balance equation for the tracer gas will be

$$V_j \frac{dC_j}{d\tau} = G_{\tau j} + \sum_{k=1}^{k'} (v_{jk}\, C_{\tau k}) - v_{\bar{0}j}\, C_j \tag{3.25}$$

where $\sum_{k=1}^{k'}$ signifies the summation of all the effects due to air flows into room j.

The concentration in room j at time τ is solved as follows:

$$C_{\tau j} = f_1(G_{\tau j}) + \sum_{k=1}^{k'} \{ v_{jk}\, f_2(G_{\tau k}) \} \qquad 3.26$$

3.3.3 Influence of indirectly adjacent rooms

The room which is connected to the room under consideration indirectly by means of another room, as in the case of rooms i and k in Figure 3.4 d, is termed a secondary connected room. Suppose that air flows exist from room k to j and from room j to i, and that there is no air flow in the reverse direction. The concentration in room i is obtained by solving the following differential equation of tracer gas conservation in room i:

$$V_i \frac{dC_i}{d\tau} = G_{\tau i} + v_{ij}\, C_{\tau j} - v_{\bar{0}i}\, C_i \qquad 3.27$$

When room i receives air flows from a number of rooms, as shown in Figure 3.4 e, all the influences must be considered as follows:

$$V_i \frac{dC_i}{d\tau} = G_{\tau i} + \sum_{j=1}^{j'} (v_{ij}\, C_{\tau j}) - v_{\bar{0}i}\, C_i \qquad 3.28$$

The concentrations $C_{\tau j}$ in the directly connected rooms are obtained by substituting Equation 3.26. Finally, the concentration $C_{\tau i}$ in room i at time τ is expressed by the following equation:

$$C_{\tau i} = f_1(G_{\tau i}) + \sum_{j=1}^{j'} v_{ij} \left[f_2(G_{\tau j}) + \sum_{k=1}^{k'} v_{jk} \{ f_3(G_{\tau k}) + \Omega \} \right] \qquad 3.29$$

Here, the effect of gas production in a secondary connected room k is

equation 3.30 : see page 88 3.30

$$f_2(G_{\tau k}) = C_{0k} \frac{V_j}{v_{\bar{0}j}V_k - v_{\bar{0}k}V_j} \left(e^{-\frac{v_{\bar{0}k}}{V_k}\tau} - e^{-\frac{v_{\bar{0}j}}{V_j}\tau} \right) + \frac{G_{pk}}{v_{\bar{0}j}V_k - v_{\bar{0}k}V_j} \left\{ \frac{V_k}{v_{\bar{0}k}} \left(1 - e^{-\frac{v_{\bar{0}k}}{V_k}\tau} \right) - \frac{V_j}{v_{\bar{0}j}} \left(1 - e^{-\frac{v_{\bar{0}j}}{V_j}\tau} \right) \right\}$$

$$+ \sum_{m=1}^{n} (H_m - H_{m-1}) \Big[\frac{1}{v_{\bar{0}j}V_k - v_{\bar{0}k}V_j} \left\{ \frac{V_k}{v_{\bar{0}k}} \left(1 - e^{-\frac{v_{\bar{0}k}}{V_k}(\tau-\tau_m)}\right) - \frac{V_j}{v_{\bar{0}j}} \left(1 - e^{-\frac{v_{\bar{0}j}}{V_j}(\tau-\tau_m)}\right) \right\} \qquad 3.24$$

$$- \frac{1}{v_{\bar{0}k} - B_k V_k} \left\{ \frac{1}{v_{\bar{0}j} - B_k V_j} \left(e^{-B_k(\tau-\tau_m)} - e^{-\frac{v_{\bar{0}j}}{V_j}(\tau-\tau_m)} \right) - \frac{V_k}{v_{\bar{0}j}V_k - v_{\bar{0}k}V_j} \left(e^{-\frac{v_{\bar{0}k}}{V_k}(\tau-\tau_m)} - e^{-\frac{v_{\bar{0}j}}{V_j}(\tau-\tau_m)} \right) \right\} \Big]$$

$$f_2(G_{\tau k}) = \frac{C_{0k}}{V_j} \tau\, e^{-\frac{v_{\bar{0}j}}{V_j}\tau} + \frac{G_{pk}}{v_{\bar{0}k}} \left\{ \frac{1}{v_{\bar{0}j}} \left(1 - e^{-\frac{v_{\bar{0}j}}{V_j}\tau}\right) - \frac{\tau}{V_j} e^{-\frac{v_{\bar{0}j}}{V_j}\tau} \right\}$$

$$+ \sum_{m=1}^{n} (H_m - H_{m-1}) \Big[\frac{1}{v_{\bar{0}k}} \left\{ \frac{1}{v_{\bar{0}j}} \left(1 - e^{-\frac{v_{\bar{0}j}}{V_j}(\tau-\tau_m)}\right) - \frac{\tau}{V_j} e^{-\frac{v_{\bar{0}j}}{V_j}(\tau-\tau_m)} \right\} \qquad 3.32$$

$$- \frac{1}{v_{\bar{0}k} - B_k V_k} \left\{ \frac{1}{v_{\bar{0}j} - B_k V_j} \left(e^{-B_k(\tau-\tau_m)} - e^{-\frac{v_{0j}}{V_j}(\tau-\tau_m)} \right) - \frac{\tau}{V_j} e^{-\frac{v_{0j}}{V_j}(\tau-\tau_m)} \right\} \Big]$$

$$f_3(G_{\tau k}) = \frac{C_{0k}V_k}{v_{\bar{0}j}V_k - v_{\bar{0}k}V_j}\left\{\underbrace{\frac{V_k}{v_{\bar{0}i}V_k - v_{\bar{0}k}V_i}\left(e^{-\frac{v_{\bar{0}k}}{V_k}\tau} - e^{-\frac{v_{\bar{0}i}}{V_i}\tau}\right) - \frac{V_j}{v_{\bar{0}i}V_j - v_{\bar{0}j}V_i}}_{\phi}\left(e^{-\frac{v_{\bar{0}j}}{V_j}\tau} - e^{-\frac{v_{\bar{0}i}}{V_i}\tau}\right)\right\}$$

$$+ \frac{G_{pk}}{v_{\bar{0}k}}\left[\frac{1}{v_{\bar{0}j}}\left\{\frac{1}{v_{\bar{0}i}}\left(1 - e^{-\frac{v_{\bar{0}i}}{V_i}\tau}\right) - \frac{V_j}{v_{\bar{0}i}V_j - v_{\bar{0}j}V_i}\left(e^{-\frac{v_{\bar{0}j}}{V_j}\tau} - e^{-\frac{v_{\bar{0}i}}{V_i}\tau}\right)\right\}\right.$$

$$\left. - \frac{V_k}{v_{\bar{0}j}V_k - v_{\bar{0}k}V_j}\left\{\underbrace{\frac{V_k}{v_{\bar{0}i}V_k - v_{\bar{0}k}V_i}\left(e^{-\frac{v_{\bar{0}k}}{V_k}\tau} - e^{-\frac{v_{\bar{0}i}}{V_i}\tau}\right) - \frac{V_j}{v_{\bar{0}i}V_j - v_{\bar{0}j}V_i}}_{\phi}\left(e^{-\frac{v_{\bar{0}j}}{V_j}\tau} - e^{-\frac{v_{\bar{0}i}}{V_i}\tau}\right)\right\}\right]$$

$$+ \sum_{m=1}^{n}(H_m - H_{m-1})U_k\left(\frac{1}{v_{\bar{0}k}}\left[\frac{1}{v_{\bar{0}j}}\left\{\frac{1}{v_{\bar{0}i}}\left(1 - e^{-\frac{v_{\bar{0}i}}{V_i}(\tau-\tau_m)}\right) - \frac{V_j}{v_{\bar{0}i}V_j - v_{\bar{0}j}V_i}\left(e^{-\frac{v_{\bar{0}j}}{V_j}(\tau-\tau_m)} - e^{-\frac{v_{\bar{0}i}}{V_i}(\tau-\tau_m)}\right)\right\}\right.\right.$$

$$\left. - \frac{V_k}{v_{\bar{0}j}V_k - v_{\bar{0}k}V_j}\left\{\underbrace{\frac{V_k}{v_{\bar{0}i}V_k - v_{\bar{0}k}V_i}\left(e^{-\frac{v_{\bar{0}k}}{V_k}(\tau-\tau_m)} - e^{-\frac{v_{\bar{0}i}}{V_i}(\tau-\tau_m)}\right) - \frac{V_j}{v_{\bar{0}i}V_j - v_{\bar{0}j}V_i}}_{\phi}\left(e^{-\frac{v_{\bar{0}j}}{V_j}(\tau-\tau_m)} - e^{-\frac{v_{\bar{0}i}}{V_i}(\tau-\tau_m)}\right)\right\}\right]$$

$$- \frac{1}{v_{\bar{0}k} - B_kV_k}\left[\frac{1}{v_{\bar{0}j} - B_kV_j}\left\{\frac{1}{v_{\bar{0}i} - B_kV_i}\left(e^{-B_k(\tau-\tau_m)} - e^{-\frac{v_{\bar{0}i}}{V_i}(\tau-\tau_m)}\right) - \frac{V_j}{v_{\bar{0}i}V_j - v_{\bar{0}j}V_i}\left(e^{-\frac{v_{\bar{0}j}}{V_j}(\tau-\tau_m)} - e^{-\frac{v_{\bar{0}i}}{V_i}(\tau-\tau_m)}\right)\right\}\right.$$

$$\left.\left. - \frac{V_k}{v_{\bar{0}j}V_k - v_{\bar{0}k}V_j}\left\{\frac{V_k}{v_{\bar{0}i}V_k - v_{\bar{0}k}V_i}\left(e^{-\frac{v_{\bar{0}k}}{V_k}(\tau-\tau_m)} - e^{-\frac{v_{\bar{0}i}}{V_i}(\tau-\tau_m)}\right) - \frac{V_j}{v_{\bar{0}i}V_j - v_{\bar{0}j}V_i}\left(e^{-\frac{v_{\bar{0}j}}{V_j}(\tau-\tau_m)} - e^{-\frac{v_{\bar{0}i}}{V_i}(\tau-\tau_m)}\right)\right\}\right]\right) \quad 3.30$$

When, in room j, the ratio of total outward airflow $v_{\bar{0}j}$ to the volume V_j of the room is equal to that in the adjacent room k, the denominator of the effect of the adjacent room in Equation 3.24 becomes zero, as

$$v_{\bar{0}j}V_k - v_{\bar{0}k}V_j = 0 \quad 3.31$$

When this is the case, this term is simplified as follows in the process of integrating the differential equation for the gas balance, 3.22, as

equation 3.32 : see page 86 3.32

The same treatment is required between directly connected rooms and secondary connected rooms in Equation 3.30.

When there is an interchange of air between rooms i and j, as shown in Figure 3.4 f, the room i can be regarded as one of the rooms which have an indirect influence on room i. In order to treat the influence of a room on itself, those parts of Equation 3.30 which are marked ϕ and Φ in the Equation must be replaced by the following expressions:

$$\phi = \frac{\tau}{V_j}e^{-\frac{v_{\bar{0}j}}{V_j}\tau} \qquad \Phi = \frac{(\tau-\tau_m)}{V_j}e^{-\frac{v_{\bar{0}j}}{V_j}(\tau-\tau_m)} \quad 3.33$$

In practice, the interchange of air between rooms must be considered as shown in Figure 3.6. The equation of concentration for a room in a building must therefore be supposed to consist of an infinite series of the effects of all the rooms. The influences due to the rooms which are connected to the room under consideration in a tertiary and more distant manner are considered not to be significant. The following term is to be substituted at the end of the effects of the secondary adjacent rooms in Equation 3.29, marked Ω , as the influence due to such rooms. By the use of this term, the effect of rooms connected in a more distant than tertiary manner is compensated for by using the history of the variations in concentration in the tertiary connected rooms.

equation 3.34 : see page 90 3.34

The deduction of this term is explained in the following section.

3.3.4 The effect of an air flow whose concentration is known at irregular time intervals

When the variation in concentration in room j is known at irregular time intervals, the effect of the air flow from this room to room i is approximated as follows. In field measurements, the timing of concentration measurements can very often deviate, for certain reasons, from the scheduled timing. It is therefore better to regard the time intervals of concentration measurement as irregularly changing. A measurement at regular intervals can be treated as a special case of this method.

Let us suppose that the concentration in room j is known only at times τ_0, τ_1, ... τ_n, as shown in Figure 3.5 a. The simplest way of approximating this curve is to keep the value of each measurement constant over a period extending from the previous measurement to the present measurement, as shown by the stepwise variation in Figure 3.5 b. This method is based on the same procedure as the finite differential equations relating to various phenomena with periodical fluctuations. However, the method is effective only when the intervals are sufficiently short compared with the rapidity of the variation in concentration.

The next simplest way is to use linear interpolation between the points of measurement, assuming that the change between intervals is linear, as shown in Figure 3.5 c. For data with long time intervals, approximation is considerably improved by linear interpolation from the rectangular approximation. When the concentration in room j at time τ_m is equivalent to C_{mj}, the concentration $C_{\tau j}$ at time τ, which is situated between times τ_n and τ_{n+1}, is expressed by the following equation:

$$\Omega = \sum_{l=1}^{l'} v_{kl} \sum_{m=1}^{n} (C_{ml}-C_{m-1,l})\frac{1}{v_{\bar{0}k}}\Big[\frac{1}{v_{\bar{0}j}}\Big\{\frac{1}{v_{\bar{0}i}}\Big(1-e^{-\frac{v_{\bar{0}i}}{V_i}\tau}\Big)-\frac{V_j}{v_{\bar{0}i}V_j-v_{\bar{0}j}V_i}\Big(e^{-\frac{v_{\bar{0}j}}{V_j}\tau}-e^{-\frac{v_{\bar{0}i}}{V_i}\tau}\Big)\Big\}$$
$$-\frac{V_k}{v_{\bar{0}j}V_k-v_{\bar{0}k}V_j}\Big\{\frac{V_k}{v_{\bar{0}i}V_k-v_{\bar{0}k}V_i}\Big(e^{-\frac{v_{\bar{0}k}}{V_k}\tau}-e^{-\frac{v_{\bar{0}i}}{V_i}\tau}\Big)-\frac{V_j}{v_{\bar{0}i}V_j-v_{\bar{0}j}V_i}\Big(e^{-\frac{v_{\bar{0}j}}{V_j}\tau}-e^{-\frac{v_{\bar{0}i}}{V_i}\tau}\Big)\Big\}e^{\frac{v_{\bar{0}k}}{V_k}\tau}\Big] \qquad 3.34$$

$$C_{\tau i} = f_1(G_{\tau i}) + \frac{v_{ij}}{v_{\bar{0}i}}\Big\{\sum_{m=1}^{n}\Big[C_{m-1,j}\Big(e^{-\frac{v_{\bar{0}i}}{V_i}(\tau-\tau_m)}-e^{-\frac{v_{\bar{0}i}}{V_i}(\tau-\tau_{m-1})}\Big)$$
$$+\frac{C_{mj}-C_{m-1,j}}{\tau_m-\tau_{m-1}}\Big\{(\tau_m-\tau_{m-1})e^{-\frac{v_{\bar{0}i}}{V_i}(\tau-\tau_m)}-\frac{V_i}{v_{\bar{0}i}}\Big(e^{-\frac{v_{\bar{0}i}}{V_i}(\tau-\tau_m)}-e^{-\frac{v_{\bar{0}i}}{V_i}(\tau-\tau_{m-1})}\Big)\Big\}\Big]$$
$$+C_{ni}\Big(1-e^{-\frac{v_{\bar{0}i}}{V_i}(\tau-\tau_n)}\Big)+\frac{C_{n+1,j}-C_{nj}}{\tau_{n+1}-\tau_n}\Big(\tau-\tau_n-\frac{V_i}{v_{\bar{0}i}}\Big(1-e^{-\frac{v_{\bar{0}i}}{V_i}(\tau-\tau_n)}\Big)\Big)\Big\} \qquad 3.36$$

$$C_{\tau j} = \frac{\tau - \tau_n}{\tau_{n+1} - \tau_n} (C_{n+1,j} - C_{nj}) + C_{nj} \qquad 3.35$$

$$\tau_{n+1} > \tau > \tau_n$$

When the differential equation of tracer gas balance 3.22 is solved by substituting Equation 3.35 into the variation in concentration in room j, $C_{\tau j}$, this results in the following equation:

equation 3.36 : see page 90 3.36

When room j receives air flows from several rooms, as shown in Figure 3.4 e, and the variation in concentration in the source rooms is known, all the effects must be treated simultaneously.

3.3.5 Comparison of equations of variation in concentration

To examine the accuracy of various methods, the variations in concentrations were calculated by the different equations, assuming a building with four rooms.

The three methods used for calculating the variations in concentration are as follows:

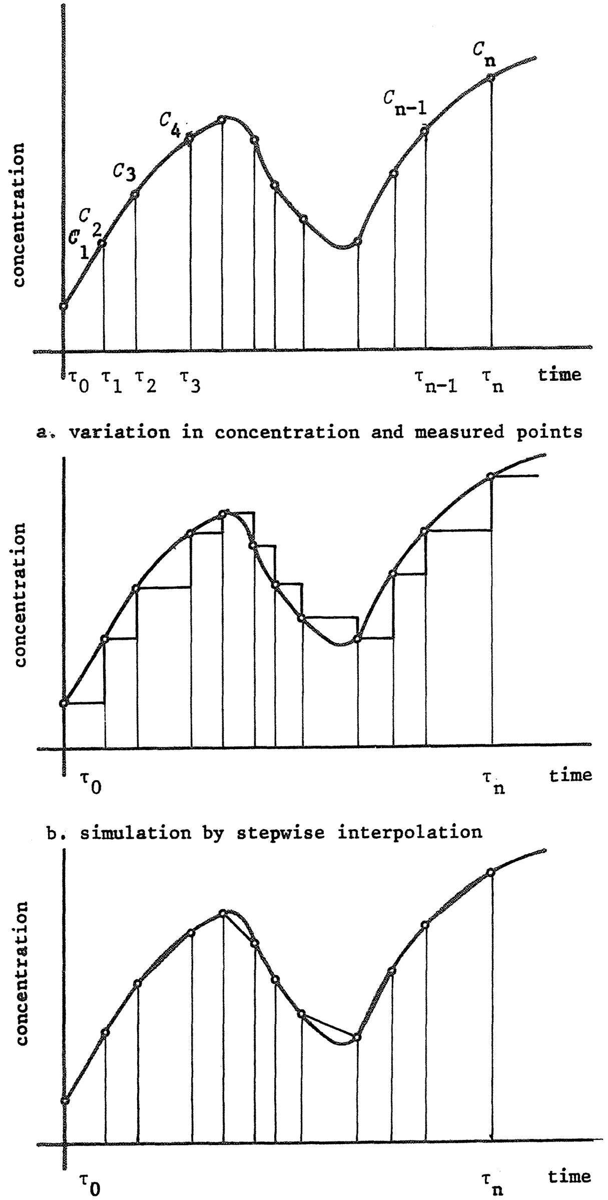
a. Finite differential equation 3.4, time interval 7.5 seconds.

b. Transient equation 3.29 without compensation for tertiary and more distantly connected rooms.

c. Transient equation 3.29 with compensation using the concentration in the tertiary adjacent rooms. Time interval 900 seconds.

The plan and air flow conditions in the building are shown in Figure 3.6. The constants of tracer gas producers in every room and the patterns of heat supply for each producer are also shown in Figure 3.6.

The concentrations in the rooms are compared in Table 3.1. After an hour's gas production, the concentration in room 4 as calculated by Method b deviates by 13% from those calculated by Methods a and c.



c. simulation by linear interpolation

Fig. 3.5 Variation in concentration and two methods of interpolati

Table 3.1 Comparison of concentration by three equations

time second	concentration in kg/m^3 room 1	room 2	room 3	room 4	equation
0	0.0	0.0	0.0	0.0	a
300	0.00459	0.00184	0.00203	0.00468	a
600	0.01046	0.00443	0.00526	0.01079	a
900	0.01524	0.00733	0.00917	0.01607	a
	0.01518	0.00718	0.00887	0.01596	b
	0.01518	0.00718	0.00887	0.01596	c
1 200	0.01888	0.01007	0.01316	0.02048	a
1 500	0.02164	0.01247	0.01691	0.02418	a
1 800	0.02375	0.01448	0.02028	0.02731	a
	0.02352	0.01341	0.01796	0.02656	b
	0.02360	0.01389	0.01897	0.02687	c
2 100	0.02193	0.02065	0.02799	0.02708	a
2 400	0.01865	0.02645	0.03586	0.02671	a
2 700	0.01625	0.02955	0.04107	0.02748	a
	0.01565	0.02722	0.03593	0.02534	b
	0.01601	0.02889	0.03965	0.02659	c
3 000	0.01469	0.03096	0.04459	0.02865	a
3 300	0.01370	0.03154	0.04684	0.02976	a
3 600	0.01307	0.03174	0.04836	0.03064	a
	0.01179	0.02890	0.04168	0.02625	b
	0.01266	0.03151	0.04773	0.02925	c

equation a : finite diferential equation, time interval 7.5 seconds
b : trancient equation
c : trancient equation + compensation of thirdly connected rooms, time interval 900 seconds

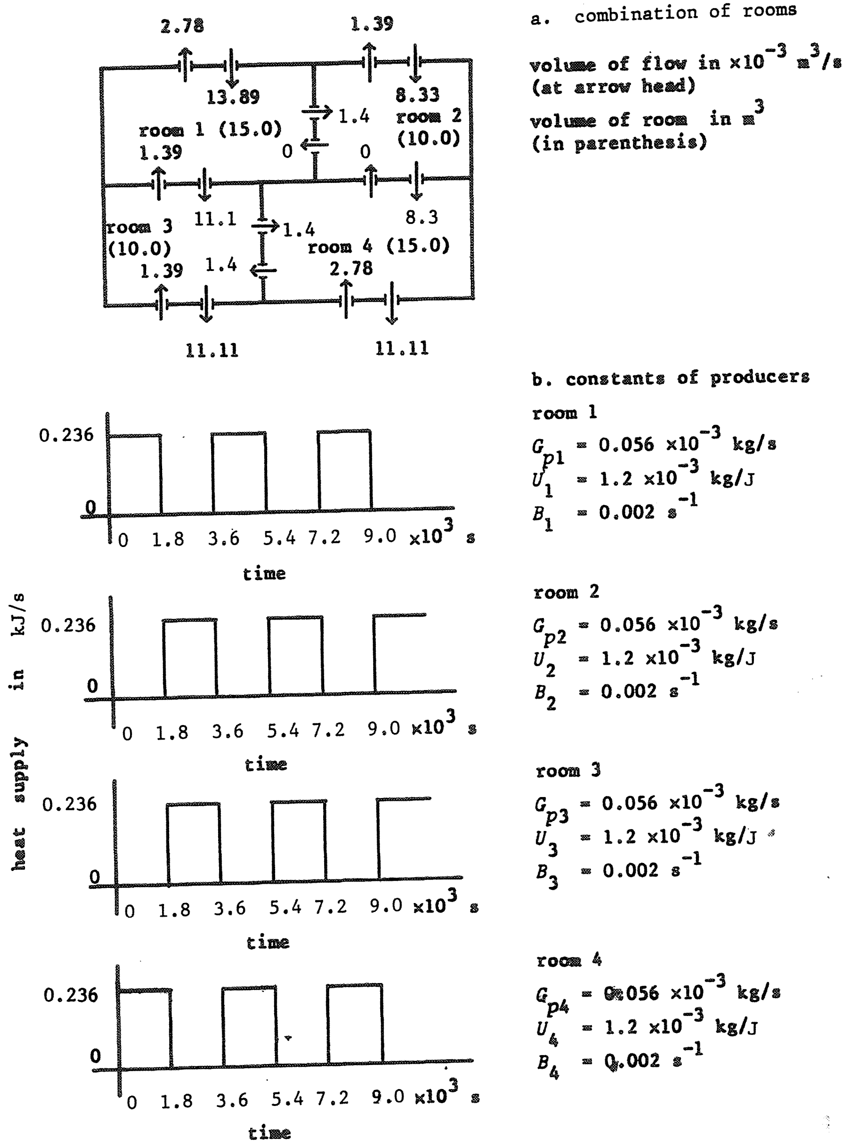


Fig. 3.6 Air flows and gas productions for examination of the program

3.4 Methods of calculating air flows

In the previous section, the equations describing the variation in concentration in each room of a building were deduced on the assumption that the air flows between rooms and between the rooms and the outside were known. On the contrary, when the variation in concentration in every room are obtained from measurements over a period, the air flows between the rooms shall be calculated by numerical analysis. Solutions of sufficient accuracy for engineering purposes are obtained by an iterative calculation, and the calculation is confirmed by a statistical method to overcome the many obstructions which may take place in the process of field measurement.

3.4.1 Linear equations by Taylor expansion

The equation which expresses the variation in concentration in a room, such as equation 3.29, consists of the terms of air flows and tracer gas production. If we know the chronological change in the intensity of gas production, then it is the air flows which are the only unknowns in the equation. Let the equation of the variation in concentration be expressed as a function of the air flows as:

$$f(v_{ij}) = C_{\tau i} \qquad 3.37$$

Suppose that an air flow v_{ij} consists of an assumed value $\dot{v}_{ij}$ and a correction term $\ddot{v}_{ij}$, as follows

$$v_{ij} = \dot{v}_{ij} - \ddot{v}_{ij} \qquad 3.38$$

Then the equation of concentration is expanded as follows by Taylor's method

$$C_{\tau i} = f(\dot{v}_{ij}) - \ddot{v}_{i1}\frac{d}{dv_{i1}} f(\dot{v}_{ij}) - \ddot{v}_{i2}\frac{d}{dv_{i2}} f(\dot{v}_{ij}) \qquad 3.39$$

$$- \cdots\cdots - \ddot{v}_{n,n-1}\frac{d}{dv_{n,n-1}} f(\dot{v}_{ij}) + \ddot{v}_{\bar{0}i}\frac{d}{dv_{\bar{0}i}} f(\dot{v}_{ij}) + R$$

Here, $df(v_{ij})/dv_{ij}$ is the first partial differential of $f(v_{ij})$ with respect to v_{ij}, and is evaluated by substituting every assumed value of $\dot{v}_{ij}$ into the differential. The last term R is the summation of the terms which are higher than the second order of the correction terms. The value of R decreases, and can be ignored when the assumed values approach the solution of the function, and accordingly the correction terms approach zero in the process of iterative calculation. When the remainder R is ignored in the first iterative calculation, equation 3.39 is changed to a linear equation of the correction terms $\ddot{v}_{ij}$

Each equation which is drawn up in each room at each stage of the concentration measurement is converted into the linear equation of the correction terms of all the air flows in a building, and all these equations make up a series of linear equations of the correction terms.

In order to avoid the influence of errors in measurement, a sufficient number of equations must be treated by the method of least squares. The normal equations which are a system of simultaneous linear equations of correction terms, the number of which coincides with the number of unknowns, are established from this sufficient number of linear equations by statistical techniques. The correction terms of every air flow are solved from the normal equation by, for instance, the Gauss elimination method. The assumed values can be supplemented by the newly solved correction terms. In this process, the following method must be used to avoid divergence due to the effect of the corrections, which may arise in the course of iterative calculation as a result of the fact that the remainder term R is ignored and that a large number of correction terms are treated concurrently during the correction process:

$$v_{ij} = v_{ij} - \delta \cdot v_{ij} \qquad 3.40$$

The constant δ is called a correction factor. Its value is between 0 and 1, and it is recommended that the value of 0.5 be used in this case.

3.4.2 Computer programs of air flow calculation and their precision

A computer program was compiled to calculate air flows from variations in concentration, using the theory of variation in concentration in a series of rooms, which resulted in Equation 3.29.

The main flow diagram of this program is shown in Figure 3. 7.

The calculation should be repeated many times, successively correcting the assumed values of air flows, until their correction terms become sufficiently small. A linear equation including correction terms should be set up for each occasion at which concentration is measured and for each room, in the form of Equation 3. 39. The equations should then be solved using the method of least squares.

At each step of the process the assumed air flows are to be corrected by using the results obtained in the previous step. Several devices were added to the program to ensure rapid and safe convergence. The iterative calculation is terminated when the sum of the absolute values of the error terms has become smaller than a reference value.

In order to examine the precision of the program, the same variations in concentration which had been calculated in 3.3.5 were applied to the program. Table 3.2 shows the process of iterative calculation, in which the variations in concentration in every room at time intervals of one minute were entered. Air flows v_{12} and v_{24} were assumed to be zero in calculating the concentration, but they were included among the unkowns, and air flows of 1.4 and 2.8×10^{-3} m^3/s were applied to each of them respectively, as the initial value. As will be seen from Table 3.2, the absent air flows result in zero or very nearly zero values. All the deviations in air flows reached less than 1% after 22 iterations.

When several field measurements had been carried out, the following point arose for modification of the program. The precision in the concentration measurements was not sufficient for treatment of the large number of unknowns which are possible in view of

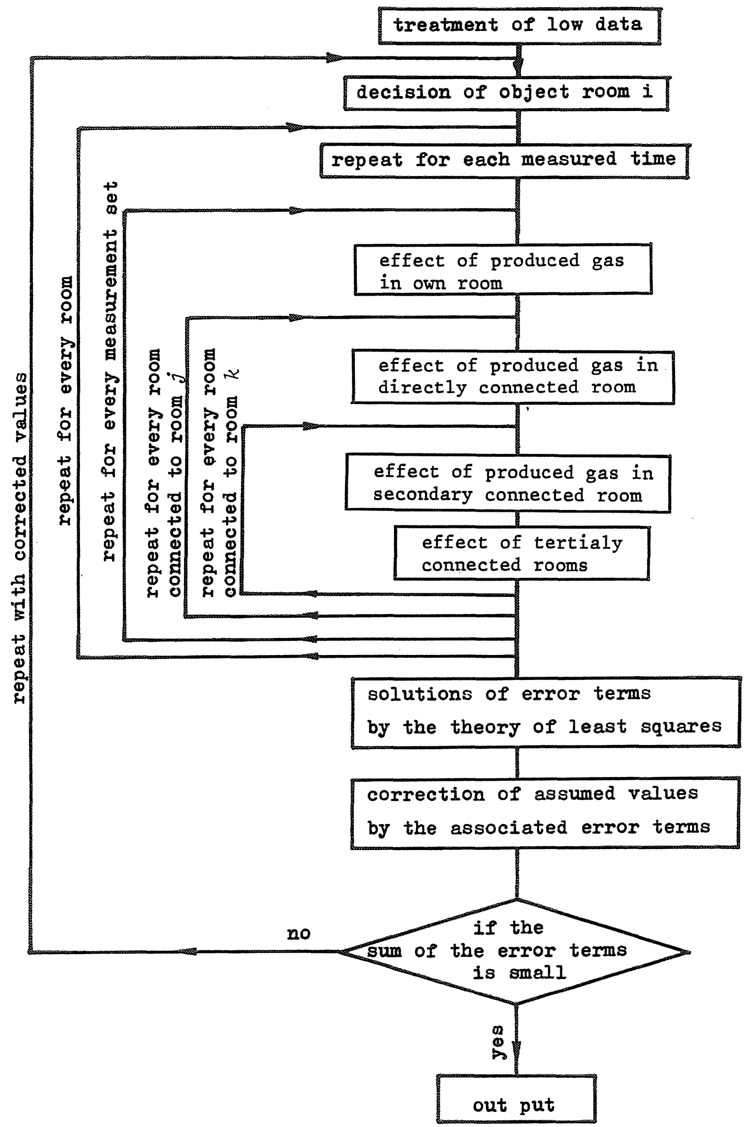


Fig. 3.7 Flow diagram of the computer program for air flow calculation

the plan of the building. Unnecessary unknowns must be eliminated from the calculation in order that reliable results may be obtained. In practice, most of the air flows between rooms are unidirectional, but this direction cannot be **predicted at the** beginning of a calculation. Trials of all possible combinations of all possible air flows may solve the problem, but the number of trials would be far too excessive. When the air flows are calculated separately for each room, the number of trials is considerably reduced.

When the variations in concentration in every room are known, the air flows are calculated separately for each room by means of Equation 3.36.

The above computer program for air flows was modified with this purpose. The loop of iterative calculation was changed so as to be completed at each room, and the effect of gas production in directly connected rooms are calculated from the assumed air flows and the measured variations in concentration. The effect of the secondary and more distantly connected rooms were removed from the first program. As explained above, it is better to cut the number of unknowns to the smallest possible value in order to gain reliable solutions from relatively coarse data of variations in concentration. Generally, the total volume of air which flows out of a room is equal to the total volume of air which flows into the room. In iterative calculation, the correction terms become very small at the end of the iterations, and the total volume of outflowing air can then be replaced by the total volume of assumed inward air flows. For this reason, an iterative calculation may proceed by replacing the term corresponding to the outflow air by the total of the assumed values of inflow air from the first step of an iteration. After this modification the program was found to function in the same way as when the outflow had been included among the unknowns.

The computing program for calculation of air flows for each room separately was examined in the following way. At first, the variations in concentration over a period of 10800 seconds were calculated assuming a flat with five rooms, which is the same type used for the field measurement of air flows, see 4.3. The

Table 3.2 Process of successive iteration

Air flow		v_{12}	v_{13}	v_{10}	$v_{\bar{0}1}$	v_{21}	v_{24}	v_{20}	$v_{\bar{0}2}$
Start value		1.4	1.4	5.6	27.8	5.6	2.8	5.6	8.3
Iteration	T.C.								
5	128.7	1.5	7.6	0.5	20.8	-5.8	2.8	10.6	10.4
10	36.7	0.7	2.3	7.5	14.9	2.2	1.3	7.7	9.4
15	4.1	0.2	1.3	13.2	15.1	1.6	-0.1	8.2	9.6
20	0.5	-0.0	1.4	13.9	15.3	1.4	-0.0	8.3	9.7
22	0.2	0.0	1.4	13.9	15.3	1.4	0.0	8.3	9.7
Mark		0.0	1.4	13.9	15.3	1.4	0.0	8.3	9.7
Air flow		v_{31}	v_{34}	v_{30}	$v_{\bar{0}3}$	v_{42}	v_{43}	v_{40}	$v_{\bar{0}4}$
Start value		2.8	2.8	1.4	27.8	5.6	2.8	1.4	8.3
Iteration	T.C.								
5		2.8	6.8	3.6	26.5	8.7	0.9	2.3	12.5
10		9.7	2.1	-1.4	19.4	8.8	1.3	2.7	12.5
15		11.6	1.3	0.9	14.1	8.5	1.2	3.1	12.7
20		11.1	1.4	1.4	13.9	8.4	1.4	2.8	12.5
22		11.1	1.4	1.4	13.9	8.3	1.4	2.8	12.5
Mark		11.1	1.4	1.4	13.9	8.3	1.4	2.8	12.5

T.C. : total of correction terms in $\times 10^{-3}$ m^3/s Air flow in $\times 10^{-3}$ m^3/s

room number 0 : doudoor

Table 3.3 Last values of iterative calculations

time interval	air flows of room 1				air flows of room 2				
	v_{12}	v_{14}	v_{10}	$v_{\bar{0}1}$	v_{21}	v_{23}	v_{24}	v_{20}	$v_{\bar{0}2}$
360	2.8	0.0	13.9	16.7	9.8	12.6	2.7	11.0	36.1
540	2.8	-0.0	13.9	16.6	9.9	12.7	2.6	10.9	36.2
720	2.9	-0.0	13.8	16.6	9.9	12.9	2.6	10.9	36.4
1080	2.9	-0.0	13.9	16.6	8.6	13.5	4.4	10.9	37.4
mark	2.8	0.0	13.9	16.7	9.7	12.5	2.8	11.1	36.1

time interval	air flow of room 3			air flows of room 4					air flows of	
	v_{32}	v_{30}	$v_{\bar{0}3}$	v_{41}	v_{42}	v_{45}	v_{40}	$v_{\bar{0}4}$	v_{54}	v_{50}
360	1.4	18.1	19.4	5.7	9.8	1.4	5.4	22.3	15.4	1.2
540	1.4	18.1	19.5	5.8	9.9	1.3	5.4	22.4	15.5	1.2
720	1.4	18.1	19.5	5.8	10.0	1.2	5.3	22.3	15.8	1.0
1080	1.4	18.1	19.6	5.4	9.9	0.7	5.5	21.4	16.4	1.1
mark	1.4	18.1	19.4	5.6	9.7	1.4	5.6	22.2	15.3	1.4

time interval in second air flow in $\times 10^{-3}$ m^3/s room number 0 :outdoor assumed air flows and the way of CO_2 gas production are shown in Figure 3.8. Four data sets of concentration, time intervals of which were 360, 540, 720 and 1080 seconds, respectively, were extracted from the variations in concentration, and air flows were calculated from the data sets. The results are shown in Table 3.3. The errors of the results were under 1 %, when the data set of time interval 360 seconds was used. The error was under, 6 %, when the air flows were calculated from the data set of time interval 1080 seconds.

About 50 iterations were necessary to calculate air flows from the variations in concentration obtained from field measurements.

The programs were written in Fortran IV, and the size of the program was about 28 thousands. Using a computer type IBM 360, the computing time was about 40 seconds for a five-room flat for all possible combinations of all possible air flows.

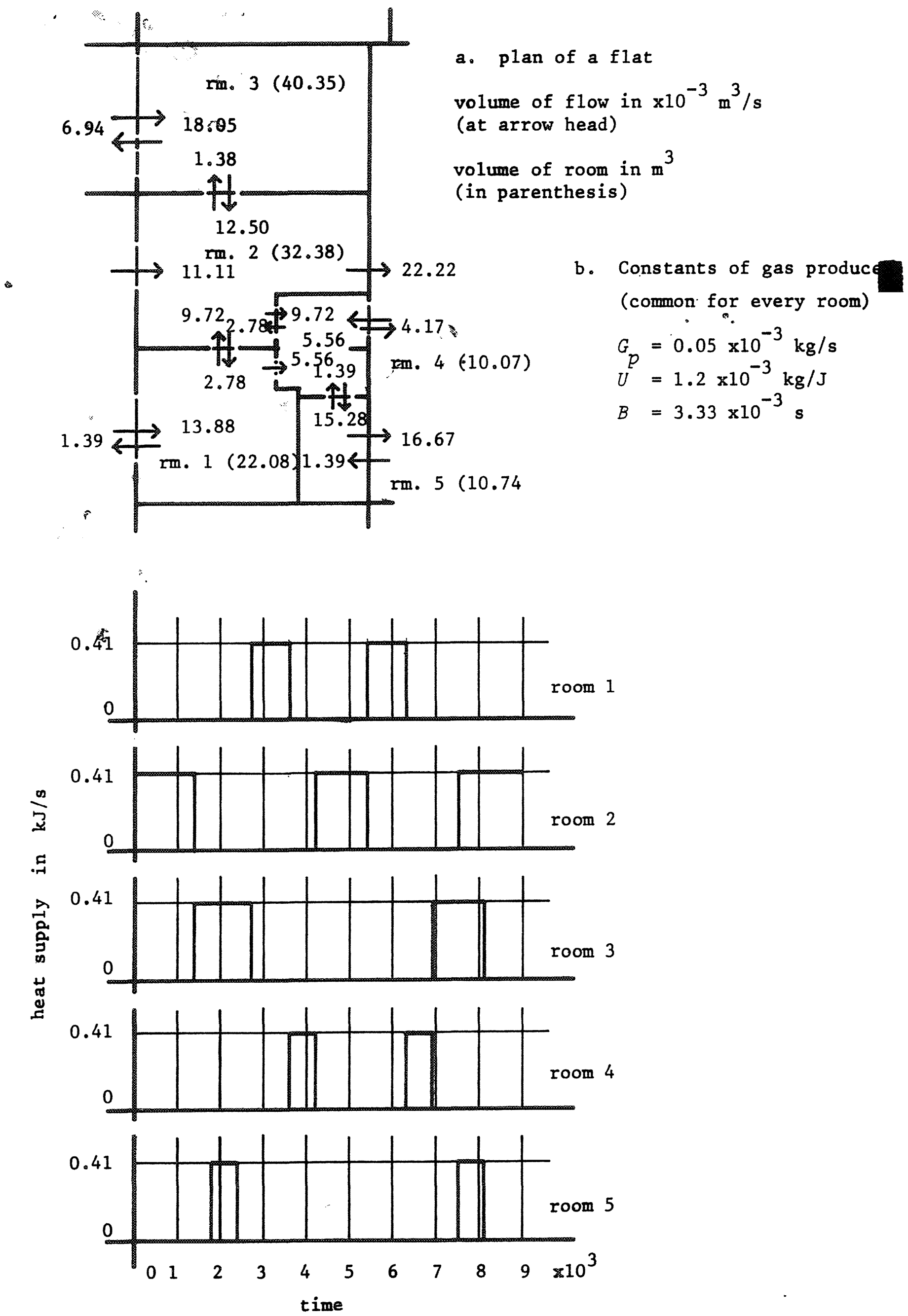


Fig. 3.8 Air flows and gas productions for examination of the program

3.5 Some simpler applications of the tracer gas technique

The theory of variation in concentration, which was set up in 3.3, can be applied to some simpler cases with a slight modification. Several examples of these are explained in this section.

3.5.1 Air change of a single room

The variation in concentration in a single room is expressed by Equation 3.21. Furthermore, when constant production G_i of a gas is obtained by the operation of a control cock, the time constant B of gas production can be assumed to be infinite, and production of gas by heat penetration into the vessel can be ignored. The equation is then simplified as follows:

$$C_{\tau i} = C_{0i}\, e^{-\frac{v_{\bar{0}i}}{V_i}\tau} + \frac{G_i}{v_{\bar{0}i}} \left(1 - e^{-\frac{v_{\bar{0}i}}{V_i}\tau} \right) \qquad 3.41$$

Further, if constant gas production commences at time 0, and the concentration at time 0 is zero in the room, the concentration in the room varies as

$$C_{\tau i} = \frac{G_i}{v_{\bar{0}i}} \left(1 - e^{-\frac{v_{\bar{0}i}}{V_i}\tau} \right) \qquad 3.42$$

When the concentration in the room is measured after a certain time interval $\Delta\tau$ from the commencement of gas production, the following relationship results between the air change and the variation in concentration:

$$v_{\bar{0}i} = \frac{V_i}{\Delta\tau} \ln \frac{C_{n-1,i} - C_{n-2,i}}{C_{ni} - C_{n-1,i}} \qquad 3.43$$

where $C_{n,i}$ is the concentration at the nth measurement at the time interval $\Delta\tau$.

The average air change during the measurement period is then decided by the following equation from the rising curve of concentration:

$$v_{\bar{0}i} = \frac{V_i}{(n'-1)\,\Delta\tau} \sum_{n=2}^{n'} \left(\ln \frac{C_{n-1,i} - C_{n-2,i}}{C_{ni} - C_{n-1,i}} \right) \qquad 3.44$$

This equation corresponds to Equation 3.1 which expresses the air change on the basis of the decay in concentration. All the terms in brackets in Equation 3.44 must be almost equal. This can be used for the examination of the precision of a concentration measurement. In this method, if it is ensured that tracer gas production is maintained at a constant rate and the character of the concentration meter is linearly proportional to the concentration, the absolute value of gas production and the absolute concentration need not be known.

Further, when gas production has been constant for a long time, concentration reaches a constant value, or theoretically the time τ in Equation 3.42 can be put equal to infinity. The air change is then calculated directly from the intensity of gas production and the constant concentration as

$$v_{\bar{0}i} = \frac{G_i}{C_{\infty i}} \qquad 3.45$$

3.5.2 Air flow from an adjacent room

Let us consider two adjacent rooms where air flows at a constant rate from room j to room i, the concentration in the two rooms is zero at time 0, and a constant gas production is commenced in room j at time 0. The following equation is derived from Equation 3.24 to express the concentration in room i by adopting the above conditions:

$$C_{\tau i} = \frac{v_{ij}\,G_j}{v_{\bar{0}i}V_j - v_{\bar{0}j}V_i} \left\{ \frac{V_j}{v_{\bar{0}j}} \left(1 - e^{-\frac{v_{\bar{0}j}}{V_j}\tau} \right) - \frac{V_i}{v_{\bar{0}i}} \left(1 - e^{-\frac{v_{\bar{0}i}}{V_i}\tau} \right) \right\} \qquad 3.46$$

Concentration in room j at time τ is expressed by Equation 3.42. Substitution of this equation into Equation 3.46 results in

$$C_{\tau i} = \frac{v_{ij}}{v_{\bar{0}i}V_j - v_{\bar{0}j}V_i}\left\{ V_j C_{\tau j} - \frac{V_i G_j}{v_{\bar{0}i}}\left(1 - e^{-\frac{v_{\bar{0}i}}{V_i}\tau}\right)\right\} \qquad 3.47$$

If concentration measurement is repeated at time interval $\Delta\tau$, then the concentration C_{ni} in room i at the nth measurement is

$$C_{ni} = \frac{v_{ij}}{v_{\bar{0}i}V_j - v_{\bar{0}j}V_i}\left\{ V_j C_{nj} - \frac{V_i G_j}{v_{\bar{0}i}}\left(1 - e^{-\frac{v_{\bar{0}i}}{V_i}n\Delta\tau}\right)\right\} \qquad 3.48$$

The following equation is obtained by subtracting the concentration at the (n-1)th measurement from that at the nth measurement :

$$\left\{ \frac{(C_{ni}-C_{n-1,i})(v_{\bar{0}i}V_j - v_{\bar{0}j}V_i)}{v_{ij}} - V_j(C_{nj}-C_{n-1,j})\right\} \frac{v_{\bar{0}i}}{V_i G_j} = -e^{-\frac{v_{\bar{0}i}}{V_i}(n-1)\Delta\tau}\left(1-e^{-\frac{v_{\bar{0}i}}{V_i}\Delta\tau}\right) \qquad 3.49$$

By treating the concentrations at the (n-1)th and (n-2)th measurements in the same way, and dividing Equation 3.49 by the latest treatment, the term of gas production in room j can be eliminated. Consequently, the air flow from room j to room i is calculated from the following equation:

$$v_{ij} = \frac{v_{\bar{0}i}V_j - v_{\bar{0}j}V_i}{V_j}\left\{ \frac{C_{ni}-C_{n-1,i} - e^{-\frac{v_{\bar{0}i}}{V_i}\Delta\tau}(C_{n-1,i}-C_{n-2,i})}{C_{nj}-C_{n-i,j} - e^{-\frac{v_{\bar{0}i}}{V_i}\Delta\tau}(C_{n-1,j}-C_{n-2,j})}\right\} \qquad 3.50$$

Here the air change in room i, $v_{\bar{0}i}/V_i$, must be obtained from a separate measurement.

When the measurements of variations in concentration in adjacent rooms, due to constant gas production, are repeated alternately in each room, the air change of each room is calculated by Equation

3.43, and if unidirectional air flow is recognized in the variations in concentration of any of the adjacent rooms, the flow is assessed by Equation 3.49 without knowing the absolute production of tracer gas.

After constant gas production has continued for a sufficiently long period and the concentrations in both of the coupled rooms reach constant values, the following equation results by putting the time in Equation 3.46 equal to infinity, and by substituting Equation 3.45 into it:

$$v_{ij} = v_{\bar{0}i}\frac{C_{\infty i}}{C_{\infty j}} \qquad 3.51$$

This equation states that the air flow from a room in which a tracer gas is being produced, into an adjacent room, is proportional to the ratio of the concentrations in the rooms. And if the rate of air change $v_{\bar{0}i}$ of the adjacent room is known, the air flow between the coupled rooms is assessed by Equation 3.51.

When a gas is blown off in room j for a very short period, and the concentration in the room reaches C_{0j}, this concentration is applied as the initial concentration in room j in Equation 3.23, and the variation in concentration in room i is given by the following equation:

$$C_{\tau i} = v_{ij}\frac{C_{0j}V_j}{v_{\bar{0}i}V_j - v_{\bar{0}j}V_i}\left(e^{-\frac{v_{\bar{0}j}}{V_j}\tau} - e^{-\frac{v_{\bar{0}i}}{V_i}\tau}\right) \qquad 3.52$$

The following equations are obtained for the calculation of the air flow from room j to room i by treating Equation 3.52 in the same way as Equation 3.50:

$$v_{ij} = \frac{v_{\bar{0}j}V_i - v_{\bar{0}i}V_j}{C_{0j}V_j}\cdot\frac{(C_{ni} - C_{n-1,i})}{e^{-\frac{v_{\bar{0}j}}{V_j}(n-1)\Delta\tau}\left(1-e^{-\frac{v_{\bar{0}j}}{V_j}\Delta\tau}\right) - e^{-\frac{v_{\bar{0}i}}{V_i}(n-1)\Delta\tau}\left(1-e^{-\frac{v_{\bar{0}i}}{V_i}\Delta\tau}\right)} \qquad 3.53$$

The accuracy of an air flow measurement by the above method is not so good as the accuracy of the statistical method set out in 3.3 and 3.4, in which the intensity of gas production and the proportionality of the gas analyser must be known. For instance, the change in concentration in a room in which a constant production of tracer gas is in progress is expressed by an exponential equation, as seen in Equation 3.42. The concentration varies steeply at the beginning of gas production; the variation becomes more gentle in time and attains an almost constant value after a certain time. Equation 3.43 is valid only during a period when there are clear differences in concentration. This means that the range of air change rate of a room, the intensity of tracer gas production, the sensitivity of the concentration meter employed, and the time interval of concentration measurement, play a more important part when this method is used than in the statistical method.

In practice, it is difficult to obtain a smooth variation in concentration from a field measurement. In order that a steady result may be obtained, it is best for the measurement of concentration to be repeated after the shortest possible time interval. A smoother variation in concentration must be assumed from the measurements at the short time intervals, and the air flow must be calculated by selecting the concentrations at the longer time intervals from the smoothed variation in concentration.

4 Air Flow Measurement in Two Tall Blocks of Flats

4.1 Carbon dioxide gas as tracer

4.1.1 Carbon dioxide gas producer

In order to calculate the air flows between several rooms, the chronological variations in concentration and the rates of production of the gas in every room must be known. Large variations in concentration are also necessary. Simple control of gas production is essential in field measurement.

When cubes of dry ice are put into a vessel capable of insulating them from the thermal fluctuations of the surroundings, it is easy to control the production of carbon dioxide gas by controlling electrically the heat supply necessary for evaporation. An apparatus was therefore designed for the evaporation of controlled quantities of carbon dioxide gas from dry ice.

The apparatus is illustrated in Figure 4.1. An electric heating element is situated at the bottom of the vessel. Dry ice placed on this will evaporate at a rate corresponding to the rate of supply of electric power. A small axial flow fan of 110 mm diameter was fitted at the top of the gas producer to mix the gas with the room air and to distribute the mixture to the whole room. Five gas producers were built to this design.

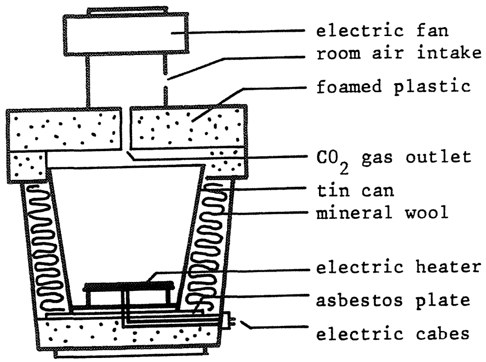


Fig. 4.1 Section of apparatus for production of carbon dioxide gas

Table 4.1 Constants of CO_2 gas producers

producer	heat supply kJ/s	production by heat penetration G_p g/s	production by unit heat supply U g/kJ	time const[redacted] B $\times 10^{-3}$ s^{-1}	
1	0.157	0.058	1.20	3.03	a
				3.44	[redacted]
	0.236	0.067	1.10	3.08	[redacted]
				3.14	b
	0.344	0.056	1.12	3.27	a
				3.78	[redacted]
2	0.234	0.058	1.13	3.44	[redacted]
				3.75	b
3	0.234	0.053	1.16	4.03	[redacted]
				4.06	[redacted]
4	0.232	0.067	1.12	3.42	[redacted]
				3.53	b
5	0.233	0.075	1.21	3.58	
				3.67	

a : onset responce

b : decay responce

The constants G_p, U and B of the five pieces of apparatus were determined in the laboratory of the Department of Heating and Ventilation. The room temperature was between 21 and 23°C. The decrease in weight of dry ice was measured at intervals of 60 seconds for determination of the time constant B, and at intervals of 120 and 300 seconds for determination of the constants G_p and U. The constants of the five pieces of apparatus are set out in Table 4.1. The electric heat supplied during each measurement are also shown in the table.

The latent heat needed for melting and evaporating the dry ice is 0.554kJ/g and 0.181kJ/g respectively. If all the heat which is generated in an electric heater is consumed to produce carbon dioxide gas, and the gas leaves the producer at its sublimate temperature of -78.5°C, then the gas produced by unit heat supply U is 1.36g/kJ. Further if the gas is warmed up in the vessel and leaves it at 20°C, the heat consumption is increased and the gas produced by unit heat supply is 1.22g/kJ. The empirical values obtained from measurements on the five producers ranged from 1.10 to 1.21g/kJ. The production of gas by heat supplies at

0.157, 0.236 and 0.344kJ/s are shown in Table 4.1 for producer No 1. The average value of the three, 1.14g/kJ, was allotted for the constant U of the producer.

The following two points must be checked before using Equation 3.9 as the index of the response of gas production.

Firstly, the production should be linearly proportional to the heat supplied under stable conditions. The relationship between gas production and heat supply for apparatus No 1, which is shown in Table 4.1.

Secondly, the constant B should be the same for a range of heat supply values. The constant B of apparatus No 1 was considered to be sufficiently concentrated. The time constant B for this apparatus was taken as 3.28×10^{-3} sec^{-1}.

The production of gas was slightly affected by the quantity of dry ice in the vessel. When it contained more dry ice, the rate of gas production by heat penetration was slightly higher than when the content was lower.

4.1.2 Distribution of the gas in room air

The specific density of carbon dioxide gas is greater than that of ordinary air at normal room temperature. The resulting variation of concentration in the vertical direction when the gas was released into a room was examined as follow.

The gas was first released from the outlet of the vessel into the room air unheated and without running the fan, and the vertical variation in concentration was measured against time. The apparatus was suspended in the centre of the room, the gas outlet being at a height of 1.8 m above the floor. The heights of the sampling points were 0.05, 0.3, 1.5, 2.0 and 2.25 m above the floor. One such arrangement of sampling points was placed in the centre of the room, and another at a distance of 0.9 m from the centre. The results are shown in Figure 4.2. The figure shows that there was a large difference between the concentration obtained above and below the outlet, due to the greater density of the gas and the low release temperature.

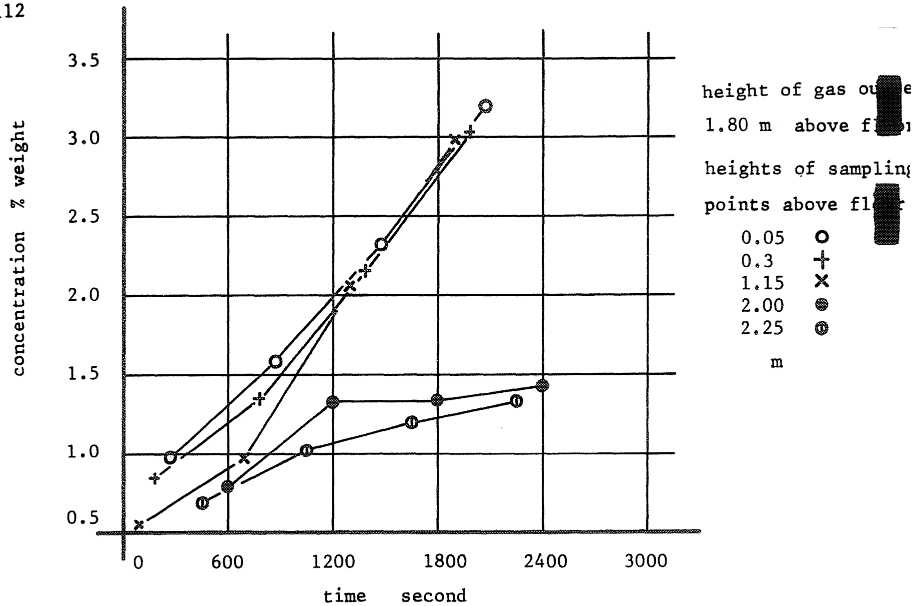


Fig. 4.2 Vertical variation in concentration due to an apparatus without heater or fan

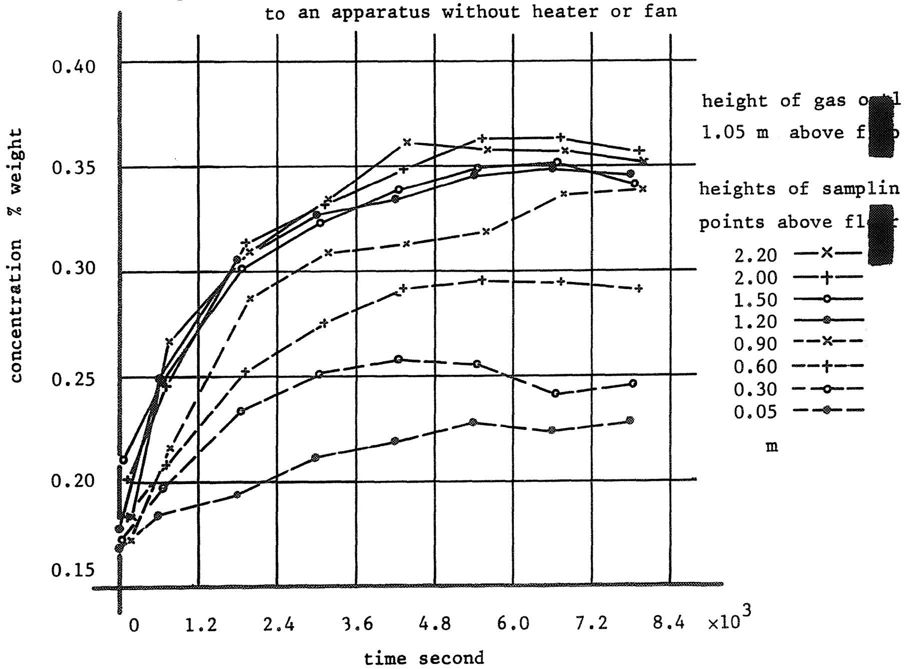


Fig. 4.3 Vertical variation in concentration due to an apparatus with a heater

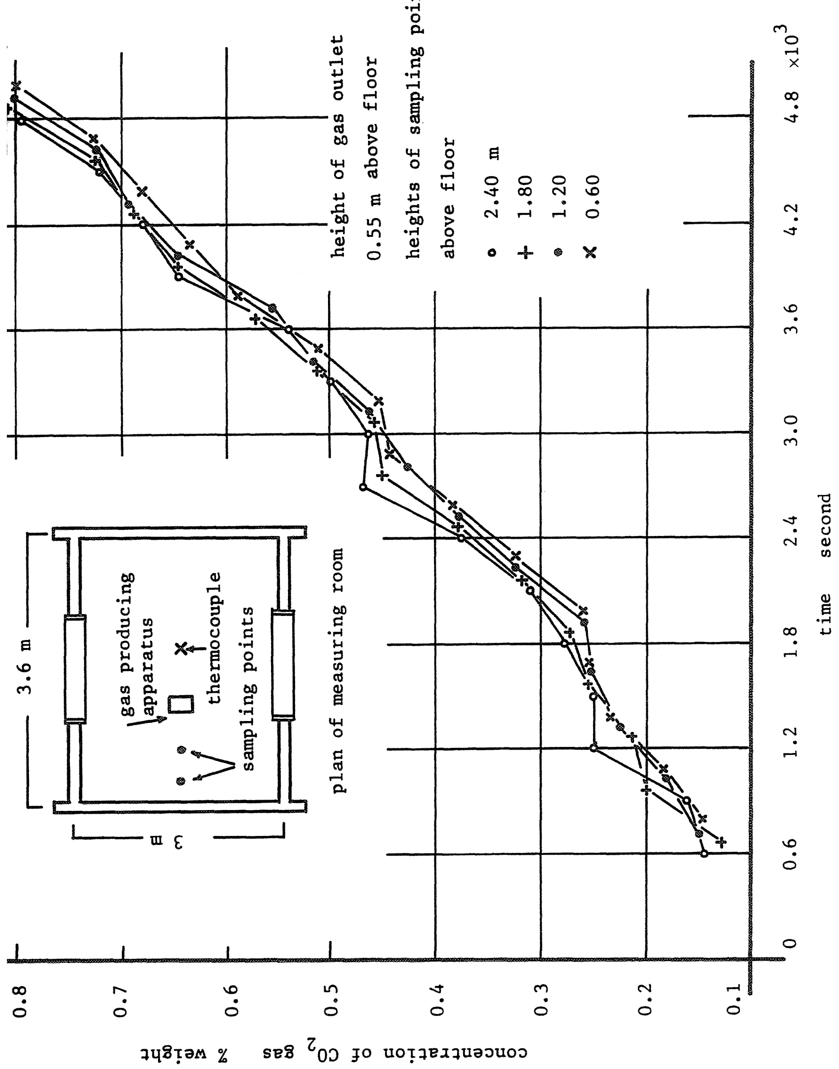


Fig. 4.4 Vertical variation in concentration due to an apparatus with a fan

In Figure 4.3 the gas concentrations as a function of height is shown, when the gas was heated to a temperature several degrees higher than that of the room air at the outlet. The height of the outlet was 1.05 m. It will be seen that in this case the gas concentration was greater above than below the outlet. The figure shows a large vertical difference below the outlet, but the difference is small above the outlet. This indicates that once carbon dioxide gas is situated in the upper part of a room, it takes a very long time for the gas to sink down to the lower part of the room.

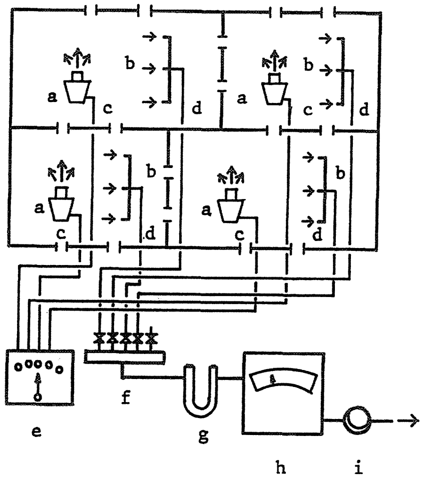
For field measurement, the gas should be distributed as quickly and homogeneously as possible in all parts of a room. To achieve this , a small fan was installed at the gas outlet to mix the gas with a large quantity of room air and to blow it upwards. The vertical variation in concentration produced by this apparatus is shown in Figure 4.4. It may be seen that the vertical difference is quite acceptably small. The formation of stable volumes of heavier gas and lighter air in a room appears to be prevented by mixing the gas with a large quantity of room air. Furthermore, when the heavier gas is mixed well with room air, it seems to take a longer time to sink down and concentrate in the lower part of the room than when a body of heavy gas is floating in the room. The outlet of the gas must be placed at the suction side of the fan.

4.2 Composition of measuring system

The equipment is mainly divided into the tracer gas producing system and the concentration measurement system. A gas producer equipped with a mixing fan is placed nearly at the centre of each room.The control of the rate of production of tracer gas is accomplished by switching on and off the electric current from outside the room.

A thermal conductivity type concentration meter was used for the concentration measurements. Sample air was sucked by a vacuum pump from each room by means of a plastic tube of 6 mm inner diameter. Three sampling points, 0.15 m below the ceiling, midway between ceiling and floor and 0.15 m above the floor, were attached to a metal pole with a

stand which was placed nearly at the centre of the room where the direct draught of the fan did not reach. The air from the three points was mixed and passed to the concentration meter. The air first passed through a set of exchange valves which enabled to select the room to be measured. The air then passed through a U tube containing calcium chloride to absorb humidity in the air and to eliminate errors due to differences in humidity. Finally, the air was transferred to the concentration meter . To avoid any disturbance from previous samples, each sampling was started about fifteen seconds before the appointed time of measurement and continued for thirty seconds. The concentration was read when the indicator became steady. This required about sixty seconds. The time intervals between concentration measurements were therefore 90 seconds. The scheme of instrumentation is shown in Figure 4.5.



a CO_2 gas producer
b sampling points
c electric cables
d plastic tubes
e exchange switch set
f exchange valve set
g water vapour absorber ($CaCl_2$)
h concentration meter
i vacuum pump

Fig. 4.5 Instrumentation for field measurement

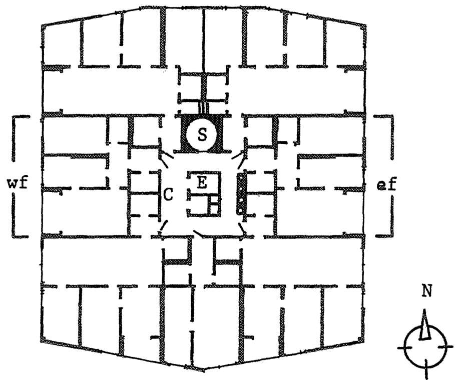
4.3 The blocks of flats used in the test

The measuring method was tested in two tall blocks of flats. The measurements were carried out during both a summer and a winter periods at various floors in each block, in order to clarify the effects of temperature differences on the ventilation systems.

The two blocks were built by prefabrication system and have 16 storeys. One of the blocks was finished in 1960 and is equipped with an extract ventilation system. The other was finished in 1968 and is equipped with a balanced ventilation system. The former block is referred to as Block a and the latter one as Block b.

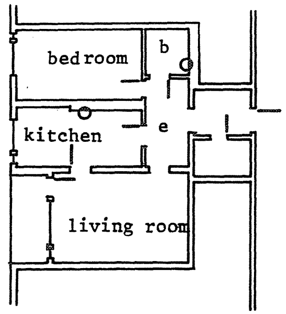
Block a: Point type block with an extract ventilation system. The plan of a standard floor is shown in Figure 4.6. Each floor of the block consists of two vertical shafts, one for the two lifts and the other for a staircase, a corridor and seven flats surrounding them. The total area of a standard floor is 620 m^2. The corridors on every floor are partitioned off from the vertical shafts by means of weather stripped doors. The corridors on each floor have no direct opening to the outside. The corridors on the ground floor and top floor are separated by double doors from the outside.

The windows are double framed wooden windows with weather strips. The total length of the cracks in a measured flat is 35.5 m along the windows and 5.7 m at the entrance door. The door is also fitted with weather strips and has a mail slot with a metal plate seal. The block has an extract ventilation system, fresh air being drawn into each flat through window cracks. The air is removed through two exhaust registers installed in the bathroom and kitchen respectively. The registers are connected to vertical ducts of the respective series. The schematic diagram of extract system is shown in Figure 4.7. Each series of the extract system is made up of two vertical ducts, one for the ventilation of the 1st to 8th floor , and the other for the ventilation of the 9th to the 16th floor. The air flow measurements were carried out in the west flats on the 1st, 7(8)th, 9th and 15th



S stair case
E lifts
C corridor
ef east flat
wf west flat

Fig. 4.6 Plan of point type block of flats



b bath room
e entrance hall
c corridor
o extract register

Fig. 4.8 Plan of west flat

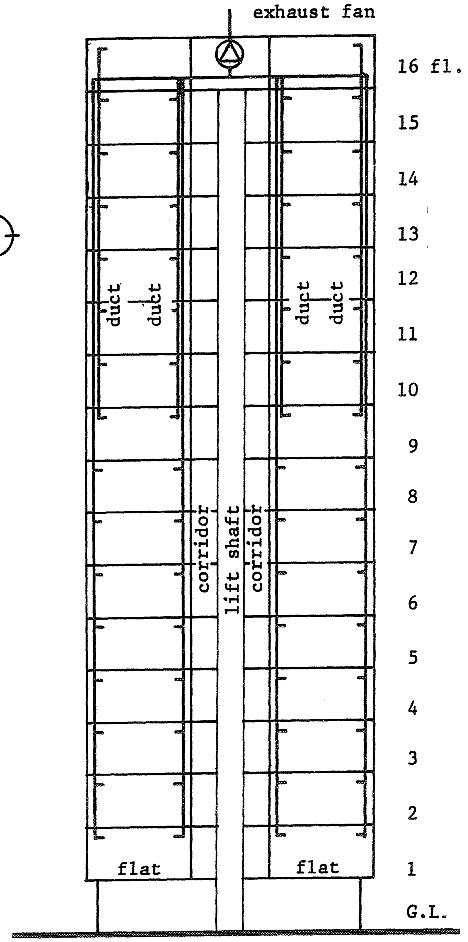


Fig. 4.7 Schematic diagram of extract ventilation system

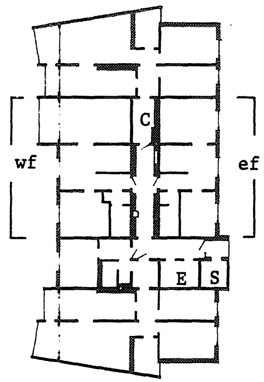
floors, and in the east flats on the 2nd, 7th, 9th and 14th floors.

Two two-room type flats were selected for measurement on each floor. One of these faced nearly west, and the other to east. The plan of a westerly flat is shown in Figure 4.8. The total floor area of each flat is 64 m^2 excluding the balcony. The total volume is 159 m^3.

The measurements were carried out at two periods, in August 1972 and January 1973. The average outdoor temperatures in the summer and winter periods were 19.4 and 1.6°C respectively. The indoor temperature during the winter period was about 22°C. The wind was mostly weak and sometimes moderate during the periods. The wind direction was mainly south-west to north-west.

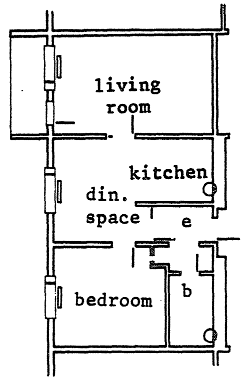
Block b : A semicircular block with a balanced ventilation system. The block is a part of a semicircular building. The plan of the block used for the measurement is shown in Figure 4.9. The plan of a west flat is shown in Figure 4.10. The floor area being 68 m^2. An east flat has the same width but its depth is smaller than that of the west flat, and its area is 54 m^2. The inside volumes are 168 and 132 m^3 respectively. The total length of the window cracks is 33.5 and 27.5 m respectively for a west and an east flat . The perimeter length of the entrance door is 5.7 m, but a mail slot with a metal plate is installed in the door. The measurements were carried out in flats on the 1st, 8th, 10th and 14th floor .

A ventilation system is installed in the penthouse, and heat and humidity controlled air is circulated by fans. The circulating air is passed into and collected from each vertical duct by the respective manifold in the attic. Air supply nozzles and exhaust registers in the same position on every floor are connected to a common vertical duct of respective series. A schematic diagram of the air circulation system is shown in Figure 4.11. Fresh air is supplied through the inlets, which are combined with the three radiators. Exhaust air is sucked



C corridor — wf west flat
E lifts — ef east flat
S stair case

Fig. 4.9 Plan of semicircle block of flats (one section)



e entrance hall — supply register
b bathroom — return register
c corridor

Fig. 4.10 Plan of a west flat

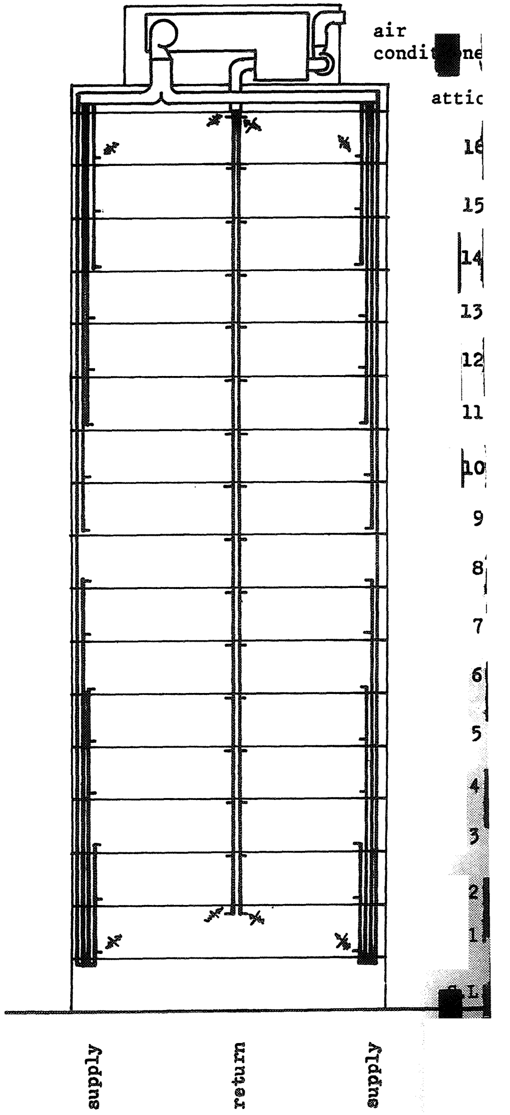


Fig. 4.11 Schematic diagram of balanced ventilation system

out through registers at the top of the wall in the kitchen and the bathroom. The positions of the air inlets and the exhaust registers are shown in Figure 4.10. The air passage between the bedroom and the entrance hall is arranged through a cabinet with grilles at both ends. The dining space and kitchen were treated as one room for purposes of measurement.

Doors for lifts on every floor are fitted with weather strips. The corridor on each floor has a door to a balcony, and the stairwell door faces the balcony. The corridors are therefore connected to the outside, and are separated from the stairwell from the point of view of ventilation.

The periods of measurement were January/February and June of 1972. The average temperatures during these periods were -1.5 and +21.3°C respectively.The main wind direction was south-west. The room temperature in the winter was about 22°C.

During measurements, all measuring and control instruments were placed in an entrance hall in order not to disturb the existing air flow between the entrance hall and the corridor by opening the entrance door. In the flats in Block a, the doors connecting the entrance hall with every other room were slightly open to allow passage for the plastic sampling tubes and the electric wires for gas production control. In the flats in Block b, the doors between the dining spaces and the bedrooms were slightly open for the same purpose. Air passages were cut open at the tops of all other doors, and the tubes and wires were passed through these. The concentration was read at intervals of 600 seconds in each room of Block a, and 450 seconds in Block b.

Carbon dioxide gas was given off by the respiration of the measurement operator. A constant gas production of 0.028 g per second was added to the term denoting the steady production by heat penetration in the gas producer in the entrance hall when the air flows were calculated,in order to correct for the production by the operator.

One measurement usually took about two and a half hours. A measurement in a flat with a large air flow rate took a longer time,and the converse is also true. During a measurement, the consumption of dry ice was 1.0 to 2.5 kg in each room. The quantities depended on the volume of a room and the rate of air flow in a room. These rates of production correspond to volumes of 0.061 - 0.156×10^{-3} m^3/sec. During measurement the doors of large closets and wardrobes were kept open.

4.4 The results of measurements

Carbon dioxide gas was produced intermittently and alternately in every room of a flat. During the same time, measurements of the variation in concentration were made in every room. An example of the pattern of gas production (heat supply to the gas producer) and the variations in concentration in every room of a flat is shown in Figure 4.12. The maximum concentration reached during measurement was about 16 g of CO_2 gas per 1 m^3 of air, which is equivalent to 1.3% by weight. The flat of the example is the west flat on the 15th floor in Block a. All the windows of the flat were closed during the measurement.

The air flows between the rooms were calculated by the second computer program from the production of tracer gas and the measured variations in concentration of the gas in every room. The calculated air flows are shown in Figure 4.13 d as air flows in the winter.

In order to examine the results,the variations in concentration in every room were calculated from the patterns of gas production and the resultant air flows by Equation 3.29 at time intervals of 180 seconds. The calculated variations in concentration are compared with the measured ones in Figure 4.12. There is good agreement between measured and calculated concentrations in the entrance hall. In the bathroom and kitchen, the measured concentrations were lower than the theoretical values; this is true particularly when the concentrations were at the tops of the waves of variation. This may mean that production of gas by the apparatus was smaller than the values calculated on the basis of the apparatus constants, or that the air flow rates were larger than those given by the calculations. There was a delay between the experimental variations in concentration in the bedroom and living room, and the theoretical variations. The volumes of these two rooms are larger than those of the others (35 and 45 m^3 respectively). There is better agreement between the empirical and theoretical values when the variation is calculated by applying a smaller time constant for the gas producer . In measuring the air flow in a single room, an average value may be obtained even when there is a time delay in the variation in concentration when the same numbers of starts and stops in gas production are repeated at equal time intervals, because positive and negative deviations balance out.

4.4.1 Air flows in Block a

Air flows in flats on each storey of the block with the extract system are shown in Figures 4.13 a-h. The measurements were carried out with every window of the flat closed. Many windows in other flats were kept open during a measurement, especially in the summer when windows were open in most of the flats. Entrance halls in the block are T-shaped in plan, and an extra fan was therefore used to ensure mixing of the tracer gas in the entrance hall. But placing of the fan in different flats was varied depending on the positions of pieces of furniture. In some of the flats an interchange of air between the entrance hall and the living room or a bedroom was found in the results. This may be due to excessively strong circulation and to incorrect placing of the extra fan. **The figure in the parenthesis in each kitchen is the total volume, which was extracted from the kitchen.**

During winter measurements, the door between the kitchen and the entrance hall was removed in the west flat on the 8th floor. The two rooms were thus treated in air flow calculation as one room, and the air flows from the corridor and from the outside could not be distinguished. The total values of these are shown in Figure 4.13 b. Air flow measurements were also carried out with the ventilator(s) of a living room and/or bedroom about 5 cm open. The ventilator is the same structure as the main window, and its size is 1.20 m x 0.20 m.

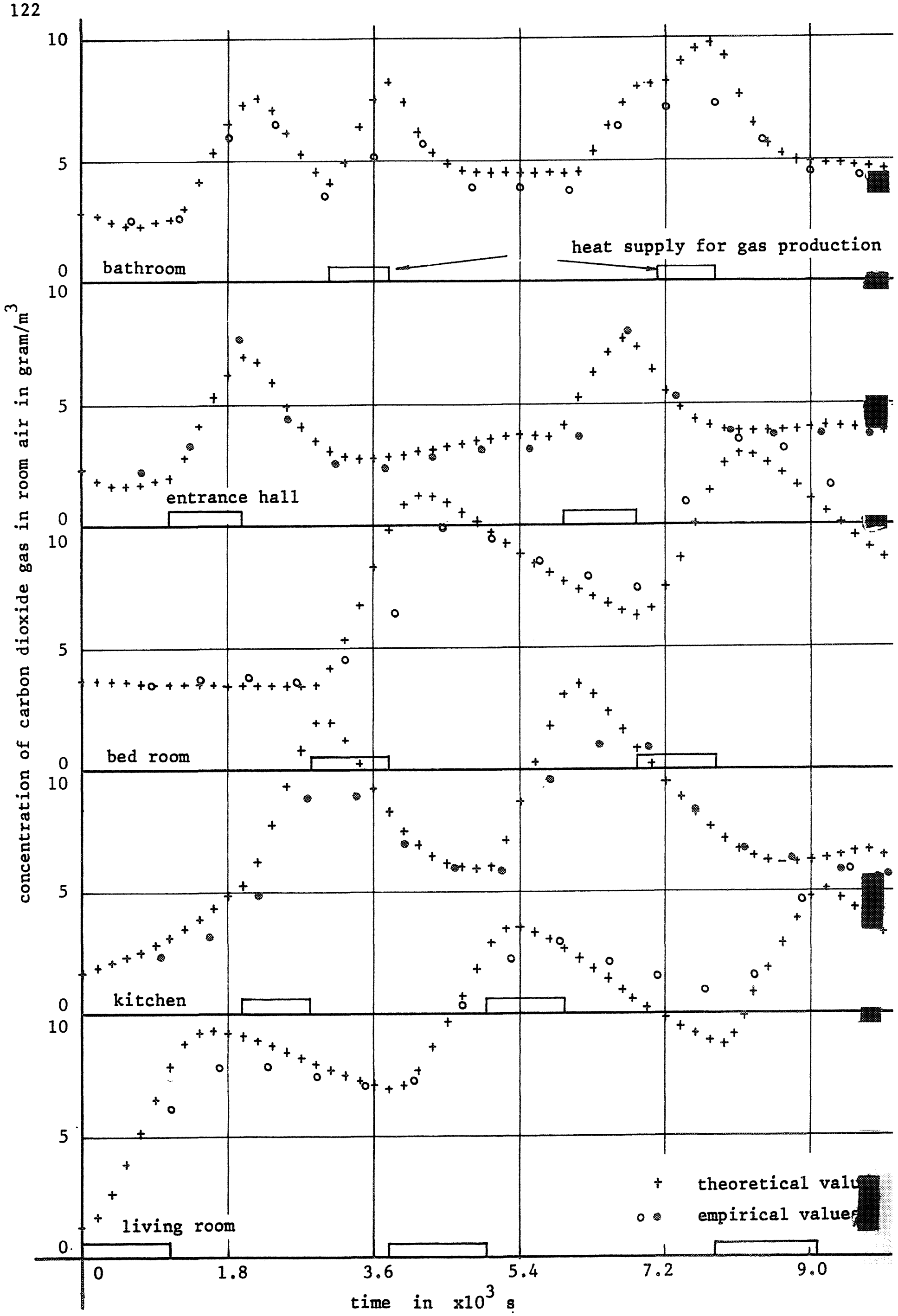
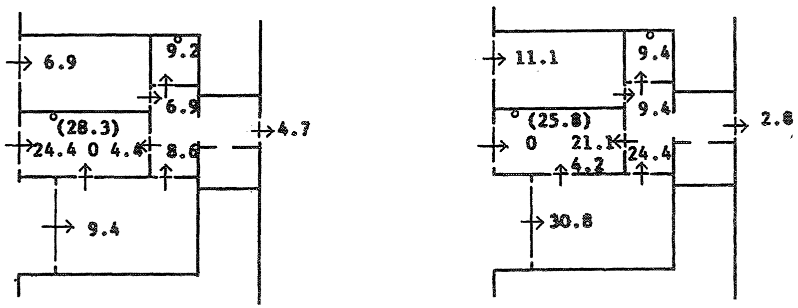


Fig. 4.12 Comparison of empirical and theoretical concentration

4.4.2 Air flows in Block b

The air flows in the flats of Block b are shown in Figures 4.14 a-f. The air flow through the passage between a bedroom and an entrance hall was smaller in every flat than the design value. The source of air flowing into a bedroom, a dining space or a living room could not be distinguished, whether it was drawn in from the outside or whether all air was supplied by the ventilation system. But observation of the results shows that in most flats the air change rates in bedrooms and living rooms were higher than those received in the dining space and the entrance hall. This seems to indicate that the pressure in the block was slightly above that outside. No measurement of the pressure was carried out. The difference between the air supplied to a room and the air received by the adjacent room(s) is shown in brackets at the window of each room.

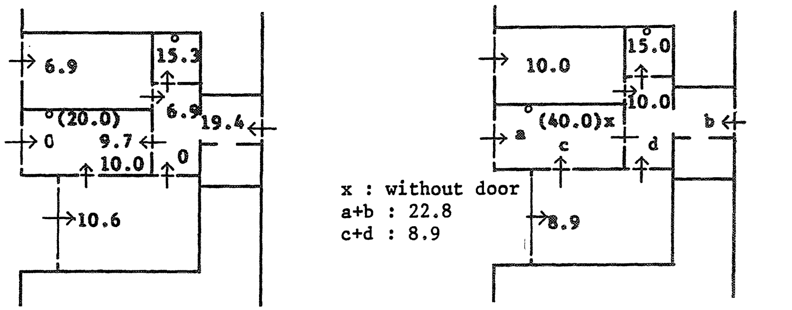


summer

winter

unit : $\times 10^{-3}$ m^3/s

Fig. 4.13 a Air flows in west flat on 1st floor

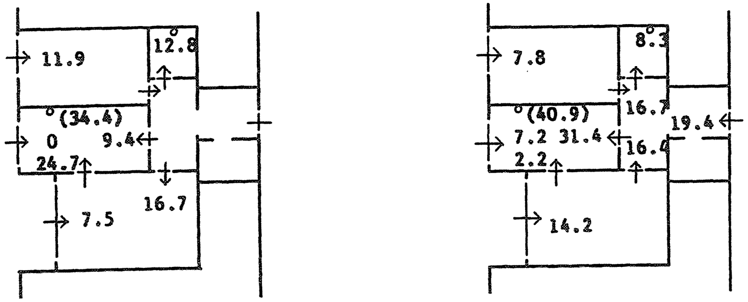


summer (7F)

winter (8F)

unit : $\times 10^{-3}$ m^3/s

Fig. 4.13 b Air flows in west flat on 7(8)th floor

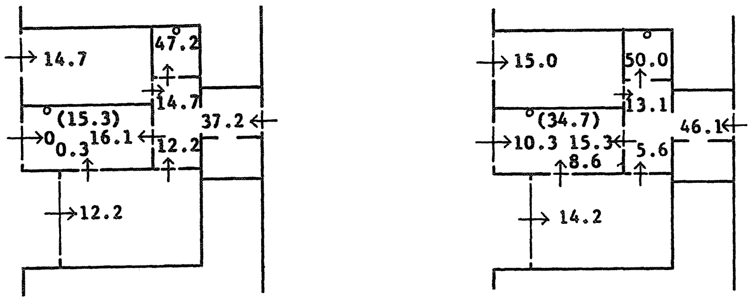


summer

winter

unit : $\times 10^{-3}$ m^3/s

Fig. 4.13 c Air flows in west flat on 9th floor

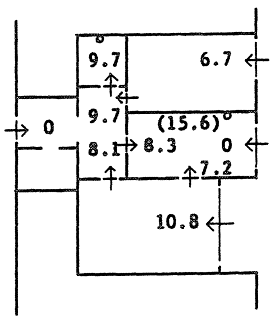


summer

winter

unit : $\times 10^{-3}$ m^3/s

Fig. 4.13 d Air flows in west flat on 15th floor



winter

unit : $\times 10^{-3}$ m^3/s

Fig. 4.13 e Air flows in east flat on 2nd floor

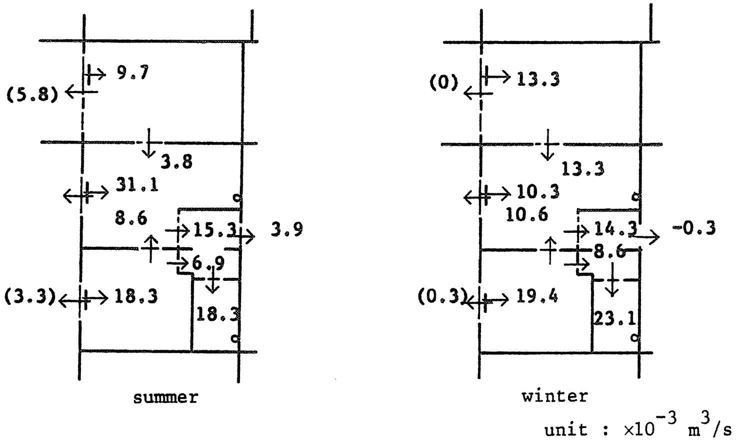


Fig. 4.14 a Air flows in west flat on 1st floor

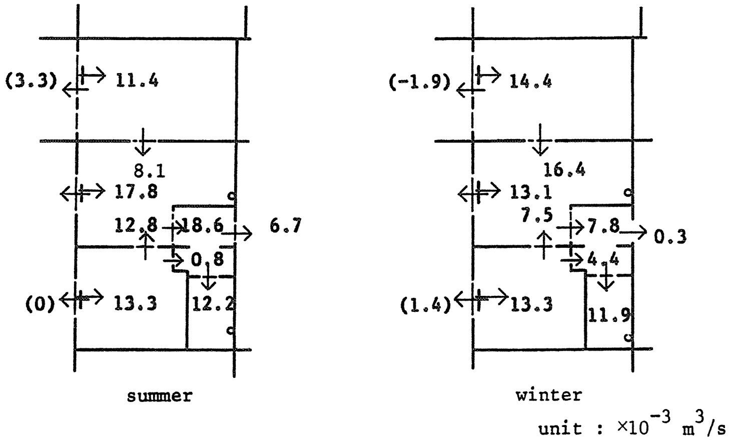


Fig. 4.14 b Air flows in west flat on 8th floor

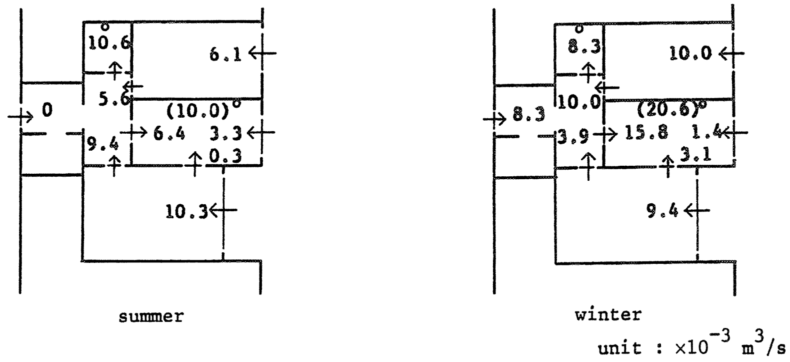


Fig. 4.13 f Air flows in east flat on 7th floor

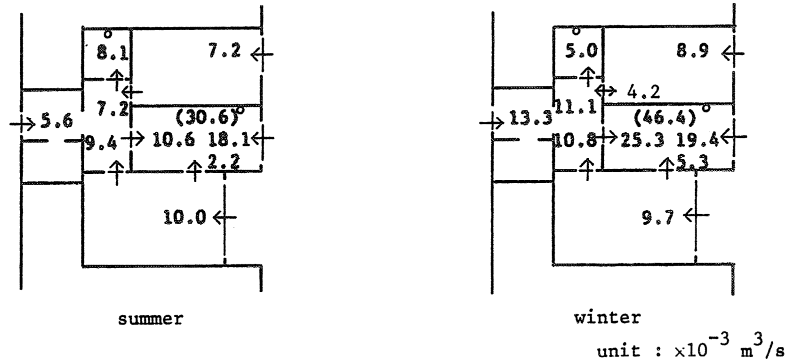


Fig. 4.13 g Air flows in east flat on 9th floor

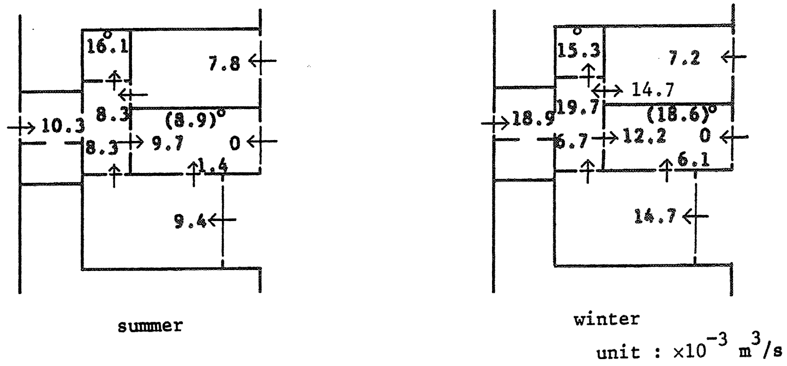


Fig. 4.13 h Air flows in east flat on 14th floor

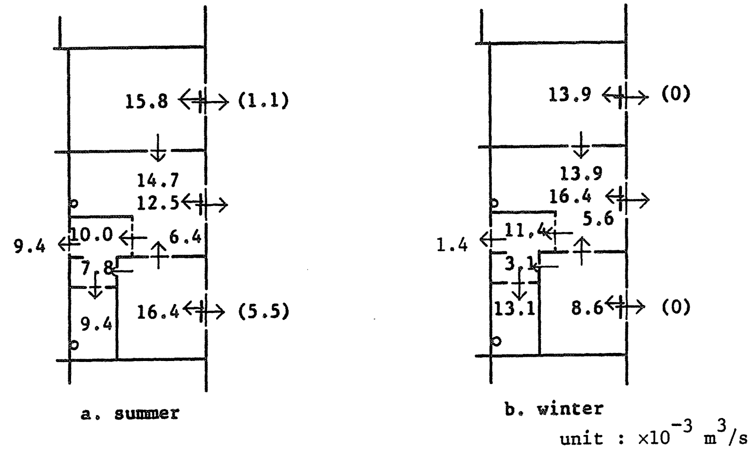


Fig. 4.14 e Air flows in east flat on 8th floor

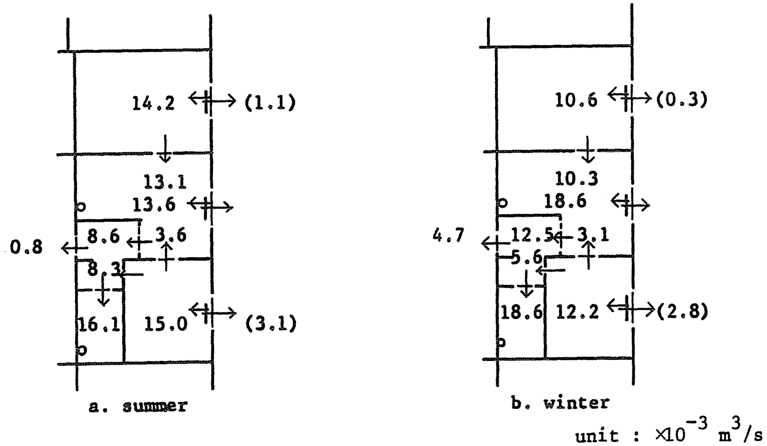


Fig. 4.14 f Air flows in east flat on 14th floor

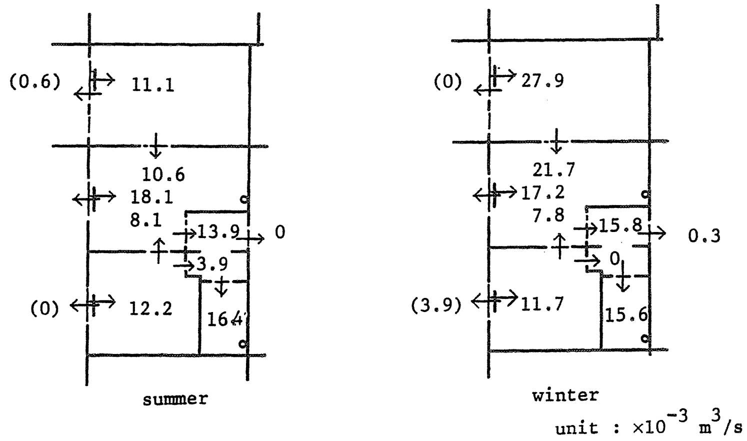


Fig. 4.14 c Air flows in west flat on 10th floor

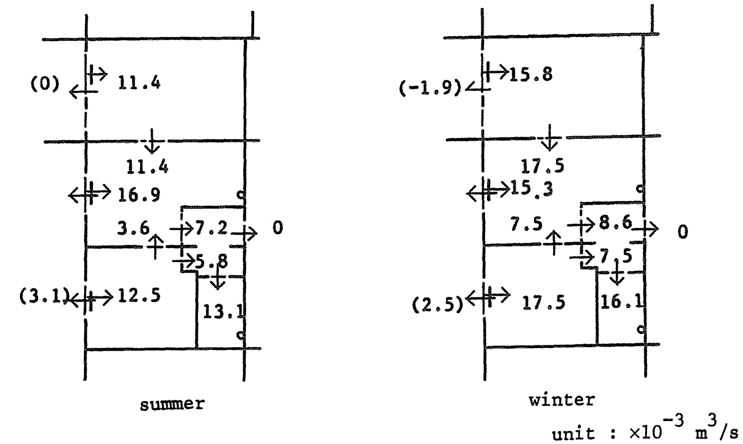


Fig. 4.14 d Air flows in west flat on 14th floor

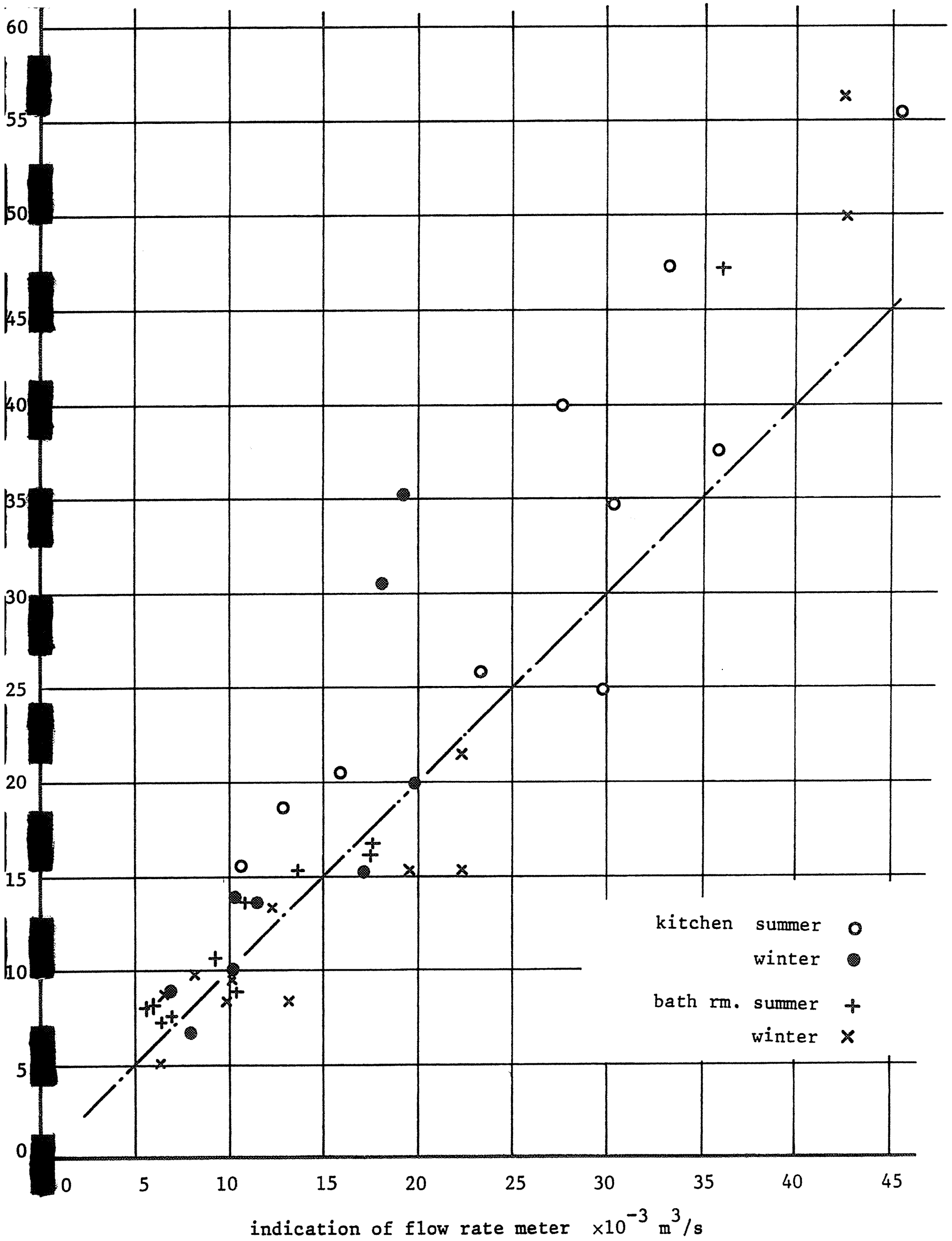


Fig. 4.15 Examination of the results of the presented method from the block with extract ventilation system

4.5 The precision of the measurements

The method presented is made up of parts such as tracer gas production, concentration measurement, and calculation of air flows by a computer program. Each of these will be subject to a certain range of errors.

As explained in 3.4, the error in calculating air flows can be brought below 3 % if variations in concentration at time intervals of 720 seconds are fed into the computer. In field measurements, concentrations were read at intervals of 600 seconds (450 seconds in Block b) in each room. In addition, the patterns of tracer gas production in every room were decided very carefully in order to distinguish clearly every air flow. The error which arose in the process of air flow calculation itself may be in the order of several per cent.

Errors in tracer gas production and concentration measurement are inevitable. Tracer gas production is slightly affected by the amount of dry ice in the producer, the standard point of the concentration meter was very often changed during a measurement, and the electric current of the hot wire was also changed during a measurement. Even if such disturbances were to be compensated for in the computer program by assuming a linear change for every one of them, the effects of such compensations should be limited.

Further, as explained in 4.4, a time lag was found in rooms with large volumes between the pattern of tracer gas production and the variation in concentration. This problem could not be solved in this study. The air flows of rooms with large volumes were instead recalculated by applying 20% smaller values for the time constants of the producers in such rooms.

In addition, air flow was not steady during each measurement. When the air flow rates at the extract air registers were observed with a hot wire flow rate meter, the indicator of the meter showed a range of fluctuation over an observation period of about one minute. This was probably due both to external and internal disturbances in the blocks. A steady state of air flow is a premise of this method of calculation. Suppose a fluctuating air flow from room a to room b. If a period of high concentration in room a coincides with a period of higher than normal rate of air flow from room a to room b, the air flow calculated will be higher than the mean during the measurement, and the converse is also true. To avoid this type of disturbance, the production of tracer gas in each room was divided into several periods. Even then the error due to fluctuations in the air flows is likely to be considerable.

The overall precision of this method must therefore be examined. No method was available the results of which can be directly compared with the results of the method presented. The air in the bathrooms of Blocks a and b was considered to be sucked out exclusively by the mechanical extract system,and no air interchange was supposed to take place between a bathroom and an entrance hall. In Block a, the air flows of kitchens can also be regarded as unidirectional. The air flow rates at the return registers in these rooms were measured with the hot wire type flow rate meter just before or after most of the measurements by this method were carried out.

The results of the method presented are compared with the readings of the flow rate meter. Figure 4.15 shows the comparison of the flow rates obtained in Block a during summer and winter measurements. The line corresponding to equality between the reading of the air flow meter and the results of the method presented is drawn in the Figure. The points are spread over a considerable range around the equivalence line. The number of points in the upper part of the equivalence line is larger than in its lower part. In cases of higher flow rates, the results of the method presented have shown a tendency to have an extreme upward deviation. The coefficient of correlation between the results of the method presented and the flow rate meter is 0.93. The regression equation is

$$v_t = 1.28\, v_r - 0.0019$$

where v_t is a flow rate by the tracer gas method and v_r is a reading of the flow rate meter.

The results of the method presented and the readings of the air flow meter are compared in Figure 4.16 by using the air flows in the bathrooms in Block b. The coefficient of correlation is 0.70 in this block. The regression equation is

$$v_t = 0.79\, v_r + 0.0056$$

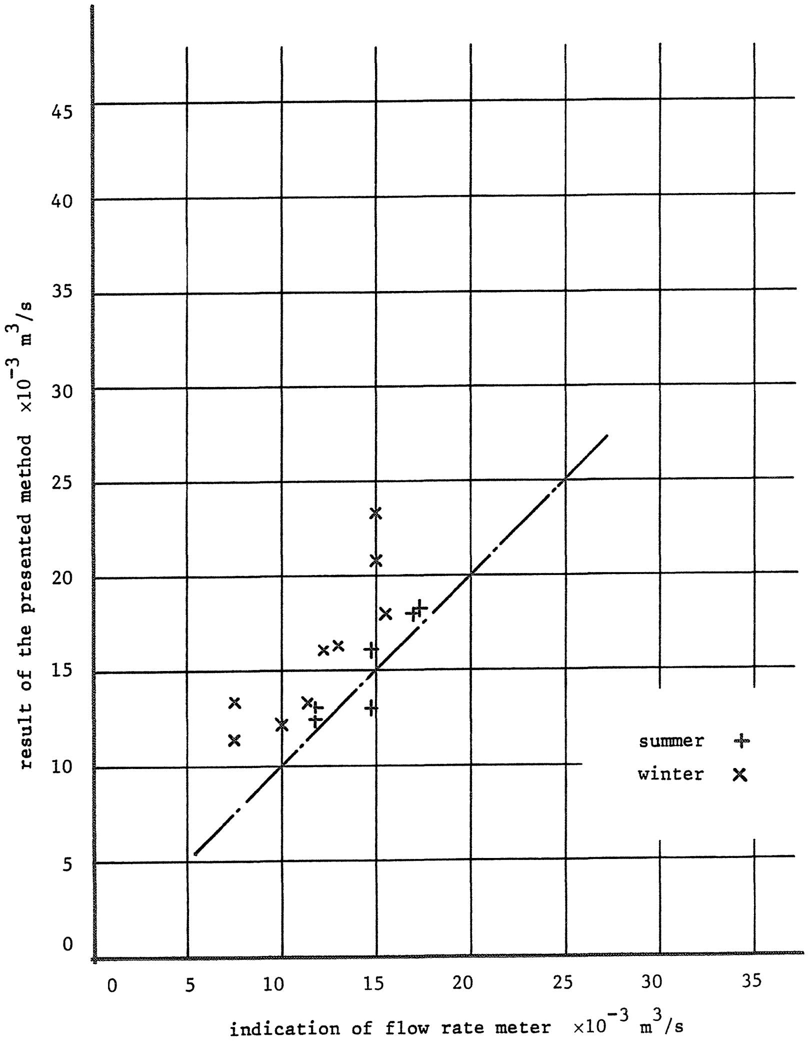


Fig. 4.16 Examination of the results of the presented method from the block with balanced ventilation system

4.6 Examination of the results

The air flow in a building varies from time to time according to the weather conditions and the living conditions of the occupants. The results of these measurements are the average values of two to three hours, and the measurement in each flat was not performed concurrently. Strictly speaking, therefore, the results of every case cannot be compared with the others. Some evident and reliable specific features of the air flows in the blocks are pointed out in this section. The air flow from an entrance hall to a corridor cannot be measured directly by this method. When the volume of air extracted from an entrance hall exceeded the total volume of air which was received by the adjacent rooms in the flat, the difference was allotted to the air flow from the entrance hall to the corridor.

4.6.1 Total air change of the flats in Block a

To examine the effect of the mechanical ventilation system and other effects, the total of the extracted rates from the bathroom and kitchen of each flat is named the total extract volume of the flat. The air flow from the entrance hall to the corridor was measured in the west flat on the first floor, but it is not included in the total extract volume of the flat.

The total extract volumes of the summer measurement are listed in Table 4.2. These figures were obtained from measurements with the windows of the flat in question closed. In the summer a stack effect would be negligibly small. The air change of the west flat on the 15th floor was very much larger than that of the other flats. The total extract volume becomes smaller when the floor on which the flat is situated is lower.

On average, the total extract volumes of the east flats were smaller than those of the west flats.

The extract registers are the adjustable type, and every register was regulated, on completion of the system twelve years ago, so as to provide an equal air flow in every flat. The unevenness of flow rates by floor level may be due to one or both of the following causes.

Table 4.2 Total extract volumes of flats of Block a

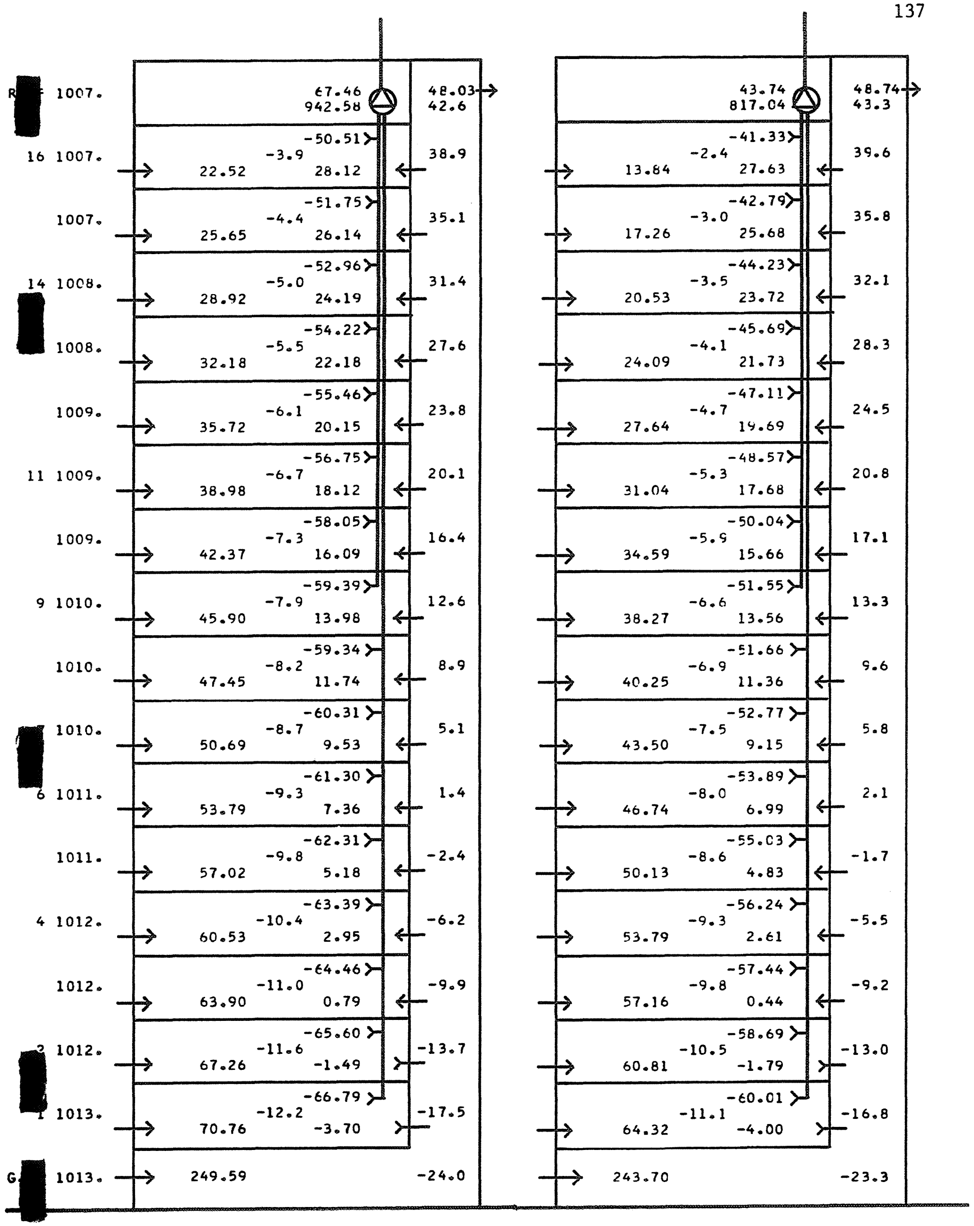
floor	orientation	summer $\times 10^{-3}\ m^3/s$	winter $\times 10^{-3}\ m^3/s$	winter/summer ratio
15	west	62.5	84.7	1.35
14	east	25.0	33.9	1.36
9	west	47.2	49.2	1.04
9	east	37.2	51.4	1.38
7(8)	west	36.1(7f)	55.0(8f)	(1.52)
7	east	20.6	28.9	1.41
1	west	37.5	35.3	0.94
2	east	-	25.3	-

a. Increase in the flow resistance of the vertical ducts due to accumulation of dust.

b. Increase in the flow rates in the vertical ducts due to windows and registers in kitchens in other flats on the common duct being opened.

By network calculation of the air flow in the block, the effects of an increase in the flow resistance of the vertical ducts can be verified. Firstly the extract ventilaion system is supposed to be regulated so as to provide a ventilation rate of 0.05 m^3/s equally in the flats of each floor. Here the pressure drops at the window crack, the extract register on the 1st floor and the vertical duct of one floor height are designated to 8.6, 50 and 2 Pa respectively at the design flow rate. The air flow in this block is changed as shown in Figure 4.17 a when the pressure drop in the vertical duct of one floor height is increased to 5 Pa. The air extraction on the 16th floor is increased slightly, but that on the 1st floor is decreased as much as 13 %.

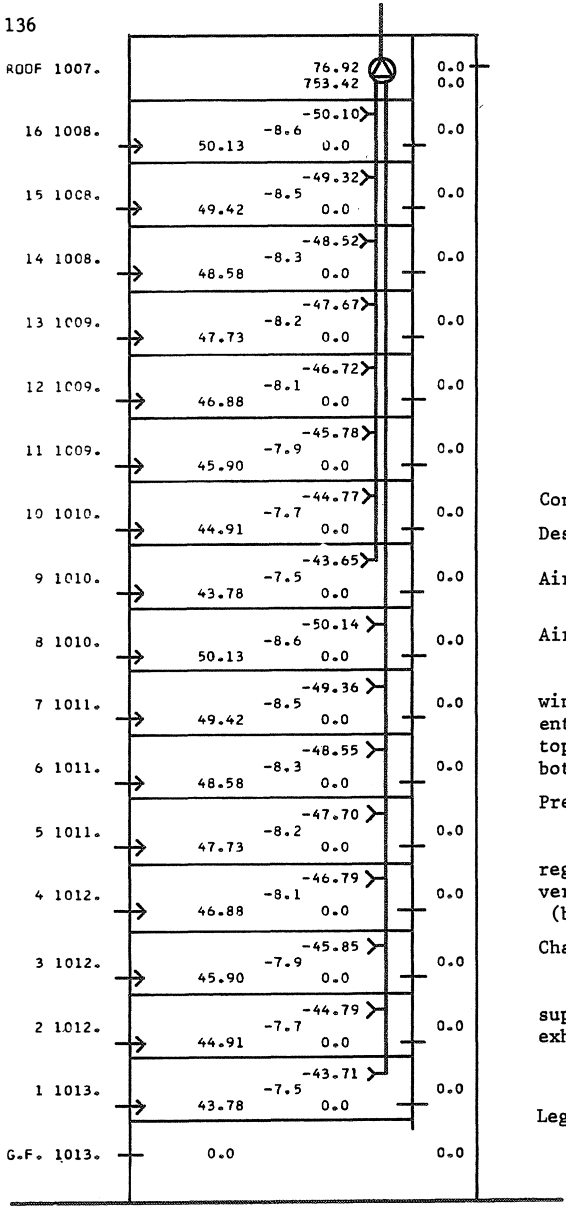
The total extract volumes of the winter measurement are also shown in Table 4.2. The average outside temperature was 1.6°C during this period. In the winter, the total extract volume was increased by 35% in the west flat on the 15th floor, and was decreased by 6% in the west flat on the 1st floor. The extract volumes of the east flats were generally increased in the winter. In the summer, most of the flats on the common vertical duct of the flat in question seemed to keep their windows open. The flow resistance of the flat under measurement was therefore very much



b. effect of air infiltration through bottom of vertical shaft

c. reduced effect of ventilation fan

Fig. 4.17 Ventilation of tall block of flats with extract ventilation system



Conditions for case a

Design ventilation rate 0.05 m^3 [redacted]
(each floor)

Air temperature	outside	20.0 °C
	indoor	20.0

Air flow coefficients of cracks [redacted]

	a $m^3/m\ s\ Pa^{1/\beta}$	[redacted]
window	0.0731×10^{-3}	80
entrance door	-	-
top of shaft	-	[redacted]
bottom of shaft	-	

Pressure drop (at design flow rate)

	supply	ret[redacted]
register (1st. floor)	- Pa	50 [redacted]
vertical duct (between each connection)	-	2 [redacted]

Charactor of fans

	κ_1 Pa	κ_2 Pa/(m^3/s)
supply	-	-
exhaust	114.6	50.0

Legend : see Fig. 1.5 on p. 35

a. changes in ventilation rates of various floors by increase of flow resistance of vertical ducts

Fig. 4.17 Ventilation of tall block of flats with extract ventilation syste[redacted] larger than those of the other flats on the common duct. This may be the reason why, on average, the summer measurements produced lower results than the winter measurements. If the airtightness of the entrance door of every flat is very high, the extract ventilation rate increases on the lower floors and decreases on the upper floors. It is deduced from this that the airtightness of the entrance doors is not sufficiently high, and that a large quantity of air flows into the vertical shaft at ground level.

The pressure differences between corridors and entrance halls was also observed during the period of winter measurement. The observation showed that the pressure in the corridor was about 40 Pa higher than that in the entrance hall on the 15th floor, and the former was about 3 Pa lower than the latter on the first floor. The level of the zero pressure difference was around the 5th or 6th floor. When the airtightness of every part of the vertical shaft is very high, the pressure in the shaft is calculated by Equation 1.2 as it is 30 Pa lower on the ground floor, and 30 Pa higher on the 16th floor, than the pressure outside at the same level. When the vertical shaft has a large opening at ground level, the pressure at this level becomes nearly equal to that outside and that at the 16th floor becomes 60 Pa higher than that outside. In the example in Figure 4.17 b, air flow coefficients ($a \times l$) of 0.012 and 0.0012 $m^3/s\ Pa^{1/\beta}$ are allotted to the ground and top levels of the vertical shaft. The flow directions at the entrance doors change between the 2nd and 3rd floor.

4.6.2 Total air supply in Block b

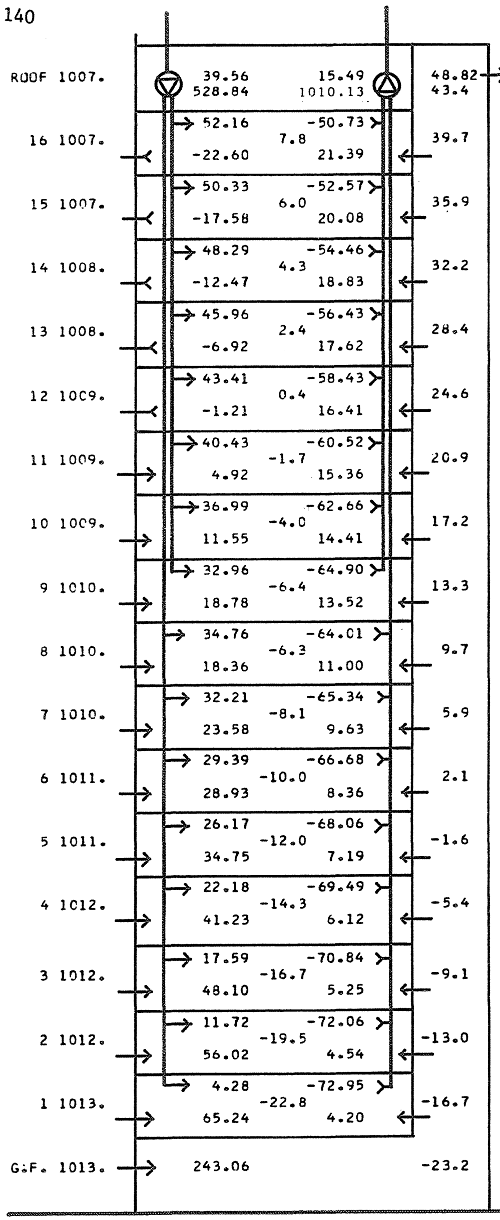
As mentioned in 4.4.2, the inside of Block b seemed to be maintained at a slightly higher pressure than that outside. The total air supply for the bedroom, dining space and living room in each of the west flats is shown in Table 4.3. The total supply volume of the west flat on the 1st floor was decreased by 27 %, while that on the 14th floor was increased by 19 % in the winter. The imbalance between the decrease and the increase was not so large in this Block as that in Block a. This shows clearly the difference between

Table 4.3 Total supply volumes of flats of Block b

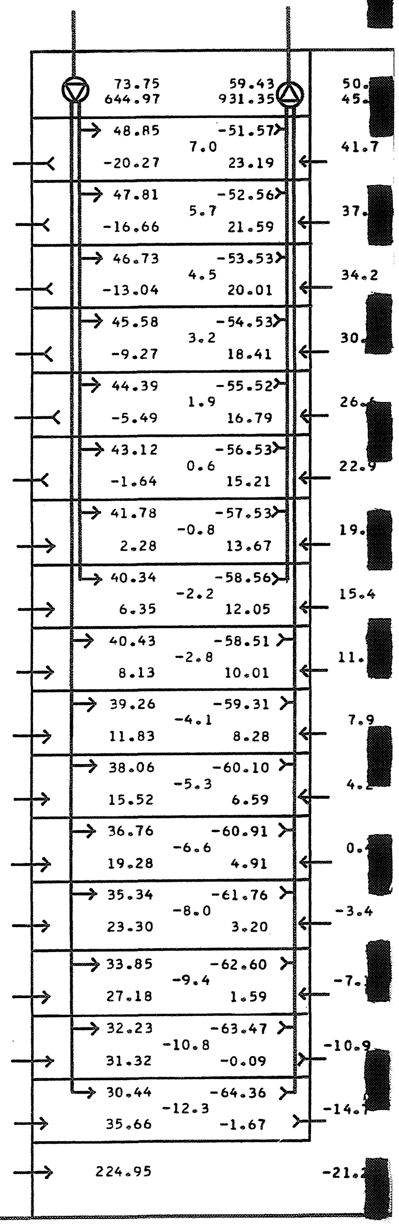
floor	orientation	summer $\times 10^{-3}\ m^3/s$	winter $\times 10^{-3}\ m^3/s$	winter/summer ratio
14	west	40.8	48.6	1.19
10	west	41.4	50.8	1.23
8	west	42.5	40.8	0.96
1	west	59.2	43.1	0.73

the ventilation systems in the two blocks. In the block with a balanced system, the opening of windows in flats on the common duct does not produce so significant an effect as in the block with an extract system. A stack effect can be the dominant cause of the change in air flows in the summer and winter in a balanced system. The increase on the upper floors and the decrease on the lower floors in the winter, were here found again. It seems like the effect of air infiltration from the corridors to the flats strongly influences these changes.

A stack effect cannot take place inside of a well thermally insulated building, because the air temperatures in the supply and return ducts and the flats are kept nearly equal. But in the cold season the relative effect of the supply fan is decreased and that of the exhaust fan increased as long as the openings for air intake and outlet are situated at the top of a block. Rydberg (1959) gave methods of restricting the air leakage across an exterior wall due to the stack effect in a tall block with a balanced ventilation system, by designing the air circulation system for a high driving pressure. The ventilation of Block b was examined according to his theory by a network method. The assumed values of the ventilation system are: design ventilation rate $0.05 m^3/s$ for each flat, design pressure loss in vertical duct 2 Pa between each connection , design pressure loss at supply and extract registers 10 Pa, and the effect of the supply and exhaust fans 26 Pa at the design flow rate. The air flows and pressures in the block are shown in Figure 4.18 a for the case when the air temperature inside the block is $20^{o}C$ and that outside is $-10^{o}C$. In the flat on the 1st floor, the air supply by the system is reduced to 9 % of the design rate, and the increase in extraction is 46 %.

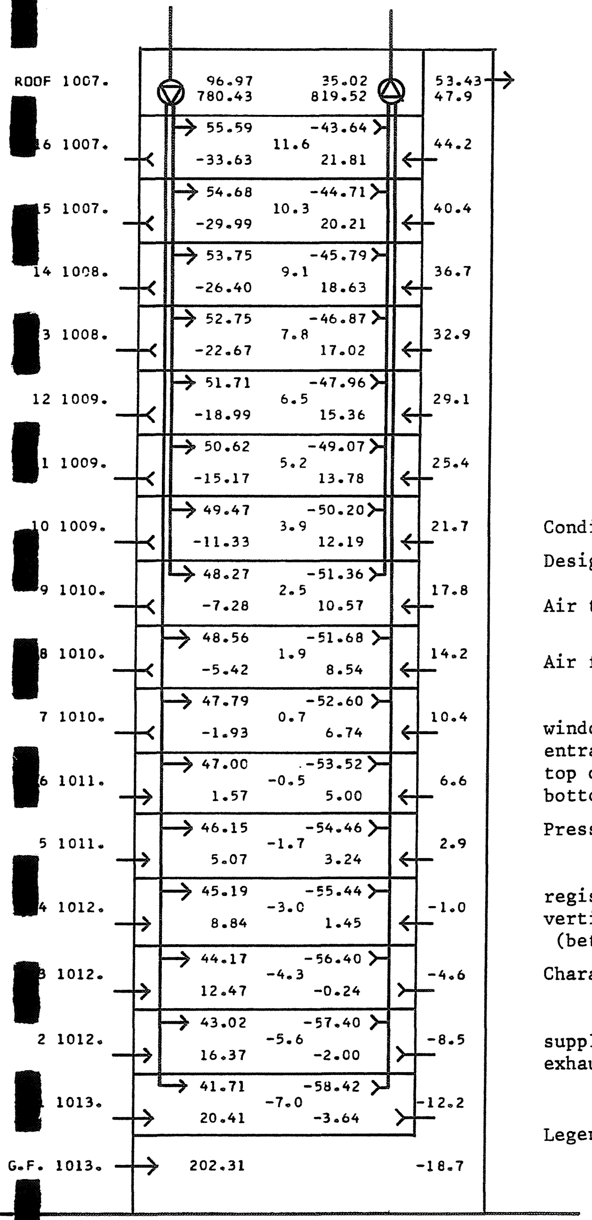


a. stack effect on ventilation system with low flow resistances at registers



b. stack effect on ventilation system with high flow resist[ances] at registers

Fig. 4.18 Ventilation of tall block of flats with balanced ventilation system



Conditions for case a

Design ventilation rate $0.05\ m^3/s$ (each floor)

Air temperature outside $-10.0\ ^{o}C$, indoor 20.0

Air flow coefficients of cracks

	$a\ m^3/m\ s\ Pa^{1/\beta}$	l m
window	0.0731×10^{-3}	40
entrance door	0.1	7
top of shaft	0.5	24
bottom of shaft	0.1	12

Pressure drop (at design flow rate)

	supply	return
register (1st. floor)	10 Pa	10 Pa
vertical duct (between each connection)	2	2

Charactor of fans

	κ_1 Pa	κ_2 Pa/(m^3/s)
supply	66.0	50.0
exhaust	66.0	50.0

Legend : see Fig. 1.5 on p. 35

c. flow condition by compensation of effects of fans for stack effect

Fig. 4.18 Ventilation of tall block of flats with extract ventilation system

At the same time, the supply on the 16th floor is increased by 4 % and the extraction is increased by 1 %.

Figure 4.18 b shows the case, when the pressure drops at the supply and extract registers are increased to 50 Pa each and consequently the effects of the fans are increased to 66 Pa each. The change on the 1st floor due to the stack effect are 39 % of decrease in the supply rate and 29 % of increase in the extract rate, and those on the 16th floor are 2 % of decrease in the supply rate and 3 % of increase in the extract rate. The air infiltration through the window cracks of the first floor is reduced from 0.065 to 0.036 m^3/s by increasing the pressure of the ventilation system.

Rydberg's paper also mentions that the change in ventilaiton in winter due to changes in outside temperature can be restricted by the installation of a pair of flap control valves at the middles of the vertical supply and extract ducts. For the same reason, the stack effect can be compensated for by changing the effects of the supply and exhaust fans. Figure 4.18 c shows the air flows in the block, calculated merely by adding this change to the above example. The effect of the supply fan was increased, and that of the exhaust fan decreased, by 30 Pa which is approximately equivalent to half the stack effect due to the temperature difference of 30°C. As a result of this change the deviations are averaged, the decrease in supply and increase in extraction on the first floor are both 17%, and the increase in supply and decrease in extraction on the 16th floor are 11% and 13% respectively. The air infiltration through the window crack on the 1st floor is reduced to 0.020 m^3/s

4.6.3 The effect of opening windows

In the block with an extract system, the total volume extracted from each flat was increased as a result of the windows in all flats being opened, with the one exception of the east flat on the 7th floor during the winter measurement, see Table 4.4. The rates of increase were under 25 %, and averaged 16%. The theoretical treatment of the change in air extraction rate due to a window being opened and the flow resistance of every part of the ventilation system is given by Rydberg and reproduced as Equation 1.9.

According to his equation, when the pressure drops at a window crack and one floor height of vertical duct are 20 Pa and 2 Pa respectively at the design air extraction rate, the pressure drop in the extract register on the first floor must be larger than the following value in order that the increase in extraction rate in the flat should be less than 16%:

$$P_r = \frac{20 - 2\cdot(0.16)^2\cdot 2.72 - 2\cdot 2\cdot 0.16\cdot 1.53}{2\cdot 0.16 + (0.16)^2} = 54.6 \text{ Pa}$$

No relationship could be found between the change in flow rate and the outdoor temperature, or the levels of the flats. Wind is considered to have a strong effect on this change.

When a window was open, almost all the air was sucked through it, and there was a remarkable decrease in the ventilation of the other rooms in a flat of the block with an extract system.

The changes in the air flow rates of the rooms which were caused by the windows in them being opened are compared in Table 4.4. Due to the windows being opened, the air flows were increased to 1.8 - 6.0 times the values with the windows closed. No difference could be found between the changes in the summer and winter measurements.

A ventilator of size 1.20 m x 0.20 m is fitted at the side of each window in the block with the balanced system. The ventilator has a rough fibrous plastic filter inside a slotted metal plate. Air flows in each flat were also measured with the ventilator in the bedroom open. The air change in the rooms was increased in three flats but it showed a decrease in the west flat on the 8th floor. The air flows from the bedroom to the dining space and the entrance hall were decreased in every flat. This may also be the reason for the slight excess pressure inside the block. No backflow from a dining space or entrance hall to a bedroom could be discerned even when the window in the bedroom was opened. The changes due to their windows being opened varied so irregularly in each flat that it was impossible to prove any effect due to the level of the flat or the outside temperature.

Table 4.4 Changes in total extract volumes by opening windows

floor	orientation	summer open/closed volume $\times 10^{-3}$ m^3/s	summer rate of change	winter open/closed volume $\times 10^{-3}$ m^3/s	winter rate of change
15	west			93.1/84.7	1.10
14	east	26.9/25.0	1.08		
9	west	57.5/47.2	1.22	61.4/49.2	1.25
9	east	43.3/38.6	1.17	55.8/51.4	1.09
7(8)	west			67.8/55.0	1.23
7	east	22.5/20.6	1.09	28.1/28.9	0.97
1	west	37.5/37.5	1.00	38.1/35.3	1.08

Table 4.5 Changes in ventilation of rooms by opening own window

floor	orientation	summer open/closed volume $\times 10^{-3}$ m^3/s	summer rate of change	winter open/closed volume $\times 10^{-3}$ m^3/s	winter rate of change
15	west			40.8/14.2	3.3
14	west	20.0/9.4	2.1		
		21.4/7.8	2.7		
9	west	16.1/7.5	2.1	50.0/14.2	3.5
		22.2/11.9	1.9		
9	east	16.4/10.0	1.6	33.9/9.7	3.5
		24.7/7.2	3.1		
8	west			53.9/8.9	6.0
7	east	20.8/10.3	2.0	22.5/10.0	2.3
		10.8/6.1	1.8		
1	west	25.0/9.7	2.6	57.8/30.8	1.9
		27.2/6.9	3.9		

Opening a window is a usual measure when the occupant feels a lack of fresh air in a room. Even though the change in total air extraction from a flat, due to a window being opened, is limited the air change rate of the room may be increased for the following reasons:

a. continuous alternations of inward and outward flow due to the pulsations of the wind,

b. a steady interchange between a room and the outside due to the temperature difference.

Van der Held (1953) proposed an equation to calculate the air change due to wind oscillation, assuming the nature of the oscillation to be sinusoidal. The theory of oscillation was applied to the method, the room air being treated as an elastic body and the flow resistance at an opening as a damper. He also gave several numerical solutions. The following conditions were assumed for purposes of calculation.

volume of room	47 m^3
flow resistance of opening	0.139×10^{-3} s Pa/m^3
intensity of oscillation	9.8 Pa
period of oscillation	1 second

The results showed that the rate of air exchange due to the pulsations was 0.141 per hour (0.0018 m^3/s).

He also explained the case where a room is adjacent to a space and the air is sucked to it. The flow resistance between the spaces was assumed to be 0.0347×10^{-3} s Pa/m^3. The following values of the air change of the room were given for three different values of the pressure difference between the outdoors and the adjacent space.

pressure difference	air change	rate of ventilation
0 Pa	0.443 h^{-1}	0.0058 m^3/s
9.8	0.487	0.0064
29.4	1.022	0.0134

The combined effect of thermal and mechanical ventilation on the air change of a room is calculated on the basis of the displacement of the neutral zone. An air interchange through a vertical slit in an exterior wall is due to a temperature difference between the room and the outside. If air is sucked out by a ventilation system from the room, the height of the neutral zone rises in a cold season in order to balance the flow rates in the room. An example of a numerical solution is given below.

conditions:	room temperature	20°C
	outdoor temperature	0°C
	specific density of outside air	1.293 kg/m^3
	room air	1.205 kg/m^3
	width of window opening	0.05 m
	height of top of window	1.8 m
	bottom of window	0.6 m
	contraction coefficient	0.8

solutions:	when a mechanical system is not in operation.		
	height of neutral zone		1.29 m
	air flows at window	inflow	0.038 m^3/s (0.033 m^3/s
		outflow	0.038 m^3/h at 0°C)
	When 0.017 m^3/s of air is sucked out of the room by the ventilation system.		
	height of neutral zone		1.41cm
	air flows at window	inflow	0.047 m^3/h (0.044 m^3/s
		outflow	0.030 m^3/h at 0°C)

The airtightness of internal doors is much lower than that of exterior windows or doors. In a flat which was equipped with an extract ventilation system, the pressure difference between the outside and the room with an exhaust register was about 110 Pa when all the windows and doors were closed, but the pressure differences across the partitions were less than 5 Pa.

4.7 Notes on the method and the results

In several cases in the process of iterative calculation, the air flow values began to diverge or to oscillate. But most of them were constrained to reach steady values by changing the initial values.

The results of experiments in the tall blocks suggest that the stack effect has an unexpectedly high influence on the ventilation of the blocks. The flow analysis using a network model of ventilation systems showed that there is a possibility of an unhealthy condition or even a health hazard when unfavourable conditions coincide. Particularly when air leakage at the lower part of a vertical shaft is large and its upper part is relatively airtight, it is very possible that the flats on the upper floors will act in winter as outlets for contaminated air from the flats on the lower floors.

The precision of the field measurements is not yet high enough, and the method needed complicated treatment in several stages of the process. The method of evaporating carbon dioxide gas from dry ice can be used conveniently for field measurements. The distribution of the constantly evaporated gas from one sealed evaporator to every room by means of a set of exchange valves is more profitable than putting a producer in each room. The reasons for this are

a. The obstructive constant evaporation by heat penetration in each room is avoided, and the variation in concentration is easier to discern.

b. The time lag between heat supply and evaporation is eliminated, and the equation of the variation in concentration is simplified.

c. The quantity of dry ice to be kept in each producer at the end of a measurement is reduced.

The method can be made more effective by employing newly developed techniques in the respective stages. There has been a remarkable development in gas analysis instruments in recent years. The continuous or very rapidly intermittent measurement of concen-

5 Conclusions

The air tightness of a window has increased as a result of high precision work and the use of weather strips. The accuracy of an air infiltration equation must be suited to the demand for better control of ventilation design. The behaviour of the exponent in the infiltration equation was examined using various experimental data over a wide range of pressure differences, and the following equation was proposed to express the reciprocal of exponent as a function of the proportionality constant a of a crack and the acting pressure difference ΔP :

$$\beta = 2.0 - e^{-5.0\, a\, \Delta P}$$

The value of the reciprocal of exponent does not exceed 1.2 for air leakage through a crack of large flow resistance, such as weather stripped windows, under pressure differences up to 500 Pa. When the pressure difference is 500 Pa, the air leakage calculated by applying 1.2 to the exponent is 182 % larger than that calculated by conventionally putting the exponent equal to 1.5.

The tracer gas technique of air change measurement was developed for simultaneous measurement of air movements between rooms. Equations were first of all constructed to express the variation in concentration in a room as a result of gas production in every room of a building. The air flows between rooms are calculated from the variations in concentrations which are determined in field measurements by a computer program. The process of computation is supplemented by a statistical method in order to counteract the errors which may arise in field measurement. The air flows can be calculated with errors less than 3 % from the theoretical variations in concentration over a time interval of 720 seconds.

Five pieces of apparatus were built to evaporate a controlled amount of carbon dioxide gas from dry ice. The production of gas is controlled by the supply of electric heat. The five producers were each placed in a room of a flat and gas was produced alternately in the rooms. Measurement of the variations in concentration in the rooms was continued during production.

tration can be accomplished by the instruments. The use of a portable data logging system for the registration of the variations in concentration and other data in a field measurement must result in an admirable advance in the accuracy of the method.

Further, when short interval registration of variations in concentration becomes available, some other mathematical method such as Fourier series must be employed to treat the variations as transient phenomena, instead of successively integrating the effects of variation.

The pattern of gas production occupies a predominant part in the precision of air flow analysis. Even if a highly mathematical treatment and improved instruments are used in the method, the observations made in conjunction with constructing the finite differential equation for the conservation of a tracer gas, Equation 3.4, must not be forgotten when a measurement is carried out.

Air flow measurements were carried out by this method in two 16 storey blocks of flats. The resulting air flows in the bathrooms and kitchens, as given by the method, were compared with the flow rates of the corresponding extract registers. The coefficients of correlation between them were 0.93 for the results of the block with a extract ventilation system, and 0.70 for those of the block with a balanced ventilation system.

Compared with the values obtained during the summer measurement, the total extract volume in the winter on the 15th floor was increased by 35 % and that on the 1st floor was decreased by 6 % in the block with an extract system. In the block with a balanced ventilation system, the changes in winter were +19 % on the 14th floor and -27 % on the 1st floor. The air infiltration from the vertical shafts to the flats on the upper floors seems to affect strongly on the changes. In a tall block, it is very possible that the flats at the upper part will in a cold season become passages for air contaminated in the lower part of the block. To compensate for such a change, the effect of the supply fan must be increased and that of the extract fan decreased in the winter.

Considerable developments have recently taken place in fields such as numerical analysis, control systems, gas analysis and data processing. This measurement method can be made into a more practical device of field work by application of these newly developed techniques.

Symbols

symbol	description of item	unit
A	constant in exponent of air infiltration equation	Pa^{-1}
a	proportionality constant of air infiltration, (§) : see chapter 2	$m^3/m\ s\ Pa^{\frac{1}{\beta}}$ (§)
B	time constant of CO_2 gas producer suffix : number of producer	s^{-1}
C	difference of concentration of tracer gas in a space above that of outdoor air 1st suffix : time 2nd " : room	kg/m^3
C'	absolute concentration of tracer gas suffixs : same to C	kg/m^3
C'_0	absolute concentration of tracer gas in outdoor air	kg/m^3
C_w	wind pressure coefficient	non
d	depth of slit	m
F	total floor area of a flat	m^2
G	production of tracer gas 1st suffix : time 2nd suffix : room	kg/s
G_h	production of CO_2 gas by electric heat	kg/s
G_p	production of CO_2 gas by heat penetration	kg/s
g	acceleration of gravity	m/s^2
H	heat supply for production of CO_2 gas suffix : time	kJ/s
H_e	electric heat supply	kJ/s
H_p	heat penetration through envelope of gas producer	kJ/s
h	hight	m
h_0	reference hight	m
l	length of slit	m
n	air change number of room suffix : room	s^{-1}

P_a	pressure loss for acceleration of air	Pa
P_C	pressure loss at window crack at design ventilation rate	Pa
P_c	pressure in corridor	Pa
P_D	pressure loss between branches of main duct at desing ventilation rate	Pa
P_e	effective pressure of ventilation fan	Pa
P_f	pressure loss by friction	Pa
P_h	pressure at hight h	Pa
P_i	pressure in room suffix : number of room	Pa
$\mathring{P}_i$	assumed pressure in room suffix : same to P_i	Pa
$\ddot{P}_i$	correction term of assumed pressure suffix : same to P_i	Pa
P_o	pressure of outdoor	Pa
P_R	pressure loss at extract register at design ventilation rate	Pa
P_r	pressure at branch point in return duct	Pa
P_S	pressure loss at supply register at design ventilation rate	Pa
P_s	pressure at branch point in supply duct	Pa
P_v	velocity pressure	Pa
P_w	wind pressure	Pa
P_0	pressure at reference hight	Pa
q_a	rate of ventilation per floor area	$m^3/s\ m^2$
q_f	flow rate at ventilation fan	m^3/s
R	resistance of flow suffix c : window crack d : entrance door r : extracc register s : supply register	$s\ Pa^{\frac{1}{\beta}}/m^3$

R_e	Reynold´s number	non
t	temperature	^oC
U	CO_2 gas production by unit heat supply suffix : number of producer	kg/kJ
u	velocity of air	m/s
u_h	wind velocity at hight h	m/s
$\bar{u}$	average velocity of air flow through slit	m/s
u_0	wind velocity at reference hight	m/s
V	volume of room suffix : number of room	m^3
v	volume of air flow 1st suffix : recieving room 2nd 〃 : serving room	m^3/s
v_t	ventilation rate by tracer gas method	m^3/s
v_r	flow rate at extract register	m^3/s
$v_{\bar{0}i}$	total volume of air extracted from room i	m^3/s
$\mathring{v}$	assumed volume of air flow suffixs : same to v	m^3/s
$\ddot{v}$	correction term of assumed volume of air flow suffixs : same to v	m^3/s
w	width of slit	m
w_m	mean face clearance of slit	m

α	power law index of wind velosity distribution by height	non
β	exponet of pressure term in air infiltration equation	non
Δ	difference in conjugated item	-
δ	correction factor of iterative calculation	non

References

Angus,T.C. (1949)
The study of air flow, ventilation and air movement in small rooms as effected by open fireplaces and ventilation ducts
JIHVE Nov. 1949

Armstrong, A.C. (1927)
Some studies of infiltration of air through windows
Transaction No. 779, ASHVE, 1927

Bahnfleth, D.R., Moseley, T.D. & Harris, W.S. (1957)
Measurement of infiltration in two residences
Transactions Nos. 1614 &1615, ASHAE, 1957

Baird, G. (1969)
Air movement control studies at the Hairmyres experimental ward unit, JIHVE, Apr. 1969

Barrett, R.E. & Locklin, D.W. (1968)
Computer analysis of stack effect in high-rise buildings
Transaction No. 2084, ASHRAE, 1968

Baschiere, R.J.,Moy, H.C., Lokmanhekim, M. & Engholm, G. (1965)
Analysis of aboveground fallout shelter ventilation requirements
Transaction No. 1917, ASHRAE, 1965

Birkeland, Ø. & Wigen, R. (1955)
Fugen mellom karm og vegg, forsøk med lufttetthet av dyttefuger of fuger uten dytt, rapport nr. 15, Norges Byggforsknings Institutt, 1955

Bursey, T. & Green, G.H. (1970)
Combined thermal and air leakage performance of double windows
Transaction No. 2157 (RP-73), ASHRAE, 1970

Cammerer, J.S. & Hirschbold, F.X. (1938)
Der Einfluß der Fensterbauart auf den Luftdurchgang
Gesundheits-Ingenieur, Jul. 1938

Carlsson, B. & Sundell, J. (1969)
Bostadsventilation - En orienterande undersökning
Examensarbete, Institutionen för Uppvärmnings- och Ventilationsteknik, KTH, 1969

Carman (1956)
Flow of gases through porous media
Butterworths Scientific Publications, 1956

Coblentz, C.W. & Achenbach, P.R. (1957)
Design and performance of a portable infiltration meter
Transaction No. 1616, ASHAE, 1957

Coblentz, C.W. & Achenbach, P.R. (1963)
Field measurements of air infiltration in ten electrically-heated houses, Transaction No. 1845, ASHRAE, 1963

Coleman, E.F. & Heald, R.H. (1940)
Building materials and structures report BMS45,
Air infiltration through windows
National Bureau of Standards, USA, 1940

Collins, B.G. & Smith, D.B. (1955)
The measurement of ventilation rates using a radioactive tracer
JIHVE Oct. 1955

η_1	variation in ventilation rate of a room by opening own window	non
η_2	variation in ventilation rate of a room by opening window of next floor	non
κ_1	constant term of pressure caused by ventilation fan	Pa
κ_2	proportional term of pressure to flow rate of ventilation fan	$Pa/(m^3/s)$
ν	kinematic viscosity	m^2/s
ξ	part of laminar flow in pressure loss	non
ρ	density	kg/m^3
τ	time	s

Davenport, A.G. (1963)
Wind effects on buildings and structures
Paper 1: The relationship of wind structure to wind loading
National Physical Laboratory, London, 1965

Dick, J.B. (1949)
Experimental studies in natural ventilation of houses
JIHVE Dec. 1949

Dick, J.B. (1950)
The fundamentals of natural ventilation of houses
JIHVE Jun. 1950

Dick, J.B. & Thomas, D.A. (1951)
Ventilation research in occupied houses
JIHVE Oct. 1951

Eriksson, B.E. (1962)
Bostadsventilation, Statens institut för byggnadsforskning
rapport 77, Stockholm, 1962

Esdorn, H. (1972)
Neubearbeitung der DIN 4701 „Regeln für die Berechnung des Wärmebedarfs von Gebäuden", HLH Dez. 1972

Foord, N. & Lidwell, O.M. (1973)
A method for studying air movement in complex occupied buildings such as hospitals: halocarbons as gas tracers using gas chromatography, BSE Jul. 1973

Gabrielsson, J. & Porra, P. (1968)
Calculation of infiltration and transmission heat loss in residential buildings by digital computer, JIHVE Mar. 1968

van der Held, E.F.M. (1953)
Der Einfluß der Turbulenz auf die Lüftung
Gesundheits-Ingenieur Heft 23/24 1953

Honma, H. & Iyoda, I. (1970)
An air change measuring method by continuous production of carbon dioxide gas, Report for annual meeting, Architectural Institute of Japan, 1970

Honma, H. (1972)
Metod för bestämning av luftströmning inom byggnader
VVS, June 1972

Honma, H.
Air flow in dwellings, part 1 Method of measurement (1972)
" part 2 Improvement of the method (1972)
" part 3 The ventilation measurement in a tall block of flats (1974)
Inst. för Uppv. & Ventil., KTH (unpublished)

Honma, H. (1973)
On infiltration of Swedish modern wood sach windows
Inst. för Uppv. & Ventil., KTH (unpublished)

Honma, H. (1974)
Relation between variables in the equation of air infiltration through window cracks, Inst. för Uppv. & Ventil., KTH (unpublished)

Hopkins, L.P. & Hansfor, B. (1974)
Air flow through cracks
BSE Sept. 1974

Horie, G. 1969)
Group heating systems in housing areas
Transaction of the Society of Heating,Air-Conditioning and Sanitary Engineers of Japan, 1969

Houghten, F.C. & Schrader, C.C. (1924)
Air leakage through the openings in buildings
Transaction No. 686, ASHVE, 1924

Houghten, F.C. & Gutberlet, C. (1937)
Heating requirements of an office building as influenced by the stack effect, Transaction No. 1069, ASHVE 1937

Howland, A.H., Kimber, D.E. & Littlejohn, R.F. (1960)
Measurements of air movements in a house using a radioactive tracer gas, JIHVE May 1960

Iyoda, A. (1971)
Tracer gas method for ventilation measurement of dwellings
Thesis for master degree, Hokkaido Univ., Japan, 1971

Jackman, P.J. (1974)
Heat loss in buildings as a result of infiltration
BSE Apr. 1974

Jennings, B.H.,& Armstrong, J.A. (1971)
Ventilation theory and practice
Transaction No. 2170 (RP-17), ASHRAE, 1971

Jordan, R.C., Erickson, G.A. & Leonard, R.R. (1963)
Infiltration measurements in two research houses
Transaction No. 1843, ASHRAE, 1963

Kaufmann, W. (1963)
Fluid mechanics
McGraw Hill, New York, .963

Kollmar, A. & Liese, W. (1949)
Die Abhängigkeit des Zuschlagsverfahrens in der Wärmeverlustberechnung nach DIN 4701 vom Luftwechsel im Raum
Gesundheits-Ingenieur, Heft 17/18, 1949

Kreith, F. & Eisenstadt, R. (1956)
Pressure drop and flow characteristics of short capillary tubes at low Reynolds numbers, Transaction of ASME, Jul. 1957

Krischer, O. (1951)
Die Druchverhältnisse in Häusern unter dem Einfluß des Winds und die Lüftungsempfindlichkeit vonRäumen, HLH März 1951

Krischer, O. (1959)
Neuerungen bei der Wärmebedarfsberechnung DIN 4701
HLH März 1959

Laschber, R.R. & Healy, J.H. (1964)
Statistical analyses of air leakage in split-level residences
Transaction No. 1900, ASHRAE, 1964

Lehmberg, W.H., Brandt, A.D. & Morse, K. (1935)
A laboratory study of minimum ventilation requirements :
Ventilation box experiments, H.P.& A.C. Jan. 1935

Lund, C.E. & Peterson, W.T. (1952)
Air infiltration through weatherstripped and non-weatherstripped windows, Bulletin No. 35, University of Minnesota, 1952

Müller, K.G. (1961)
Bestimmung der erforderlichen Zuluftmenge bei lufttechnischen Anlagen, HLH, Jul. 1961, Aug. 1961 & Sept. 1961

Müller, K.G. (1964)
Notwendige Lufterneuerung bei intermittierender Verunreinigung HLH Jul. 1964

Newberry, C.W.,,Eaton, K.J. & Mayne, J.R. (1968)
Wind loading of a tall building in an urban environment
Current paper 59/68, Building Research Station, Watford, 1968

den Ouden, H. (1967)
Prediction of the anticipated air volume passing through buildings by means of the air current analogue, Publication No. 272, Research Institute for Public Health Engineering TNO, Delft, Holland, 1967

Maeda, T. (1961)
Calculation method of ventilation rate for multi-room system
Report for annual meeting, Architectural Institute of Japan, 1961

Raisch, E. (1922)
Die Wärme- und Luftdurchlässigkeit von Fenstern verschiedener Konstruktion, Gesundheits-Ingenieur, 9. Heft, 1922

Raiss, W. (1966)
Research on heating installations in tall blocks
JIHVE Jun. 1966

Roedler, F. & Schlüter, G. (1968)
Untersuchungen an einer Sammelschachtlüftung ohne Motorkraft (DIN 18017) in einem Hochhaus, Gesundheits-Ingenieur Heft 1,1968

Rydberg, J. (1959)
Disturbances in ventilation systems
Heating and Ventilation Engineers, May 1960

Rydberg, J.
Om ventilationssystems funktion
VVS Feb. 1968

Sasaki, J.R. & Wilson, A.G. (1965)
Air leakage values for residential windows
Transaction No. 1950, ASHRAE, 1965

Schrader, C.C. (1924)
Air leakage around window openings
Transaction No. 704, ASHVE, 1924

Schüle, W. (1961)
Heizwärmeverbrauch bei Hochhäusern
Gesundheits-Ingenieur, Heft 9, 1961

Schutrum, L.F., Ozisik, N.,Baker, J.T. & Humphreys, C.M. (1961)
Air infiltration through revolving doors
Transaction No. 1760, ASHRAE, 1961

Shaw, C.Y., Sander, D.M. & Tamura, G.T. (1973)
Air leakage measurements of the exterior walls of tall buildings
Transaction No. 2280, ASHRAE, 1973

Shellard, H.C. (1963)
Wind effects on buildings and structures
Paper 2: The estimation of design wind speeds
National Physical Laboratory, London, 1965

Shoda, T., Terasawa, T. & Katayama, T. (1970)
Experimental study on air and water tightness of metal window sa sashes, report for International Congress for Heating, ventilating and Air-conditioning, Kopenhagen, 1971

Sigwart, K. (1932)
Luftdurchlässigkeit von Holz- und Stahlfenstern
Gesundheits-Ingenieur Nr. 43, 1932

Tamura, G.T. & Wilson, A.G. (1964)
Air leakage and pressure measurements on two occupied houses
Transaction No. 1869, ASHRAE, 1964

Tamura, G.T. & Wilson, A.G. (1966)
Pressure differences for a nine-story building as a result of chimney effect and ventilation system operation
Transaction No. 1973, ASHRAE, 1966

Tamura, G.T. & Wilson, A.G. (1967)
Pressure differences caused by chimney effect in three high buildings, Transaction No. 2046, ASHRAE, 1967

Tamura, G.T. & Wilson, A.G. (1967)
Building pressures caused by chimney action and mechanical ventilation, Transaction No. 2047, ASHRAE, 1967

Tamura, G.T. & Wilson, A.G. (1968)
Pressure differences caused by wind on two tall buildings
Transaction No. 2085, ASHRAE, 1968

Thomas, D.A. & Dick, J.B. (1953)
Air infiltration through gaps around windows
JIHVE, June 1953

Wigen, R. (1958)
Vinduer av tre, undersøkelser av tetthet mot vind og regn
rapport nr. 28, Norges Byggforsknings Institutt, 1958

Wilson, A.G. & Nowak, E.S. (1959)
Condensation between the panes of double windows
Transaction No. 1685, ASHRAE, 1959

Wise, A.F.E. & Curtis, M. (1968)
Development of ventilation systems for multi-storey housing
JIHVE, May 1968

Yaglou, C.P. Riley, E.C. & Coggins, D.I. (1936)
Ventilation requirements
H.P.&A.C. Jan. 1936

Yaglou, C.P. & Witheridge, W.N. (1937)
Ventilation requirements, part 2
Transaction No. 1068, AHVE, 1937

Yaglou, C.P. (1955)
Ventilation requirements for cigarette smoke
Transaction No. 1524, ASHAE, 1955

van Zuilen, D. & van Gunst, E.
Wärmebedarf unter besonderer Berücksichtigung der Luftdurchlässigkeit von Fenstern
Gesundheits-Ingenieur, Heft 12, 1958

Rättelser till TM nr 57-62

Nr	sid.	rad	står	skall stå
57	15	bild 8	d_h	d_e
"	"	bild 9	d_h	d_e
58	20	7 nedifrån	mestadels	huvudsakligen
"	21	överst	-	tabell för molnig himmel
"	29	bild 3	- I bilden fattas beteckningarna A-D i vänstra kolumnen och E-H i högra	
"	30	18 nedifrån	15 mm	1,5 mm
"	40	13 o. 14	(kg/dm^2)	(kg/dm^3)
"	"	4 nedifrån	tryckökning	tryckminskning
"	"	sista	strypflänsens tryck-ökning	strypflänsens tryck-förlust
"	43	6 nedifrån	väg	vägg
"	"	5 nedifrån	$\frac{d_y}{d_i} \cdot \frac{1}{\alpha_y A_y}$	$\frac{d_y}{d_i} \div \frac{1}{\alpha_y A_y}$
"	"	längst ned, tillägg:		l = cylinderns längd
"	44	4 nedifrån	värmeöverförings-	värmegenomgångs-
"	45	4 o. 11	värmeöverförings-	värmegenomgångs-
"	"	bild 13	I bilden skall siffrorna 3 och 20 cm byta plats	
59	52	5	ammoniak	ammoniakavgivning
"	63	9 nedifrån	ämnet har högt	ämnet skall ha högt
61	72	10	transistenta	transienta
"	74	sista	$\frac{0{,}023\ \nu}{d_h}$	$\frac{0{,}023\ \lambda}{d_h}$