VENTILATION OF DOMESTIC BUILDINGS

No. 6 in the series
"Studies in Energy Efficiency in Buildings"

BRITISH GAS

INTRODUCTION

The series of technical publications 'Energy Efficiency in Buildings' was introduced by British Gas during the International Energy Conservation Month, October 1979, as part of the industry's support of Government energy initiatives. The publications consist of a compilation of technical studies mainly prepared by the staff of British Gas Watson House Research Station. The aim of 'Energy Efficiency in Buildings' is to contribute to the training and development of those concerned with building design and the efficient use of fuel. It is also aimed at assisting the Royal Institute of British Architects in its mid-career training programme for architects.

VENTILATION OF DOMESTIC BUILDINGS

The proper level and control of ventilation in buildings are important aspects of energy efficient design. This sixth publication in the series 'Studies in Energy Efficiency in Buildings' focuses on the ventilation requirement of domestic buildings. The effects of different methods of construction and methods of measuring ventilation rates are discussed.

Other titles in this series are:

No. 1 "Studies in Energy Efficiency in Buildings"
No. 2 "Energy Demand and System Sizing"
No. 3 "Selection of Boiler Plant and Overall System Efficiency"
No. 4 "People and Buildings"
No. 5 "Insulation, Heat Services and Housing Design"

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CONTENTS

Ventilation: Design Considerations

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Building Services Engineer, Vol. 42, 132-141 (1974)

1. BACKGROUND:

The provision of openings to allow fresh air into a room of a house is required for health and comfort reasons, and where an open flued appliance is installed, air for combustion and flue dilution is required. The former requirements are based on the following needs:-

i) To satisfy the respiratory needs of the occupants
ii) To remove contaminated air (body odours, tobacco smoke, cooking smells, etc)
iii) To maintain a thermally comfortable environment unless artificial cooling is provided).

This paper indicates the level of ventilation that is required for all these needs in an average living room and discusses the practicability of providing the required fresh air supply by either a natural or mechanical system. As the presence of an appliance can have a significant effect on the ventilation conditions in a room the interaction between appliance performance and ventilation conditions is reviewed.

2. VENTILATION REQUIREMENTS

2.1 Basic Needs

2.1.1 Respiratory Needs of the Occupants:

As little as 0.5 1/s (1 cfm) of fresh air will meet the oxygen requirements of one person and between 1.0 and 1.6 1/s per person (i.e. 2.1 to 3.5 cfm) is required depending on the activity of the person involved to meet a desired limit value for carbon dioxide of 0.5% which is satisfactory for long term exposure. As these values are lower than those needed to remove body odours, a ventilation rate that is sufficient to remove body odours satisfies the respiratory requirements.

2.1.2 Removal of Contaminated Air

The air in a room can be contaminated either by body odours or what may be termed "artificial" contaminants such as tobacco smoke and cooking smells.

The amount of fresh air needed to remove body odours depends on a number of factors, the main being available air space per person, personal hygiene and type of activity. Many standards have been suggested and the proposed values vary significantly. Typical values accepted in this country of minimum and recommended minimum ventilation rates are quoted in the 1970 IHVE Guide. These values are represented in Table 1.

TABLE 1

MINIMUM VENTILATION RATES

Air Space per Person (m^2)	Fresh Air Supply Per Person (litre/s)		
	Minimum	Recommended Minima	
		Smoking not permitted	Smoking permitted
3	11.3	17.0	22.6
6	7.1	10.7	14.2
9	5.2	7.8	10.4
12	4.0	6.0	8.0

If the recommended minima is applied to a typical living room and the ventilation requirement and air change rate are plotted against the number of occupants a graph as shown in Figure 1 is produced. The ventilation rate is given as a band where the upper limit represents a situation where all the occupants are smokers and the lower limit non-smokers. The dashed line in the centre of the band represents a mixed population of 50% smokers and 50% non-smokers.

2.1.3 Heat Balance

The human body produces heat, the amount varying mainly with activity but also with height, weight, age and sex and this heat must be dissipated to the surrounding environment. The sensible part of the heat emission varies from 100 W for a person at rest to 220 W for a person working hard. Internal heat gains are welcome in winter but can cause an excessive rise of the internal temperature in the summer. If the rise of air temperature due to occupancy and for that matter from any other miscellaneous gains is to be controlled, then a supply of fresh air is required.

Not all sensible heat emission is transformed into a convective load. A part is stored in the structure and reappears later, but a substantial part constitutes a direct convective load. The diagram in Figure 2 shows the variation of ventilation rates with the number of occupants for a range of temperature differentials superimposed on the ventilation requirements as plotted in Figure 1. A mean value of 100 W of direct convective load per person has been used in the diagram. The value included proportionately all other heat gains. A surprisingly high volume of air is needed to remove a 100 W load, e.g. for a temperature differential of $\Delta t = 1^\circ C$ between the room and external atmosphere a flow of 83 1/s, representing 7 air changes per hour in a typical living room, is required.

2.1.4 Combustion Air

The quantity of fresh air required to maintain the satisfactory and safe operation of an appliance depends on appliance type and its heat input. When the appliance is fuelled with any fossil fuel and has an open flue air for combustion and flue dilution, if a draught diverter is fitted, has to be supplied directly from the associated room. An appliance that is often installed in an existing hearth of a living room is a combined gas fire/central heating boiler. Typically these appliances have a total maximum heat input of 20 kW (68,000 Btu/h) with an associated air requirement of 15.0 1/s assuming 4% CO_2 in the secondary flue.

2.2 Combined Requirements

A comparison of air requirements to meet individual needs in a living room is given in Table 2, which illustrates the large variation in fresh air supply that will be necessary to satisfy these needs.

The air supply can be provided either by natural ventilation or mechanical means and the conditions that such a ventilation system has to meet are:

i) Room occupied
ii) Room unoccupied
iii) Summer ventilation

Reference to Table 2 suggests that a range of air flows between 20 1/s and 80 1/s (1.8 – 7.2 air changes per hour) is required in a living room having a volume of $40m^3$, and if these rates of fresh air supply are achieved then there will be more than adequate air for a fuel burning appliance. Although, in practice, natural ventilation will be both uncontrolled and variable, and economic consideration will limit the range of air flows achieved by a mechanical system, a compromise solution has to be reached. The following sections of the paper consider in more detail the effect of building construction and mechanical ventilation if fitted, on ventilation rates and the performance of conventionally flued appliances.

TABLE 2

INDIVIDUAL AIR REQUIREMENTS (LIVING ROOM)

Requirement		Air Supply litres/second	Air changes/hour*
Open Flued Appliance (heat input 20 kW)		15	1.3
Respiration	3 people	6	0.55
	6 people	12	1.1
Contamination	3 people	25	2.2
(odours, tobacco smoke)	6 people	80	7.2
Heat Balance	3 people	60	5.4
($\Delta t = 4^{\circ}C$ in the summer)	6 people	120	10.8

*Based on a room volume of 40m^3

3. NATURAL VENTILATION

3.1 Appliance not Installed

Natural ventilation is the air flow through openable windows, ducts, etc., and can be controlled to some extent by the occupant. Infiltration, often termed adventitious ventilation, is the uncontrolled leakage of air through a building. Leakage occurs mainly through cracks around windows, doors and even through the structure itself.

As natural ventilation and infiltration are dependent on natural forces the rate of air flow through a building is governed by the pressure differential across the building and the resistance offered by apertures and cracks. The pressure differential may be caused either by wind or by the "stack effect" i.e. the pressure differential caused by differences in density of air due to the indoor-outdoor temperature difference. The "stack" effect is generally small in relation to wind pressures, except in the case of tall buildings with vertical shafts, or where an open flue is installed. Typical ventilation rates measured in a room having a suspended floor and containing a single window and door are presented in Table 3.

The above results indicated the large variation in fresh air supply that can be expected with changing wind speed and degree of air "tightness" associated with the structure of a building. Only one measurement of ventilation rate, 17 1/s, when the room was in its original condition approached the minimum rate of 20 1/s indicated in the previous section. To achieve this value and higher rates of fresh air supply it will normally be necessary to open a door or window.

TABLE 3

EFFECT OF PROGRESSIVE SEALING ON VENTILATION RATE: NO FLUE

(Bedroom, Property No. 5: Paper 2)

Degree of Sealing	Open Area m^2	Ventilation Rate 1/s	Wind Speed (Variable direction) m/s
Room in original condition	0.056	6	0.7
		17	5.0
Door and window sealed Floor unsealed	0.030	3	2.7
Door and window unsealed Floor sealed	0.028	3	0.5
		7	2.6
Door and window (weatherstripped) Floor sealed	0.003	1	3.3

1 in^2 = 0.0064 m^2 — Sealing was undertaken with adhesive tape

1 m.p.h. = 0.45 m/s — Weatherstripping was undertaken with plastic foam strip

1 ft^3/h = 0.0078 1/s — Room volume 36 m^3

3.2 Appliance Installed

The installation of an open flued appliance, taking air from a room, provides a ventilation opening that promotes increased ventilation by both the "stack effect" and the wind. This is illustrated in Table 4 where it can be seen that ventilation rates can be more than doubled by the presence of an open flue.

TABLE 4

EFFECT OF PROGRESSIVE SEALING ON VENTILATION: APPLIANCE INSTALLED

Living Room (40m^3), Property 2: Paper 2
(one window, two doors, solid floor)

Degree of Sealing	Open Area m^2	Appliance Condition	Ventilation Rate 1/s	Wind Speed m/s
Room in original condition	0.025	OFF	11	4.3
		ON	25	4.8
Completely weather-stripped with double glazing	0.0065	OFF	5*	4.2
		ON	17	2.6

*A ventilation rate of 2 1/s was measured for this degree of sealing when the flue was also sealed

Living Room (33m3), Property 5: Paper 2
(one window, two doors, suspended floor)

Degree of Sealing	Open Area m^2	Appliance Condition	Ventilation Rate 1/s	Wind Speed m/s
Room in original condition	0.079	OFF	14	3.2
		ON	21	3.4
Completely weather-stripped with double glazing and floor	0.008	OFF	4	2.6
		ON	16	2.7

The results in Table 4 and other measurements presented in Figure 3 are representative of the ventilation rates measured in the living rooms of the properties, that contained a flued gas appliance. The resultant values for fresh air entry into the room when the appliance is operating are much closer to the minimum ventilation rate indicated in section 2 although they are still subject to large variations caused by changes in wind speed and direction. Having quantified the effect of an open flue on the ventilation rate in a room it is necessary to determine how the open area in a room influences appliance and flue performance.

Combined fire/central heating boiler units were installed, Figure 4, in the living rooms of properties 2, 3 and 5 listed in Appendix 1 Paper 2 and the minimum open area necessary with in the room to ensure complete clearance of combustion products up the flue established. When these rooms were in their original condition no spillage of combustion products could be detected for the operation of any combination of the appliances. Smoke tube tests indicated that there was an in-flow of air from the cracks around the doors and windows and out-flow of diluted combustion products through the flue. During these tests no air was detected leaving the room except through the flue hence Figure 3 also gives the variation of flow rate in the flue with increased sealing of a room. The open area at which spillage of combustion produces was visually detected is indicated in Figure 3, and is one of the values included in Table 5 below.

TABLE 5

MINIMUM OPEN AREA TO ENSURE CLEARANCE OF COMBUSTION PRODUCTS

Appliance A Fire, input 4.8kW (16,500 Btu/h) Boiler, input 15.5kW (53,000 Btu/h)	Minimum area to avoid continuous spillage m^2	
	Property 2	Property 5
Boiler	0.005	
Combined Unit	0.005	0.006
Appliance B Fire, input 5.3 kW (18,000 Btu/h) Boiler, input 14.6 kW (50,000 Btu/h)*	Property 3	
Boiler	0.002	
Combined Unit	0.002	

Comparison of the open areas given in Table 5 with values obtained for the rooms before they were weatherstripped indicates that in normal circumstances there will be more than adequate provision of open area for satisfactory performance of a gas appliance without ventilators being installed. However, in the unlikely event that internal doors are sealed as well as the windows being weatherstripped, then the addition of the residual adventitious open area, see Table 4, to that specified in the Building Regulations will ensure that spillage of combustion produces does not occur.

4. MECHANICAL VENTILATION

The previous section of the paper has illustrated the difficulty associated with achieving reasonable design levels of reliable ventilation by natural means. This suggests that a mechanical system is worth consideration as it would offer the advantage of maintaining a controlled supply of fresh air. At the present time many commercial buildings in the United Kingdom are provided with comprehensive mechanical ventilation systems but in domestic premises controlled ventilation is normally restricted to a simple extract system. However, there are trends in both building construction and public attitudes in the country that suggest there culd be an increasing use of mechanical systems in the future, particularly as the concept is being adopted elsewhere in Europe.

Building standards in the future will be influenced both by higher comfort standards and the need to conserve energy which will result in higher thermal resistance of the outside walls. A consequence of this trend will be to increase the proportion of heat lost by natural ventilation and infiltration and unless action is taken to control the level of fresh air supply there will be a steadily diminishing return in conservation of heat as the thermal resistance of the structure is increased. In certain locations, close to sources of high noise output such as airports and motorways, there are already grant schemes for enabling householders to sound insulate rooms in their houses and Paper 1 describes a natural draught ventilator to be used in these circumstances. However, in a room that has been substantially sealed, it has already been shown that natural ventilation is not adequate for supplying fresh air requirements, especially for eliminating odours. This is recognised in the sound insulation schemes (1) that specify the installation of mechanical ventilators that are acoustically treated to prevent ingress of noise. Living habits can also affect the mode of ventilation. An increase in the number of working housewives, which can lead to unoccupied and therefore unheated homes, is one of the contributory factors in the increase of problems caused by condensation. One effective method of overcoming this problem is to control ventilation.

4.1 Effect of Mechanical Extract on an open-flued gas appliance

Included in the study of gas appliance and flue performance while rooms were progressively sealed, discussed in section 3.2, were tests designed to provide information on the interaction between an extract fan and flue performance for use in conjunction with a mathematical procedure for predicting the onset of spillage of combustion products. The operation of an extract fan will obviously increase the maximum area required in the room to prevent spillage, as illustrated in Table 6, but the extra open area required to balance the action of the fan above the natural ventilation requirement will depend on the operating characteristics of the fan. This situation can most satisfactorily be assessed using a theoretical model of the ventilating system.

TABLE 6

EFFECT OF AN EXTRACT FAN ON THE MINIMUM AREA TO PREVENT SPILLAGE OF COMBUSTION PRODUCTS

Appliance 'B' Fire: heat input 5.3 kW (18,000 Btu/h)
Boiler: heat input 14.6 kW (50,000 Btu/h)

Appliance	Minimum Area m^2*	
	Natural Ventilation	6 in diameter axial fan
Boiler	0.0018	0.0053*
Combined Unit	0.0030	0.0065

*Area specified in Building Regulations = 0.007 m^2

The theoretical study of this type of system has received a small amount of attention most of which has been based upon the comparison of the fluid system with an electrical analogue. This approach was first suggested for ventilation work by Scott (2) who produced a working model, using electric lamps, for a single room. More recently, Billington (3) has used a similar technique to consider the gas appliance. In his work he has considered the problem as two sections, namely:-

a) The quantity of fresh air required for combustion and the removal of combustion products.
b) The calculation of ventilation under a variety of operating conditions (i.e. the size and disposition of fresh air openings, the action of the wind and the effect of an extract fan).

Billington concludes that his method shows that a fresh air vent of an appropriate size as recommended by the Gas Industry provides an adequate supply of air to ensure clearance, through the flue, of the combustion produces of a 12 kW or 24 kW boiler installed in an otherwise sealed room. The adventitious cracks which exist around the windows and doors of a typical room provide an additional inlet area and, with this crackage, the recommended open areas are sufficient even when a typical extract fan is used. In general the work carried out by British Gas on theoretical models incorporating gas appliances would verify these conclusions. However, attempted correlation with results obtained in the field has so far not been satisfactory and it is thought that this can be attributed to the difficulty of defining a criterion for spillage of combustion products.

In Billington's model there is no provision for including effects of spillage but it must be clearly defined if flue performance is to be correctly modelled and it is in this area that future work on this topic is to be concentrated. It is not difficult to define the situation where 100% spillage occurs but the mechanism operating before this condition is reached are not easily specified. However, the following approach is to be tried as it offers the possibility of defining a theoretical criterion for spillage that is of more practical use than complete flue reversal.

When spillage of the combustion products commences the flow through the secondary flue can be considered as undiluted and equal to the volume flow rate of the combustion products from the appliance. The flue draught prior to this condition has been a function of the volume flow rate through the flue, but with the onset of spillage, this draught may be taken as a constant dependent only on the inlet temperature and the efficiency of the flue. Thus an expression for the volume flow rate of the combustion products may be written in terms of the pressure differences across the flue for the onset of spillage. It is hoped that this expression combined with the other flow equations in the room should give a solution. However, the results obtained from such an approach would have to be substantiated with controlled laboratory experiments and further field measurements.

4.2 Controlled Ventilation Systems

To obtain guidance on the general application of controlled (mechanical) ventilation to dwellings, it is necessary to consider how such systems have been utilised in many Continental countries (4, 5, 6) as the number of such installations in the United Kingdom is small (7, 8). However, tests on a combined heater/ventilation unit installed in a maisonette, Figure 5, has been shown to be effective in the control of condensation. A natural extension of the simple extract system for one room described in section 4.1 is to extract a predetermined volume of air from a dwelling via grilles situated in the kitchen, bathroom and lavatories. This reduces the air pressure in the premises and air is drawn in through cracks and special intakes. The air passes through the dwelling in a manner that prevents local generation of odour spreading to other rooms. Where an appliance is installed it is possible to exhaust the combustion produces through the extract system and such a combined system has been developed by Gaz de France (9). The gas appliances, see Figure 6, are connected to the extract system by means of a thermostatic exhaust register that ensures a constant mass flow.

The extract only ventilation system, because of the methods of fresh air supply, has certain disadvantages. In winter the incoming cold air may cause draughts in the proximity of the windows and may be a source of dirt from the outside. Added to this there is no complete control of the air flow through each dwelling. When exposed to strong winds cross-draughts may affect the comfort of the occupants.

To achieve complete control of the air flow pattern each occupied room should be supplied with a predetermined volume of air. This air should be filtered, heated and admitted to the rooms in such a way that the required standards of comfort are attained and because similar quantities of air can be admitted and removed under full control the ventilation of each dwelling is balanced. The importance of being able to fully control the air flow in a dwelling has led to a substantial rise in the number of houses equipped with both an extract and supply system, especially in countries with more extreme weather conditions than those experienced in the United Kingdom. The trend for Sweden is shown in Figure 7 and the latest statistics show that a full 25% of new houses being built have a balanced ventilation system. A more detailed picture of such a system is shown in Figure 8. The building has a conventional exhaust system, including a smoke ventilator in the staircase and a supply system where the fresh air is preheated to about 15°C and filtered in a make-up unit and ducted to supply nozzles situated behind radiators. Individual room temperatures can be adjusted by varying the output of the radiator, e.g. by means of a thermostatic valve.

The actual volume of air that is to be extracted will inevitably be a compromise between the ideal requirements suggested in section 2 and such factors as capital cost, noise levels and ventilation heat-loss. Although this situation requires further study some indication of what is practicable can be obtained from the Swedish Standard SBN67 where the ventilation rate for the whole dwelling is given by the following equation:-

$$q = 2.2 - 0.004\ G$$

where q = ventilation rate m^3/h per m^3 of floor area
G = total floor area in m^2

The equation is presented graphically in Figure 9. A typical dwelling having a floor area of 90 m^2 would require 45 1/s which would be sufficient for four people, under normal circumstances.

5. CONCLUSIONS

5.1 In a living room a typical range for a demand of fresh air is 20 1/s – 80 1/s and this is more than adequate for safe operation of open-flued gas appliances having a heat input of up to 20 kW (68,000 Btu/h).

5.2 The fresh air supply required in a dwelling can be more effectively controlled by a mechanical ventilation than by natural means, but in the latter case it is more adequate if there is an open flue.

5.3 The open areas available for ventilation where an open-flued central heating appliance is installed, as specified in the Building Regulations, will be adequate in a number of situations where an extract fan is fitted in the same room. However, further work is required before a complete assessment can be made of the interaction between fans and flues.

5.4 The experience gained in Continental countries on mechanical ventilation of domestic premises will provide a sound basis for application of such systems in the United Kingdom.

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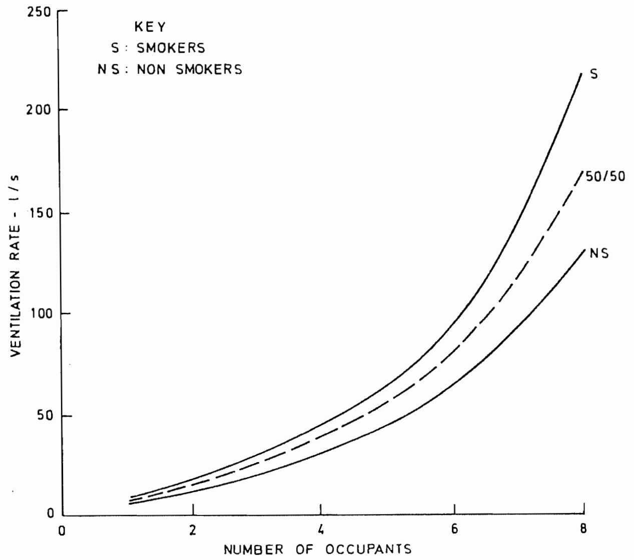


Figure 1

Ventilation requirements in a typical living room containing smokers and non-smokers

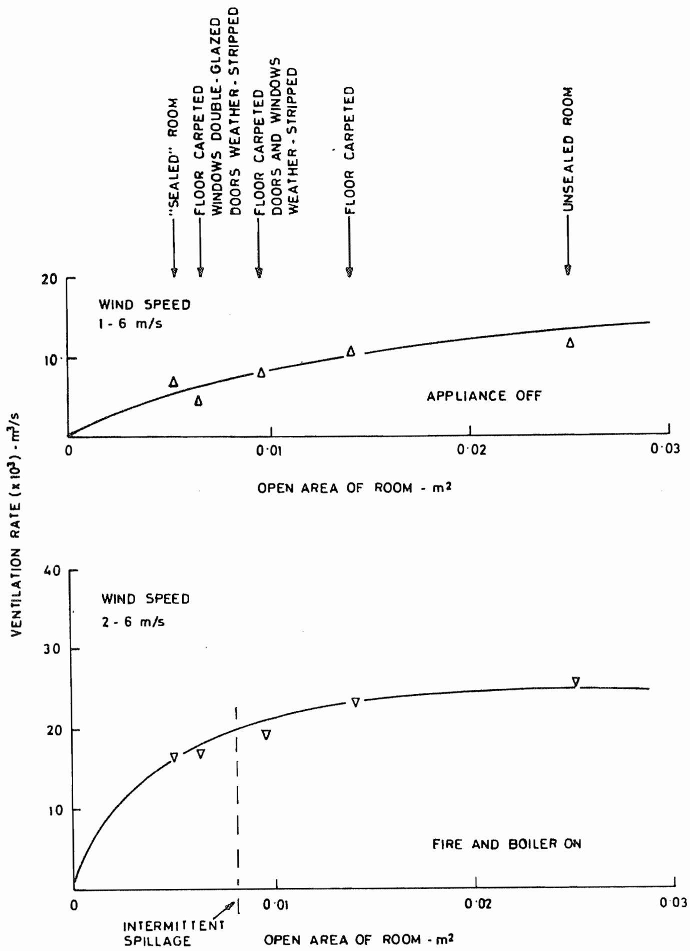


Figure 3

Effect of progressive sealing of room on ventilation rate

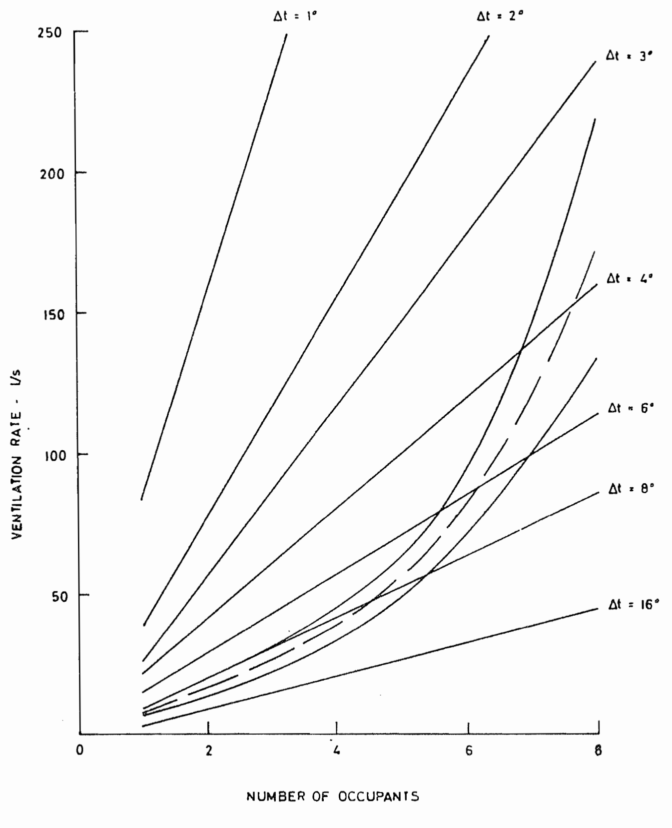


Figure 2

Ventilation requirements in a typical living room to maintain a temperature differential Δt between indoor and outdoor temperature

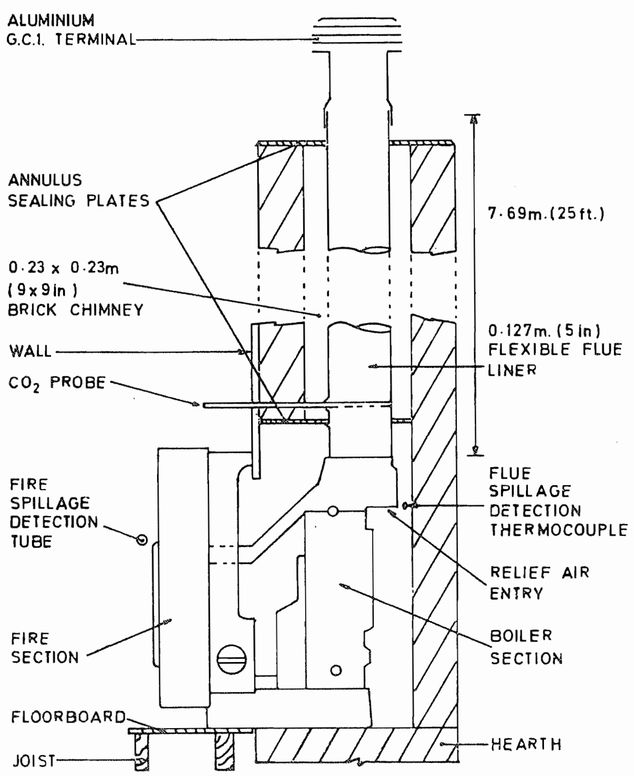
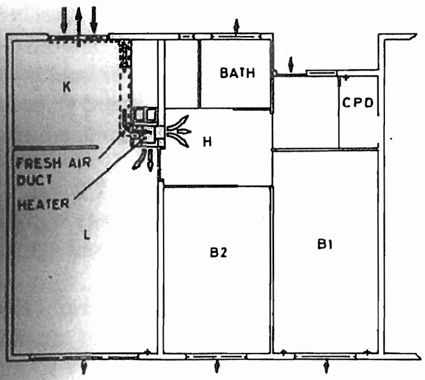


Figure 4

Typical installation of back-boiler during ventilation tests



Key

- HEATER / VENTILATOR INTAKE
- VENTILATOR
- WARM AIR 30% FRESH AT HEATER
- KITCHEN EXTRACT FAN

Figure 5

Ventilation of maisonette using adapted warm-air heating system (Reference 7)

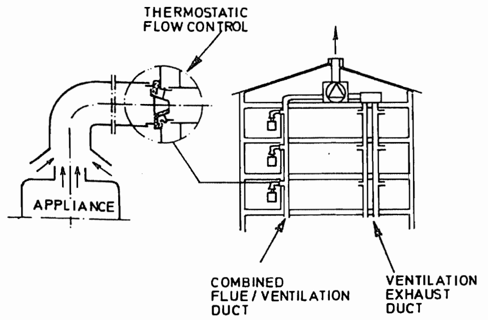


Figure 6

Thermostatic exhaust ventilator (Gaz de France)

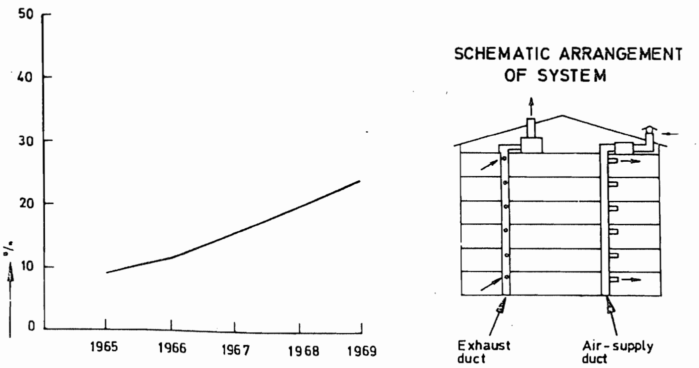
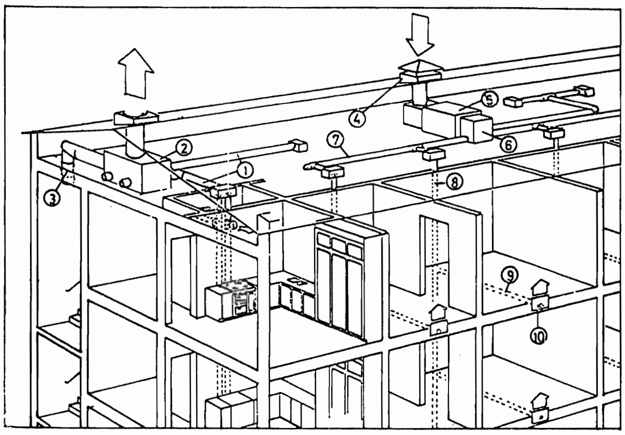


Figure 7

Percentages of new housing units equipped with mechanical inlet and exhaust ventilation in Sweden



Key

1.	SPIRAL DUCTING
2.	PREFABRICATED FAN CHAMBER
3.	SMOKE VENTILATOR
4.	AIR INTAKE
5.	MAKE-UP AIR UNIT
6.	PLEMIUM CHAMBER
7.	INSULATED SPIRAL-TYPE DUCTWORK
8.	VERTICAL DISTRIBUTION RISERS
9.	HORIZONTAL BRANCH DUCTS
10.	SUPPLY NOZZLE

Figure 8

Block of flats with mechanical supply and exhaust systems

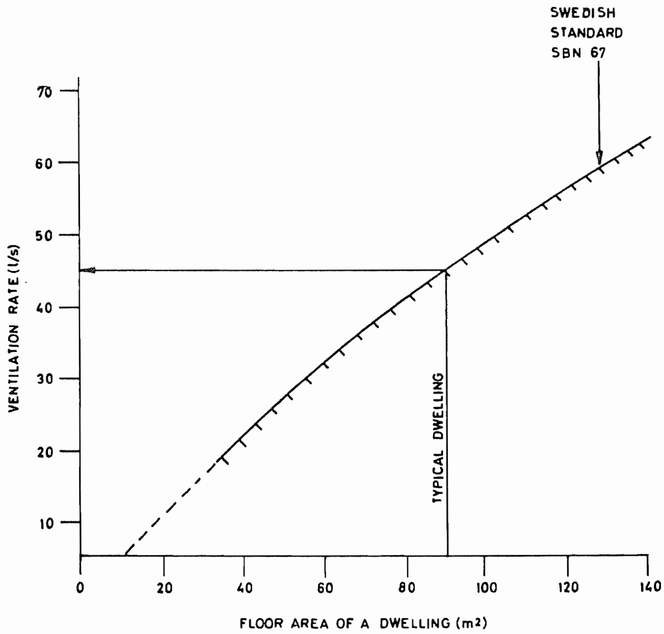


Figure 9

Ventilation rates of dwellings depending upon floor area according to Swedish Standard SBN67

#21

Crack flow equations and scale effect

by D. W. Etheridge

Building and Environment, Vol. 12, pp 181-189, 1977

1. INTRODUCTION

A major source of ventilation in many dwellings is that arising from air flow through cracks. To date, much use has been made (e.g. see references [1, 2]) of equations of the form

$$\Delta p = \text{Constant} \,.\, V^n \tag{1}$$

for describing crack flows. In this equation Δp is the pressure drop across the crack, V is the volume flow rate through the crack and the exponent has a value of about 1.6. Equations of this type lack generality because they are not dimensionally homogenous. That is, they are in conflict with a fundamental law of fluid mechanics — Reynolds law of similitude.

From an earlier laboratory investigation of crack flows [1], improved semi-empirical equations have been derived. A method of applying them was proposed, but it has been found to have limitations and the present article is an attempt to provide an improved method.

Essentially, crack flow equations are required for two purposes.

(i) For use in a prediction method for investigating the effects of dwelling configuration, mechanical systems and external wind on ventilation rates.

(ii) For estimating the open areas of room components (e.g. doors, windows) when direct measurement is not possible.

The use of the equations in the above two ways and the implications of the equations with regard to the use of model-scale measurements of ventilation rates are discussed and some recent experimental results are presented.

2. DERIVATION OF EMPIRICAL EQUATIONS

Dimensional analysis indicates that for cracks with exact geometric similarity, the flow can be described by the functional relationship (see Appendix I)

$$C_z = f(R_{e_y}) \tag{2}$$

where C_z is the discharge coefficient

$$C_z \equiv \frac{V}{A}\sqrt{\frac{\rho}{2\Delta p}} \tag{3}$$

and the Reynolds number, R_{e_y}, is

$$R_{e_y} \equiv \frac{\bar{w}\, d_h}{\nu}. \tag{4}$$

In the above equations, ρ and ν are the air density and kinematic viscosity respectively and $\bar{w}$ is a mean velocity defined by

$$\bar{w} \equiv \frac{V}{A}. \tag{5}$$

The symbol A denotes an area of the crack corresponding to some specified cross-section and d_h is a typical dimension defined by

$$d_h \equiv \frac{4A}{\text{wetted perimeter}}. \tag{6}$$

For most cracks likely to be encountered in ventilation work

$$d_h = 2y \tag{7}$$

where A is the thickness of the crack as illustrated in Figure 1.

In practice, cracks take a variety of shapes and this implies a large number of distinct functions f. By introducing a geometric parameter one would hope to reduce this number to a manageable value. By analogy with pipe flows the ratio z/d_h suggests itself as a parameter, where z is the distance through the crack, as illustrated in Figure 1. Hence the crack flow equation is assumed to take the form

$$C_z = f\left(R_{e_y}, \frac{z}{d_h}\right). \tag{8}$$

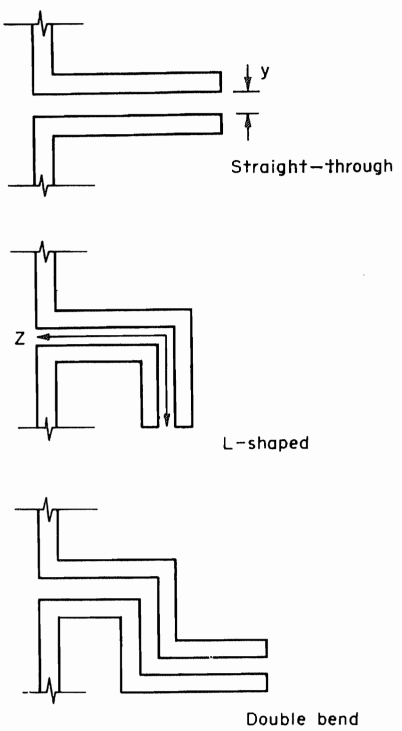


Figure 1

Common types of cracks encountered in buildings

The feasibility of using such an equation has been investigated [1] by carrying out tests on a large number of simulated cracks for ranges of R_{e_y} and z/d_h which are likely to be encountered in practice. The cracks were divided into three basic types — straight-through, L-shaped and double-bend (see Figure 1). For each type of crack it was found that the experimental results could be reasonably correlated by

$$\frac{1}{C_z^2} = B\frac{z}{d_h}\cdot\frac{1}{R_{e_y}} + C. \tag{9}$$

Figures 2 (a—c) show the experimental results and the empirical values of the constants B and C chosen for the three crack types.

Equation (9) can be derived by equating the pressure drop across the crack to the sum of the losses due to skin friction and end effects (see Appendix II). In the above form, the skin friction contribution corresponds to laminar flow. The equivalent equation for turbulent flow is

$$\frac{1}{C_z^2} = B\cdot\frac{z}{d_h}\cdot\frac{1}{R_{e_y}^{0.25}} + C \tag{10}$$

and the results for the double-bend cracks are plotted in Figure 3 as $1/C_z^2 \sim z/d_h \cdot 1/R_{e_v}^{0.25}$. Comparing this with Figure 2 it is clear that equation (9) is more appropriate than equation (10) for correlating the experimental data.

In reference [1] the use of equation (9) in combination with equation (1) is proposed. However, the introduction of (1) is inconsistent and unnecessary, because the flow is completely and rigorously described by (9) alone and it is therefore now proposed that it should be used alone.

Undoubtedly, equation (9) is a simplified representation of a complex flow situation. Bearing in mind the wide variety of crack types which are met in practice, however, it seems that some degree of simplification is unavoidable. Indeed, it is uncommon to find a room component for which the crack type and the crack dimensions are everywhere constant.

Despite these problems it will be seen from Section 3.2 that equation (9) has been found to be successful both for estimating the open areas of real full—scale components and for describing the flow through them.

The advantages which equation (9) has relative to equation (1) are two-fold. Firstly, it is dimensionally homogenous so that the value of the constants do not vary with the particular system of units used. Secondly, it takes into account the effect of Reynolds number (i.e. scale effect) which is an important effect (see Sections 3.1 and 4). As a result of this, the estimated values of open area are independent of the flow rate through the open area. This also means that where an open area can in fact be obtained by direct measurement, the value can be used directly in the equation. Equation (9) is thus claimed to have wider validity than equation (1).

3. USE OF CRACK FLOW EQUATIONS

3.1 Prediction of ventilation rates

The crack flow equation (9) has been incorporated into a computer program for predicting the flow rates through cracks in a multi-cell dwelling. An iterative procedure is used to solve the crack flow and continuity equations.

Figure 4 shows some results obtained from this program in the form of a plot of the total air change rate of an actual nine-roomed house against reference wind speed, U_{ref}. The open areas of this house have been estimated experimentally as described in Section 3.2. For these calculations the pressure distribution over the house was estimated, but for future calculations a scale model has been constructed to enable more accurate distributions to be obtained in a wind tunnel. These calculations will be compared with ventilation measurements made in the house in order that the prediction method can be fully assessed.

Also shown in Figure 4 are the corresponding results obtained assuming that the discharge coefficient of the open areas is independent of Reynolds number and crack type. It can be seen that this simple square-law approximation gives significantly higher air change rates than the crack flow equations.

3.2 Indirect measurement of open areas

For the prediction of ventilation rates for a full-scale dwelling it is necessary to know the open areas of each room component. In most cases a direct measurement of the full-scale area is not possible and an indirect method is employed. The pressure drop Δp across the component (at some point on the component) is measured, together with the corresponding volume flow rate. It can be shown that the flow equation can be written in the form of a simple cubic

$$A^3 \frac{2\Delta p}{\rho V^2} - C \cdot A - \frac{B z L^2 \nu}{4V} = 0 \qquad (11)$$

which enables A to be calculated knowing Δp, V, the crack type (which determines B and C), z and L, where L is the length of the crack. Figures 5—7 show the solutions of this equation for the three types of cracks. Hence, knowing the above quantities from full-scale tests it is possible to obtain estimates of open areas, A.

In the following the results of tests on a door in the house which were carried out to verify the above procedure are described. The tests also constitute a check on the suitability of equation (9).

The tests were carried out in the small bedroom of the house. Prior to the tests, vinyl floor covering was laid and the room was extensively sealed with adhesive tape. An orifice plate was used to measure the extraction rate of the air which was exhausted through the vent of the installed warm-air system. The static pressure difference across the door was measured with a micromanometer.

The dimensions of the door are shown in Figure 8. It should be noted that the quoted dimensions of the crack thickness, y, are only approximate since there was some variation of this quantity over the door.

Three separate $V \sim \Delta p$ characteristics were recorded. Firstly, the characteristic corresponding to the whole door was obtained. The lower gap of the door was then sealed and the second characteristic recorded. Finally, the door was completely sealed and the base level characteristic was obtained.

The base level characteristic was used to correct the other characteristics. That is, at a given value of Δp, denote the base level extraction rate by V_{base} and the extraction rate for the test with the door unsealed by V_1. The flow rate through the door at that value of Δp is then given approximately by $(V_1 - V_{base})$.

Figure 6 was then used to estimate the open area of the component from each measured point on the $V \sim \Delta p$ characteristic. These estimates are given in Figure 9. Also shown here are the estimates of A which are obtained from the $V \sim \Delta p$ characteristics on the assumption that C_z is constant ($C_z = 0.60$). It is clear that these latter estimates are far from satisfactory. The estimated open areas for the two door configurations are tabulated below, together with the values obtained from the direct measurement of y (see Figure 8).

	Area, A m^2	
	From crack flow equation	From direct measurement
Whole door	0.0157	0.0171
Lower gap sealed	0.0087	0.0095
Difference	0.0070	0.0076

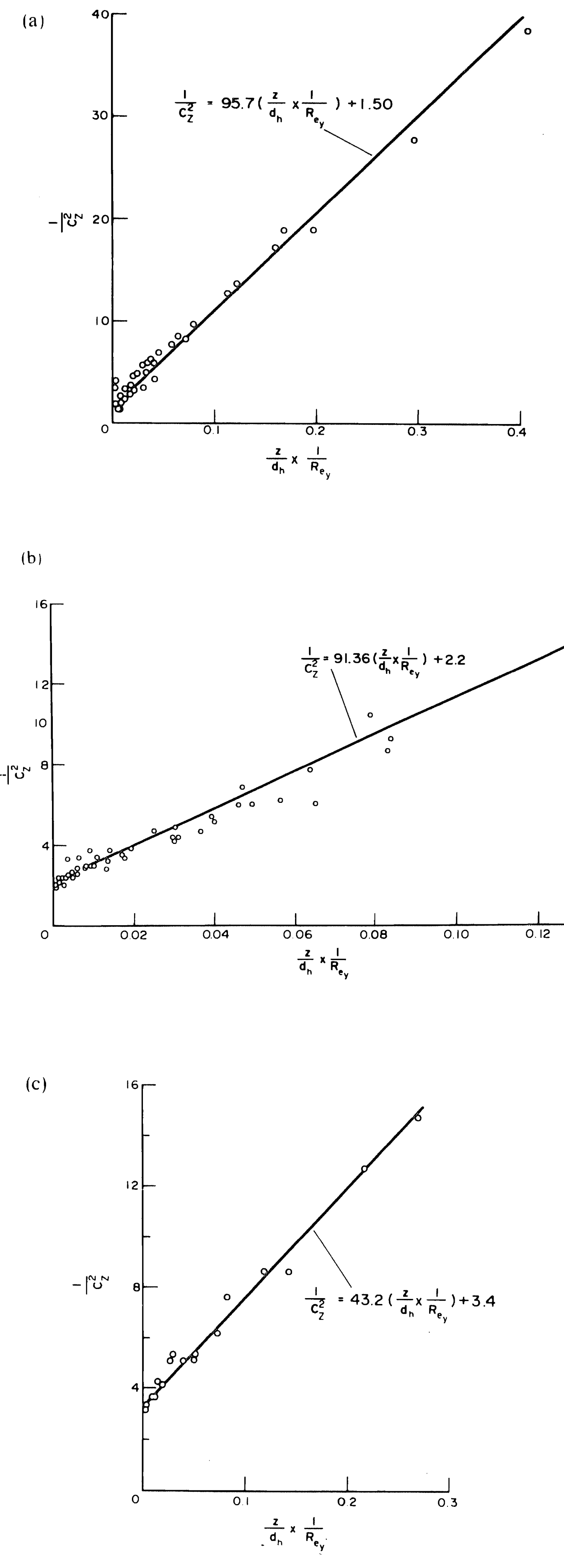


Figure 2

Experimental results and empirical lines chosen for the three crack types. (a) straight-through, (b) L-shaped, (c) double-bend

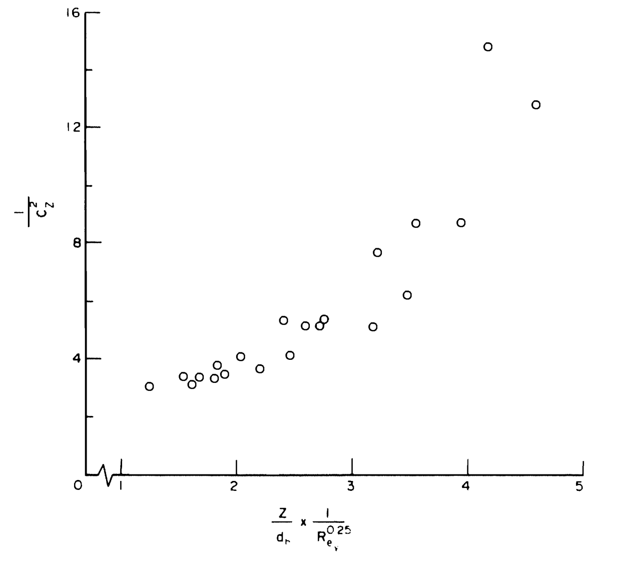


Figure 3

Experimental results for double-bend cracks plotted in accordance with turbulent flow

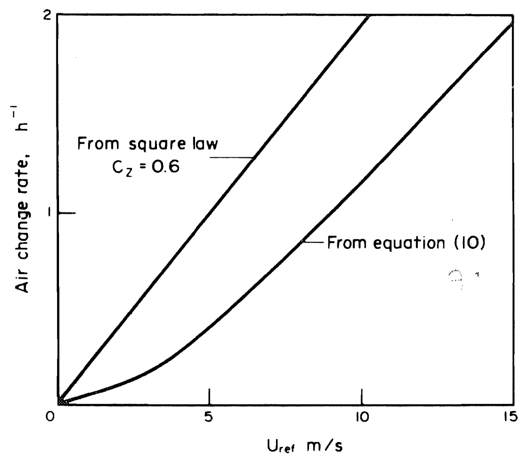


Figure 4

Calculated effect of wind speed on air change rate of a test house

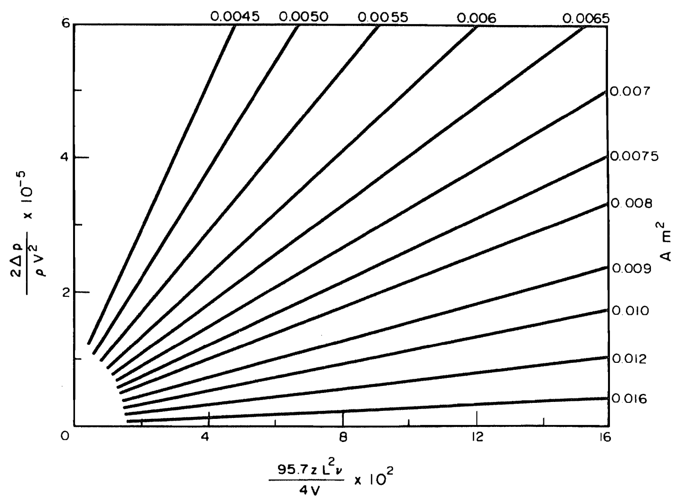


Figure 5

Solution of crack flow equation for straight-through cracks

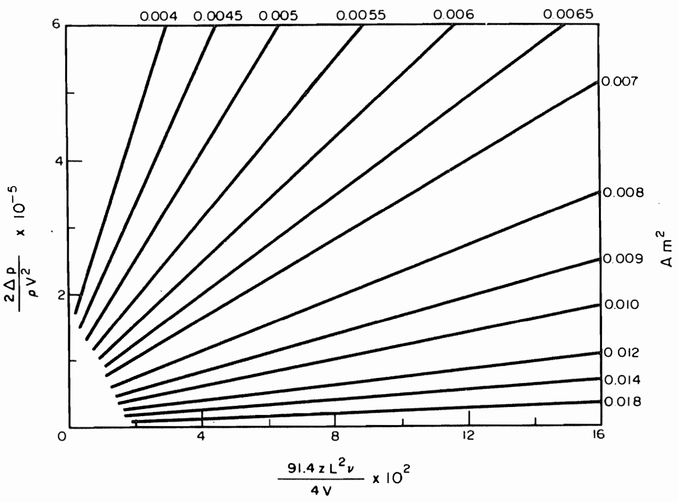


Figure 6

Solution of crack flow equation for L-shaped cracks

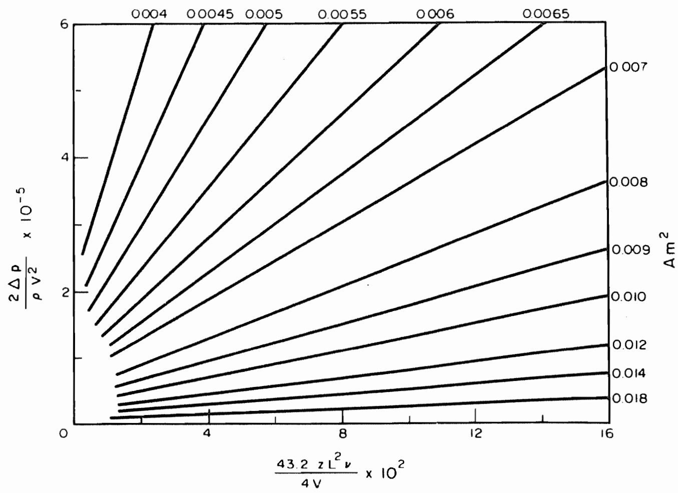


Figure 7

Solution of crack flow equation for double-bend cracks

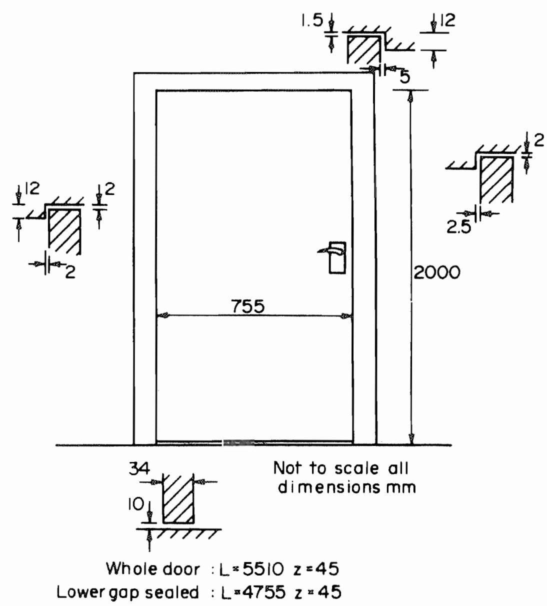


Figure 8

Dimensions of door in test house

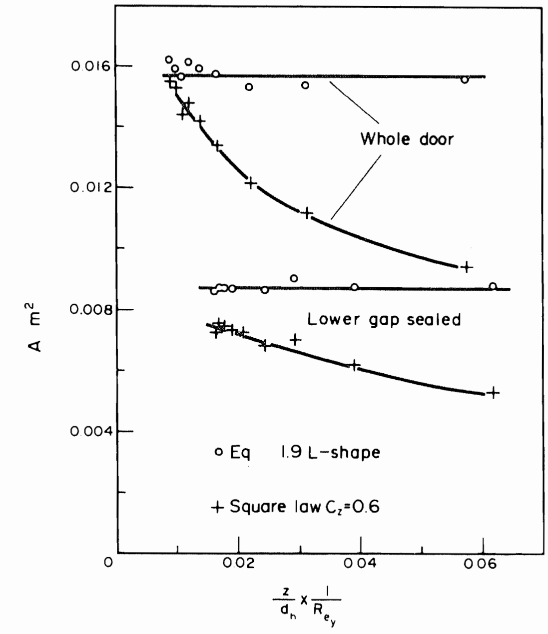


Figure 9

Estimated values of open area of door

It must be borne in mind when comparing the above values that it was not generally possible to measure y directly with great accuracy because of variations over the door. The thickness of the lower gap did not vary appreciably, however, and thus the good agreement between the difference values is encouraging.

All of the doors and windows of the house have had their open areas estimated in the above manner and the results obtained are qualitatively similar to those given in Figure 9. This means that the flow through these components can be accurately calculated knowing the pressure difference across the components.

When using the above technique for obtaining leakage data of houses one is often faced with room components (doors, windows) for which the crack type varies. One therefore has the choice of either dividing each room component into smaller parts according to crack type, or treating the component as a whole. If the first choice is adopted, the open area of each part can be measured by the pressure/extract technique (by sealing the other parts) and the total open area is then obtained by summing the individual open areas. If it were desired to use the open areas for prediction of flow rates for the house, then each part of the component would be treated separately. For reasons of economy, however, it might be desirable to treat each room component as a whole. Tests on all of the components in the house have shown that it is permissible to do this. For example the results for the whole door given in Figure 9 show that the flow characteristics of the whole door are adequately described by the equation for L-shaped cracks. In general, the open area obtained in this way must be treated as an effective open area, since it will not generally be equal to the sum of the individual parts. However when the component consists of straight-through and L-shaped cracks (for which the flow equations do not differ greatly) with similar values of z, the results from the tests on the door indicate that the effective open area is approximately equal to the total open area.

An extreme example of the use of the flow equations for estimating open areas is given in Figure 10. Here the 'background' open area of a room has been estimated from the measured base level flow characteristics, $\Delta p \sim V_{base}$. For the measurements the doors and windows in the room were sealed, so that the flow occurred through the remaining cracks in the surfaces of the room. Since it was not possible to identify the geometry of these cracks, the values of L, z, B and C used for estimating open area were arbitrarily chosen. Nevertheless, the fact that the estimated values of background area do not vary greatly with flow rate, indicates that the crack flow equations can be usefully employed to describe very complex crack formations.

To summarise, it can be seen that the pressure/extract technique coupled with the crack flow equations given here, can be used to obtain open areas of room components. When applied to whole components with varying crack geometry, the open area obtained is an effective total open area, which may differ from the actual total open area. However this is of little consequence when the results are used as data for predicting flow rates, since the equations describe the flow characteristics of whole components very well. Even for background areas the equations give an adequate description of base level flows.

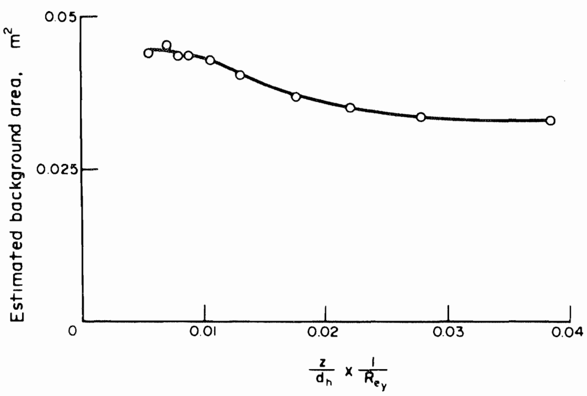


Figure 10

Estimated values of background area of a room

4. USE OF SCALE MODELS FOR DETERMINING VENTILATION RATES

Ventilation rates can be determined from model tests in wind tunnels either by measuring the external pressure distribution and using this as data for a theoretical prediction or by measurement of ventilation rates in the model. The use of a model for obtaining external pressure distributions for the purposes of design, development and/or research is an established procedure. The problem is whether or not the ventilation rates are best determined theoretically or by measurement at model scale.

A strong argument in favour of model scale measurement is that the available theory does not take account of the effects of wind turbulence and internal air movements. Both of these phenomena are at least present in model tests and there is evidence (e.g. references [3, 4]) that the former effect can be very significant.

There are however quite strong arguments against the use of model scale measurement. Firstly, it is difficult to model full-scale cracks accurately unless these are large or the model scale is large (1/25 scale, say). Secondly, it is generally not possible to achieve full-scale Reynolds number at model scale because of limitations on tunnel speed. Since the flow through cracks tends to be laminar, Reynolds number effects are likely to be significant. Figure 11 shows the theoretical variation of flow rate, $V/(U_{ref}A)$, through typical windows with reference wind speed U_{ref}. It can be seen that even with a large crack thickness of 5 mm, Reynolds number effects are significant up to a speed of about 5 m/s. To achieve the same Reyholds number with a 1/25 scale model would require tunnel speeds up to 125 m/s. For the smaller crack sizes, wind tunnel simulation over the required Reynolds number range appears to be even less feasible.

The relative importance of the effects of Reynolds number and turbulence has been investigated experimentally in a wind tunnel of the type described in reference [6]. A grid of horizontal slats was used to generate the simulated atmospheric boundary layer. Figure 12 shows the resultant profiles of mean velocity, U/U_{ref}, and turbulence intensity, U_{rms}/U_{ref}. These measurements were made with a pitot-static tube which was considered adequate for the present purposes. The intention was to obtain profiles appropriate to an open-country site, and the corresponding power law for the mean velocity is shown for comparison. A single-cell model fitted with identical model windows on two opposite faces, as shown in Figure 13, was chosen for the tests. Ventilation rates were obtained from tracer gas decay records measured by a katharometer installed inside the model. A full report on the tests will be published at a later date, but some relevant results are presented in Figure 14. The measured variation of $V/(U_{ref} \cdot A)$ with U_{ref} is shown for two wind directions $\alpha = 0^{\circ}$ and 90°. At $\alpha = 0^{\circ}$ the mean pressure difference across the windows is at a maximum, whereas at $\alpha = 90^{\circ}$ the mean pressure difference is negligibly small and the ventilation rate is due to turbulence effects alone. The U_{ref} axis can be considered as a Reynolds number axis.

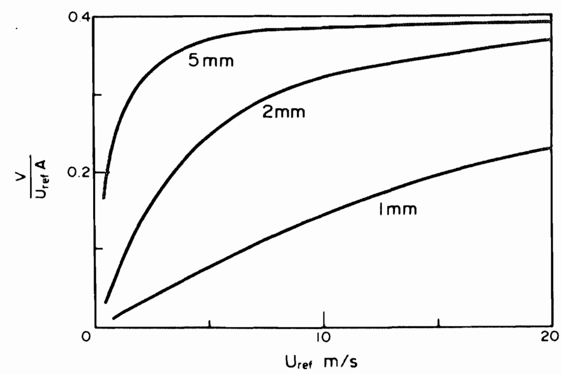


Figure 11

Theoretical calculations of scale effect for typical full-scale crack sizes

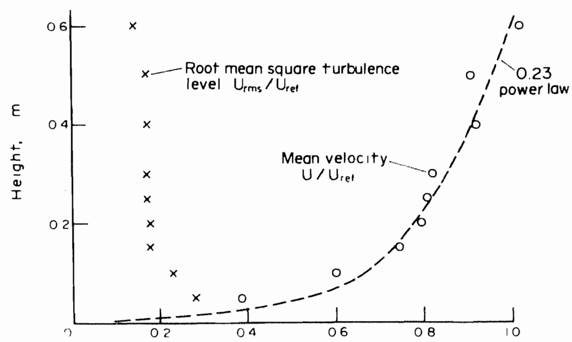


Figure 12

Profiles of mean velocity and turbulence intensity in the wind tunnel

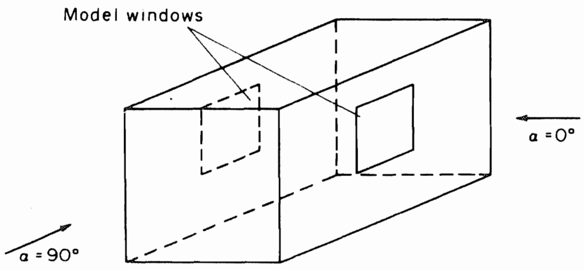


Figure 13

Wind-tunnel model used for measurement of ventilation rates through model windows and circular holes

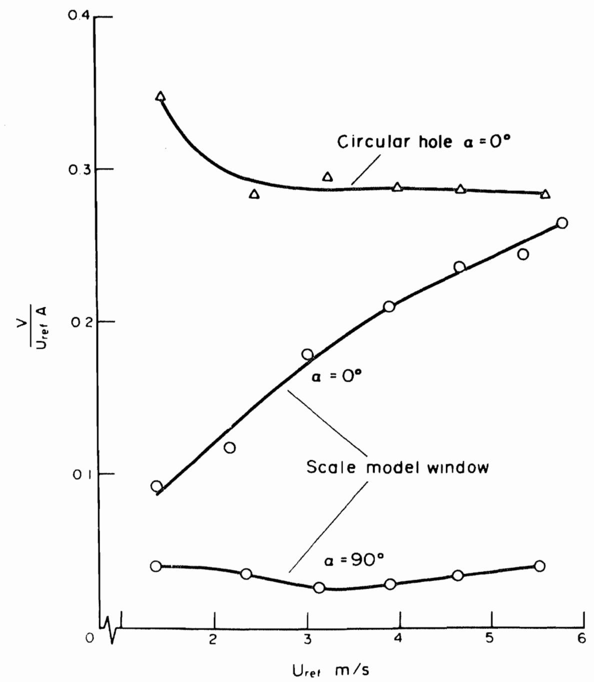


Figure 14

Scale effects for model windows and circular holes as measured with the wind-tunnel model

It can be seen that the ventilation rate due to turbulence alone ($\alpha = 90^\circ$), when suitably non-dimensionalised, is roughly independent of Reynolds number. In contract the dimensionless ventilation rate for $\alpha = 0^\circ$ increases rapidly with Reynolds number i.e. scale effect is significant. As far as magnitudes are concerned, at the highest Reynolds numbers tested, the ventilation rates at $\alpha = 0^\circ$ are much larger than those at $\alpha = 90^\circ$. However, at the lowest Reynolds numbers the ventilation rate due to turbulence alone is appreciable in relation to the ventilation rate for $\alpha = 0^\circ$.

Also shown in the figure are the results obtained for $\alpha = 0^\circ$ when the model windows are replaced by circular holes, as used by other workers (e.g. reference [5]). The flow characteristics of the two types of open area are clearly different, with the non-dimensional ventilation rate of the circular holes tending to a constant value at a relatively low Reynolds number. This behaviour indicates that circular holes can be used to model the behaviour of real windows in the limit of high Reynolds number. The open area of the circular holes should not be simply scaled geometrically, however, but allowance should be made for the different values of the discharge coefficients of full-scale cracks and of circular holes.

5. CONCLUSIONS

Crack flow equations of the form of equation (1) are not satisfactory because they do not satisfy Reynolds law of similitude. Earlier investigations (reference [1]) have shown that the flow through simulated full-scale cracks can be adequately described by the semi-empirical equation (9). It is now confirmed that this equation should not be used in the manner described in reference [1], but should be used on its own for obtaining the discharge coefficient.

Full scale tests on a door in a house have been found to support the use of equation (9) for estimating open areas of real full-scale room components. It has also been found from tests of all the room components that the equation satisfactorily describes the flow through a wide range of components.

The usefulness of ventilation rates measured at model scale for the design of full-scale dwellings is open to doubt, because of the large effect of scale implied by the crack flow equations. Calculations indicate that scale effect is apparent even at the Reynolds numbers associated with full-scale dwellings.

Tests carried out on a model in a wind tunnel have illustrated the significance of scale effect when the ventilation rate is due mainly to the existence of a time-mean pressure difference across a crack. When the time-mean pressure difference is zero, and the ventilation rate is due solely to turbulent pressure fluctuations, the scale effect is much smaller. *At the lowest Reynolds numbers tested, the ventilation rate due to turbulence alone is about one half of the ventilation rate which exists when the time-mean pressure difference is maximised.*

The model tests have also illustrated the different flow characteristics of scale model cracks and circular holes. Circular holes can be used to model the behaviour of real cracks in the limit of high Reynolds number, provided that the open areas of the holes are correctly determined.

ACKNOWLEDGEMENTS

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*See Build. and Env., **14**, 53–64 (1979) for further discussion of this conclusion.

APPENDIX I

The dimensionless groups which describe the flow through geometrically similar pipes can be obtained by dimensional analysis in the manner shown for example in reference [7]. One obtains

$$\frac{\Delta p}{\rho \bar{w}^2} = f_1\left(\frac{\bar{w} \cdot D}{\nu}\right) \equiv f_1(R_{e_D}),$$

where D is the pipe diameter and R_{e_D} is the corresponding Reynolds number. The analysis for flow through geometrically similar cracks is the same, and one obtains

$$\frac{\Delta p}{\rho \bar{w}^2} = f_1\left(\frac{\bar{w} \cdot d_h}{\nu}\right) \equiv f_1(R_{e_y}),$$

which, on putting $\bar{w} = V/A$, can be expressed in a form involving the discharge coefficient, i.e.:

$$\frac{1}{C_z^2} = f_1(R_{e_y})$$

or

$$C_z = j\,(R_{e_y}).$$

APPENDIX II

The total pressure drop across the crack can be considered as the sum of pressure drops due to skin friction (of the form encountered in long straight cracks) and due to bends and end effects. This is the procedure which is generally adopted for pipe flows (e.g. see reference [7]). It is assumed that the pressure drops caused by bends and end effects are simply proportional to $\rho \cdot \bar{w}^2/2$.

In the following, equation (9) is derived using the above procedure, in a similar way to that given in reference [1].

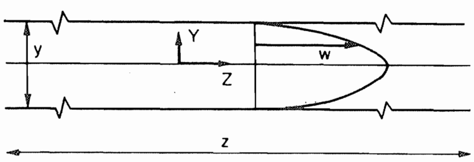

Fig. A1.

For steady laminar flow between parallel walls, the velocity distribution is parabolic (see e.g. reference [8]), i.e.:

$$w(Y) = \frac{1}{2\mu}\,\frac{dp}{dZ}\left(\left(\frac{y}{2}\right)^2 - Y^2\right) \qquad \text{(A1)}$$

In the present notation Y and Z denote distance in the crosswise and streamwise directions, respectively, with the origin of Y at the centre-line. The crack has breadth y and length z. $w(Y)$ denotes the streamwise velocity at Y and since the flow is parallel everywhere it follows that the pressure gradient is constant, so one can put

$$\frac{dp}{dZ} = \frac{\Delta p}{z}.$$

From (A2), the mean velocity $\bar{w}$, defined by equation (5) is given by

$$\bar{w} = \frac{V'}{y} = \frac{1}{y} \cdot \int_{-y/2}^{y/2} w(Y).\mathrm{d}\,Y,$$

where V' is the volume flow rate per unit width.

On integration one obtains

$$\bar{w} = \frac{1}{12} \cdot \frac{\Delta p \,.\, y^2}{\mu z}. \qquad \text{(A3)}$$

Now

$$\frac{1}{C_z^2} \equiv \frac{2\Delta p}{\rho \bar{w}^2}$$

and since

$$R_{e_y} \equiv \frac{\bar{w}\, d_h}{\nu} \quad \text{and} \quad d_h = 2y$$

we can express (A3) in the form

$$\frac{1}{C_z^2} = 96 \cdot \frac{1}{R_{e_y}} \cdot \frac{z}{d_h}. \qquad \text{(A4)}$$

Proceeding in the manner described above, the total pressure drop, Δp_T, is given by

$$\Delta p_T = \frac{\rho \bar{w}^2}{2} \cdot 96 \cdot \frac{1}{R_{e_y}} \cdot \frac{z}{d_h} + \frac{\rho \bar{w}^2}{2} \cdot C$$

where C is a constant. Thus the flow through the crack is described by

$$\frac{1}{C_z^2} = 96 \cdot \frac{1}{R_{e_y}} \cdot \frac{z}{d_h} + C$$

which is the required form.

It is interesting to note that the empirical values of B obtained for the straight-through and L-shaped cracks are close to the value 96. This supports the assumption that the pressure drop can be considered as two additive components. The value of B obtained for the double-bend crack type, indicates that the assumption becomes less tenable as the number of bends is increased. Nevertheless the flow is still adequately described by an equation of the same form.