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Ventilation of an Enclosure Through a Single Opening

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Building ventilation is affected not only by the steady mean effect of air pressures and temperatures around and within the building, but also by the turbulent nature of wind causing a diffusion of air through openings and cracks in the building envelope.

The ventilation of an enclosure with a single opening subjected to a turbulent impinging airstream is studied and simple theoretical models are derived to assist in understanding the physical phenomena causing airflow through the opening. These are compared with the results of experiments on a large scale model. The need for further work on this problem is stated.

NOMENCLATURE

A	opening area
c	concentration of tracer gas
f	mixing coefficient
k_1	wave number component in direction of X-axis
K	roughness coefficient
n	frequency
p	pressure
q	flow rate
$S_x(n)$	spectral density function
t	time
$\mathbf{u}$	velocity vector
u_x, u_y, u_z	velocity components in x, y and z directions
u'_x, u'_y, u_z	fluctuating components of velocity in x, y and z directions
$\hat{\mathbf{u}}_x, \mathbf{u}_y, \hat{\mathbf{u}}_z$	root mean square components of velocity in x, y and z directions
V_t	enclosure volume
v	imaginary volume of air at opening
x, y, z	cartesian coordinate axes
β	coefficient of discharge for opening
γ	ratio of specific heats of air
θ	angle of incidence of airstream
ρ	density of air
ω	radian frequency.

DESIGN ASSESSMENT AND VENTILATION

In addition to providing information relating to the adequacy of the environmental conditions experienced by building occupants, design assessment facilities have an important rôle in the estimation of the energy cost implied by alternative design decisions. The consideration of this latter objective is being increasingly called to our attention as the necessity for the conservation of scarce energy resources becomes crucial. When the savings accrued by even marginal reductions of energy consumption are significant, the accuracy of the predictive techniques we use is of primary importance. With the gradual improvement in methods of assessing the thermal performance of buildings, the lack of a satisfactory method of evaluating ventilation rates for design assessment purposes becomes an acute problem. Prediction of natural ventilation, infiltration and air movement in buildings has always had to rely on uncertain and imprecise techniques. There are several reasons why this situation persists. Firstly, the physical relationships that govern the movement of air through a building are non-linear; this tends to undermine attempts at a simple solution. Secondly, the air flows are directly motivated by variable external forces such as wind speed and direction, and modified by the operation of openable windows, doors, lifts and other components, and the effects of heating and ventilating services. Thirdly, the actual characteristics of building components can vary considerably between samples, due to manufacturing and installation tolerances. The combination of all these factors makes a precise evaluation of air flows a practical impossibility. However, for design appraisal purposes, it should be possible to develop a form of statistical analysis to allow for the parameters which cannot be precisely determined, and to use this as part of a computerised system simulation for relatively precise assessment of building performance, in the sense of "climatic modification". Before this approach can be advanced further, a fuller understanding of the more complex physical processes involved must be achieved, in order that the desired degree of precision be attained.

When air flows around a building with openings to its interior, air flow through the openings will be encouraged by a variety of factors which can be broadly classified into two groups; those which induce steady flow by virtue of their mean value, and those whose effect is due to their fluctuating nature. Factors which fall into the first group include the mean wind pressures on the building surfaces and the effect of temperature differences between inside and outside the building. This paper is concerned with one of the second group of factors whose effect becomes apparent if one considers the ventilation that takes place in a sealed room with only one window open to the outside. The mechan-

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ism whereby air exchange takes place across such an opening due to fluctuating external air velocity is obviously fairly complex and consists of a combination of effects. A low frequency variation in pressure at the opening will induce a fluctuating flow due to the compressibility of the air in the internal space. Higher frequency fluctuations will enable air exchange at the opening due to smaller scale turbulence with eddy sizes comparable with, or smaller than the opening size, providing an additional contribution to ventilation in the internal space. If more than one opening exists then a fluctuating flow will be set up dependent on the correlation between the pressures generated at each opening. Each of these factors will depend on a variety of parameters, e.g. the sizes and shapes of openings, the nature of the external airflow at the openings, the size of the internal space, and the magnitude of any net mean airflow through the openings.

Malinowski's experiments[1] using a model with airflow passing across the openings in a direction parallel to the wall containing the openings demonstrated some of these effects. He showed that ventilation rate increases as the spacing between two coplanar holes is increased, as mean air velocity is increased and, for a given mean air velocity, as turbulent intensity is increased. He also showed the effect of different wall thicknesses, and various permeable materials in place of the simple hole. There was no attempt, however, to quantitatively correlate the airflows with turbulence in the external airstream. Harris-Bass *et al.*[2] used a model consisting of a cuboid with an opening in each of two opposite faces parallel to the air stream. Experiments were carried out both in the wind tunnel and full scale and good correlations were obtained between the two sets of results. Again no measure of turbulence or correlation with ventilation rates was made other than on an empirical basis. It would be generally useful to be able to predict, for any given configuration, the airflows that will result under sets of external conditions. This study was aimed at examining the feasibility of generating such a predictive facility.

2. DERIVATION OF A SIMPLE THEORETICAL MODEL

Consider the case of an enclosure with only one opening in a plane wall, and an airstream moving with velocity $\mathbf{u}$ directed towards and impinging on the wall containing the opening (Fig. 1) with components of velocity u_x, u_y and u_z in the x, y and z directions respectively. We define $\mathbf{u} = \bar{\mathbf{u}}+\mathbf{u}'$ where $\bar{\mathbf{u}}$ is the mean velocity and $\mathbf{u}'$ the fluctuating component with zero mean. Assume initially that $\bar{u}_y = \bar{u}_z = 0$, i.e. the airstream is directed along the normal to the plane surface containing the opening. As the airstream approaches the surface containing the opening, the x-component of velocity will decrease and the static pressure will increase. At the opening itself the mean velocities, denoted by $\bar{u}_x$, $\bar{u}_y$, $\bar{u}_z$, will all be zero, but the fluctuating components u_x', u_y', u_z' will not. There will therefore be a fluctuating transfer of air through the opening in either direction. The low frequency content of the fluctuating velocity will produce a pulsating flow through the opening which will depend on the compressibility of the air in the enclosure, in other words, the size of the enclosure. Also at low frequencies, there will be a high correlation between velocities at different points within the opening. On the other hand, at high frequencies there will be a low correlation between velocities at different points within the opening. These high frequency fluctuations will produce a turbulent diffusion of air through the opening which will be less dependent on compressibility effects. The relative contributions to airflow through the opening due to fluctuations in different bands of the frequency spectrum will not be independent. For example a large low frequency pulsating flow will tend to minimise the contributions due to smaller scale turbulence. For ventilation to take place within the enclosure, i.e. for a net exchange of air between the enclosure and outside to occur, some fraction of the fluctuating airflow passing

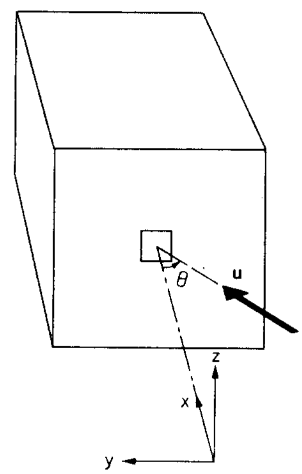

Fig. 1. Schematic diagram showing coordinate axis representation.

through the opening must be mixed with the bulk of air within the enclosure, the remainder passing back out without mixing. This fraction will depend on the magnitude of the fluctuations, and on the spectrum of the turbulence at the opening, as well as the intensity of turbulence within the enclosure.

In view of these several effects, an intensive study of the problem is required, leading to the definition of a mathematical model that will adequately describe their combined effect on ventilation of the enclosure. Rather than attempt such a complex description at this stage, it was decided to neglect for the present some of these factors and formulate a simple model to correlate external airflow with ventilation. Since it was expected that most of the turbulent energy would be contained in the low frequency part of the spectrum, and since the detailed analysis of the turbulent air motion at the opening would require quite an involved investigation and fairly extensive experimental work, it was decided to concentrate on the study of the low frequency pulsating airflow through the opening.

With the configuration of Fig. 1 and the mean air-flow directed along the normal to the plane of the opening, we will assume that the flow is quasi-steady, and that the stagnation pressure of the airstream is generated immediately outside the opening. Using the suffix a to represent conditions in the free airstream and o for conditions immediately outside the opening, the stagnation pressure of the air outside the opening p_o is given by

$$p_o = \tfrac{1}{2}\rho u_{x,a}^2 + p_a \qquad (1)$$

where ρ is the density of the air. If we assume that the opening behaves as a sharp orifice and that the Reynold's number is high enough for Bernoulli's theorem to apply, then using the suffix i to represent conditions inside the enclosure, and applying a coefficient of discharge β, the flow rate of air through the opening, q, is given by

$$q = \pm A\beta(2|p_o - p_i|/\rho)^{\frac{1}{2}} \qquad (2)$$

where p_i is the static pressure inside the enclosure and A is the area of the opening. If we further assume that p_i is constant, and equal to $\overline{p_o}$, we obtain from (1),

$$p_o - p_i = p_o - \overline{p_o} = \tfrac{1}{2}\rho(u_{x,a}^2 - \overline{u_{x,a}^2})$$

Now, since $u_{x,a} = \overline{u_{x,a}} + u'_{x,a}$ and $\overline{u'_{x,a}} = 0$, we can obtain

$$u_{x,a}^2 - \bar{u}_{x,a}^2 = 2\overline{u_{x,a}}u'_{x,a} + u'^2_{x,a} - \overline{u'^2_{x,a}}$$

and so

$$2|p_o - p_i|/\rho = |2\overline{u_{x,a}}u'_{x,a} + u'^2_{xa} - \overline{u'^2_{x,a}}|$$

Hence substituting into equation (2)

$$q = \pm A\beta(|2\overline{u_{x,a}}u'_{x,a} + u'^2_{x,a} - \overline{u'^2_{x,a}}|)^{\frac{1}{2}}$$

and

$$\hat{q} = (\overline{q^2})^{\frac{1}{2}} = A\beta(2\overline{u_{x,a}}\,\overline{|u'_{x,a}|})^{\frac{1}{2}} \qquad (3)$$

where $\hat{q}$ is the root mean square value of q.

As will be seen later, measurements of the mean airflow rate $\overline{|q|}$ into the enclosure are most readily obtained and we therefore require to relate the r.m.s. of q, as expressed above, to $\overline{|q|}$. Now if q is assumed to vary randomly and to be describable in terms of a Gaussian probability distribution, it can be shown that $\overline{|q|}$ is estimated by $(2/\pi)^{\frac{1}{2}}\hat{q}$; therefore

$$\overline{|q|} = A\beta(2/\pi)^{\frac{1}{2}}(2\overline{u_{x,a}}\,\overline{|u'_{x,a}|})^{\frac{1}{2}}.$$

On the other hand, measurements of $\hat{u}_{x,a}$, the root mean square of $u'_{x,a}$ are more readily obtained than the mean $\overline{|u'_{x,a}|}$ contained in the above equation. Therefore by assuming that $u'_{x,a}$ also varies randomly in a Gaussian manner, we can estimate $\overline{|u'_{x,a}|}$ by $(2/\pi)^{\frac{1}{2}}\hat{u}_{x,a}$ and thus,

$$\overline{|q|} = A\beta(2/\pi)^{\frac{3}{4}}(2\overline{u_{x,a}}\hat{u}_{x,a})^{\frac{1}{2}}. \qquad (4)$$

Clearly the assumption that both $u_{x,a}$ and q can be described in terms of Gaussian probability distributions cannot be strictly justified—they are related non-linearly as is shown in the derivation of equation (3). However for the sake of simplicity we will use this assumption in order to obtain a simple function correlating these variables. Since $\bar{q} = 0$, the mean flow into and out of the enclosure being the same, the mean flow rate into the enclosure q_i is equal to $\frac{1}{2}\overline{|q|}$. Clearly all the air flowing into the enclosure, as represented by q_i does not necessarily contribute to the effective ventilation of the enclosure. Some of the inflowing "pulsations" of air will be exhausted before mixing has a chance to occur. This proportion will not be fixed by any means. As the frequency of the pulsating flow increases, with $\overline{u_{x,a}}$ and $u'_{x,a}$ fixed, the amount of air mixed with the bulk of air in the enclosure will decrease, as the volume of the injected "slugs" of air reduces. In fact the frequency spectrum of the turbulent airstream will determine, in a complex way, the actual proportion of air involved in ventilating the internal space. We will call this proportion f. Therefore the net mean effective flow rate of ventilating air $\overline{q_{eff}}$ into the enclosure is given by

$$\begin{aligned}\overline{q_{eff}} &= f\overline{q_i} \\ &= \tfrac{1}{2}f\overline{|q|}. \end{aligned} \qquad (5)$$

The value of f would be expected to lie in the range $0 < f < 1$.

Measurements of $\overline{u_{x,a}}$ and $\hat{u}_{x,a}$ can be readily obtained using conventional instrumentation, and some measure of correlation with $\overline{q_{eff}}$ should be possible. However, since modern instrumentation allows us to record and electronically process varying signals, a slightly more sophisticated version of this simple model can be constructed which will still be capable of direct comparison with experimental results. This will overcome some of the inherent shortcomings of the foregoing analysis, avoiding the more gross approximating assumptions. It was stated previously that since low frequency turbulence was expected to be dominant, pulsating flow would occur through the opening, and that this would depend on the compressibility of the air within the enclosure. The effect of compressibility has so far been neglected, and led to the simplicity of the above result. We will now develop the analysis further to include this effect. If we assume the enclosure to be an adiabatic system,

$$p_i(V_i - v)^\gamma = p_a V_i^\gamma \qquad (6)$$

where V_i is the total volume of the enclosure, and $v = \int_0^t q\,dt$ represents the decrease in volume of the original mass of air inside the enclosure when $p_i = p_a$ at time $t = 0$. Since $|v/V_i| \ll 0$ and $(p_i - p_a)/p_i \ll 1$, equation (6) can be simplified to

$$(p_i - p_a)/p_a - \gamma v/V_i = 0,$$

so from equation (1),

$$p_o - p_i = \tfrac{1}{2}\rho[u_{x,a}^2 - (2\gamma p_a/\rho)v/V_i]$$

and substituting into equation (2) gives

$$q = \frac{dv}{dt} = \pm A\beta|u_{x,a}^2 - (2\gamma p_a/\rho)v/V_i|^{\frac{1}{2}}. \qquad (7)$$

Equation (7) is a non-linear differential equation which can be solved for q given the velocity $u_{x,a}$. Hence q_{eff} can be obtained from equation (5).

There is one more adaptation which can be made where, in the context of building ventilation, the effect of variations in the direction of mean flow of the free airstream is of interest. Suppose the mean velocity is directed in the x–y plane towards the centre of the opening at an angle θ to the normal to the plane of the opening (Fig. 1). Then

$$\overline{u_{x,a}} = \overline{|\mathbf{u}_a|}\cos\theta \qquad (8)$$

and this value can be substituted into equation (4). If only moderate values of θ are allowed, then, retaining our assumption of isotropic turbulence, we can substitute on equation (7) for

$$u_{x,a} = \overline{|\mathbf{u}_a|} \cos\theta + u'_{x,a}. \quad (9)$$

At greater angles of incidence than, say, 60°, the complexity of the flow is outwith the scope of this study.

In the foregoing analysis we have neglected to allow for the fact that there is considerable distortion of the flow in the vicinity of the opening, and this will undoubtedly affect the relative values of the components of turbulent velocity in the directions of the three coordinate axes. The above equations can only be described as approximations and, at the level of this analysis, it is unlikely that we can proceed much further.

3. EXPERIMENTAL STUDY

In an attempt to assess these models for flow through an opening in a sealed enclosure it was decided to carry out an experimental investigation to correlate ventilation in the enclosure with turbulence in the impinging airstream with the intention of proceeding to full scale at a later stage. A large wooden box was constructed with dimensions approximately 1·2 × 1·2 × 2·4 m. This was completely sealed except for a 15·2 cm square opening in the centre of one of the small faces. The box was laid with this face lying in a vertical plane. A variable speed fan with an outlet 40·7 cm wide and 25 cm high was placed with the plane of the outlet situated 2·18 m away from the opening and the centreline of the outlet lined up to the centre of the opening. When it was desired to reduce the intensity of turbulence in the airstream a 1·68 m length of 40 cm dia. duct was inserted at the fan outlet. A D.I.S.A. Hot-wire anemometer operating in constant temperature mode was placed on the centre line of the fan, 30 cm from the enclosure. This had a uniform frequency response up to 250 kHz, which was adequate for the purposes of this study. The instrument gave direct readings of mean and r.m.s. velocity to an accuracy within ± 3 %. The ventilation rate was measured by releasing a small quantity of nitrous oxide gas into the box, allowing this to mix with the air in the box, and then measuring the decay in concentration of gas using an i.r. gas analyser and recording the output on a chart recorder. The effective mean ventilation flow rate q_{eff} was then obtained to an accuracy within ± 5 % by fitting the plotted values to a curve of the form

$$c_t = c_0 \exp(-t/t_c) \quad (13)$$

where c_0 = gas concentration at time $t = 0$

c_t = gas concentration at time t

t_c = ventilation rate time constant

whence $q_{\mathrm{eff}} = V_i/t_c$.

A small circulating fan was placed on the floor near the centre of the box in order to maintain a uniform gas concentration within the box. The airstream from the fan was directed away from the opening. The air sample point was located at the inlet to this fan. The magnitude of ventilating airflow which occurred at the opening resulting from the action of this fan and the ambient turbulence in the laboratory was measured as 0·3 l/s, the highest velocities being about 50 mm/s in the region of the opening. This was considered to be an acceptably small background level. A Solartron HS7-3A analogue computer was programmed to solve equation (7). The anemometer signal was transmitted to the computer which evaluated q_t to an accuracy within ± 6 %.

In order to assess the assumptions made regarding the spectra of the velocity fluctuations, recordings were made of the anemometer output signal during a number of test runs. These were later analysed using a B&K sound spectrum analyser. In addition, a sample of data was digitised and the power spectral density function evaluated by a digital computer program using the Blackman–Tukey method[3] and smoothing using the Hamming window function. The digital data was also used as input to a program which was written to effectively solve equation (7) using a finite difference technique. This provided a useful check on the operation of the analogue computation and was more easily adaptable to some of the test conditions set up.

Experiments were conducted to assess, over a range of mean air velocities, the effects of:

(a) two different intensities of turbulence;

(b) two different sizes of opening;

(c) airflow impinging normal to and at two angles to the plane of the opening.

The spectral density function of a sample of digitised data is shown in Fig. 2. There are no sharp peaks evident in the record. Certainly most of the energy is contained in large scale eddies, although smaller scale turbulence is present at frequencies up to 100 Hz. This could be interpreted as being indicative of the relative quantities of air transferred by a low frequency pulsating flow, and high frequency eddy diffusion.

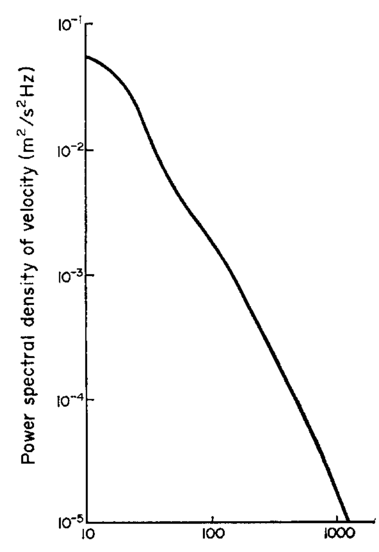

Fig. 2. Power spectral density of impinging air velocity, r.m.s. value = 1·6 m/s.

The velocity magnitude spectra, expressed as dB *re* the r.m.s. velocity, were plotted against wavenumber k_1, which was obtained using the approximate relationship $k_1 = \omega/\overline{u_{x,a}}$ where ω is the radian frequency. It was found that, for each of the two experimental configurations—one for high and one for low turbulent intensity—the spectra obtained for each record over the range of mean velocities fitted the same curve. The two curves, corresponding to mean values of $\hat{u}_{x,a}/\overline{u_{x,a}}$ of 0·252 and 0·104 for high and low turbulent intensities

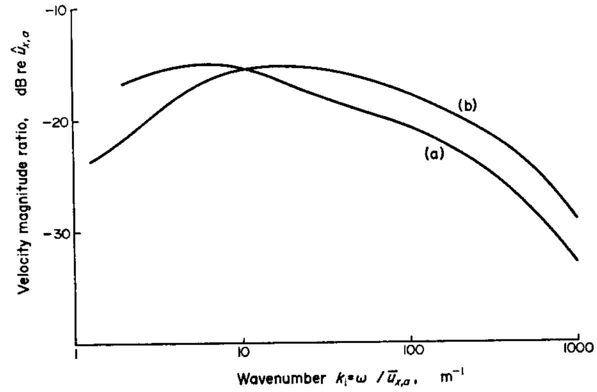

Fig. 3. Velocity magnitude spectra. (a) "high" turbulent intensity, (b) "low" turbulent intensity.

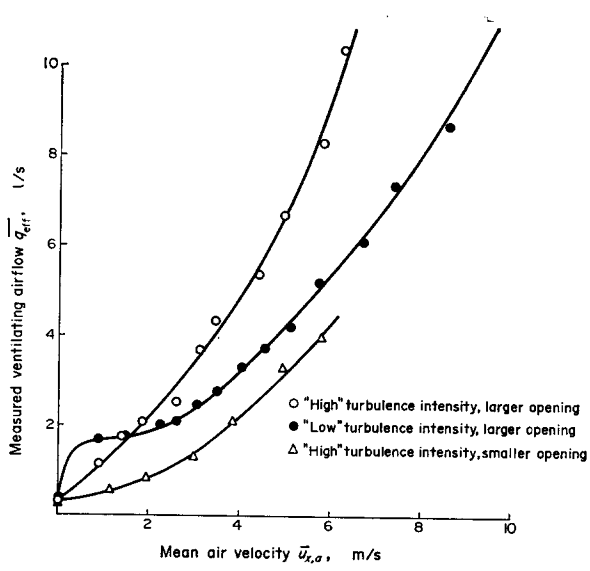

Fig. 4. Experimental results plotted as measured mean ventilating airflow $\overline{q_{eff}}$ against mean impinging air velocity $\overline{u_{x,a}}$.

respectively, are shown in Fig. 3. The sample of digitised data was therefore suitably rescaled for each test and input to the digital program described previously.

Figure 4 shows the measured ventilating flow rates $\overline{q_{eff}}$ plotted against $\overline{u_{x,a}}$. $\hat{u}_{x,a}$ was approximately proportional to $\overline{u_{x,a}}$ for each experimental configuration. The ventilating airflow was less at corresponding values of $\overline{u_{x,a}}$ when the turbulent intensity was reduced and less still with the smaller opening area. Clearly the mean velocity alone cannot give much guide to ventilation rates.

If the length of the opening side is taken as a characteristic dimension, then at a velocity of 0·5 m/s a Reynold's number of about 5000 is obtained. Therefore, within the limits of our assumptions, the models derived previously ought to be valid over the range of variables tested.

The results for airflow normal to the opening using the simple model and the analogue and digital computer versions of the adapted model are shown in Fig. 5. A value of $\beta = 0{\cdot}65$ was assumed. The computed flow inwards through the opening $\overline{q_i}$ is plotted against $\overline{q_{eff}}$ as measured by gas concentration decay. The curves obtained for the analogue and digital computer models agree well although there is a substantial difference between the correlation obtained for these models, and

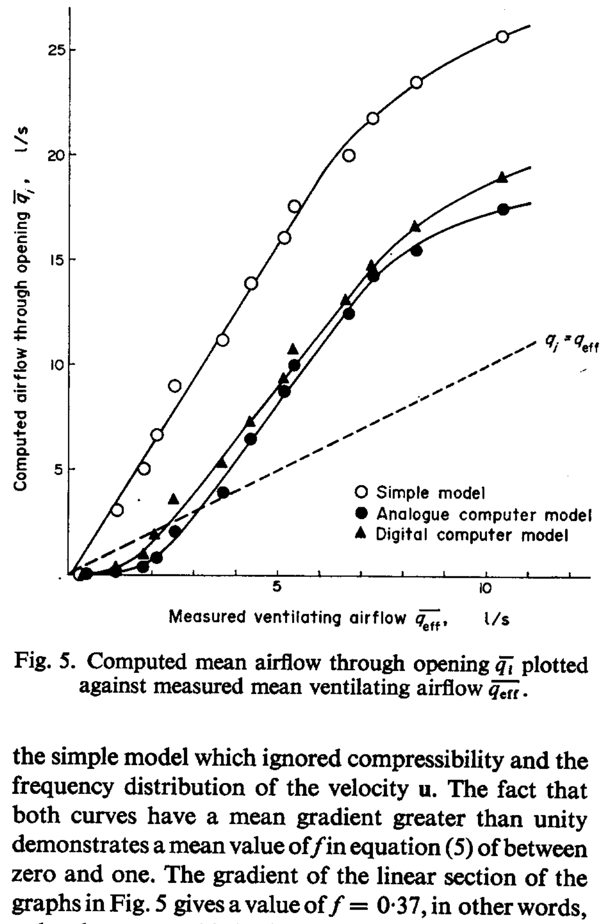

Fig. 5. Computed mean airflow through opening $\overline{q_i}$ plotted against measured mean ventilating airflow $\overline{q_{eff}}$.

the simple model which ignored compressibility and the frequency distribution of the velocity **u**. The fact that both curves have a mean gradient greater than unity demonstrates a mean value of f in equation (5) of between zero and one. The gradient of the linear section of the graphs in Fig. 5 gives a value of $f = 0{\cdot}37$, in other words, only about one third of the airflow passing into the enclosure is ultimately mixed with the bulk of the air within the enclosure.

The interpretation of the shape of the curve for the adapted model may shed some light on the physical processes involved in the air transfer through the opening. As the air velocity increases from zero, the measured ventilation rate increases quite rapidly. At these very low speeds the pulsating flow will be minimal and small scale eddies will have maximum effect, encouraging turbulent diffusion at the opening. This is not accounted for in the model and hence the initial gradient of the curve $(1/f)$ is low. As the velocity increases this smaller scale turbulent diffusion is masked by the increasing pulsating flow due to low frequency velocity fluctuations. Therefore, although the relative quantities of energy in the wavenumber spectrum remain unaltered, the higher frequencies have less effect on $\overline{q_{eff}}$, and hence the errors inherent in the model are reduced. Also the gradient of the curve exceeds unity indicating the proportion of the airflow entering the enclosure which mixes with the air inside the enclosure. A possible cause for the levelling-off effect at very high velocities could be the fact that, as the turbulence within the enclosure increases due to the increased activity at the opening, so the value of f is increased.

Figure 6 shows the effect of reducing turbulence intensity by inserting a length of duct at the fan outlet. These results correlate reasonably well with those for the high turbulence case. The results obtained when the size of the opening was reduced to 10·15 cm square are also shown in Fig. 6. This corresponds to an area of

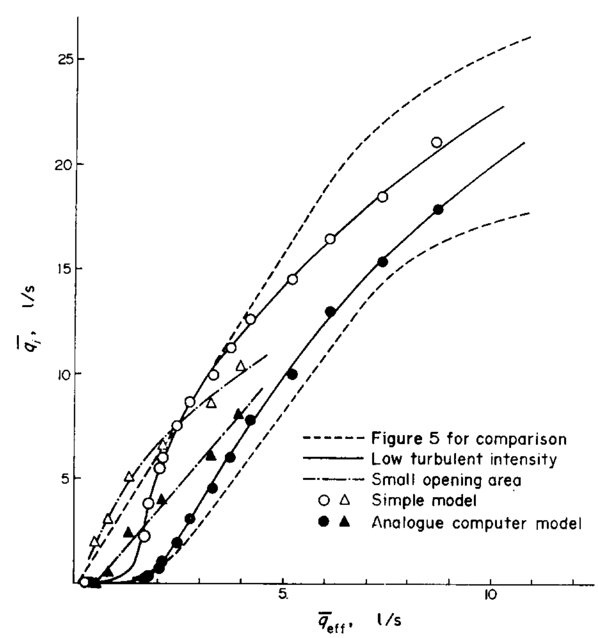

Fig. 6. Computed mean airflow through opening $\overline{q_i}$ plotted against measured mean ventilating airflow $\overline{q_{eff}}$; ○● low turbulent intensity, △▲ smaller opening area.

44·5% of the original opening. Although the shapes of the curves differ from the results using the larger opening, the mean gradients remain unaltered. There was good agreement between the analogue and digital computer models—the analogue computed results only are shown here. One effect of reducing the opening area is to reduce the effect of small scale turbulent diffusion throughout the velocity range. The absence of a low gradinet portion at low $\overline{q_{eff}}$ in the curve of Fig. 6 for the adapted model supports this argument.

For experiments using airflow from two different directions—one at 30° and one at 60° to the normal to the plane of the opening—the modification discussed previously was made to the simple model and the digital computer model to account for the change in direction of the impinging airstream. The results are shown in Fig. 7. Again a reasonably good agreement with the original curves was obtained.

4. EXTENSION OF MODEL TO FULL SCALE

Davenport[5] has proposed a standard spectrum of horizontal gustiness and by re-scaling the digitised velocity data it was possible to simulate a turbulent wind whose spectrum approximately matched Davenport's function. This was effected for mean wind speeds

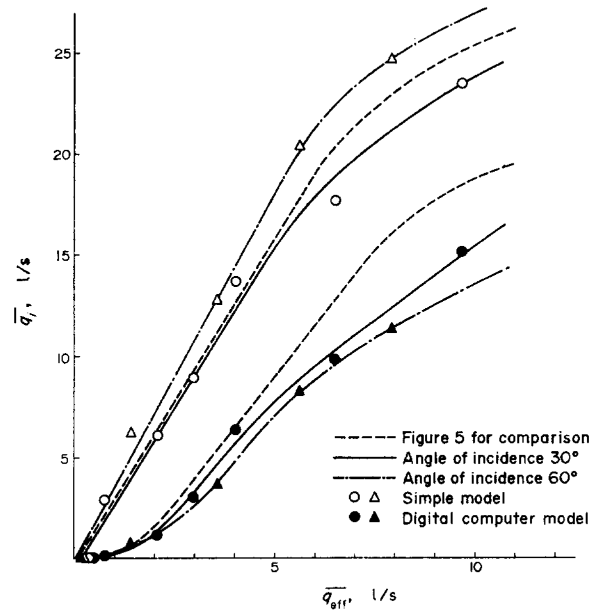

Fig. 7. Computed mean airflow through opening $\overline{q_i}$, plotted against measured mean ventilating airflow $\overline{q_{eff}}$; ○● angle of incidence 30°, △▲ angle of incidence 60°.

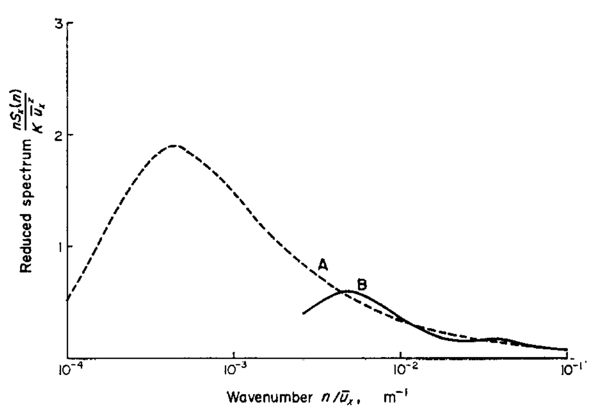

Fig. 8. Reduced power spectral density function, (a) due to Davenport (b) re-scaled velocity data.

of 2·5, 5 and 7·5 m/s, and a roughness coefficient K of 0·015, corresponding to terrain uniformly covered with obstacles 9–15 m in height, e.g. residential suburbs. Figure 8 shows the spectrum of the rescaled data superimposed on Davenport's standard curve, and the total record length in this case represents a wind run of about 3000 m. It is not possible to simulate lower frequencies but it can be shown that their effect on pulsating ventilation is minimal compared to the higher frequencies present in the spectrum. Table 1 shows the mean airflow rates, obtained from the digital computer model,

Table 1. Ventilation of enclosure and 48 m³ room due to turbulence in natural wind. The last column gives an air change rate that includes the effect of a mixing coefficient $f = 0{\cdot}37$

	Opening area (m^2)	Mean wind velocity (m/s)	Air flow rate (l/s)	Air changes (per h)
3·48 m^3 test enclosure	0·0233	5	0·06	0·024
48 m^3 sealed room	0·02	2·5	0·11	0·003
		5	0·85	0·024
		7·5	2·76	0·077
	0·2	2·5	0·11	0·003
		5	0·86	0·024
		7·5	2·92	0·081
Unsealed room	0·02	2·5	4·7	0·13
		5	12·6	0·35
		7·5	20·8	0·58
	0·2	2·5	18·3	0·51
		5	71·7	1·99
		7·5	137	3·8

into the test enclosure in this simulated wind, into a room measuring $4 \times 4 \times 3$ m, completely sealed except for window openings of 0·02 and 0·2 m^2, and into an unsealed room with no through airflow and the same window openings. The effective air change rates are also shown taking a value of mixing coefficient $f = 0{\cdot}37$. It is interesting to note that in the case of a sealed room, there was little increase in ventilation when the opening area was increased ten-fold. At the relatively low frequencies being simulated, the internal pressure follows the external pressure generated at the window opening. The ventilating airflow is therefore the amount of air required to maintain the appropriate mass of air in the room, and is independent of opening area. On the other hand, with the more realistic case of an unsealed room with relatively low resistance to airflow to other parts of the building, much higher flows were obtained. These figures relate to fluctuating pulsations at the window only and if a net flow of air into or out of the window were present, a more complex situation would result.

5. CONCLUSIONS

Clearly the physical processes involved in this type of airflow are much more complex than has been given credit for in this paper, and more theoretical and experimental work would be necessary to investigate the relative magnitude of the various effects discussed here. Nevertheless the experimental work did provide consistent results which suggest that further work would be rewarded with better correlation between theory and experiments. It is interesting to note that in this investigation the more complex computer model did not provide significantly better overall correlations than the simple model, despite the apparent shortcomings and assumptions inherent in the latter. The work described does show that about one-third of the fluctuating air flow into the enclosure was finally mixed with the bulk of air in the enclosure.

The re-scaling of our test data to simulate full scale wind was not perfect, only a portion of the complete spectrum being obtainable. Table 1 can be regarded as showing the orders of magnitude obtained in full scale situations. Therefore, although the present study provides an indication of the magnitudes of ventilating airflows which may be generated by turbulent wind, the extension of the results to real buildings depends on further analysis and experiment to determine more precisely the effects of shape, size, number and spacing of openings, interactions within a building due to interconnecting air flow paths, and other relevant variables.

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