VENTILATION AND COOLING

18TH ANNUAL AIVC CONFERENCE
ATHENS, GREECE, 23-26 SEPTEMBER, 1997

NON-DIMENSIONAL GRAPHS FOR NATURAL VENTILATION DESIGN

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Non-dimensional Graphs for Natural Ventilation Design

Synopsis

There are certain conditions which are of interest when designing for natural ventilation of commercial buildings. These are:-

- summer cooling
- indoor air quality in winter
- night-time cooling.

For the first two conditions it is necessary to determine the distribution of open areas to give the desired distribution of flow rates. Since one is dealing with openings whose position and basic geometry are known, the problem is relatively simple compared to general ventilation problems. When buoyancy acts alone the position of the neutral layer can be specified and the size of the openings can be determined explicitly, as described by other authors. The paper takes this explicit approach further.

First it is shown how, for the summer and winter design conditions, one non-dimensional graph covers the buoyancy alone case with a uniform temperature and then how non-uniform temperatures can be covered by a few extra graphs. The approach is also extended to include the sizing of stacks as distinct from sharp-edged openings.

For the winter design condition where the openings are small it is important to estimate the effects of adventitious openings. It may also be desirable to determine the effects of wind. Suitable procedures and graphs for doing this are described.

Finally the possibility of using similar graphs in the evaluation of night-time cooling is briefly discussed.

List of symbols

a	leakage coefficient [Pa.s/m^6]	
b	leakage coefficient [Pa.s/m^3]	
A	area of opening [m^2]	
A_o	area of outlet opening [m^2]	
Ar	Archimedes number, U_b/U [-]	
C_p	wind pressure coefficient [-]	
C_d	discharge coefficient of opening [-]	
d	diameter of stack [m]	
h	height of outlet, see Fig, A1 [m]	
H	height of occupied space [m]	
L	length of stack [m]	
Q	flow rate through opening [m^3/s]	
Re	stack Reynolds number [-]	
T	temperature [K]	
U	wind speed [m/s]	
U_b	$\equiv \sqrt{\Delta\rho . gH / \rho}$	[m/s]
z	height [m]	

ΔC_p	pressure coefficient difference [-]
Δp	pressure difference across envelope [Pa]
ΔT	temperature difference between interior and exterior [K]
$\Delta\rho$	density difference between interior and exterior [kg/m^3]
ρ	density [kg/m^3]
ν	kinematic viscosity [m^2/s]

Subscripts

i	opening number
E	exterior
I	interior
H	at z = H
0	at z = 0
L	leakage measurement

1. Introduction

Although sophisticated numerical methods such as CFD and envelope flow models are available to the designer (e.g. Refs. 1 and 2), in the early stages of a design it may be helpful to use simple manual calculations. Such calculations are facilitated by non-dimensional graphs, which present the results of numerical predictions in a general and easily used form (Ref. 3). Examples of this approach for natural ventilation design, using results from envelope flow models, form the subject of this paper.

2. Summer cooling

The usual aim of summer design is to provide sufficient fresh air entry to each space to prevent overheating of the occupants. The basic procedure is:- (i) decide an acceptable temperature rise above ambient e.g. $\Delta T = 3$ K, (ii) calculate the flow rates required for each space, Q_i, and (iii) calculate the area of openings required for each space when the wind speed is zero i.e. buoyancy alone. The third part of the procedure is of concern here.

The openings will be windows or air vents and they can be treated as sharp-edged with a square-law flow characteristic

$$Q = C_z A \sqrt{\frac{2\Delta p}{\rho}} \qquad (1)$$

Since the positions of the openings are known (for buoyancy alone it is only the height z_i which is relevant), the determination of the values of A_i for the required Q_i can be found by a relatively simple calculation for open-plan buildings with negligible internal flow resistance. This has been referred to as an explicit calculation (e.g. Ref. 2).

A typical problem is illustrated in Figure A1 in the Appendix. The height of the occupied part of the building is H and the height of the uppermost opening is (H + h). The values of Q_i and z_i are known and it is required to calculate the A_i such that fresh air enters each opening at the required rate, with the exception of the uppermost outlet opening where the air leaves the building (for convenience only a single outlet opening is considered here). This means that the neutral level, i.e. the height, z_n, at which the pressure difference across the envelope is zero, must lie somewhere close to H. By choosing a reasonable value for z_n, z_n = H say, the pressure difference across each opening is known, because the relation between Δp and z is determined by the variation of ΔT with z and the height at which $\Delta p = 0$ is z_n. Using this relation, Δp_i for each opening can be found and substituted with Q_i in Equn. (1) to determine A_i (C_z can be taken as 0.6). The area of the outlet opening is found in the same way, since the flow rate through it is given by the sum of the other flow rates.

The explicit procedure has been described in Ref. 2 for the case of uniform internal temperature. In the following the explicit procedure is expressed in non-dimensional form and is extended to include the case of non-uniform temperature and the use of a stack in place of a square-law outlet opening.

2.1 Buoyancy alone, uniform temperature

With a uniform internal temperature ($\Delta T_H / \Delta T_0 = 1$), the expression for the non-dimensional envelope pressure difference takes a very simple form

$$\frac{\Delta p}{\Delta\rho.gH} = 1 - \frac{z}{H} \qquad (2)$$

Equation (2) is shown in Figure 1 for values of z/H up to 2.0. It is unlikely that the outlet opening will lie beyond z/H = 2.0.

2.2 Buoyancy alone, non-uniform temperature

With tall buildings such as atria, it is probably unrealistic to assume a uniform temperature. For design purposes a simple non-uniform temperature distribution can be assumed, whereby the temperature difference varies linearly between values of ΔT_0 and ΔT_H at z = 0 and z = H respectively (see Appendix).

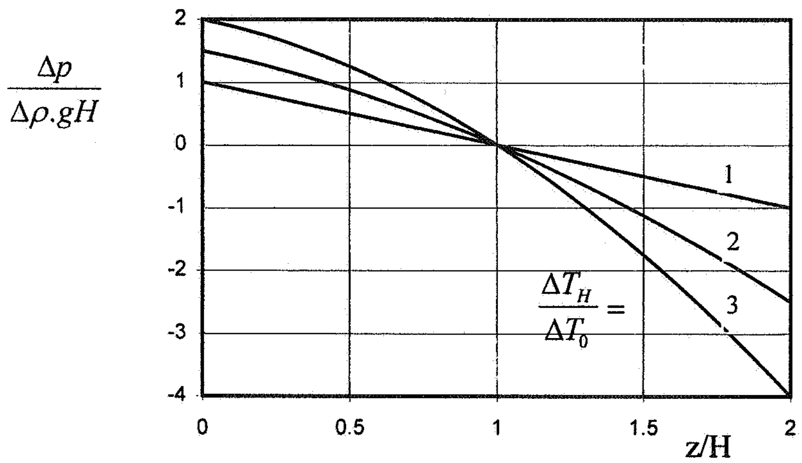

Figure 1 Non-dimensional pressure as a function of z/H for different temperature distributions.

The corresponding expression for the non-dimensional envelope pressure difference is

$$\frac{\Delta p}{\Delta\rho.gH} = 1 - \frac{z}{H} + 0.5\left(\frac{\Delta T_H}{\Delta T_0} - 1\right)\left(1 - \left(\frac{z}{H}\right)^2\right) \qquad (3)$$

Equation (3) is also shown in Figure 1, for values of $\Delta T_H/\Delta T_0$ of 2 and 3.

Figure 1 is particularly helpful for preliminary design of the outlet opening, because it can quickly show the trade-off between open area and height and the effect of non-uniform temperature. For example, raising the outlet height from z/H = 1.25 to 1.50 leads to an increase in Δp and a corresponding decrease of 30 % in the required area. With z/H = 1.25, a non-uniform temperature corresponding to $\Delta T_H / \Delta T_0 = 2$ will allow a similar reduction in area compared to that with a uniform temperature.

2.3 Buoyancy alone, with stack

Rather than use a simple opening for the outlet it may be desirable to use an internal or an external stack, as illustrated in Figure A2 in the Appendix.

It may often be reasonable to assume that the temperature distribution in the stack is the same as that in the building. This means that for the case of a uniform temperature the stack can be treated as a pipe in a uniform density flow. The effect of the stack is then simply that it alters the area required to obtain the given outlet flow by virtue of the change in discharge coefficient. The discharge coefficient for a pipe with a circular cross-section and a bellmouth inlet is given by (see e.g. Section 3.2 of Ref. 4)

$$C_z = \sqrt{\frac{1}{\frac{0.316L}{R_e^{0.25}d}+1}} \quad \text{and} \quad C_z = \sqrt{\frac{1}{\frac{96L}{R_e d}+1.67}} \quad \text{for turbulent and laminar flow}$$

respectively, where C_z and Re are defined by $C_z = \frac{Q}{A}\sqrt{\frac{\rho}{2.\Delta p}}$, $Re = \frac{Qd}{A\nu}$.

For the design condition, the value of Δp is Δρ.gh i.e. the same as that for the square-law outlet at height (H + h), as shown in the Appendix.

The benefit of using a stack under these conditions can be seen in a plot of C_z against Re (Figure 2). The value of C_z for sharp-edged openings such as windows and vents is 0.6, which is significantly less than the values for a stack with turbulent flow, therefore requiring a larger opening area for a given Q. However this disadvantage of sharp-edged openings would be overcome if the opening were fitted with a short bellmouth inlet which would increase C_z to a value close to 1.0.

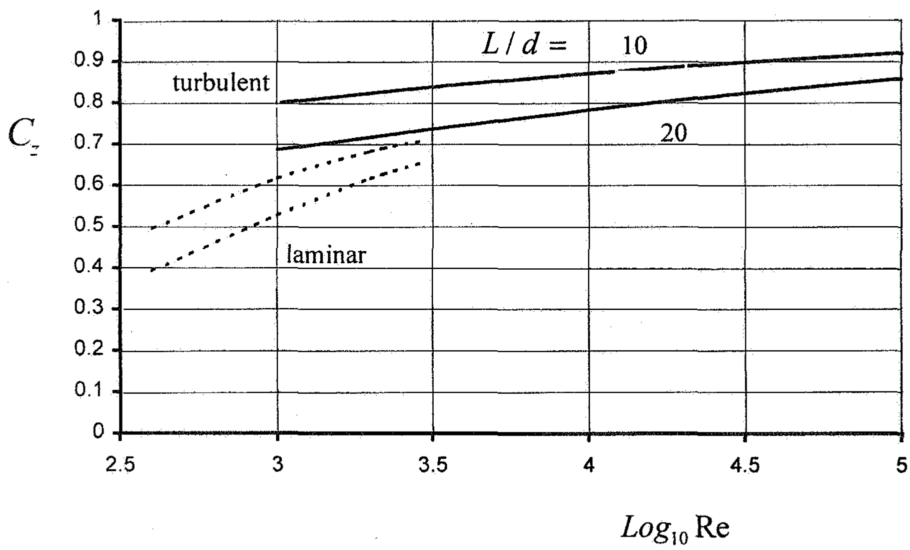

Figure 2 Discharge coefficient of stack as a function of Re and L/d.

3. Winter design

The usual aim of the winter design condition is to ensure adequate indoor air quality without excessive ventilation. This can be achieved by choosing air vents such that a fixed minimum area is always available and the area can be increased beyond this by the occupants as necessary. The design condition remains the same as for summer (i.e. z_n = H) and the procedure for determining the fixed minimum open area for each floor is the same i.e. Figure

1 can be used. The value of ΔT will be larger, but the values of Q_i will be much smaller than for the summer condition so that the vent areas, A_i, will be much smaller.

3.1 Buoyancy alone, effect of adventitious leakage

In view of the above it is quite possible that the adventitious leakage of the building will be significant i.e. it could exceed the leakage associated with the fixed minimum areas. Account should be taken of the adventitious leakage if excessive ventilation heat loss is to be avoided. One approach to the problem is to estimate the adventitious leakage which will give the design outlet flow rate with the same pressure distribution as the air vents. This equivalent leakage can be compared with either the measured value or the range of values likely to be encountered. If it is less than either of these values, there may be no need for a fixed minimum opening. If it is much less, it may be desirable to tighten the building envelope.

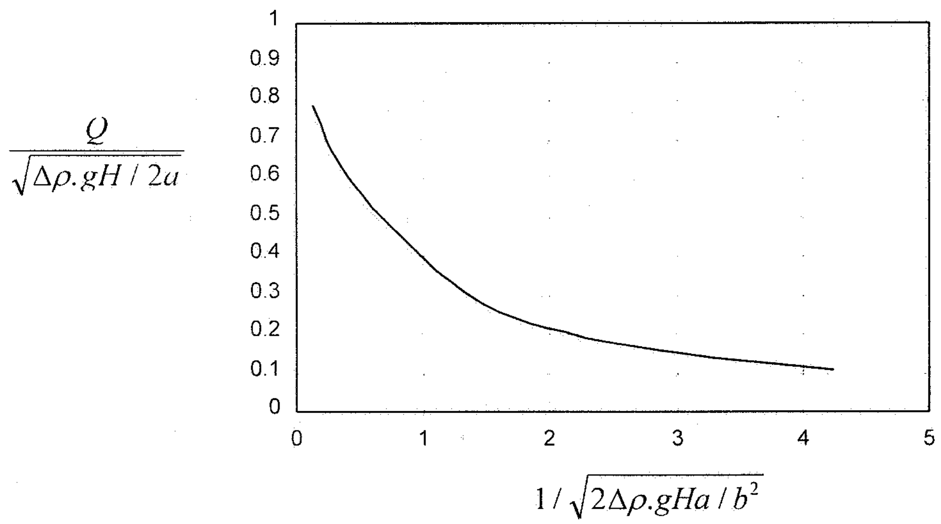

Figure 3 Non-dimensional ventilation rate for the winter design condition with a uniform distribution of adventitious openings.

The estimation of the equivalent adventitious leakage can easily be done with results from an envelope flow calculation method, such as *VENT* (Ref. 3), which assumes a quadratic leakage characteristic

$$\Delta p_L = aQ_L^2 + bQ_L \qquad (4)$$

where a and b are the leakage coefficients.

Figure 3 shows how the non-dimensional ventilation rate varies with the non-dimensional leakage parameter when the neutral level is at z = H. For this example it is assumed that the openings are uniformly distributed on the walls but curves for other distributions can easily be produced.

Knowing a/b^2, H and $\Delta\rho$, the parameter $2\Delta\rho.gH$ is known and Figure 3 then gives the corresponding value for $Q/\sqrt{\Delta\rho.gH/2a}$. The value of Q is given by the design outlet flow rate and this enables the leakage coefficient a to be determined. Knowing a and a/b^2 the leakage at a given Δp_L can be found from Equn. (4). As an illustrative example, for H = 30 [m], ΔT = 20 [K] and $a/b^2 = 0.1$, the parameter $1/\sqrt{2\Delta\rho.gHa/b^2}$ = 0.42 and hence

$Q/\sqrt{\Delta\rho.gH/2a}$ = 0.59. If the design ventilation rate is 10,000 [m3/h], the equivalent leakage is 25,530 [m3/h].

3.2 Effect of wind

If the building is in an exposed position, the designer may wish to investigate the effects of wind. This can be done quite easily using non-dimensional graphs of results for the design opening distribution. Figure 4 shows results from *VENT* for a building with h/H = 0.1 and with the openings on two walls. ΔC_{pl} is the difference between the wind pressure coefficients on the two walls and ΔC_{p2} is the difference between the coefficients on the windward wall and the roof. The effect of the Cp distribution can clearly be seen. It is a simple matter to generate curves for other values of h/H.

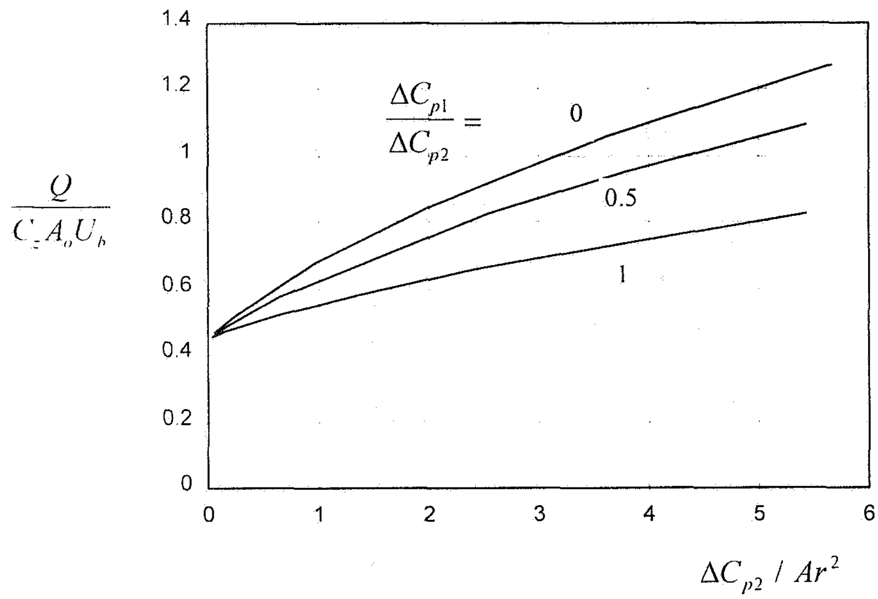

Figure 4 Effect of wind on non-dimensional ventilation rate for h/H =0.1.

4. Night-time cooling

Night-time cooling in summer will rely on large openings so that the calculations are again simplified by (a) the form of the flow characteristic, Equn. 1, and (b) the ability to neglect adventititious openings. For a given building layout and opening distribution it is relatively easy to generate a non-dimensional graph for the variation of Q with ΔT and wind speed, similar to Figure 4. This enables quick estimates to be made of ventilation rates under the conditions likely to be encountered during cooling. These could give a preliminary indication of whether or not the openings are of a sufficient size, prior to a full calculation with a combined thermal/ventilation model.

5. Conclusions

Results from envelope flow models can be expressed in a general non-dimensional form and thereby provide graphs which can be used for quick manual calculations in natural ventilation design. This is particularly true when the design involves large purpose-provided openings, because of the ability to neglect adventitious openings.

References

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APPENDIX.

Equations for buoyancy alone.

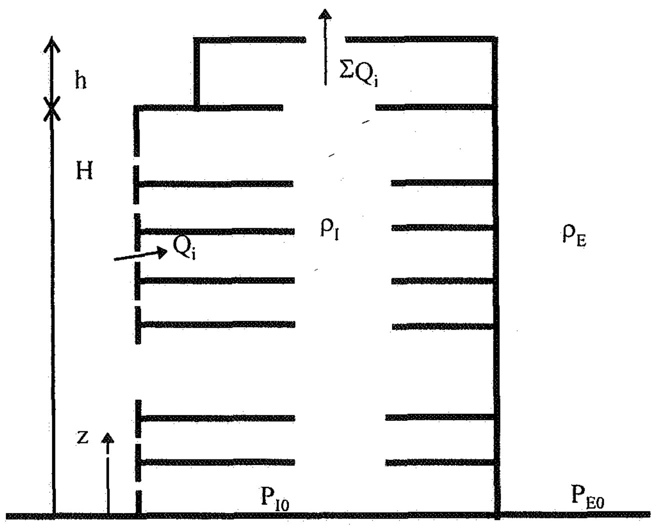

Figure A1

The absolute pressures at points outside and inside the building at height z are respectively

$$P_E\{z\} = P_{E0} - \rho_E gz \quad \text{and} \quad P_I\{z\} = P_{I0} - \int_0^z \rho_I\{z\}g.dz$$

and the pressure difference across the envelope is given by

$$\Delta p\{z\} = P_{E0} - P_{I0} - g[\rho_E z - \int_0^z \rho_I\{z\}.dz] \qquad \text{A.1}$$

The design condition is $\Delta p\{z\} = 0$ at $z = H$, so

$$P_{E0} - P_{I0} = g[\rho_E H - \int_0^H \rho_I\{z\}.dz] \qquad \text{A.2}$$

and substituting in Equn. A.1 gives

$$\Delta p\{z\}/g = \rho_E(H - z) - \int_z^H \rho_I\{z\}.dz$$

$$\Delta p\{z\}/\rho_E gH = (1 - z/H) - \int_{z/H}^1 \rho_I\{z\}.dz/H\rho_E \qquad \text{A.3}$$

For a uniform internal temperature Equn. A.3 becomes

$$\Delta p\{z\}/\Delta\rho.gH = (1 - z/H) \qquad \text{A.4}$$

For a linear variation of internal temperature given by

$$\Delta T\{z\} = \Delta T_0 + (\Delta T_H - \Delta T_0)z/H \qquad \text{A.5}$$

Equn. A.3 becomes

$$\Delta p\{z\}/\Delta\rho.gH = (1 - z/H) - 0.5(\Delta T_H/\Delta T_0 - 1)(1 - (z/H)^2) \qquad \text{A.6}$$

Stack with uniform density

The pressure difference across the stack arises from the motion of the air. The implicit assumption is made that the flow establishes itself in an upward direction. There are two expressions for the absolute pressure at the top of the stack. Referring to Figure A2

$$P_2 = P_{EH} - \rho_E gh \qquad \text{A.7}$$

and

$$P_2 = P_{IH} - \rho_I gh + p_2 \qquad \text{A.8}$$

where p_2 denotes the pressure arising from the motion of the air. For the design condition $\Delta p = 0$ at $z = H$, so

$$P_{IH} = P_{EH}$$

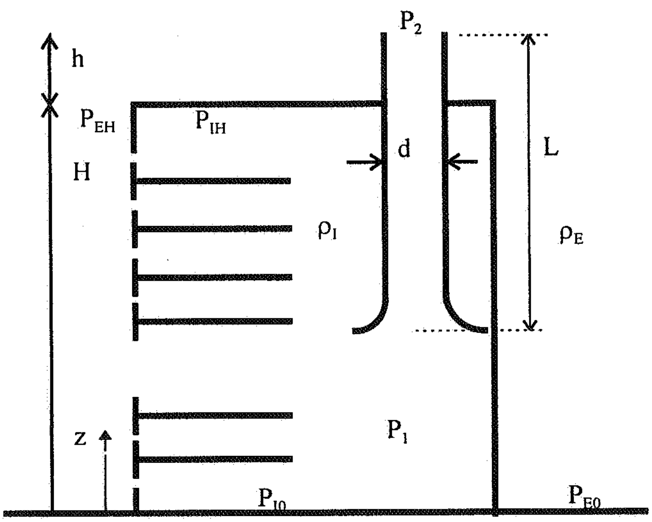

Figure A2

Thus from Equns. A.7 and A.8 it follows that

$$-\rho_E gh = -\rho_I gh + p_2$$

or

$$p_2 = \Delta\rho.gh \qquad \text{A.9}$$

The pressure due to motion in the room away from the inlet to the stack, p_1, is equal to zero, so the pressure difference across the stack is

$$\Delta p = p_1 - p_2 = -\Delta\rho.gh \qquad \text{A.10}$$