Estimation of Flow Rates in Naturally Ventilated Buildings Using Simplified Method

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Abstract: A simplified design method has been derived for use in the estimation of the flow rates in naturally ventilated buildings. The method is based on a one-dimensional "loop analysis" in which the buoyancy forces are balanced by the pressure drops to due friction. Multiple zones, multiple floor buildings are catered for with each zone characterised by its internal gains and its discharge coefficients. Wind effects at the entrance and exits are also taken into account. The procedure yields the zone mass flow rates and temperatures directly by the solution of a simple cubic equation for each loop. Interaction between loops requires a few iterations. The procedure is straight forward and simple enough to be put on a Spreadsheet. This methodology allows the architect to explore various building configurations at little expense and hence to focus on those designs which warrant further detailed analysis using a full building simulation package. In this paper, the fundamental theory behind the loop analysis is described together with the first results at its validation using an emulation of the simple single stack PV cladding arrangement.

Keywords: loop analysis, natural ventilation, multiple zone, single stack PV cladding

1. Introduction

Heating and natural ventilation have been given attention in the architecture of early civilisations. Visual and structural aspects have dominated the mind of architects. Buildings used to be cold but over the last hundred years many of them are too hot. People over the ages have chosen their clothing more to suit the climate than for any other consideration, and have fashioned their buildings to allow the passive control of energy between the inside and the outside. The large increases in internal heat production in commercial buildings are due to the information technology (e.g. computers) revolution in the 1980’s and the tendency to build deep-plan office buildings. As a result there has been considerable use of air conditioning systems to maintain the comfort temperature of such buildings.

Passive solar design is one of the most important strategies for replacing the conventional fuels and reducing the pollution. A wide range of passive techniques for heating and cooling is available for new or refurbished buildings depending on the local climate with little or no extra-cost compare to conventional constructions methods.

Passive ventilation for commercial buildings has not been popular in the past mainly for the following reasons (Awbi and Gan, 1992);

1) the control of ventilation rates is difficult because the air flow is largely influenced by external environmental conditions.
2) because of the lack of control, the energy consumption can be greater than for mechanical ventilation systems and thermal comfort could also be more difficult to achieve.
3) outdoor air is usually only accessible to perimeter zones if less than 6m from or ventilation openings.
4) supply of clean air may be difficult in air-polluted cities.

All these problems could be solved with intelligent and proper designs and advanced control at low cost.
Basically ventilation is a means to replace the contaminated and/or over-heated/cooled indoor air

with quality fresh outdoor air. In practice, ventilation plays a role in the creation of a healthy and comfortable indoor environment involving a number of others aspects, e.g., room geometry, outdoor climate and indoor heat and contaminant sources and sinks. Ventilation also contributes an important part of occupancy comfort and efficient working practice.

The value of ventilation lies on how well basic needs are fulfilled, such as an indoor environment that does not endanger the health and comfortable and promotes productivity of the occupant. General requirements for advanced ventilation can be simplified as to create a healthy indoor air quality and a comfortable indoors thermal condition with as low as possible energy consumption.

The primary reasons for ventilating a space with air are:

a) oxygen is needed for human life process;
b) the air acts as a dilutant; the amount of air required depends on the permissible contaminant level for the room. The contaminant may be CO_2 from respiration, odours secreted through the human skin, cigarette smoke, due to body odours in a crowded room or emission from any other process;
c) ventilation promotes and directs air movement in the space, this is one major environmental comfort factor;
d) control airborne contamination, i.e. industrial ventilation.

In addition the circulation of air throughout a space can remove heat or moisture as required or provide air for combustion (Aynsley, 1977). Fresh air may enter a building by infiltration, i.e. the leakage of air through a building due to imperfections in the structure such as cracks around doors, windows or infill panels. Natural or mechanical ventilation systems supply air, usually with a suitable proportion of fresh and recirculated air through design apertures.
The stack effect and the wind pressure influence the movement of air through a building. It occurs in all buildings but at rates that are determined by the geometry and construction detail of each building.

As an alternative to mechanical ventilation the naturally driven pressure force acting on the envelope of a building can be used to provide a cheap and effective means of ventilating buildings of few zones, e.g. residences. Natural ventilation can be used not only to provide fresh air for the occupants but also for cooling in arid climates.

The drive for low energy consumption in buildings serves to motivate architects, and building services engineers to seek new techniques both for retrofit and new construction. The replacement of mechanical ventilation with natural ventilation is one such important technique made difficult by the dependency on climatic conditions, building use, and lack of simplified design guidelines and rules of thumb.

A procedure is presented below in which the mass flow rate within each zone of a building can be computed without recourse to a complicated and time consuming CFD code. It is based on a "loop analysis" around which the buoyancy force balances the friction. The building is represented as a closed pipe or duct and pressure losses are calculated in a similar fashion to those in a pipe circuit. Pressure losses across ventilation openings and exits are easily calculated using conventional pressure loss fitting factors, Cd, called discharge coefficients herein. But friction factors for real, occupied rooms are more difficult to estimate as the room structure constitutes a poor representation of one-dimensional flow assumed for conventional pipework and ducts. Still, friction can be represented by a Reynolds number dependency with a coefficient, Z, whose value can be established by experiment and used for estimation of the building performance. Wind pressure coefficients, CP1 and CP2, and hence effects, too, can be calculated by conventional methods thus revealing the benefits of wind pressure applied normal to the inlet and tangential to the exit.

2 Energy sources for natural ventilation

There are two energy sources that create air pressure differences used to promote natural ventilation

1. pressure differences due to variation in air density with height, created by differences in air temperature, referred to as the 'stack effect '.
2. pressure differences due to the distribution of wind pressure on building.

The two forces may act independently or in combination. Pressure differences from the stack effect and wind pressure distribution can act simultaneously, the effective pressure being their algebraic sum. However, when wind is present the pressure difference due to wind is generally dominant. Application of the stack effect to ventilation is limited to situations where there is a significant difference between internal and external air temperature and a difference in height between inlet and outlet openings in the building envelope. Favourable conditions for ventilation utilising the stack effect, consistent with thermal comfort of building occupants, occur in multi-storey buildings in regions with cold winter climates. Heating of the building interior for winter comfort creates a significant indoor-outdoor temperature difference, which can be used to provide minimum ventilation rates by the stack effect. Multi-storey buildings increase the height differences between wall openings for ventilation and reduce the area of external wall and roof surfaces exposed to low outdoor temperatures. In cold climates with low external temperatures and heated buildings interior temperature differences can be large, of order of 30^o C. This potential for natural ventilation is sometimes used to provide minimum ventilation by infiltration through small cracks around doors and windows, to prevent carbon dioxide and odour levels reaching uncomfortable levels.

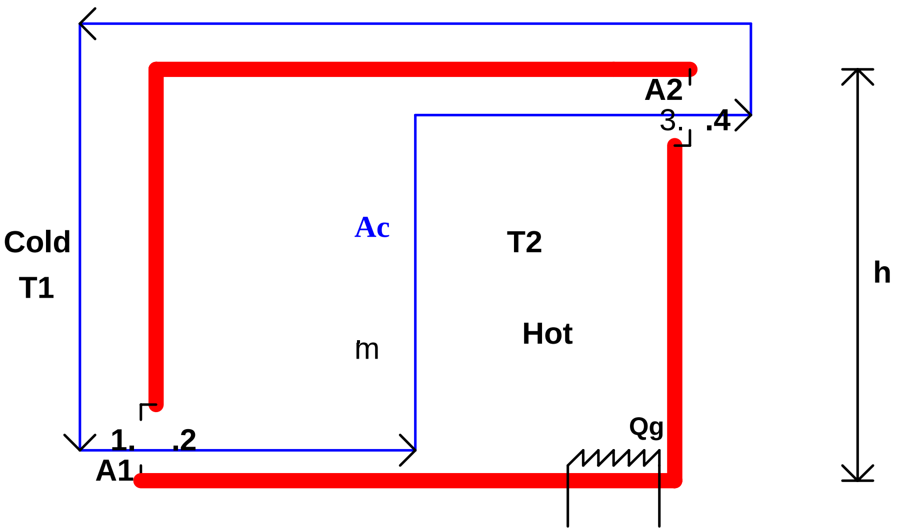

Figure 1 A simple loop

Buildings in warm humid climates often use natural ventilation due to wind to improve the thermal comfort of occupants as well as to supply fresh air for building spaces. The stack effect in buildings in hot climates is usually insignificant as openings are larger and ventilation due to wind ensures temperature differences between internal and external air temperatures are small, of the order of 2^{o}C. Elongated buildings, orientated normal to the prevailing winds, increase the wall area available for openings and decrease the resistance to airflow created by a small opening in internal partitioned walls. Extended eaves and projecting end walls on the windward face improves cross ventilation through rooms near the ends of the buildings.

Buildings in temperate climates often utilise the stack effect in winter and wind pressure in summer. As climatic extremes are less in these regions, building forms are more varied with less emphasis on optimising for summer airflow or winter heat loss.
The stack effect in buildings in hot climates is usually insignificant as openings are larger and

ventilation due to wind ensures temperature differences between internal and external air temperatures are small, of order of 2^o C. Exceptions can occur in spaces where high temperature processes are being performed such as foundries, steel-works or commercial kitchens but the airflow due to the stack effect in such cases is unlikely to offset the high heat stress on workers.

In general, the kinetic energy of the air motion is related to the driving pressure difference by an equation of the form $\Delta P= ½ C_pU^2$. Where C is a coefficient whose value depends on the shape and size of the opening.

If the air flow results from temperature differences, C is called the discharge coefficient, indicated as C_d (ASHRAE, 1989, Karakatanis, 1986).

The authors, e.g. (Awbi, 1995) and (Aynsley, 1977) have tended to use the discharge coefficient C rather than the dynamic loss coefficient C_d in studying natural ventilation but in our formulation we use the dynamic loss coefficient C_d to estimate the air flow rate. These coefficients are related by $C_d=1/C^2$

Airflow rates through openings in series are estimated using the discharge coefficients from Table 4.1.

If the flow is caused by wind blowing into or across an opening, C is called the static pressure coefficient, denoted by C_p. The pressure difference generated by the wind also depends on the direction of the wind relative to the surface of the openin

3. Single Storey, Single Loop Analysis

Consider the single storey building shown in Fig. 1. In steady state, the mass flow rate, **m**, is constant everywhere in the loop flow path denoted by the thin line. Around the loop, the momentum equation states that the sum of the pressure drop around the loop due to friction must equal the buoyancy force developed due to a change in density, hence temperature. The buoyancy force is generated due to the temperature rise as a result of internal gains due to solar gains, heating, lighting, equipment, and occupancy minus any fabric heat loss. In the following, Qg denotes the **net** internal heat gain. This is often easily estimated in the design stage using various building design guidelines or from energy audits on existing buildings.

The building of height h, is characterised by the cross sectional area A_1 at inlet and A_2 at outlet. Inside, the air is heated to temperature T_2 such that it has an average density ρ_2. The external air has a density ρ_o at an average temperature T_1. From the Fig. 1 the pressure balance around the loop reads $\Sigma \Delta P = (P_1-P_2) + (P_2 - P_3) + (P_3-P_4) + (P_4-P_1) = 0$ where (P_1-P_2) and (P_3-P_4) denotes the sum of all the internal friction and flow resistance pressure caused by the stack on the air flowing through the control volume, of cross sectional area Ac. The Bernoulli equations read, $P_2 = P_3 + \rho_2 gh + F$ and $P_1 = P_4 + \rho_o gh$ where F denotes the head loss due to internal friction along the flow path, with $F=f(L/D_h)\rho_o U^2/2$. Hence $(P_2 - P_3) + (P_4-P_1) = gh(\rho_2 - \rho_o) + F$ or rearranged, $(P_1-P_2) + (P_3-P_4) = gh(\rho_o-\rho_2) - F$. The pressure drops (P_1-P_2) and (P_3-P_4) can be expressed in terms of a pipe fitting pressure drop loss coefficient, C_d, $\Delta P=\rho U^2/2$ where **U** = local mean velocity across an opening. Since $\mathbf{U}=\mathbf{m}/A\rho$, and **m** is the loop mass flow rate, which by conservation is constant around the loop, the pressure drop around the loop must equal the buoyancy force, i.e.

$$gh(\rho_o - \rho_2) = \tfrac{1}{2} C_{d1} \frac{\rho_o m^2}{\rho_o{}^2 A_1{}^2} + \tfrac{1}{2} C_{d2} \frac{\rho_2 m^2}{\rho_2{}^2 A_2{}^2} + \tfrac{1}{2} f \frac{L}{D} \frac{\rho_o m^2}{\rho_o^2 A_c^2} \quad (1)$$

The flow is basically incompressible so that in the pressure drop terms $\boldsymbol{\rho_2 \approx \rho_o}$ is assumed and the Boussinesq approximation is made in the buoyancy term so that

$$(\rho_o - \rho_2) = \rho_o \beta (T_2 - T_1) \quad (2)$$

with $\beta=1/T$ where $T=(T_1+T_2)/2$ and T is the absolute temperature in Kelvin.

From the energy equation, see Fig. 1, $\mathbf{m}C_p(T_2-T_1)=Q_g$ so that

$$gh\beta(T_2 - T_1) = \frac{gh\beta Q_g}{mC_p} \quad (3)$$

and therefore, the final result becomes

$$m^3 = \frac{2\rho_o^2 gh\beta Q_g}{C_p\left(\frac{C_{d1}}{A_1^2} + \frac{C_{d2}}{A_2^2} + \frac{f(L/D_h)}{A_c^2}\right)} \quad 4)$$

Equation 4 is the basic fundamental "loop equation" used as the starting point for all subsequent derivations. Note that in the absence of friction, F=0, equation 4 predicts that the mass flow rate is proportional to the net internal gain, $(Qg)^{1/3}$.

4. Formulation for Single Storey with Stack and Wind Effects

Since the pressure created by the wind and the buoyancy forces act at the same time, it is necessary to estimate the resultant air flow. The total pressures at the inlet and exit are re-written $P1'=P1 \pm CP1\rho_o U^2/2$ and $P4'=P4 \pm CP2\rho_o U^2/2$. Following through the derivation leads to,

$$gh\rho_o\beta\frac{Q_g}{mC_p} \pm \tfrac{1}{2}\rho_o CP1W_{in}^2 = \tfrac{1}{2}C_{d1}\frac{\rho_o m^2}{A_1^2\rho_o^2} +$$

$$\tfrac{1}{2}C_{d2}\frac{\rho_o m^2}{A_2^2\rho_o^2} \mp \tfrac{1}{2}\rho_o CP2W_{out}^2 \quad (5)$$

A positive wind pressure coefficient, +CP1, at the inlet and a negative (suction) wind pressure coefficient,-CP2, at the exit will enhance the mass flow rate. The value of the pressure coefficient, C_p at a point on the building surface is determined by:

1. the building geometry;
2. the wind velocity (i.e. speed and direction) relative to the building
3. the location of the opening with respect to elevation.
4. the location of the building relative to other buildings and the topography and roughness of the terrain in the wind direction.

The values of the pressure coefficient, CP are available for rectangular buildings in Mcdonald (1975).

Rearranging yields the fundamental single loop equation with wind effect which can be solved for **m**

$$m^3\left[\frac{C_{d1}}{A_1^2} + \frac{C_{d2}}{A_2^2}\right] - \rho_o^2 m\left[\pm CP2W_{out}^2 \pm CP1W_{in}^2\right]$$

$$-\frac{2\rho_o^2 gh\beta Q_g}{C_p} = 0 \quad (6)$$

Equation 6 is a cubic equation which predicts that for very strong wind pressures, in at the inlet and tangential to the exit, the mass flow rate is linearly proportional to the net internal gain, **m ∝Qg,** and **m ∝W** for constant Qg, see Figs. 2 and 3.

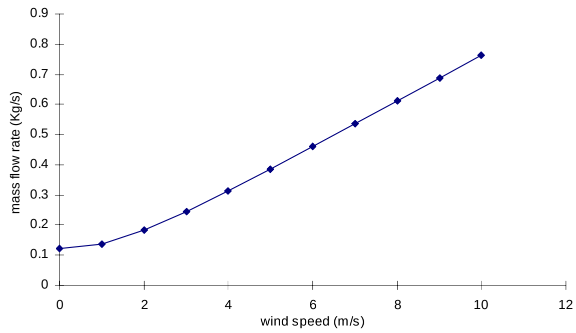

Figure 2: Mass flow rate in the single storey

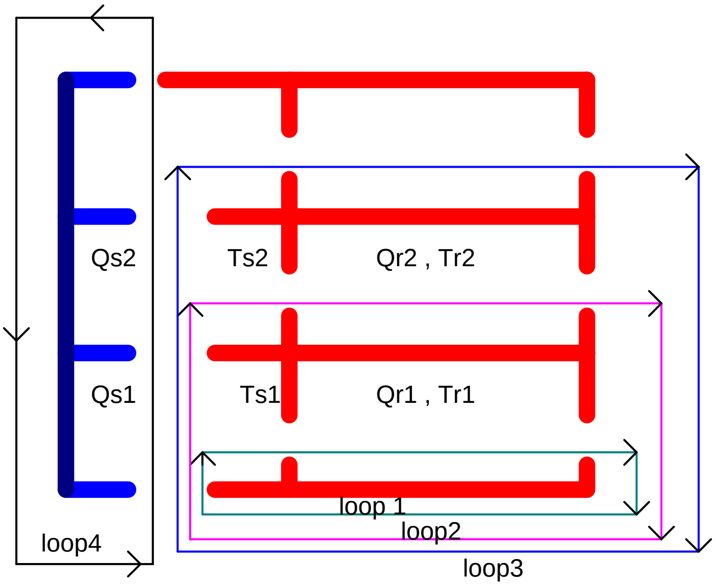

Figure 4: 2D model multiloop solution.

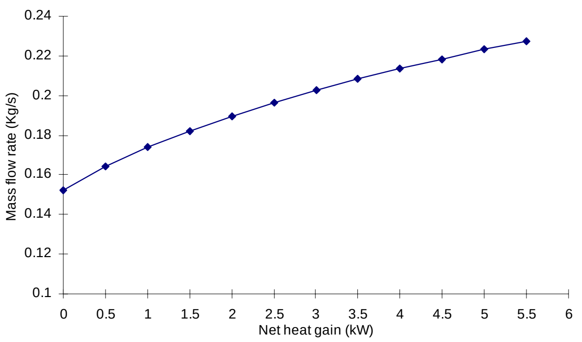

Figure 3: Mass flow rate for single storey building for variation of the wind speed versus net heat gain at 2m/s wind speed

5. Formulation for Multi Storey Buildings

The single loop analysis can be extended to include a multiple storey building with different internal gains for each room zone (denoted with "i") and its adjacent stack zone (denoted with "j") as depicted in Fig. 4. Again, for reasons of brevity only the main features of the derivation are included.

Each loop must be considered individually. But each loop mass flow rate is different and the pressure drops at each inlet and exit depend on the local mass flow rate through that opening. Likewise the internal heat gains in each loop and the stack (Loop4) depend on the flow rates through each zone. It is convenient, therefore, to introduce an "effective" heat gain or area based on the ratio of the local mass flow rate to the total mass flow rate at the entrance to the building, see. Fig. 4. By denoting of Qsi as the equivalent heat gain, $Q'_{si}=Q_{si}(m_{ri}/m)$ and A'_{si} as the effective area, $A'_{si}=A_{si}(m_{ri}/m)$ the generalisation to a multi-loop system including friction can be expressed as;

$$m_i^3 = \frac{2\rho_o^2 gh\beta\left[\left(\sum_{k=1}^{k=j} Q'_{sk}\right)+Q_{ri}\right]}{C_p\left[\left(\sum_{k=1}^{k=j}\frac{C_{dsk}}{A'^2_{sk}}\right)+\left(\sum_{k=1}^{k=j}\frac{f_{ik}\frac{L}{D}}{A^2_{ik}}\right)+\frac{C_{dri,1}}{A^2_{ri,1}}+\frac{C_{dri,2}}{A^2_{ri,2}}\right]} \quad 7)$$

where s_j denotes the stack at floor j, r denotes the room number and i denotes the loop number. From this equation the direct solution for the mass flow rate into the room can be found once the total mass flow rate, **m**, is known.

6. Solution for Multi Storey with Wind Effects and Friction

The governing equation 7 can be extended as for the single loop case to include wind pressure effects at the building inlet and the flow loop exit, see Fig. 4. As the notation becomes somewhat unwieldy we illustrate the application to Loop 1 which becomes,

$$m_1^3\left[\frac{C_{ds1}}{A_{s1}^{'2}}+\frac{C_{dr1,1}}{A_{r1,1}^2}+\frac{C_{dr1,2}}{A_{r1,2}^2}+\sum_{k=1}^{k=j}\frac{f_{1k}L/D}{A_{1k}^2}\right]-$$
$$\rho_o^2 m_1\left[\pm CP_{out}W_{out}^2 \pm CP_{in}W_{in}^2\right]-$$
$$\frac{2h\rho_o^2 g\beta\left[Q_{s1}^{'}+Q_{r1}\right]}{C_p}=0 \qquad (8)$$

Equation 8 is of the form $Am_1^3 + Bm_1 - C = 0$, i.e. a cubic equation. These equations are then solved for each loop in turn using the Newton Raphson technique. The corrected mass flow rates into the rooms are then found by iteration of equation 8, since the effective area and equivalent heat gain depend on the ratio of the room mass flow rate for loop *i* to the total stack mass flow rate, **m.**

The model for a multi storey building was run which assumed that the net heat gain in a room of 15m^2 floor area is 1.5 kW, with entrance and exit areas of 2.5m^2 and 0.2 m^2 respectively. The opening to every floor through a stairwell is taken as 3.75 m^2. The direction of the wind speed on the building is taken as normal to the entrance (positive pressure effect) and tangential to the exit (beneficial suction pressure effect). The results of the model can be seen in Fig. 5 depending on the wind speed. The mass flow rate for the first floor is slightly higher than floors above it because the path through the first floor to the room has less resistance in its loop and therefore more air flows through this path.

7. Friction

7.1 Existing correlation

Within each building zone the through flow velocity can be expected to be low and therefore the pressure drop can be dominated by the zone entrance and exit vent resistances, the Cd's. In sharp contrast, the flow through a PV ventilated stack with its narrow gap and tall structure can be expected to be dominated by the internal flow resistance, F, and the nature of the flow structure. In both areas the real flow structure will be 3 dimensional, and characterised by entrance lengths and transition to turbulence. We can retain the simplicity of the 1 dimensional duct flow analogy in three ways;

Firstly, one can use a textbook analogy whereby $F= f(L/D)\rho U^2/2$ with the friction factor, f, determined by the simplest fully developed laminar flow Reynolds number relationship, e.g. f=64/Re (Cengel,1997) or extended to a general turbulent flow correlation of the form $Re=CRe^{-m}$. Actually the flow becomes turbulent at Re>400 and is highly complex. The friction factor in turbulent flow depends on the roughness of the surface. The roughness is causing a greater loss of energy than laminar flow. For smooth tubes, McAdams (1954) states that the friction factor in fully developed turbulent flow can be determined from $f = 0.18Re^{-0.2}$ and for the rough surface the simple, explicit equation (Kreider and Rabl,1994) is

$$f=\frac{1.325}{\left\{\ln\left[\frac{\delta}{3.7D_h}+\frac{5.74}{\mathrm{Re}^{0.9}}\right]\right\}^2} \qquad (9)$$

Equation (2.29) is as accurate as the more commonly used, implicit Colebrook equation (Kreider, 1994) for *f* given by

$$\frac{1}{\sqrt{f}}=-0.87\ln\left(\frac{\varepsilon}{3.7D_h}+\frac{2.52}{Re\sqrt{f}}\right) \qquad (10)$$

In the hydraulically rough regime the Von Karman rough pipe formula is

$$\frac{1}{\sqrt{f}}=4\log_{10}\left(3.7\frac{D}{\varepsilon}\right) \qquad (11)$$

Batti and Shah (1987) present several friction factor correlations and a number of explicit equation for calculating *f* for fully developed turbulent flow in smooth and rough surface.

The friction factor for flow in tubes with smooth as well as rough surfaces over a wide range of Reynolds number is given in Moody diagram. This diagram is based on the work of Colebrook and White and Moody (Incropera, 1990).

The Moody diagram covers four flow regimes: laminar flow, critical zone flow, transitional flow, and complete turbulent flow. If Re<2300, the flow is laminar, and the friction factor is inversely proportional to Reynolds number. The critical zone is that regime where 2000<Re<4500. Here the flows will either be laminar, or it can be transitional, where the effect of the roughness produces a greater loss of energy than in laminar flow for the same Reynolds number. In the transitional zone, the friction factor *f* is dependent upon both the Reynolds

The Reynolds number in turn can be re-written as

$Re=UD/\nu = \{\dot{\mathbf{m}}/\rho A_c)\}D_h/\nu$ so that for the simplest analogy

$$F = \left[\frac{64\rho A_c \upsilon}{\mathbf{m}D_h}\right]\left(\frac{L}{D_h}\right)\rho\frac{U^2}{2} \qquad (12)$$

This, when inserted in the simple loop equation 4, leads to a quadratic equation for **m** The dependency of **m** on the net internal gain is therefore $\mathbf{m}\propto(Qg)^{0.5}$ for negligible duct entrance/exit effects.

7.2 Hollands correlations

For the PV ventilated stack the entrance lengths, transition to turbulence, turbulence, and aspect ratio must be included. Hollands (1981) has determined a number of correlations which characterise air collectors operating under a fixed 60 Pa, pressure drop and forced convection. These take the form whose coefficients are summarised in Table 1, $f=f_o + f_1/(L/D_h) + f_2/Re^m$

	f_o	f_1	f_2	m
Laminar	0	0.9	24	1
Transition	0.0094	$2.92Re^m$	0	-0.15
Turbulent	0	0.73	0.059	0.2

Table 1. Summary of the Hollands (1981) friction coefficients.

These correlations can be implemented provided the appropriate characteristic hydraulic diameter, D_h, is used, 4*perimeter/area.

7.3 Emprical Z*f

Since a real building or a real PV ventilated stack does not resemble perfectly the ideal one dimensional duct flow and within the duct there are entrance effects, laminar flow, transition to turbulence and turbulent flow we can introduce an empirical constant, Z, to lump all these effects together so that $f'=Z*f_{\text{theoretical}}$. The factor Z becomes an empirical constant for each individual case to be determined by experiment and parameter identification. Z can be expected to depend on the net heat gain, Q_g, hence the flow regime, hence Reynolds number, the duct aspect ratio, L/D_h, the entrance lengths and inlet/exit vent design, hence flow resistances, and on the wind effects. Further study is warranted to establish a simple form of this dependency. In the following results, seen in Fig. 6, a simple relationship $\mathbf{Z\approx 0.8L/D_h}$ was found suitable. Here, the experimental results of Sandberg and Moshfegh (1995) are shown for a simple, well controlled experiment of a simulated ventilated PV stack of height 6.5m and aspect ratio about 25 without wind effects and for a uniform heat flux of up to 300 W/m^2 (i.e. the net internal gain). Both the laminar flow textbook constant 64 and the Hollands constants lead to theoretical mass flow rates of 2.1-2.4 the measured values. This is expected due to the entrance lengths and flow regimes encountered for which the only recourse would be to model the flow using a CFD package.
But, by applying the empirical factor Z=14.5 to the simplest laminar flow analogy, Z*64/Re, or Z=11 as a multiplier of the Holland's formulation good agreement between the empirical and the experimental results are obtained. There is no need for a CFD code. As seen, Fig. 6, there is about a 25% error in the predicted mass flow rate at the lowest heat gain, $Q_g=50$ W/m^2 but much better as Qg increases. Also, the simplest Re analogy and the Hollands's coefficients lead to the same results throughout. Just a simple Z factor correlation could be proposed, $Z=z_0+z_1*Q_g$. Such correlations are left to future work.

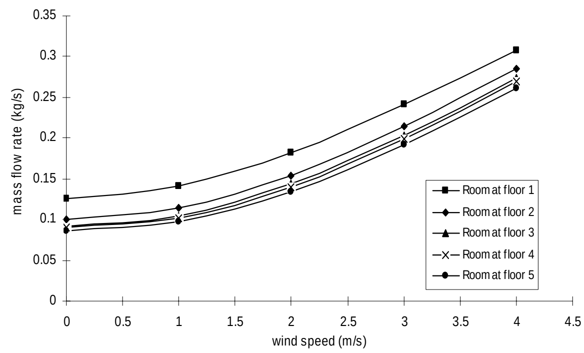

Figure 5: Mass flow rate of the room in each heat gain

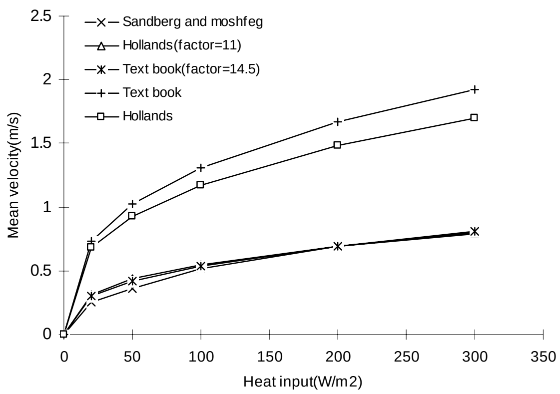

Figure 6: Mean velocity in air gap versus thefloor against wind speed

8. Conclusions

A direct solution for the mass flow rate in a multi storey naturally building with wind pressure effects was developed. The theory predicts the zone mass flow rates (and hence temperatures) given the net internal gain to each zone. It was shown that the ventilation mass flow rates depend on the gain to the power 1 for wind dominated conditions, 0.5 for internal friction dominated conditions, and 0.33 for entrance/exit duct domination over internal friction.
The loop solution can be implemented easily as a Spreadsheet programme and used to assess viable designs for new or retrofit applications without recourse to detailed CFD codes. The methodology could also be implemented in a building simulation code as a beginning-of-hour calculation to establish the ventilation heat loss/gain.

The stack effect and wind effect can be used to reduce temperatures of building integrated PV panels to increase their efficiency. Comparison with existing experimental work shows good agreement for the predicted mass flow rates using existing textbook and forced convection internal friction calculations provided that an empirical constant, Z, can be established which makes the analogy to a simple one-dimensional flow in a duct a viable model possible.

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