QUANTIFYING THE ACOUSTIC CHARACTERISTICS OF OPEN WINDOWS AND VENTS

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1 ABSTRACT

Natural ventilation is often a preferred solution in low-carbon building strategies. This type of ventilation can compromise the acoustic attenuation of the building envelope, meaning that it is not always possible to implement.

Literature on the actual acoustic performance of vented facades is limited, where published data is often presented in different and non-comparable parameters. This shortfall means that when environmental noise is at or above desirable noise levels for natural ventilation, engineers move towards sealing the building facade, often resulting in increased energy usage. The current guidance does not provide design information that could be used to enhance vented façades to accommodate higher external noise levels.

This paper therefore looks at current literature and ISO standards, evaluating the key characteristics of open windows. From here, early modelling results using FEM and FDTD provide a starting point in the characterization of the acoustic performance of openings within façades, highlighting the influence of sound field conditions and the angle of incidence to the window/vents. This paper concludes with a basic model able to approximate the actual window performance, whilst providing a reference to a test methodology and the characterisation of windows.

2 INTRODUCTION

Engineers have developed modelling tools, enabling temperature and ventilation rates to be modelled to astonishing levels of accuracy. Internal noise level targets are well specified for different rooms and buildings. To meet these precise targets, accurate characterisation of windows and vents is needed. The importance of accurately assessing the acoustic performance of vented facades is illustrated below.

Sound Reduction dBA	Internal Noise Levels		
	35 dB	40 dB	45 dB
10 dB	5%	22%	46%
15 dB	22%	46%	65%
20 dB	46%	65%	85%
25 dB	65%	85%	-

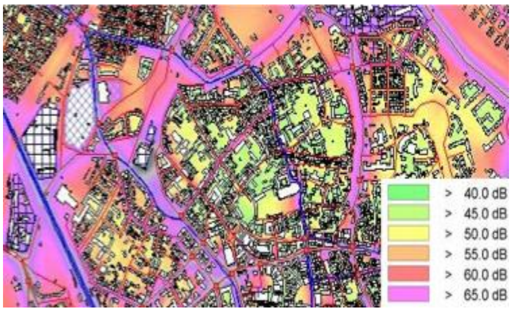

Table 1: Potential number of buildings to be naturally ventilated (%) as a function of the sound insulation performance of an open window

Table 1 above shows the percentage of sites held in the noise map above that achieve an internal noise level of 35 to 45 dBA, with a façade achieving 10 to 25 dBA of sound reduction. The table shows that a 5 dB error in under estimating the performance of a façade, can result in a 20% reduction in the chances of using natural ventilation. This difference is illustrated in Figure 1, where the image to the left shows the area available for natural ventilation based upon a vented façade achieving a sound reduction of 10 dBA and the image to the right assumes a sound reduction of 15 dBA. Alternatively, it can also be said that if the façade is enhanced by 5 to 10 dB, there will be a 20 to 40% increased chance of using natural ventilation.

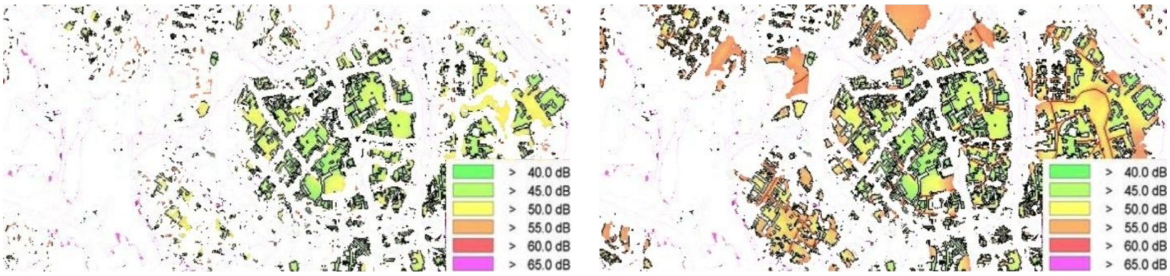

Figure 1: Difference in sites available for natural ventilation, assuming 10 dBA sound reduction for the façade (left) and 15 dBA (right)

The state of the art in predicting the performance of an open window comes from Planning Policy Guidance 24 1994[1] which states that a partially open window provides between 10 to 15 dBA of sound reduction. PPG24 makes reference to Transportation Noise[2], which in turn states that a ‘single window open’ offers 5 to 15 dB of sound insulation, referencing Laurence and Burgess[3]. Note this reference is now 32 years old.

More current guidance, BB93 ‘Acoustics Design for Schools’[4], states ‘For partially open single-glazed windows or double-glazed windows with opposite opening panes, the laboratory measured airborne sound insulation is approximately 10-15 dB R_w. This increases to 20-25 dB R_w in the open position for a secondary glazing system with partially open ventilation openings, with the openings staggered on plan or elevation, and with absorbent lining of the window reveals’. BB93 does not provide a reference to the source of this data and this statement is challenging to interpret.

It is important to note at this stage that the three sets of data above are all different, not only in number but in acoustic terminology: 5 to 15 dB becomes 10 to 15 dBA and then 10-15 dB R_w. In addition, these commonly used references relate to general scenarios, which do not consider; opening area, source direction, field conditions and so on.

The aim of this paper is therefore to identify the key characteristics of open windows and to quantify their acoustic performance in the form of a basic mathematical reference model.

3 VENTILATION AND ACOUSTICS

The proposed building ventilation strategy can have a significant impact upon the required open area in a façade, which will ultimately affect the sound insulation of a facade. To illustrate this, vent open areas based upon four different ventilation schemes have been calculated as per the method given in Appendix A. Table 2 therefore provides the open vents sizes such to achieve a ventilation rate of 8/l/p for an occupied classroom under buoyancy-driven conditions, i.e. in the absence of wind. As shown, the opening vent areas vary from 3.9 m^2 for a single-sided single-vent to a 2.6 m^2 if two openings are used with a height difference of 1 m. When using a ventilation scheme based on a multi-storey stack, the vent size reduces to 0.3 m^2. It is clear that going from an open area of 3.9m^2 to 0.3 m^2 will have a significant impact upon the acoustics performance of the façade. However the current state of the art does not provide an agreed method for scaling the change of acoustics performance based upon the area of an open vent.

Ventilation type	**Inlet [m^2]**	**Outlet [m^2]**
Single sided -one opening	3.9	-
Single sided - top and bottom vents	1.3 and 1.3	-
Cross ventilation	1.3	1.3
Single stack across 3 floors:1st / 2nd / 3rd	0.3 / 0.4 / 0.8	2.4

Table 2: Open areas required to achieve sufficient ventilation for a classroom of 31 people, to achieve a ventilation rate of 8l/s/p based on a windless scenario

4 QUANTIFYING THE ACOUSTIC CHARACTERISTICS OF AN OPEN WINDOW

One possible reason for the shortfall in information relating to the acoustics performance of windows/vents is the lack of a test method which assesses windows/vents accurately, presenting information which can be used by engineers. As an example, each of the papers referenced below assesses the performance of windows to different parameters: Carl Hopkins, $D_{ne,w}$; PPG24, dBA; Transportation Noise Reference Book, dB; BB93,R_w; Napier, $D_{2m,nT}$, $D_{0.01m,nT}$, R' and other terms. These terms are not interchangeable since they are measured and assessed in different ways.

ISO standards are intended to prevent this, the reason for using different test methodologies, is that current ISO standards do not describe how to quantify the acoustic properties of open windows and vents. If a dedicated ISO standard where to be provided, this would encourage the market to test their products, enabling the acoustic understanding of vented façades to grow quickly.

The two relevant standards when measuring windows and vents are ISO 140-10[5] and ISO 140-5[6]. Based upon MACH Acoustics experience, the shortfalls of these standards when measuring windows is presented below.

ISO 140-10 is the reference standard of measuring airborne sound insulation of small building elements under laboratory conditions, i.e. in diffuse field. This standard requires test elements to be less than $1m^2$, the issue being that the elements within the vented façade are typically larger than this. ISO 140-10 states that the methodology is not suitable when measuring the performance of windows and doors. In addition, under diffused conditions, it is not possible to measure acoustic effects that occur in the field such as directivity, local screening, a sound source being a point source or line source and other free field acoustic effects, etc.

ISO 140-5 defines two measurement procedures, the element method and the global method, indicating that they are suitable to assess the in-field airborne sound insulation of facade elements and whole facades. ISO 140-5 states that the element method can be used to estimate the sound reduction index of a facade element, e.g. a window, stating that the most accurate method of assessing an element, is to place a loudspeaker as an artificial sound source at 45° to the element (in both the vertical and horizontal planes combined). However, measuring the acoustic performance at multiple angles to the windows would provide a much better characterisation of the directivity of windows. ISO 140-5 also claims that element loudspeaker measurements under certain circumstances are comparable to those from laboratory measurements, yet these circumstances are not specified. The global method on the other hand aims to estimate the outdoor/indoor sound level difference, ideally using traffic noise as the source, however the use of a loudspeaker is stated to be permissible. In this case, external sound pressure is measured at 2m in front of the facade. Results are expressed in terms of the standardized level difference $D_{2m,nT}$ or normalised level difference $D_{2m,n}$.

Either of these methods does not express the acoustic performance of openable windows as D_{ne}, which could be considered the reference standard for ventilation elements. Element methods assess the performance in terms of the sound reduction index, the global method results in D_n for the whole facade. No indications are given in the standard as to how to convert field parameters to D_{ne} values. Moreover, for either method in ISO 140-10 and ISO 140-5, a diffuse field is assumed in the receiving room, which rarely occurs in field measurements. Hence a suitable reference standard to measure natural ventilation elements is required.

5 ESTABLISHING THE ACOUSTIC CHARACTERISTICS OF AN OPEN WINDOW

At this stage, only very early modelling results are presented. The aim of this modelling is to illustrate the importance of selecting the correct source field conditions, as well as illustrating the geometric effects of a window on its directional properties.

5.1 Finite Element Modelling

2D Finite Element Methods (FEM) has been used to represent the sound transmission through a slit of 0.9m wide by 0.3m deep. Both source and receiving rooms have been modelled as fully reflective and fully absorbent (apart from the separating boundary where the opening is placed). The results for three different scenarios are shown in Figure 2, Figure 3 and Figure 4. In all cases, a point source has been used as a source.

When both the source and receiving room are reverberant (Figure 2), the sound field is fully diffuse only at higher frequencies. In this case, the sound pressure in both source and receiving rooms is seen as constant. When the source room is anechoic (Figure 3), the sound field in the receiving room at low frequencies shows the modes of the room. However for higher frequencies, the sound field is not purely diffusive, as seen, where some beaming can be observed. This proves that the angle of incidence has an effect on the sound field in the receiving room only, when the source room is not diffuse. If both rooms are fully absorptive apart from the separating wall where the opening is placed, the sound field in the receiving room is dominated by diffraction at low frequencies. As shown, sound radiates hemispherically from the opening as a virtual point source and the sound pressure decreases isotropically. As the frequency increases, the spreading of transmitted sound pressure narrows, beaming towards the direction of the sound source. The sound is scattered in other directions, as a result of diffraction through the opening.

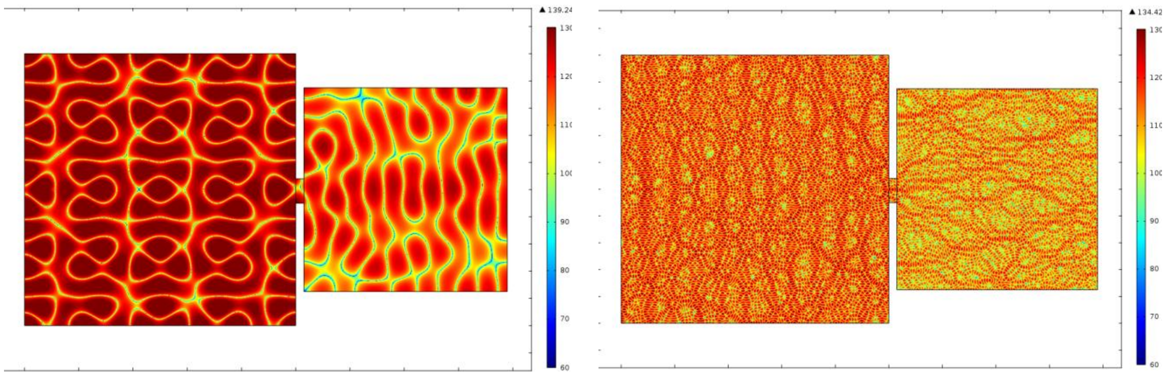

Figure 2: SPL (dB) for reverberant-reverberant case at 250 Hz (left) and 2 kHz (right)

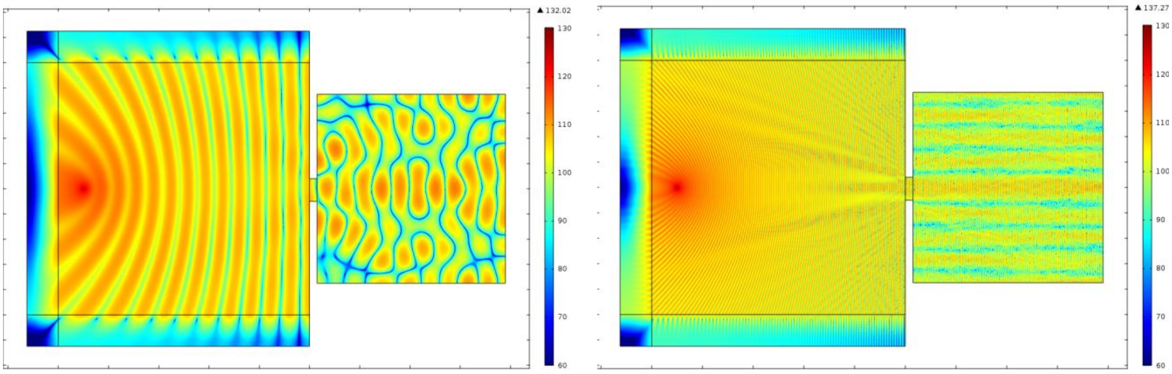

Figure 3: SPL (dB) for anechoic-reverberant case at 250 Hz (left) and 2 kHz (right)

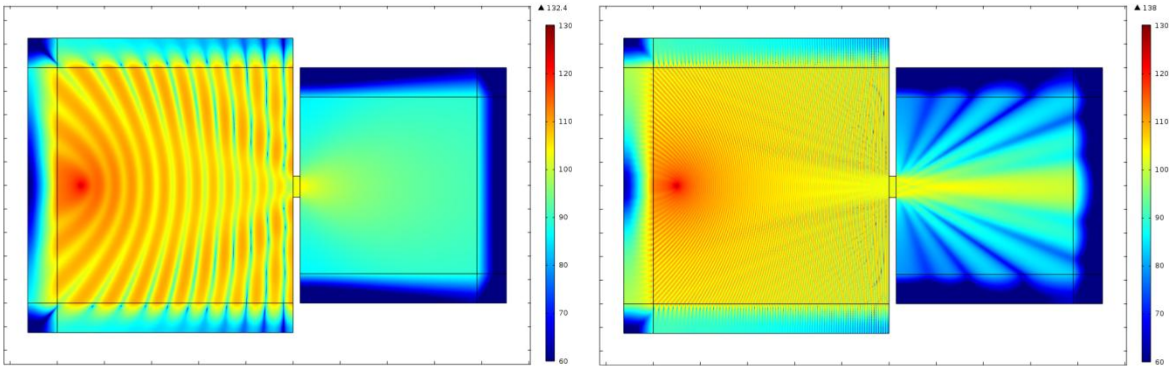

Figure 4: SPL (dB) for anechoic-anechoic case at 250 Hz (left) and 2 kHz (right)

5.2 Finite-Difference Time Domain

Finite-Difference Time Domain(FDTD) has been used to model the acoustic wave propagation in time domain. FDTD is particularly suited to visualise the passage of sound through an opening.

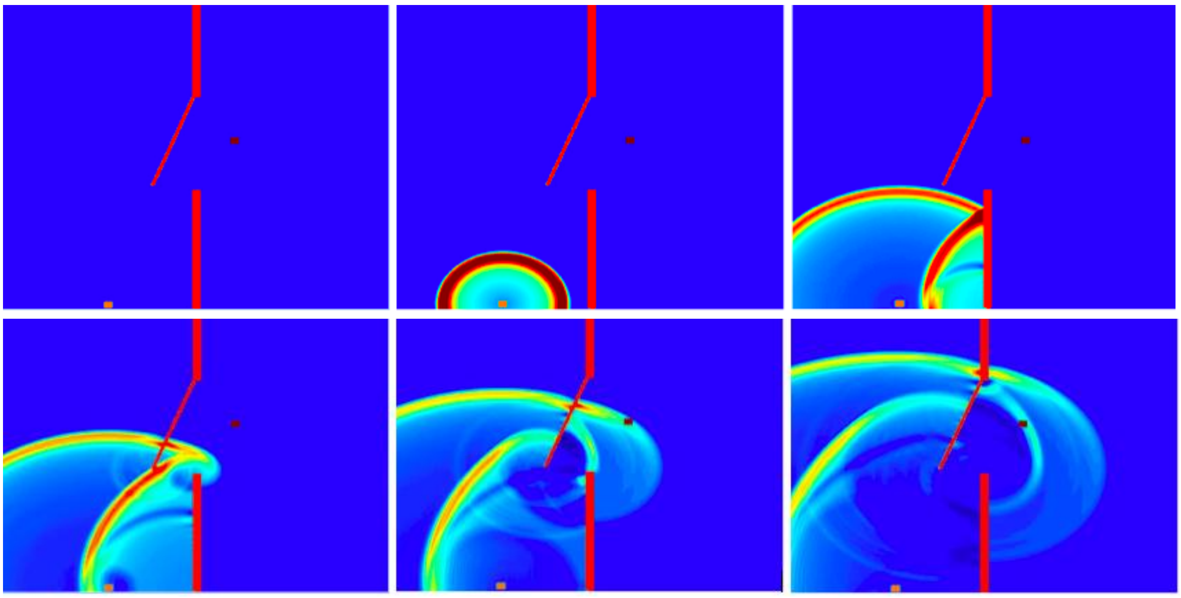

Figure 5: FDTD simulation with sound source in line with the receiving room

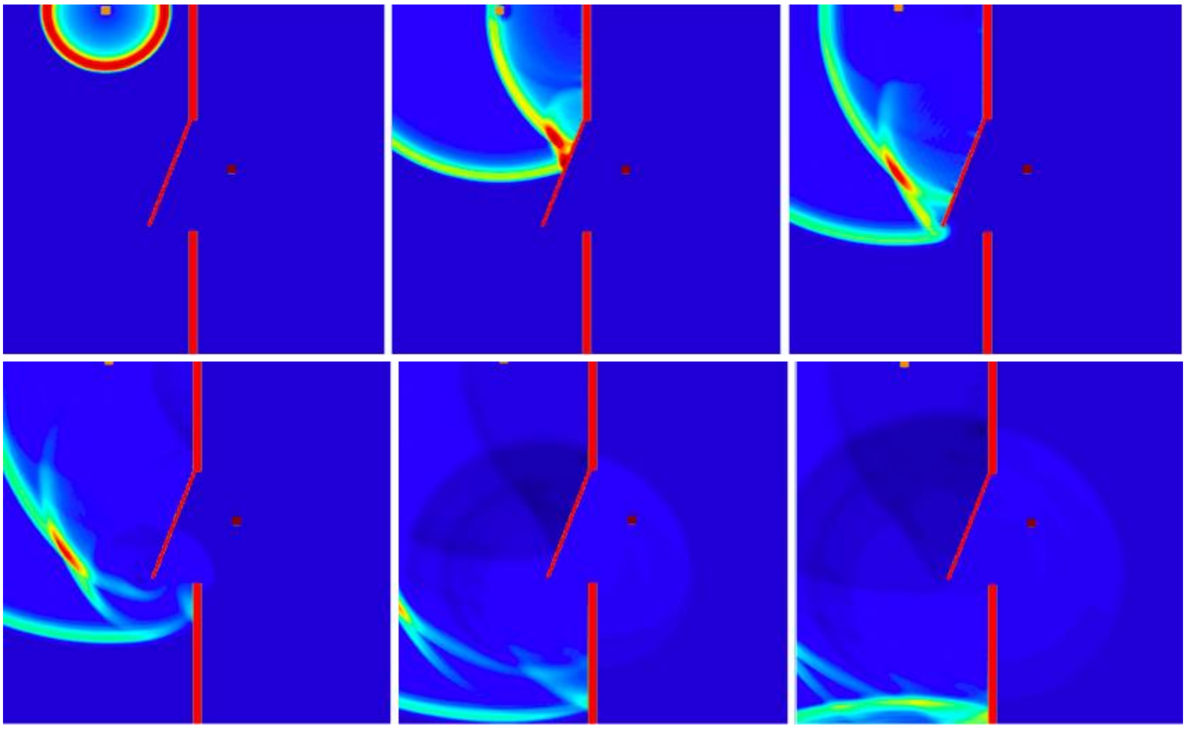

Figure 6: FDTD simulation with sound source out of line of sight with the receiving room

In this case, FDTD has been used to assess how the orientation of a source to a side-hung window affects the levels of sound reduction through the window. An impulse from a point source is used as the excitation signal and two source positions are compared: a source in line with the receiving room at the bottom and a source out of line of sight of the receiving room at the top. From Figure 5 and Figure 6, it can be seen that the acoustic wavefront in the receiving room is more dominant when the source has line of sight into the room.

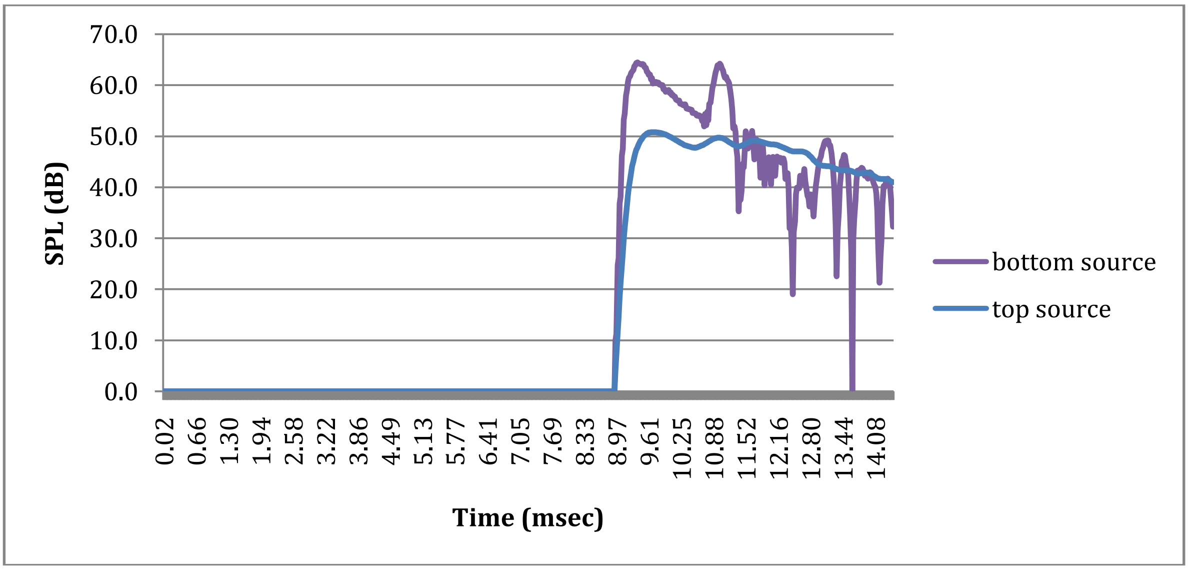

Figure 7: SPL of the direct sound in the receiving room with source positions in and out of line of sight

The SPL at the point in the receiving room shown in Figure 5 and Figure 6 is compared for both source positions in Figure 7. This includes the direct sound and the diffraction and reflections off the window and the separating partition, but not any reflections from other boundaries. Note that for the bottom source, the sound pressure level has been shifted in time due to the difference in travel time between the source and the opening. The average SPL_{RMS} of the graphs shown in Figure 7 has also been calculated, giving an SPL_{RMS} of 52 dB for the bottom source case, compared to 43 dB for the top source case. This shows a significant reduction of direct sound depending on the source orientation.

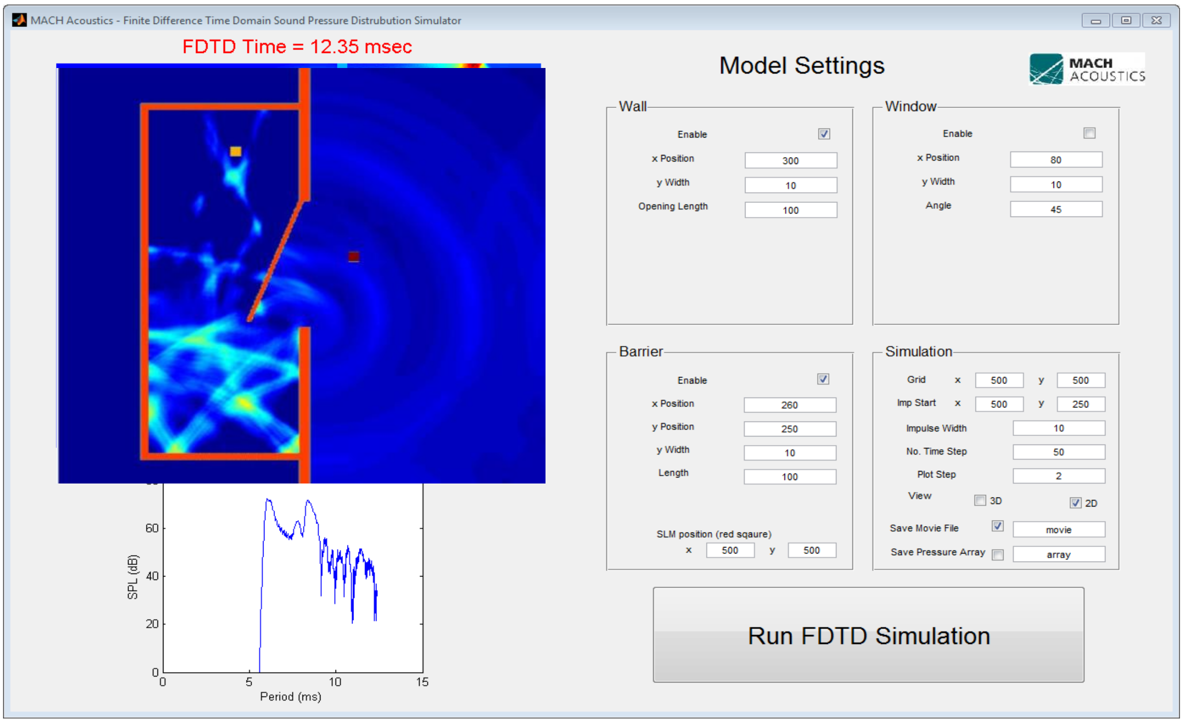

Figure 8: MACH Acoustics in-house FDTD modelling software interface

Finite Element modelling and Finite Time Difference modelling are two very powerful tools, however their complexity and cost is considerable. This type of modelling is also exceptionally time consuming and requires care in ensuring that the output is correct. At this point in time, MACH Acoustics are using the above in house tool to assess the performance of different windows and source conditions. However at this stage, a simple method of establishing the performance of a window is seen to be a fundamental part of our work.

6 SIMPLE MODEL TO ESTABLISH THE PERFORMANCE CHARACTERISTICS OF AN OPEN WINDOW

6.1 Influence of Acoustic Field

It is widely recognised that as the area of an element increases, the sound insulation through the element will decrease logarithmically. The equation below is therefore regularly used to scale the sound insulation between different sized units.

$$D(A_2) = D(A_1) - 10\,Log\,(A_2/A_1) \quad (1)$$

where $D(A_1)$ and $D(A_2)$ is the sound level difference for opening areas A_1 and A_2 respectively. Note that the same equation can be applied to D, D_{ne}, D_{nT} etc.

Nunes[7] has provided a comparison between the acoustic performance of a window in diffuse and free field conditions. A Velfac 200 window was measured for different opening distances by Hopkins[8] under laboratory conditions, and by MACH Acoustics in field conditions. The rearranged results are shown in Figure 9, which indicates that the acoustic performance of the window measured 'in-situ', is at least 5dB above those in diffuse field conditions. In addition, the decay of sound insulation as a function of opening area is also reduced. The equations that govern the decay of sound are:

$$D_{ne\ lab}\ (A) = -\,9.3\ Log\ (A)\ +\ C_1 \ldots\ldots \quad (2)$$

$$D_{ne\ in-situ}(A) = -\,7 Log\ (A)\ +\ C_2 \ldots\ldots \quad (3)$$

where C_1 and C_2 are constants.

Equations (2) and (3) can also be presented of the form of equation (1). Thus, the sound insulation for an opening A2 is:

$$D_{ne\ lab}\ (A_2) = D_{ne\ lab}\,(A_1)\ -\ 9.3\ Log\ (A_2/A_1) \ldots\ldots \quad (4)$$

$$D_{ne\ in-situ}\,(A_2) = D_{ne\ in-situ}\,(A_1)\ -\ 7\ Log\ (A_2/A_1) \ldots\ldots \quad (5)$$

As seen from equation (4), the decay of sound for this window in laboratory conditions is in the form of -9.3 log(A_2/A_1). This is relatively similar to the theoretical behaviour given by equation (2). The in-field performance given by eq. (5) however, decays as -7log(A_2/A_1). This means that the in-situ sound insulation reduction of this open window when doubling the distance, is closer to 2dB compared to the theoretical 3dB figure.

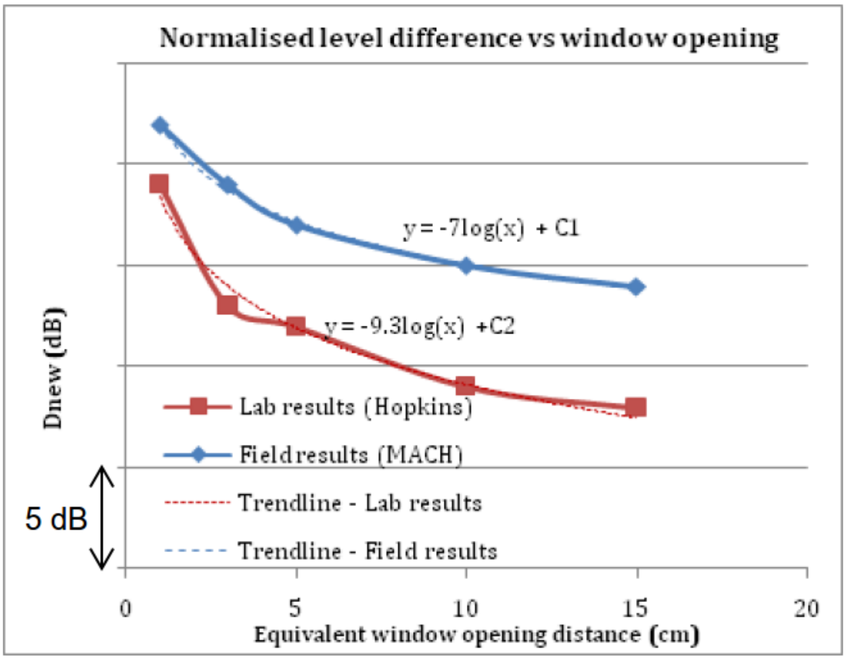

Figure 9: Decay of sound insulation with opening distance for a Velfac 200 window

Going back to Table 2 above, It is seen that typical opening areas range from 3.9 m^2 for single-sided ventilation to 0.3 m^2 for stack ventilation. Hence, the associated sound insulation based upon eq. (1) will under predict the acoustic performance by over 5 dB compared to that by eq. (5).

6.2 Window types

Napier University[9] assessed the acoustic performance of fourteen different window types. The source room was anechoic apart from the wall, where the windows were installed and the receiving room was supposed to replicate a typical room with an average reverberation time of 1.4 seconds.

MACH Acoustics has focused its analysis on five of these windows that are thought to be the most common and relevant to natural ventilation. Napier University[9] data is based upon an external microphone at 2 m in front of the window. The windows sound reduction is then determined using an open area of 0.05 m^2, 0.1 m^2 and 0.2 m^2.

Since the w-weighting curve in accordance with ISO 717-1[10] gives a single figure in the form of an integer, there is an uncertainty when calculating the slope in sound reduction. This is especially significant when only a few points (opening areas) are used to calculate the windows decay slope, and when these points are relatively close to one another.

Hence, an alternative method is shown here to calculate the weighted element-normalised level difference, which is given with a precision of 0.1dB. This was achieved by shifting the reference curve by 0.1 dB rather than at intervals of 1dB. Figure 10 thus compares the results of using these two methods. From the right hand image, it can be seen that the predicted decay of sound (curves) matches more closely the actual acoustic performance for the different windows, rather than using the standard integer figure.

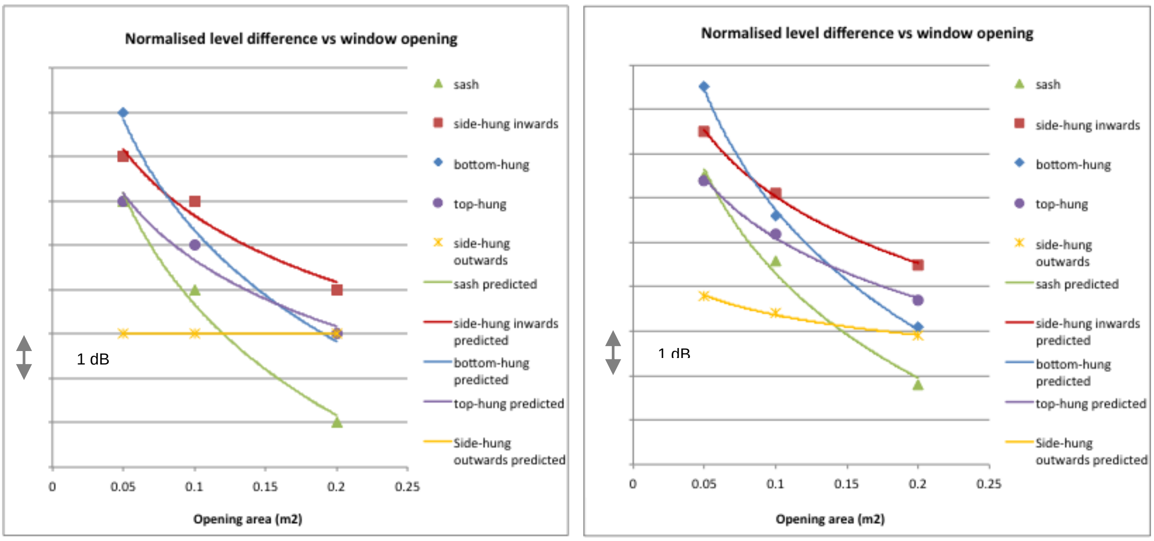

Figure 10: Sound insulation decay for five different windows based on traditional ISO 717-1 D_{new} (left) and the alternative 0.1dB accuracy figure (right).

The root-mean-squared error (RMSE) has also been calculated in both scenarios and the results are given in Table 3. It is seen that the RMSE decreases when using the 0.1dB w-weighting figures.

Window	side-hung inwards	sash	bottom-hung	top-hung	side-hung outwards
1dB	0.4	0.4	0.4	0.4	0
0.1dB	0.1	0.4	0.2	0.1	0.0

Table 3: RMSE for five different open windows using 1dB and 0.1 dB w-weighting figures

The slopes for the five windows are shown in Table 4. The slopes vary significantly from one window type to another as shown in Figure 10. Thus, bottom-hung and sash windows seem to have the steeper slopes, approaching to that of the theoretical representation -10log(A). Side-hung inward tilt and top-hung are close to -5log(A), whereas side-hung opening outwards has a minimal decay of -1.5 log(A). Although the measurement setup by Napier[9] is unclear as to where the external microphone is placed compared to the window openings, whose heights vary with window type, it can be said that the decay of sound insulation depends significantly on the type of window. This shows that the general use of equation (1) is not of sufficient accuracy.

Window	side-hung inwards	sash	bottom-hung	top-hung	side-hung outwards
1dB	-5.0	-8.3	-8.3	-5.0	0.0
0.1dB	-5.0	-7.8	-9.0	-4.5	-1.5

Table 4: Slopes for five different open windows using 1dB and 0.1 dB w-weighting figures

6.3 Directivity

The angle of incidence of the source to the window is another parameter that should be considered when characterising the acoustic performance of open windows. The sound reduction values at angles of -75°, -35°, 0°, 35° and 75° in the horizontal plane, are shown in Figure 11for two windows: a horizontal inward tilt and a side-hung outward tilt. The former shows a symmetrical pattern due to its horizontally symmetrical geometry. On the other hand, the side-hung shows a decrease of up to 7 dB when the source is in line of sight with the receiving room compared to that at angles out of sight. Hence, by having a directivity component in the model, the accuracy of the actual performance of open windows can be improved.

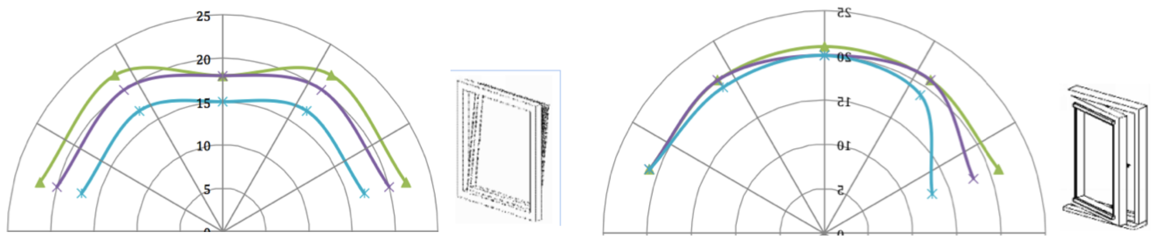

Figure 11: Sound insulation performance (dB) as a function of source angle of incidence for a horizontal inward tilt (left) and a side-hung outward tilt (right) window

6.4 Simple Model

Based on results given in Sections 6, 6.2 and 6.3,a simple model that characterises the performance of open windows is provided here. This includes the decay of sound with opening area and the directivity effect due to the angle of incidence of the source to the window opening. Thus the sound insulation of open windows can be estimated as:

$$D_{ne}\ (A) = -XLog\ (A)\ +\ C\ +\ D \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (6)$$

where X is the slope of the sound decay, A is the opening area, D is the directivity component associated to the angle of incidence and C is a constant relating to a specific windows type.

In other words, the parameters in the above equation are dependent on the window type and hence the slopes, constants and directivity components will vary accordingly depending upon window type.

7 CONCLUSION

Current guidance in acoustics refers to the sound insulation provided by open windows with imprecise numbers varying from 5 to 15 dBA/R_W. However, accurately assessing the acoustic performance of naturally vented facades has been shown to be of significant importance, with an approximate increase of 20% to 40% chance to naturally ventilate buildings when the facade sound insulation is assessed to significantly higher levels of accuracy.

Hence, a simple model to characterise the acoustic performance of open windows has been presented here. The model includes the sound decay of open windows with opening area, as well as the directivity component of the source angle of incidence.

The influence of the acoustic field in which the acoustic performance is measured has been shown. Thus, the sound insulation of open windows measured in-situ, decays less steeply with opening area than that measured in diffuse field. In addition, different windows have very different decays of sound which ultimately result in their acoustic performance at several opening distances. Finally, the angle of incidence which is also dependent on the window type can have a significant effect on the actual performance, depending on where the source is located. All these components are included in the proposed simple model which represents a starting point to characterise the acoustic performance of open windows.

8 FUTURE WORK

This paper is a result of the work between MACH Acoustics and the University of Bath researching into methods to assess and improve the sound insulation of vented façades, using different modelling techniques. The aim of this paper is to establish how to assess and quantify the acoustic performance of a window. Since, the decay of sound and directivity effects are window-type dependent, the next step is to create a database with several window types and as many opening distances as possible. This will then be used to provide an excel model estimating the performance of different openings. Although the proposed model is currently based on the single w-weight figure, this will be expanded to evaluate the acoustic performance for octave or third octave bands

Over time, MACH Acoustics are looking to increase their modelling abilities based upon Finite Element and Finite-Difference Time-Domain methods. This includes quantification of the performance of windows in diffuse and non-diffuse fields, including variation of angle of incidence. This data will be used to assess the frequency characteristics of vents aimed at determining parameters to enhance our current model.

Finally, a new procedure to test open windows/vents will be proposed in the absence of a standard methodology to test such elements. This aims to overcome the limitations of ISO 140-10 and ISO 140-5.

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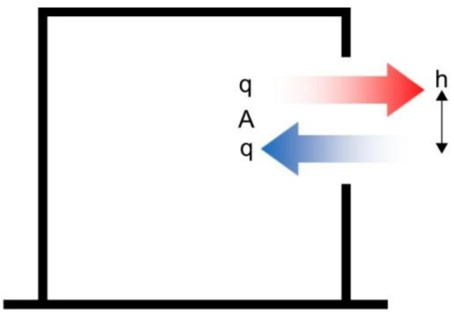
APPENDIX A - VENTILATION DESIGN

Once the ventilation strategy has been designed and the flow rates through openings have been quantified, all the openings can be sized to achieve the designed flow rates. The general steps are:

- Identify isolated rooms and spaces: when the openings on the internal part of the envelope need to be much smaller than the external ones, rooms can be assumed to be isolated in terms of ventilation. The two basic cases are single-sided and cross ventilation.
- Determine which parts can be treated as a single space. This can be assumed if internal openings are much greater than those in the external envelope and hence all openings are considered simultaneously when sizing them. If internal and external openings are of the same size order, they could be sized based on isolated space and single-cell methods, choosing the one with larger areas.
- Input data: parameters such as positions of the openings, their discharge coefficients and wind and temperature conditions will be needed to obtain a solution
- Size the openings based on relevant equations
- Check practicality of the obtained opening sizes.

6.5 Isolated spaces

6.5.1 Single-sided, single vent, buoyancy driven

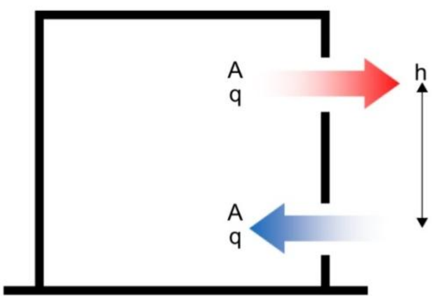


The area A of each opening IS required to give a ventilation rate q for a specified value of h is:

$$A = \frac{q}{C_d}\sqrt{\frac{(T_I + 273)}{\Delta T\ g\ h}} \qquad (A.1)$$

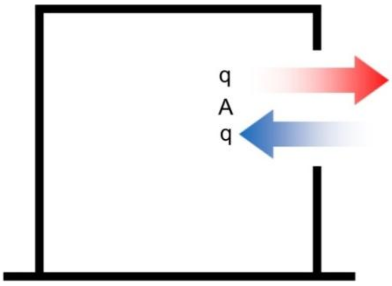
where$A$ is the area of each opening (m^2), h is the ventilation rate (m^3·s^{-1}), C_d is the discharge coefficient, T_I is the internal temperature (°C), ΔT is the difference between the internal and external air temperatures (K), g is the gravitational force per unit mass (m·s^{-2}) and h is the height between the openings (m).For a single opening, the discharge coefficient is typically 0.25.

6.5.2 Single sided, 2 vents, buoyancy driven



The same equation as above governs the ventilation flow with two identical openings, one above the other. However, since the air comes in through the lower opening and exts through the upper opening, the discharge coefficient is typically 0.61, i.e. larger than for a single opening since the flow is unidirectional through each opening.

6.5.3 Single-sided, single vent, wind driven

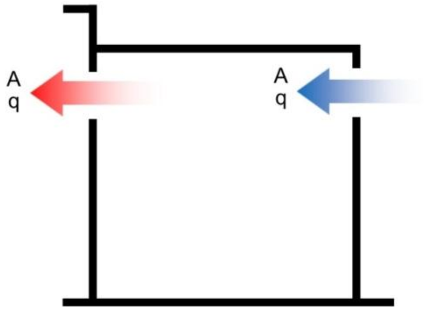


When ventilation is driven by wind for a single opening, the required area to achieve the flow rate is given by:

$$A = \frac{q}{C\,U} \tag{A.2}$$

Where C is a coefficient that depends on the geometry of the opening, the position at which the reference wind speed is measured and the flow field around the building. Typical values vary from 0.01 to 0.05.

6.5.4 Cross-ventilation, wind driven

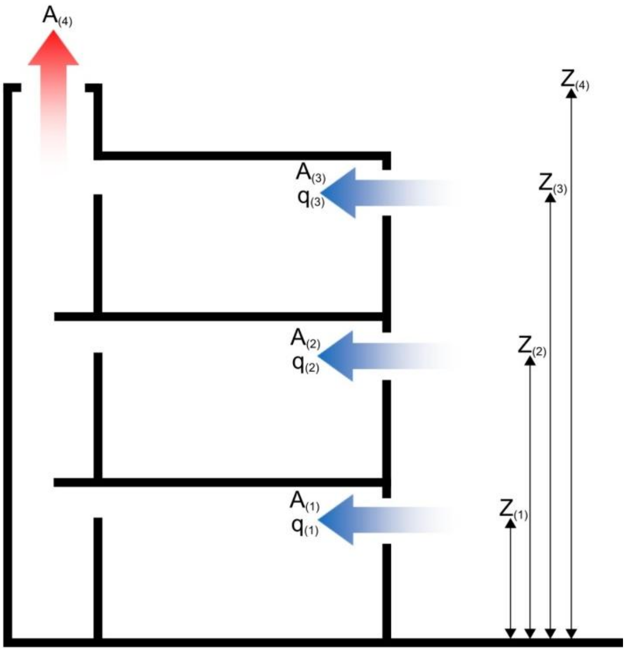


The ventilation area required for a wind-driven, cross-ventilated space is given by:

$$A = \frac{q}{C_d\, U\sqrt{\frac{\Delta C_p}{2}}} \tag{A.3}$$

Where A (m^2) is the total ventilation area for each wall and ΔC_p is the difference between the wind pressure coefficients C_{p1} and C_{p2}.

6.6 Multiple spaces



6.6.1 Stack – buoyancy only

The procedure then to estimate the area of the openings is as follows. First the desired air flow through the openings is specified as well as its direction. It is common practice to choose air entering through the external facade and exhausting through the top of the stack. After selecting the airflow directions, the height at which the pressure difference must change sign will be specified. The neutral height z_n is that at which $\Delta p_i = 0$. Thus, in the absence of wind

$$\Delta p_0 = \Delta \rho_0 \, g \, z_n \qquad (A.4)$$

The pressure difference at any height can be calculated as

$$\Delta p_i = \Delta \rho_0 \, g(z_n - z_i) \qquad (A.5)$$

The required areas can then be calculated as

$$C_{d\,i} A_i = \frac{q_i}{S_i}\sqrt{\frac{\rho_0}{2|\Delta p_i|}} \qquad (A.6)$$

The value of Δp_0 completely determines all the values of Δp_i. In this explicit model a change in the height and flow rate of one of the inlets will not affect those at other inlets.

6.6.2 Stack – wind only

Assuming the external and internal temperatures are the same, the required flow patter would be achieved if the internal pressure were such that the pressure difference across the upper opening is half of the total pressure difference across the building, i.e.

$$\Delta p_4 = -0.5 \, \rho \, U^2 \Delta C_p \; / \; 2 \qquad (A.7)$$

where ΔC_p is the total pressure coefficient across the whole building. Δp_0 can be calculated setting $\Delta\rho_0 = 0$ (i.e. no buoyancy effect) as

$$\Delta p_0 = -0.5\,\rho\,U^2\left[\left(\Delta C_p\ /\ 2\right) + C_{p\,4}\right] \qquad (A.8)$$

Thus, the pressure difference at any height is given by

$$\Delta p_i = -0.5\,\rho\,U^2\left[\left(\Delta C_p\ /\ 2\right) + C_{p\,4}\right] + 0.5\,\rho\,U^2 C_{p\,i} \qquad (A.9)$$

6.6.3 Results

The parameters included in the case study are:

- External temperature: TE=25
- Internal temperature (the same as external temperature for wind-only case), TI=28
- Wind speed (except for buoyancy case), U=3m/s
- Reference density, $\rho_0 = 1.2$
- Density difference, $\Delta\rho_0 = 0.01028$ kg/m^3
- Pressure coefficients, $C_{p\,1} = C_{p\,2} = C_{p\,3} = 0.25$; $C_{p\,4} = -0.1$
- Discharge coefficient, $C_{d\,1} = C_{d\,2} = C_{d\,3} = C_{d\,4} = 0.61$
- Number of people: 31
- Flow rate target: 8l/s/p
- Total inlet flow rate, q_i = 0.25 m^3/s
- Opening height above floor: 1.5 m
- Floor height: 3m
- Neutral height, Zn=10 m

		Area (buoyancy only)		Area (wind only)	
Case	**Ventilation type**	**Inlet (Outlet too for case 1)**	**Outlet**	**Inlet (Outlet too for case 1)**	**Outlet**
1	Single sided	3.2	-	4.1	-
2	Single sided 2 vents	1.3	1.3	-	-
3	Cross-vent	1.3	1.3	0.3	0.3

Table 5: Opening areas of inlets and outlets for isolated spaces

Opening	Floor	Opening height [m]	Flow rate [m/s]	Buoyancy only	Wind only	Buoyancy and wind
Inlet	1	1.5	0.25	0.3	0.3	0.2
Inlet	2	4.5	0.25	0.4	0.3	0.2
Inlet	3	7.5	0.25	0.6	0.3	0.3
Outlet	4	11.5	-0.75	2.2	1	0.9

Table 6: Opening areas of inlets and outlets for stack space

APPENDIX B - THEORY FINITE TIME DIFFERENCE MODELING

The fundamental part of the FDTD method is to approximate well known the physical wave equations by using finite difference equations to replace the time and space derivatives within (in this case) the equations for momentum and mass continuity,

$$\nabla p = -\frac{\partial}{\partial t}\vec{u}\rho \qquad (\mathbf{1.1})\frac{1}{\rho c^2}\frac{\partial}{\partial t}p = \frac{q(r,t)}{\rho} - \nabla.\vec{u} \qquad (\mathrm{B.1})$$

where$\vec{u} = u_x\hat{x} + u_y\hat{y} + u_z\hat{z}$ is the particle velocity vector, p is the acoustic pressure, ρ and c are the acoustic density and sound speed of the surrounding medium respectively and $q(r,t)$ defines the rate of creation of fluid in kg/m^3s. Suppose that a spatial grid size and a discrete time interval are Δx and Δt respectively, differential terms in space and time can be defined by,

$$\frac{df}{dx}(x_0) \approx \frac{f(x_0+\Delta x) - f(x_0-\Delta x)}{2\Delta x}(\mathbf{1.3})\frac{df}{dt}(t_0) \approx \frac{f(t_0+\Delta t) - f(t_0-\Delta t)}{2\Delta t} \qquad (\mathrm{B.2})$$

known as the 'centered' difference equations. To represent each uniform element in a 2D or 3D grid requires matrices to contain the directional components of the velocity vector $\vec{u} = u_x\hat{x} + u_y\hat{y} + u_z\hat{z}$, as well for the acoustic pressure p. To obtain an expression for the x component of particle velocity requires the substitution of equations 1.3 and 1.4 into 1.1, rearranging to solve for $u_x\left(x,y,z,t+\frac{n}{2}\right)$ to give

$$u_x\left(x,y,z,t+\frac{n}{2}\right) = u_x\left(x,y,z,t-\frac{n}{2}\right) + \frac{n}{\rho h}\left(p\left(x-\frac{h}{2},y,z,t\right) - p\left(x+\frac{h}{2},y,z,t\right)\right) \qquad (\mathrm{B.3})$$

where$n = 2\Delta t$ and $h = 2\Delta x$ as this case is using a staggered leap frog method of calculation. Similarly again approximating derivatives in equation 1.2 obtains pressure

$$\begin{aligned} p\left(x,y,z,t+\frac{n}{2}\right) = p\left(x,y,z,t-\frac{n}{2}\right) + \frac{\rho k c^2}{h^3}Q(x,y,z,t) - \frac{\rho k c^2}{h}\left[u_x\left(x+\frac{h}{2},y,z,t\right) - u_x\left(x-\frac{h}{2},y,z,t\right)\right] \\ - \frac{\rho k c^2}{h}\left[u_y\left(x,y+\frac{h}{2},z,t\right) - u_y\left(x,y-\frac{h}{2},z,t\right)\right] \\ - \frac{\rho k c^2}{h}\left[u_z\left(x,y,z+\frac{h}{2},t\right) - u_z\left(x,y,z-\frac{h}{2},t\right)\right] \qquad (\mathrm{B.4}) \end{aligned}$$

where the source term q has been substituted for a volume velocity function Q, related by $q = Q(\rho/\Delta V)$ where $\Delta V = \Delta x \cdot \Delta y \cdot \Delta z = h^3$.

6.7 Boundaries

The previous equations (1.5 and 1.6) show that knowledge is required of the particle velocity pressure at all surrounding elements (i.e. the element is in free space), this is not the case at a boundary. For this 'right' and 'left' difference equations are required:

$$\frac{df}{dx}(x_0) \approx \frac{f(x_0+\Delta x) - f(x_0)}{\Delta x}(\mathbf{1.7})\frac{df}{dx}(x_0) \approx \frac{f(x_0) - f(x_0-\Delta x)}{\Delta x} \qquad (\mathrm{B.5})$$

In a similar way to before, equations 1.7 (or 1.8) and 1.4 are substituted into 1.1. Rearranging to solve for $u_x\left(x,y,z,t+\frac{n}{2}\right)$, using the fact that the normal acoustic impedance $Z_n = p/u_n$ and approximating that $u(x,y,z,t) \approx \left(u\left(x,y,z,t+\frac{n}{2}\right)+u\left(x,y,z,t-\frac{n}{2}\right)\right)/2$, an expression for a 'right' or 'left' hand boundary (in the x direction) is finally obtained:

$$u_{x\frac{R}{L}}\left(x,y,z,t+\frac{n}{2}\right)=\frac{(\rho h \mp nZ_n)}{(\rho h \pm nZ_n)}u_x\left(x,y,z,t-\frac{n}{2}\right)\pm\frac{2n}{(\rho h+nZ_n)}p\left(x\mp\frac{h}{2},y,z,t\right) \quad (\mathrm{B.6})$$

whereZ_n (the normal acoustic impedance of the boundary surface) can be manipulated by plane wave analysis to relate to normal incidence absorption coefficient a_0:

$$Z_n=\rho_0 c_0\frac{1+\sqrt{1-a_0}}{1-\sqrt{1-a_0}} \quad (\mathrm{B.7})$$

6.7.1 Defining a Source

A simple source can be made by applying a Gaussian pulse centred at time t_0 with a width σ to a single point in the pressure matrix

$$p(x,y,z,t)=e^{-\left((t-t_0)^2/\sigma^2\right)} \quad (\mathrm{B.8})$$

Such a source has inherent problems as the pressure point does not follow the acoustic equations when the source pulse has ended. However this is only apparent when receiving reflected sound at the source, which in this case is not of interest.

6.7.2 Stability and Accuracy

For stability the choice of time step Δt and grid spacing Δx must satisfy the Courant condition $c(\Delta t/\Delta x)\leq 1$. To obtain accurate results requires a fine enough resolution such the shortest wavelength of interest is represented by at least 7-8 nodes, i.e. to simulate 500 Hz with a cell size of $\lambda/7$ would require $\Delta x = 10\ cm$.