The acoustics and ventilation method: a simple design tool.

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Summary

The acoustics and ventilation method allows the designer to explore natural ventilation and acoustics issues at an early design stage. Few input parameters are necessary, like the number of people the space must accommodate and the volume of air per person appropriate to the acoustic function. While allowing the comfort temperature to rise during occupation by some 10 to 15%, the natural ventilation and acoustics can be discussed via the volume of the space.

Introduction.

Misconceptions about natural ventilation are inhibiting architects and developers, according to a Building Research Establishment publication, February 1998 edition of "News of Research and Innovation", although the topic is explored in many texts (1) (2) (3). Natural ventilation allows air into and through buildings so it also allows noise and air pollutants to enter. It has always been important to encourage designers to imagine the sound and noise in their design proposals. Imagining these acoustics levels should be placed alongside the visualisation of the interior. Rassmussen (4) points out that buildings do not radiate light, but are seen, so spaces that do not generate sound nevertheless can be heard. Sugden (5) reminds us that deafness reduces our contact with the world more greatly than blindness.

The Acoustics and Ventilation method, AV method, is proposed as an aid to the designer at the sketch design stage. The AV method allows the designer to explore the possibility of naturally ventilating a space, it confronts issues of size, shape, reverberation times and background noise levels. The AV method gives a sketch design tool for looking at (i) the simple acoustics of a space and (ii) the possibility of naturally ventilating the same space, starting with some simple environmental factors and comfort assumptions. The connection between these two issues being the static and dynamic air volumes. This method is based upon simple routines for thermal air movement and the acoustics of spaces, which are readily available in many undergraduate texts (3), (6), (7).

This paper will show the AV method applied to a number of spaces. Spaces like concert halls, courtrooms, lecture theatres, opera houses, seminar rooms, theatres, and town halls are considered. However, each space has its own set of additional issues that the designer will need to resolve. For example, domestic size spaces, like seminar rooms may only require that a window be opened to provide the natural ventilation. For a small urban workshop theatre it will be impossible to open the windows to allow ventilation to take place, since the illusion will be lost as noise enters. Again, in a city centre office block the interior might become polluted by noise and vehicle exhaust fumes simply by opening windows. Here a natural ventilation strategy would require pollution control. Natural ventilation is therefore more than just opening windows. The incoming air may need warming, cooling, cleaning and quietening.

This paper has three sections. Following this brief introduction the AV method spread-sheet is presented. This shows the various design parameters and issues, applied to a range of quite different spaces, together with the main assumptions. There follows a section where each line of the method is explained and discussed. Finally, there is a brief discussion of the AV method together with some observations on the difficulties to be resolved.

The Acoustics and Ventilation method.

The spread-sheet for the AV method calculations is illustrated in the figure below. The design and comfort issues, which are in the numbered rows 1 to 17, are given in the second column, while the output for the various spaces are shown in the various columns, the spaces identified at the column heads. The method can best be described by looking at each design issue in turn. In the following description of the Excel spread-sheet, the various rows of data will be divided into two categories, (1) those where the designer must input data, and (2), those where some assumptions and calculations provide the output, usually based on some previously gathered data in the sheet. Data is only entered in the bordered boxes.

ACOUSTICS AND VENTILATION SPREAD-SHEET

Spread-sheet to explore natural ventilation and acoustics issues at an early design stage.
Begin with the number of people to occupy the space, select the sound source type (below), and the optimum volume of air per person.

YOU MUST ONLY ENTER DATA INSIDE the BORDERED BOXES

Issue	Design input or output	symbol	seminar or boardroom	court room	lecture theatre	civic hall	theatre	opera house	concert hall
1	Number of people to occupy space	n	20	50	100	200	500	1000	2000
2	Volume of air per person $/m^3$	Z	3	4	4	6	6	5	8
3	Volume of space $/m^3$	V	60	200	400	1200	3000	5000	16000
4	Floor area, assuming rectangular $/m^2$	A	15	37	75	150	374	748	1495
5	Length, golden section /m	l	5	8	11	16	25	35	49
6	Width of space /m	w	3	5	7	9	15	21	30
7	Height of room /m	h	4	5	5	8	8	7	11
8	Total surface area in space $/m^2$	S	92	206	333	690	1357	2196	4587
9	Additional chimney added to height /m		0	0	0	0	0	2	3
10	Reverberation time /s	RT	0.61	0.70	0.78	0.93	1.11	1.23	1.62
11	Acoustic absorption due to people (0.56) $/m^2$	AP	11	28	56	112	280	560	1120
12	Area-weighted absorption coefficient	α	0.05	0.08	0.08	0.14	0.11	0.04	0.10
13	Acoustic absorption due to fabric $/m^2$	a	4.5	17.5	27	95	153	88	463
14	Air change rate per hour	N	11.2	8.4	8.4	5.6	5.6	6.7	4.2
15	Stack effect pressure $/Nm^{-2}$	Δp	1.6	2.2	2.2	3.3	3.3	3.5	5.6
16	Air inlet (and outlet) grille area $/m^2$	G	0.2	0.5	1.1	1.8	4.4	8.4	13.4
17	Grille inlet (and outlet) area to floor area %	R	1.7	1.4	1.4	1.2	1.2	1.1	0.9

AGAIN, MAKE YOUR SELECTION IN THE BORDERED BOXES in this assumption section.

INPUT DATA AND ASSUMPTIONS:

Floor area per person $/m^2$ — 0.65 — For an office this should be larger, say 1.5 m^2.
Floor area is increased by factor — 1.15 — For circulation and performance area ADD 15%.
Golden section: Length/width is — 1.62

Stephens and Bate sound types: speech 4
orchestral 5 — Place your sound type selection here.... 4
choral 6

Heat generated per person is /Watts — 85 — which is increased by a factor of 1.25
to allow for other incidental gains, like lighting and equipment.

Internal initial air temperature t1 / C = 20 which will be allowed to rise by t2 / C = 3
External air temperature t3 / C = 12

The temperature difference driving the stack effect is taken as t1 + 0.5 t2 - t3 = 9.5 C
The factor of 0.5 has been introduced to distinguish between temperature and comfort.

Row 1. INPUT: The number of people to occupy the space, n. When thinking about the design of a space a fundamental parameter must be the number of people that the space must accommodate. The local village hall may be used for PCC meetings, or for a village summer fayre. In the first case the main hall space might have 20 members at the meeting, while the fayre might lead to full hall with 150 people. The designer might choose 100 as being an optimum occupancy for the hall.

Row 2. INPUT: The volume of air per person, Z m^3 per person. For a space to work acoustically, the volume of air that each person has in the space must be chosen with care. Choral music is more acceptably reproduced in large volumes like cathedrals, while a law court, where speech intelligibility is paramount, requires a smaller volume. Egan (8) and Smith (9) have both reported optimum volumes of air per person for various spaces. Here, it is proposed that the following m^3 per person be used; for rooms for speech 2.5 to 3.5, for cinemas 3 to 4, for Italian style opera houses 4 to 5, for churches where speech is important 5 to 9, for churches where music is important 6 to 10, for concert halls 8 to 12, and for multipurpose spaces 6 to 11 m^3 per person.

Row 3. OUTPUT: The volume of space, V m^3. This row is simply the product of row 1 with row 2, and the volume of the various spaces can been seen for comparison.

Row 4. OUTPUT: The floor area, assuming the space is a rectangular box, A m^2. Below the table of issues and results there are a number of input parameters. The first assumption is the floor area occupied by each individual in the space. For an auditorium this value is relatively straight-forward, and is determined by seating geometry. If a seat density of 0.65 m^2 per seat is assumed, it is a simple matter to increase this figure by a factor of 1.15, or an increase of 15%, to allow for circulation and performing area. The floor area is the product of the number of people and the seat density adjusted for circulation and performing area. In the very large spaces, these simple assumption will require further consideration, since keeping the audience at reasonable distances from the performers, and ensuring suitable sight lines are crucial. Again, in an office the seat density needs to be increased by a larger fraction to account for the equipment and furniture.

Row 5. OUTPUT: The length of space, assuming rectangular shoe box shape with golden ratio floor, l m. There are four basic types of auditorium, fan, shoe box, vineyard and arena. However, for the purposes of this paper a simple geometry for the floor area is chosen, to calculate the room dimensions. Acousticians, like Sugden (5), suggest that auditoria with a rectangular shoe box shape usually give the warmth and clarity required of a concert hall, while others pursue more exciting forms. Here a ratio of room length to width of 1.6180 has been chosen, which leads to a golden plan. A golden plan would be a rectangular floor "the sides of which are in the approximate ratio 3:5, or 8:5 is presumed to have the most pleasing proportions aesthetically speaking." Anon (10). The length of the space can now be calculated since the room length is equal to the square root of 1.6180 times the floor area. The value of the length for the largest space in the table, the concert hall, is not unreasonable, although in all the spaces the activities versus solid geometry would need revisiting.

Row 6. OUTPUT: The width of space, w m. This is simply 0.6180 times the room length.

Row 7. OUTPUT: The height of space, h m. Given that the space is a rectangular shoe-box the ceiling height above the floor is the volume divided by the floor area. This height is important since the stack-effect, used to determine the ventilation rate, relies on the temperature difference across this height. To achieve ventilation it will be assumed that the cool air is at floor level, while the warm air is at ceiling height. In some spaces it may be appropriate to add some height to improve the stack-effect. This is achieved by adding a chimney.

Row 8 OUTPUT: The total fabric surface area in the space, S m^2. This row gives the total wall, floor and ceiling area in the rectangular box, and this area will be used later to calculate the acoustic absorption required in the space to achieve the optimum reverberation time.

Row 9. INPUT: The additional chimney height. This additional height was introduced above.

One major advantage in adding a chimney, or duct through which the air must flow to escape the main space, is that the walls of this duct may be lined with acoustically absorbing material. External ambient noise may be attenuated in the chimney, ensuring that suitable background noise levels may be achieved.

Row 10. OUTPUT: The optimum reverberation time in seconds, RT. The Stephens and Bate (11) empirical expression for the optimum reverberation time in seconds at a frequency of 500 Hz is,

$$RT \quad = r\,[\;0.0118\,(\,n\,Z\,)^{1/3} \;+\; 0.1070\;]\;\text{sec}$$

where the value of r depends upon the type of sound to be produced in the space. For speech r is taken as 4, for orchestral music 5 and r is 6 for choral and organ music. We will use r as 4 for human speech. Row 10 gives this optimum reverberation time at 500 Hz. Another run of the sheet would demonstrate the difference if the value of r was taken as 6 for choral music. The values in the present table show that for speech only the last space is so large that the reverberation time becomes difficult, and would lead to a possible reduction in speech intelligibility. Some form of speech reinforcement system would be required in this large space.

This optimum reverberation time RT is quoted at a frequency of 500 Hz. At 125 Hz the RT should be increased by about 30%, while the RT at 4000 Hz should be decreased by about 20%, after Egan (8), page 125. The magnitude of bass increase is often a matter of taste, while the decrease at high frequency can reflect the air absorption. For the example, the opera house has an optimum reverberation time shown as 1.11 seconds. It follows that the reverberation times at 125 and 4000 Hz should be 1.60 and 0.98 seconds respectively. Reverberation times must be treated with caution. The values were once thought to be the only measure of the acoustics of a space.

Row 11. OUTPUT: The acoustic absorption due to the people present in the space, AP m^2. Before Beranek (12) the average human absorption at 500 Hz was taken as 0.46 m^2 per person. However, this value is now regarded as an underestimate and, moreover, we now use the effective audience floor area together with an absorption coefficient of 0.8 in more sophisticated calculations for concert halls. To enable this simple calculation we will follow Beranek and take the average human absorption coefficient as 0.56 m^2 per person at 500 Hz. Row 11 gives the acoustic absorption provided by the people present in the space.

Row 12. OUTPUT: The area-weighted absorption coefficient α. The Eyring (13) expression for reverberation time is used to assess the absorption necessary in the space. The total absorption has two components, (neglecting the air absorption), (i) the absorption by the fabric surfaces, and (ii), that due to the people. Eyring derived his expression for the reverberation time of a space assuming the acoustic energy was lost, or absorbed, when it was incident on a surface. The expression is,

$$RT = 0.16\, V / [- S \log_e(1 - \alpha) + AP]$$

where S is the total surface area in the space, and AP is the total absorption provided by the people. This expression can be re-arranged to give a value of α. It would be simpler to use the Sabine expression, except that the Sabine expression is not valid if the value of α is greater than 0.2.

Row 13. OUTPUT: The acoustic absorption provided by the fabric in the space. This is calculated from the area-weighted acoustic absorption coefficient times the total surface area in the space, and suggests how the surfaces finishes in the space might be chosen in order to achieve the optimum reverberation times.

Row 14. OUTPUT: The air-change rate per hour, N per hour. The incidental heat gains from the occupants, equipment and lighting causes the air temperature to rise. Ventilation cooling allows fresh air from outside to remove this heat and cool the interior. If this ventilation can be achieved without the use of energy, then clearly this has an advantage. Obviously, the method only works well if the outside air is cooler than the internal air. In winter the incoming air may need warming. Assume that the external ambient air is at a temperature of θ_e, the initial temperature in the space is θ_s, with $\theta_e < \theta_s$, and the instantaneous temperature at any time is θ. Each person in the space will be generating heat at a constant rate W Watts, or 3600W Joules per hour, and the heat will be lost by the ventilation. Suppose the ventilation rate provides N air-changes per hour, then we can write a differential equation for the rate of change of temperature as the heat gain and heat loss rates compete. It will be assumed that the thermal exchange takes place continuously and that the fabric of the space is, therefore, not heated in the process. In other words, it will be assumed that the heat given to the air is removed via the ventilation.

The volumetric heat capacity of air is taken as 1200 J/m^{3}degC, since this magnitude divides simply by the number of seconds in one hour. Therefore, the rate of change of heat content in the air per hour in the space is 1200 n Z ($d\theta/dt$). This rate of change of heat content in the air of the space must be equal to the rate of arrival of heat minus the rate of loss, or

$$1200\, n\, Z\ (d\theta/dt) \quad = \quad n\ W\ 3600 \quad - \quad 1200\, N\, n\, Z\, (\theta - \theta_e).$$

This expression shows that the result is independent of the number of people in the audience, and note that it is per hour. Therefore, we may write:

$$d\theta/dt \quad = \quad (3\,W/Z + N\,\theta_e) \ - \ N\,\theta$$

After an infinite time a state of equilibrium will be reached, the rate of change of temperature with time becomes zero, and a final steady temperature will be established, say θ_f. Here, the heat gain in the space exactly balances the heat loss, and the air-change rate N is,

$$N \quad = \quad 3\,W\ /\ Z(\ \theta_f\ -\ \theta_e\ )$$

Suppose the final temperature in the auditorium is the initial temperature plus some rise θ_r, or θ_f = $\theta_s + \theta_r$, then the ventilation air-change rate corresponding this maximum temperature for a given external air temperature θ_e is,

$$N \quad = \quad 3\,W\ /\ Z\,(\theta_s + \theta_r\ -\ \theta_e\ )$$

This rise in temperature θ_r, and the distinction between air temperature and effective comfort temperature, is discussed later, but its value is selected in the spread-sheet towards the bottom. Sufficient to say here is that the value of θ_r used in the spread-sheet calculations is half the input value. This recognises the difference between sensible and latent heats, and is a simple way of allowing the effective temperature in the space to rise by 3 C, but allowing only 1.5 C to be the sensible rise in air temperature to drive the stack effect.

Once the air-change rate N is obtained, then the volume of air required per second can be determined, and if an assumption about the pressure difference driving this air-change is made then the cross-sectional area of the ventilation inlet and outlets systems can be calculated.

Row 15. OUTPUT: The pressure difference giving the stack-effect, Δp Pa. Infiltration and natural ventilation result from two causes often acting separately on different sides of a building. The first cause is the stack effect, while the second is the wind effect. The stack effect pressure difference driving the air through a building comes about because of a temperature difference. For the stack effect, which considers two columns of air, both of height h, but at different temperatures, say $T_a > T_b$ K, and since in the built environment both these temperatures are not greatly different from 10 C, the pressure difference Δp may be written to a reasonable approximation as,

$$\Delta p \quad = \quad 0.043\, h\, (\theta_b - \theta_a)$$

The pressure difference increases with both height and temperature difference, so in a naturally ventilated building the fresh air inlets and the stale air outlets should be separated by maximum height. This is the reason why row 9 was included, so that we may add a chimney. Again, in temperate climates it may be necessary to have some means of increasing the temperature difference to improve the overall efficiency of the ventilation scheme. At the Queens Building at De Montfort University, Leicester, only the stack-effect is required to give the necessary and sufficient air-changes. It is assumed that including the wind effect is unnecessary, but when there is a wind effect it will provide additional drive for ventilation.

Row 16. OUTPUT: The input - output ventilation grille area, G m^2. Practical studies have been carried out on the movement of air through typical building constructions under ordinary weather conditions. Together with these studies and laboratory tests, it has been shown that for buildings the quantity of air flowing dQ/dt m^3/sec through a hole of area A m^2 is approximately proportional to the square root of the pressure difference Δp N/m^2 across the hole, and it is assumed that dQ/dt = 0.83 A $(\Delta p)^{1/2}$ m^3/sec. In buildings the air flow will occur through many openings of different sizes, some in parallel, some in series. However, here it is assumed that the air will be displaced at a rate dQ/dt m^3/sec, when a temperature difference $\Delta\theta$ C exists between an inlet and outlet opening, or grille, each grille being of area G m^2.

For an air volume flow rate of N n Z / 3600 m^3s^{-1} through two equal grille areas G m^2 placed in series, then G is given by,

$$G = 4.75\ 10^{-4}\ N\ n\ Z\ (\Delta p)^{-0.5}\ m^2$$

Practical details of the size and distribution of all the ventilation inlets and outlets will need to be determined later in the design process, along with considerations of the flow resistance of the ventilation ducting.

Row 17. OUTPUT: The grille area to floor area ratio, R. A recent design guide by Penz (14) has suggested that the area of fresh air intakes to a naturally ventilated law court should be 1% of the floor area, the outlets having an area 2% the floor area. For the Queens Building at De Montfort University the design has incorporated inlet and outlet grille areas both equal to 4.8% of floor area. The values of R in the spread-sheet are less than 2%, and therefore acceptable.

Discussion.

The first issue and major assumption in this AV method is the magnitude of the rise in temperature experienced in the space by the occupants as the space warms due to the incidental heat gains. Now our thermal comfort preferences are contextual, relative and subjective. They vary with age, gender, the time of day and year. Moreover, these thermal preferences are paralleled in the areas of environmental lighting and acoustics. At a recent meeting entitled, "Indoor air temperature standards for the 21st Century", both Forwood (15) and Rowe et al (16) suggested the subjective perception of thermal comfort was not fully documented and understood. Baker (17) underlines the present ambiguity associated with thermal comfort studies, people's perceptions of comfort and the more exact analysis suggested by Fanger. Baker concludes that when people are allowed to make small adaptive changes to their environments, these small adaptive mechanisms produce significant reductions in the percentage of people predicted as dissatisfied (ppd) with a thermal environment, the ppd falling from two-thirds to one sixth of the occupancy. Here, it is assumed that people may tolerate wider environmental fluctuation than previously accepted, given they appreciate they are in a naturally ventilated space. Moreover, comfort is a function not only of air temperature, but also of air speed, mean radiant temperature, and relative humidity, together with other subjective factors. Instead of the air temperature being maintained at 20 C by complex and costly hardware, suppose an audience at a concert were happy to enjoy a romantic symphony while the air temperature increased by 10%. If the audience knew that they were in a naturally ventilated auditorium they would be happy to tolerate changes in the thermal environment.

There are three areas that require further study. Firstly, natural ventilation cooling requires that the external air has a lower temperature than the internal air. Secondly, the incoming air will be polluted, for example with pollen, vehicle exhaust fumes, and ozone. Finally, the incoming air will bring noise pollution, and the entry of noise through the air exit route has already been mentioned. Evans (18) reports on a proposal by Hampshire County Architects to use underground concrete pipes to ventilate a theatre at Bedales School. He goes on to offer a brief analysis of the feasibility of the scheme. In a later article Evans (19) describes the BRE latest low energy office with a ventilation strategy as loose and robust, where air is moved through a variety of horizontal and vertical routes at low velocity. The new Glyndebourne opera house has four 50m long 2m by 2m concrete ducts buried in the ground to carry air from the ventilation plant to the auditorium. Air flowing in large underground ducts can be moved slowly to minimise noise and maximise heat transfer. Computational fluid dynamic studies applied to buried ducts might investigate the cooling of the incoming air, the flow through filters to remove pollution, while the lining of these ducts might be chosen carefully to attenuate the noise. An interesting conflict must be resolved, since the length of the duct must be short, with minimum air flow impedance, to allow the volume of air into the interior, while long to achieve sufficient noise attenuation and thermal adjustment.

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