OPTIMISATION OF BUILDING VENTILATION OPENING SIZE IN A NOISY ENVIRONMENT

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ABSTRACT: This paper addresses the problem of achieving noise reduction in buildings that are exposed to road traffic noise while satisfying ventilation requirements. A design process has been developed that optimises the size of an opening in a building for ventilation while estimating the attenuation of external traffic noise inside the building. A fan with associated duct work is fitted to an opening in the building facade to provide ventilation. Therefore the optimisation of the opening size is used to determine the necessary fan characteristics which best suits the opening. The process allows the user to specify particular environmental conditions including traffic noise spectra, required ventilation rates, ventilation fan type, ventilation duct characteristics and building facade properties. Model results are presented indicating an optimum ventilation opening size with the associated internal noise levels. The work is part of a larger study into reducing noise through ventilation openings.

1. INTRODUCTION

The noise from road traffic in urban areas is a well known problem throughout the world. While there is an obvious requirement to reduce the level of external noise entering the building enclosure, ventilation standards must also be adhered to. Unfortunately, the methods of obtaining satisfactory levels of noise reduction and ventilation conflict. Therefore there must be a point of compromise between these two functions.

Extensive research has been carried out into the shielding of external noise provided by buildings [1]-[4]. While the need for ventilation is acknowledged, a method of ventilation and the noise characteristics associated with it has not been included in the research. Similarly, many references concerning heating, ventilation and air conditioning systems (including fan engineering) concentrate on the resulting sound levels in enclosures due to ventilation fans or devices performing the ventilation process, without taking into account other external noise sources (such as road traffic) affecting the enclosure being ventilated [5]-[9]. An important combination of these two functions involves a design process that optimises the size of an opening in a building to satisfy ventilation requirements while attenuating external noise from road traffic and noise generated by the particular ventilating device. This paper outlines the development of such a design process. Two case studies are presented to demonstrate the process.

2. THE DESIGN PROCESS

A design process has been developed that optimises the size of an opening in a building for ventilation while estimating the attenuation of external traffic noise and ventilation fan noise level inside the building. To provide ventilation, a fan with associated duct work is fitted to the opening in the building facade containing a window. Therefore the optimisation of the opening size is used to determine the necessary fan characteristics which best suits the opening. A spreadsheet program was used to provide a range of opening sizes in the wall for ventilation so as to obtain an optimum hole size.

Initial Conditions

Some initial conditions had to be established to begin the design process. Although particular conditions have been chosen, these could be set at any reasonable values according to the desired environment. This allows a sensitivity analysis of the results of the design process to be carried out.

The dimensions of the room to be ventilated were set to be similar in size to a normal bedroom, measuring 3x5x2.4 m high. These dimensions were used by Lawrence and Burgess [2] for a bedroom in an experimental building used to measure the reduction of traffic noise by facades containing windows. The facade exposed to traffic noise measured 5x2.4 m high. The design process was developed to allow a road traffic noise spectrum in octave bands from 63 Hz to 8 kHz to be used as the source of external noise. Typical values were taken from a relevant reference [10]. The ventilation rate of the room was required. Appropriate ventilation requirements were taken from Australian Standard AS1668 Part 2-1980 [11]. Different ventilation rates could be chosen according to the class of occupancy involved. The option was made for an inlet duct to attenuate fan and traffic noise entering the building through the ventilation opening (assuming that the air intake was on the noisy side of the building and the exhaust on the quiet side). The length of duct, number of bends and type of duct lining could be chosen as required. Calculations could be carried out without an inlet duct attached. The duct was assumed to be on the inside of the building and that no duct breakout occurred into the room.

Appropriate values for the absorption coefficient of the duct lining in octave bands from 125 Hz to 4 kHz were taken from a relevant reference [5]. The transmission loss characteristics of the walls of the building were chosen to correspond to single leaf brick masonry walls and the windows were chosen to be double glazed with an area of 10% of the total wall area [12].

Fan Selection

To select a suitable fan, the fan pressure needed to circulate the required flow Q had to be determined. The total pressure drop of the inlet duct system is estimated in a similar manner to that adopted in common fan engineering references [5], [6] [8], [9] i.e. the total pressure drop is obtained by adding together the elements of total pressure drop around the system including:

- Losses at entry to the inlet duct from atmosphere
- Losses due to friction along the duct length
- Losses at bends
- Losses at discharge from the system to atmosphere

The total pressure drop due to air flowing into and out of the room is calculated using the equation:

$$\Delta P_{TOTAL} = \left(\frac{0.02L}{D} + K_{entry} + nK_{bends} + \frac{0.02L_{wall}}{D} + K_{exit} \right) \frac{1}{2} \rho V_{air}^{2} \quad \text{[Pa]} \qquad (1)$$

where

L= length of inlet duct (m), D= diameter of duct to fit opening (m), K_{entry} = coefficient accounting for losses at entry of inlet duct [6], K_{bends} = coefficient accounting for losses at the bends of inlet duct [6], L_{wall} = wall thickness, n = number of bends, K_{exit} = coefficient accounting for losses at discharge to atmosphere [6] and ρ = air density (kg/m³).

This pressure is the fan pressure needed to circulate the required flow Q. Normally there would be further resistance due to air flow through the interior of the rest of the building (as the room would be joined to other rooms). It has been assumed, however, that the air flowing out of the room flows to the surrounding atmosphere. Since the average velocity of air through the ventilation opening is inversely proportional to the area of the opening, the total fan pressure required to circulate the flow decreased rapidly at a rate of $1/A^2$ as the area of the opening A increased.

Obviously the most reliable method of determining the sound power level of a particular fan would be to obtain sound power levels in each octave frequency band from a fan manufacturer. In this design stage of the ventilation system, however, only the required duty has been established. The type of fan to be used has not been decided upon. Therefore an empirical formula must be used to estimate the likely sound power level that will be produced. An approximate idea of the sound power level of the chosen fan was gained from the following equation [7]:

$$L_w = 10 \log Q + 20 \log \Delta P_{total} + 37 \quad \text{[dB]} \qquad (2)$$

There are correction factors for different fan types in different frequency octave bands. These have been incorporated into the design process to illustrate the difference in optimum opening size according to fan type [7]. The application of these correction factors allows the sound power level of a fan, for a particular ventilation opening size, to be calculated in octave bands.

Attenuation of Noise Provided by Lined Inlet Duct

The level of attenuation provided by the inlet duct is a function of the absorption coefficient of the duct lining. For a straight length of duct L metres, variations of Sabine's equation have been used to provide reasonable estimates.

Estimates of the attenuation provided by lined bends were very difficult to find in relevant references. The data that were found are difficult to compare because many different measurements techniques are used and the fact that there is no consistent nomenclature for the noise reduction provided by bends. Field measurements indicate that the attenuation due to a 90° bend in the duct work would be 10 dB higher than that for the same length of lined duct [13]. This is valid provided the lining extends at least two diameters of the duct each side of the bend. Therefore the total attenuation provided by a lined duct of diameter D metres with n 90° metre bends can be calculated using the following equation:

$$\text{Total attenuation} = 4.2\alpha^{1.4}\left(\frac{L}{D}\right) + n\left(8.4\alpha^{1.4} + 10\right) \qquad (3)$$

where n = number of bends and α = absorption coefficient of duct lining.

The absorption coefficient is a function of frequency. Typical values in octave frequency bands were taken from a relevant reference [5] to obtain the total attenuation provided by the inlet duct.

Equivalent Transmission Loss of Wall with Inlet Duct Attached

The transmission loss of a wall with a window and an opening for ventilation with an inlet duct attached was calculated in two stages. Firstly, the equivalent transmission loss of the building facade with a hole and window in it, calculated in octave bands, was determined as an overall area-weighted average of the wall, window and duct cross sections. For this calculation, equation (4) was used.

Obviously as the percentage open area increased, the equivalent transmission loss of the wall, window and hole decreased. The area of the hole dictated the level of transmission loss provided by the composite combination. The equivalent transmission loss with the inlet duct attached was then calculated using an equation similar to equation (4). The equivalent transmission loss decreased in a similar fashion, but at a slower rate, however, than that for a wall with only a hole it since the inlet duct has the ability to attenuate noise.

$$TL_{eq} = -10 \log\left(A_{wall}\tau_{wall} + A_{hole}\tau_{hole} + A_{window}\tau_{window}\right) / A_{total} \quad \text{[dB]} \qquad (4)$$

where $\tau_{wall} = I_{t\ wall} / I_i$
$\tau_{hole} = I_{t\ hole} / I_i$
$\tau_{window} = I_{t\ window} / I_i$

The terms I_t, and I_i, are the transmitted and incident sound intensities respectively

Noise Entering Room

The noise entering the room will be a combination of fan and traffic noise that has been attenuated by the lined inlet duct and building facade.

The sound pressure level of traffic noise entering the room is determined by subtracting the equivalent transmission loss in each octave band calculated in the equation similar to equation (4), from the traffic noise spectrum in octave bands set in the initial conditions. Since the attenuation provided by the composite building facade decreased as the area of the opening for ventilation increased, the level of traffic noise entering the room increased as the area of ventilation opening increased. The traffic noise entering the room, however, was largely low to mid-frequency noise (A-weighted). The noise levels in the 250 Hz to 1 kHz octave bands were the only significant contribution to the overall traffic noise level in the room. Hence the contribution of the traffic noise in the room to the overall sound level in the room is only significant for large ventilation openings in the building facade because of the lower equivalent transmission loss provided by the wall, window, hole and inlet duct at larger ventilation opening sizes.

In addition to the fan noise being attenuated by the lined inlet duct, end reflection occurs at the end of the duct run, unless the duct diameter is very large (compared to the wavelength of sound). The maximum end reflection occurs at low frequency as the wavelength is greatest compared with the size of the opening. Equation (5) was used to calculate the end reflection loss depending on frequency and diameter of the duct being used [8]:

$$\Delta L = 10\log\left[1+\left(\frac{c_o}{\pi f D}\right)^{1.88}\right] \quad \text{[dB]} \qquad (5)$$

where c_o is the speed of sound in air (344m/s), f is the octave band frequency and D is the diameter of the duct.

The sound power level of the fan entering the room in dB will therefore be:

$$L_{w\ room} = L_{w\ fan} - attenuation\ by\ inlet\ duct - end\ reflection \qquad (6)$$

where the attenuation by the inlet duct in each octave band is calculated from equation (3) and the end reflection loss in each octave band for a particular duct area is calculated in equation (5).

The sound power level of the fan in the centre of the room is converted to an equivalent sound pressure level using the standard equation for a point source located in the centre of the wall, assuming the fan noise radiates hemispherically on a single reflecting surface. The average absorption of the interior of the room was also taken into account:

$$L_{p\ fan} = L_{w\ room} + 10\log\left(\frac{D}{2\pi r^2}+\frac{4(1-\bar{\alpha})}{S\bar{\alpha}}\right) \qquad (7)$$

where $L_{w\ room}$ = fan L_w calculated in equation (6), D = directivity of fan, $\bar{\alpha}$ = average room absorption coefficient, S = surface area of room (m²), r = distance from duct exit to room centre (m).

The sound pressure level of the fan in the centre of the room decreased as the size of the ventilation opening increased in a similar fashion to the sound power level since all the terms in equation (7), except the fan sound power level $L_{w\ room}$ are constants.

Total Sound Pressure Level in Room

The total sound pressure level in the room could then be determined by adding the contributions made by the traffic noise entering the room and the noise from the fan entering the room. The overall sound pressure level of the traffic noise in the room was determined by adding the equivalent squared pressures of the traffic noise in the eight octave bands and then converting back to a total sound pressure level. Similarly, the squared pressures of the fan noise in the room were combined and converted to a total sound pressure level. The overall sound pressure level in the room due to the fan and traffic could then be calculated using a similar technique.

The fact that the sound pressure level of traffic noise in the room increases as the size of the ventilation opening increases and the fact that the sound pressure level of the fan in the room decreases as the size of the ventilation opening increases indicates that there will be a particular ventilation opening size where the total sound level in the room due to these two external noise sources is a minimum.

3. CASE STUDIES

Variation of Fan Type

The correction factors for different fan types previously mentioned indicate that the noise level in the room and hence the optimum ventilation opening size is greatly influenced by the type of fan used for ventilation. Figure 1 shows the total sound pressure level in the room as a function of percentage open area (corresponding to the size of the opening for ventilation) for different types of fans. The different fan types considered were forward curved centrifugal, backward curved centrifugal, radial, axial and propeller fans.

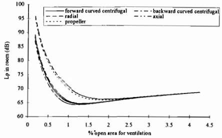

Fig.1 Total sound pressure level in room (different fan types) vs percentage open area for ventilation.

Forward curved and backward curved centrifugal fan types were the quietest while the radial type was the noisiest for the same volume flow and fan pressure. Correspondingly, the optimum ventilation opening size for forward curved and backward curved centrifugal fans is smaller, allowing a more compact size for a given duty than other fan types. The smaller ventilation opening size also increases the effective transmission loss of the building facade to the external traffic noise. Therefore the type of fan used will also affect the amount of traffic noise entering the room in addition to the amount of fan noise entering the room. From Figure 1, the optimum ventilation opening size for the quietest fans (forward and backward curved centrifugal) were approximately 1% of the total facade area exposed to external noise and 1.6% for the noisiest fan (propeller). From the nature of the curve for areas greater than the optimum percentage area, it is reasonable to expect that any opening size between 1 and 4% of the total wall area could be considered for ventilation purposes.

Variation of Traffic Noise Characteristics

The design process required that the traffic noise spectrum be set in the initial conditions. The typical spectrum chosen [10] characterised urban flow traffic (below 60 km/h) directly outside the building facade, which is dominated by the large amount of acoustic energy concentrated in the 63 Hz and 125 Hz octave bands (due to exhaust noise generated by heavy diesel commercial vehicles). Obviously this type of traffic noise will not always be present. For traffic flowing at steady speed (greater than 80 km/h) the spectrum contains energy concentrated at higher frequencies due mainly to tyre/road interaction noise (not present at low speeds) and mechanical noise from power train components. The nature of the noise will also vary with distance from the building facade. Hence it is necessary to investigate the effect of different traffic noise characteristics on the optimum ventilation opening size in the building facade exposed to the traffic noise.

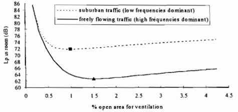

Fig. 2 Total sound pressure level in room (different types of traffic noise) vs percentage open area for ventilation.

A traffic noise spectrum with acoustic energy concentrated in the low frequency bands was used to simulate suburban traffic. Likewise, a spectrum with energy concentrated in the high frequency bands was used to simulate freely flowing traffic. To allow a comparison to be made between the two forms of traffic, the dB(A) values of the two spectra were kept equal. Figure 2 shows the total sound pressure level in the room for various ventilation opening sizes predicted using the design process. The optimum ventilation opening size for the suburban traffic noise was 1.0% of the total wall area and was 1.5% for the freely flowing traffic (both points are shown on Figure 2). This indicates that a larger opening size can be used for the freely flowing traffic because the building facade and attenuating duct are able to attenuate the high-frequency noise more effectively than the low-frequency dominant suburban traffic noise. The difference in attenuation of the two types of traffic flow can be seen by the distance between the two curves (the ventilation opening sizes where the two curves overlap is the region where the fan noise is dominant).

4. CONCLUSIONS AND DISCUSSION

From this work it is concluded that the optimum size fan for a ventilation opening, when there is a fan and a short length of lined duct, is about 1-1.5% of the total wall area. This compares to about 5-10% of openable area when natural ventilation is utilised. The difference in noise level inside a building, with brick veneer construction, facing a road will be approximately 17 dB(A).

If only a fan is used (without a lined duct), the optimum opening area for minimal sound level will be 0.75% with a sound level reduction of 9.2 dB(A) over the case for 10 % of the facade open for natural ventilation. Compared to the reduction due to a barrier this is significant.

The work described forms part of a project to reduce noise entering building ventilation openings. Future work will be concerned with developing alternative methods such as 'intelligent' openings, noise cancellation and systems where there is a road barrier between the inside and outside of the building.

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