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Aeroacoustics of automotive vents

S. Guérin[a,*], E. Thomy[b], M.C.M. Wright[a]

[a]*Institute of Sound and Vibration Research, University of Southampton, Southampton SO17 1BJ, UK*
[b]*Research Division, Renault, 1 Avenue du Golf, 78288 Guyancourt Cedex, France*

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Abstract

This paper studies the generation of noise by car ventilation systems whose outlet rates are controlled by a butterfly valve and whose directions are controlled by grilles. First the noise created by the valve alone is analysed with the theory formulated by Nelson and Morfey for spoiler-generated noise in-duct flow. To confirm this theory the fluctuating force experienced by the valve is measured experimentally and the mean drag force is deduced from analytical work presented by Sarpkaya. Then the noise generated by the grille and its effect on sound transmission is investigated. Finally, it is shown that a strong and complex interaction between the wake shed behind the valve and the grille occurs when both elements are placed close together. This is responsible for an overall increase in the noise level although some sound reduction is measured at low frequency. It is found that moving the valve further upstream can reduce the noise by several decibels.

1. Introduction

The noise generated by ventilation systems can be a significant component of interior noise in cars. Many components contribute to aerodynamic sound generation by car ventilation units, but the outlet part is of great importance. Usually, this is equipped with a butterfly valve placed upstream of a cascade of horizontal and vertical deflectors that form a grille, a typical example of

*Corresponding author. Müller-Breslau Str. 8, 10623 Berlin, Germany.
E-mail addresses: sebastien.guerin@dlr.de, mcmw@isvr.soton.ac.uk (S. Guérin), emmanuel.thomy@renault.com (E. Thomy).

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Nomenclature

a, b	test duct cross-section dimensions ($a>b$) (m)
A	duct cross-section area (m^2)
c_0	speed of sound (m/s)
C_{CU}, C_{CL}, C_C	contraction coefficient
d	spoiler width (m)
f	frequency (Hz)
f_0	cut-on frequency of the first transverse mode (Hz)
$\bar{F}_{\mathrm{DRAG}}$	steady-state drag force (N)
$\tilde{F}_{\mathrm{DRAG}}$	fluctuating drag force (N)
$K(St)$	ratio $\tilde{F}_{\mathrm{DRAG}}/\bar{F}_{\mathrm{DRAG}}$
M_C	contraction Mach number
$R(f)$	pressure reflection coefficient
s	microphone separation (m)
St	Strouhal number
$\mathrm{SWL}_{\mathrm{rad}}$	sound power level radiated outside the duct (dB)
$\mathrm{SWL}_{\mathrm{duct}}$	in-duct sound power level (dB)
t	butterfly valve thickness (m)
U	in-duct flow velocity (m/s)
U_c	constriction velocity (m/s)
W_{rad}	power radiated into free space (W)
α	butterfly valve angle of attack (angle of opening)
α_H,α_V	angle of inclination of the horizontal and vertical deflectors
Δp_s	static pressure drop created by spoiler (Pa)
ρ_0	mean density (kg/m^3)
ω	radian frequency (rad/s)

which is shown in Fig. 1. This particular ventilator is taken from a Renault Clio; its dimensions are similar to those of the experimental model described later. These devices generate high levels of broadband noise when they obstruct the flow path; i.e. when the valve is partly closed in order to reduce the flow rate or the deflectors are inclined to change the flow direction at the exit. The other main sources of noise in the ventilation system are the blower and other discontinuities in the system such as pipe junctions, etc.

This study deals with noise generation in low-speed flow. Therefore, the force fluctuations over the bodies are expected to be the major sources of sound. When the butterfly valve is almost closed, it is likely that the sheared stress fluctuations in the two jets formed in the lower and upper constrictions will be significant sources of aerodynamic noise as well, but this situation is not treated here.

Previous work, which was developed with the purpose of estimating the noise produced in a ventilation system obstructed by a single rigid obstacle, may be applied to this problem. Iudin [1] derived a first rule of prediction deduced from dimensional analysis and measurements carried out on various airduct elements. This rule indicates that sound power is proportional to the cube of the pressure drop across this element and the square of the geometric dimensions. Neglecting the effect of confinement on the sound propagation, a similar relationship was found by Gordon [2,3] who replaced the noise sources by aerodynamic dipoles whose strength is proportional to the pressure drop.

The role of the confinement was highlighted by Heller and Widnall [4] and Nelson and Morfey [5]. Both have demonstrated that aerodynamic dipoles behave like free field monopoles at frequencies below that of the first transverse mode, with a consequent U^4 velocity dependence of the sound power radiated by a spoiler in an infinite duct. The effect of enclosure on the sound power radiation may be ignored at higher frequencies when the acoustical wavelength is shorter than the duct cross-sectional dimensions; in this case the sound power is proportional to U^6.

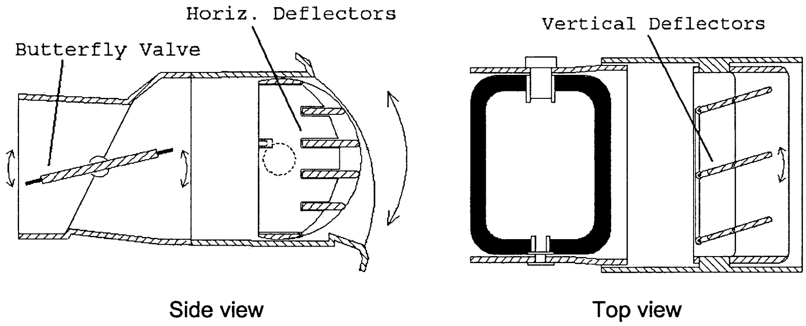

Fig. 1. Sketch of the ventilation outlet system mounted on RENAULT CLIO II.

In a semi-infinite duct, the end-reflection at low frequency reduces the sound power that passes out of the exit by a factor proportional to the inverse square of the frequency. This restores the original velocity dependence of the noise generated at low frequency to that of an aerodynamic dipole.

Nelson and Morfey [5] modelled the sound sources generated by a vertical spoiler by a vertical distribution of axial dipoles. Under realistic hypotheses on the turbulence correlation length and the spatial distribution of the forces, the sound power radiated is related to the total fluctuating drag force acting on the spoiler. At constant Strouhal number, this force is assumed to be proportional to the mean drag force, with a constant of proportionality $K(St)$. This factor, characteristic of the airduct component, is deduced from acoustic and mean flow measurement in the present study. In Section 3, the fluctuating drag is directly measured on an inclined valve with miniature force transducers. The result is introduced into the Nelson–Morfey model in which the mean drag force is calculated with the free streamline theory. The results are compared to acoustic measurements in a reverberant chamber.

The deflectors which make up the grille behave to a certain extent like the valve. But when they are placed directly after the butterfly valve the disturbances contained in the wake shed behind the valve may increase the noise radiation by the grille, even if it is not configured to alter the flow direction. However, the grille will also reflect a part of the sound coming from the valve and so may reduce the noise radiation due to the valve. Results from these valve–deflector interaction mechanisms are given in Section 4.

2. Acoustical rig

An outline drawing of the acoustical rig used in these experiments is shown in Fig. 2. Air is supplied by a centrifugal blower whose flow rate is controlled by an adjustable conical inlet insert. The pressure loss in the entrance is adjusted by varying the opening inlet section. In order to isolate the downstream part of the system from the noise generated by the fan, the fan is connected via a flexible pipe to a large muffler containing basalt as an absorbing material. A flexible pipe carries airflow to the test duct. This is located in the reverberant

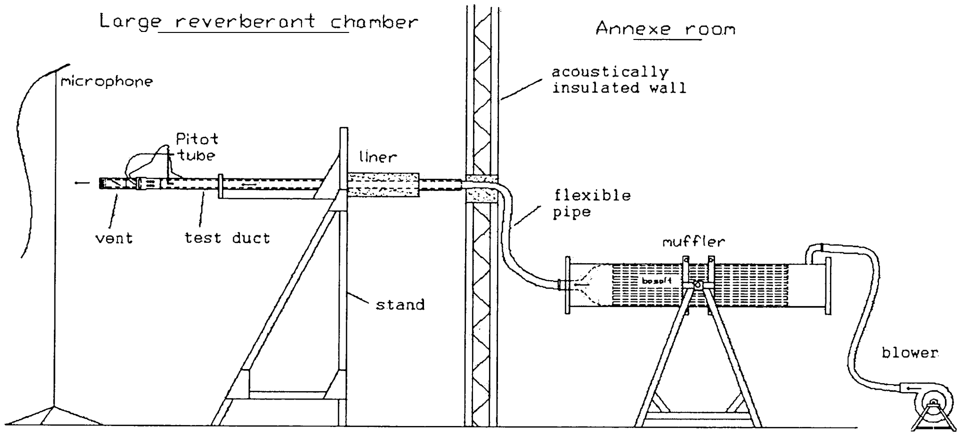

Fig. 2. Acoustic rig.

chamber where the acoustic field is measured, while the muffler and the fan are housed in a separate room.

The test duct is a 2 m long, 10 mm thick hard wooden duct at the free end of which the vent is mounted. The inner cross-sectional dimensions are $a = 70$ mm and $b = 50$ mm. In its upstream part, the duct has been lined with 0.5 m of sound-absorbing material in order to obtain nearly anechoic boundary conditions. The attenuation provided by the liner is only 1.4 dB at 125 Hz but increases rapidly with frequency; for instance at 250 Hz, the attenuation is already about 14 dB.

Considerable efforts were made to smooth the inner surface of the wooden duct. The mean velocity along the duct centreline was measured with a thin Pitot tube placed 1 m downstream of the lining. At this location, the stream is fully turbulent, well established and the wall boundary layer may be considered thin enough to assume that mean velocity in a vertical section is approximately the velocity measured along the duct centreline. The absolute precision of the pressure meter used to measure the dynamic pressure was ± 3 Pa, giving 5% error in the velocity at $7\,\mathrm{m\,s^{-1}}$ but only 1% at $16\,\mathrm{m\,s^{-1}}$.

The experiments were performed with a simplified full-scale model of an automotive vent composed of two modules made of Plexiglas; the first one contained a butterfly valve and the second one a grille of three vertical and four horizontal deflectors. A picture of this model is shown in Fig. 3. The valve is idealized by a flat plate 70 mm long, 50 mm wide and 3 mm thick. The coordinate system is defined by (O, x, y, z) where the origin O lies on the centreline at the junction between the two modules and x, y and z are, respectively, the longitudinal (i.e. streamwise), transversal and vertical directions. In this coordinate system, the valve can be moved from $x = -113.5$ to -26 mm (x being positive towards the exit). The angle of incidence with the flow α can be varied from -40° to $+40^\circ$.

The grille module (40 mm long) is composed of three vertical deflectors and four horizontal deflectors. The chord of these deflectors is 15 mm and their thickness is 3 mm. The length of the vertical and horizontal deflectors is the same as the duct height (50 mm) and the duct width (70 mm), respectively. The vertical and horizontal deflectors are symmetrically arranged at $x = 16$

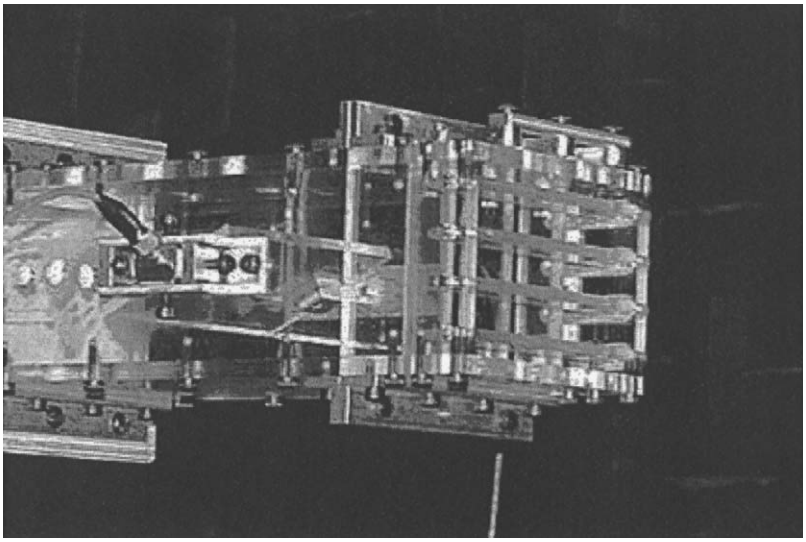

Fig. 3. Simplified model of vent (equipped here with force transducers).

and 32 mm, respectively in the coordinate system defined above. A system of rods allows all the deflectors of a same row to be moved together.

3. Sound generation due to the butterfly valve

3.1. Model of sound prediction

The noise radiated by the valve alone may be approximated by the theory formulated by Nelson and Morfey [5]. In this theory, two expressions are derived for the spectral density of the acoustic power $W(\omega)$ radiated in one direction of a semi-infinite square duct obstructed by a vertical spoiler. The first one is valid below the cut-on frequency of the first transverse duct mode f_0 (plane wave propagation) and the second one applies to frequencies higher than f_0 (multimodal propagation)

$$\begin{aligned} W(\omega) &= \frac{\tilde{F}^2_{\mathrm{DRAG}}(\omega)}{4A\rho_0 c_0}, && f < f_0, \\ W(\omega) &= \frac{\omega^2 \tilde{F}^2_{\mathrm{DRAG}}(\omega)}{24\pi\rho_0 c_0^3}\left[1 + \frac{3\pi c_0}{4\omega}\frac{(a+b)}{A}\right], && f > f_0. \end{aligned} \tag{1}$$

In the two preceding expressions, a and b are the cross-sectional dimensions, A is the duct cross-section area, ρ_0 the ambient density and c_0 the ambient sound speed (a nomenclature is provided). The total fluctuating drag force acting on the spoiler is $\tilde{F}^2_{\mathrm{DRAG}}(\omega)$, which is in direct proportion to the steady-state drag force $\bar{F}_{\mathrm{DRAG}}$ as observed by Heller and Widnall [4]. Within 1/3-octave

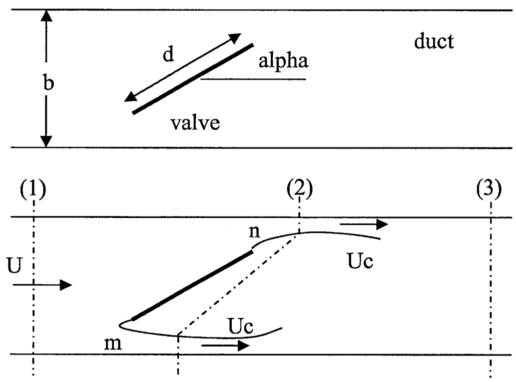

Fig. 4. Butterfly valve in-duct flow [6,7].

bands, it may be further assumed [5] that

$$\tilde{F}_{\mathrm{DRAG}} = K(St)\bar{F}_{\mathrm{DRAG}}, \tag{2}$$

where St is an appropriate Strouhal number.

Sarpkaya [6,7] showed that the steady force exerted by flow impinging obliquely upon an infinitely thin plane placed between two infinite parallel planes can be found, in a two-dimensional case, by the free streamline theory. The case is sketched in Fig. 4 in which U represents the velocity at upstream, U_c is the constriction velocity, b the height of the conduit, d the length of the valve, α the angle of opening, and m and n are the thicknesses of the jets in the lower and upper contractions, respectively. The angle of closure β ($=90^\circ$ when $d \equiv b$), the contraction coefficients at M and N, respectively, C_{CL} and C_{CU}, the total contraction coefficient C_C and the contraction area ratio σ are defined below.

$$\begin{aligned}
&\beta = \sin^{-1}\frac{b}{d},\\
&n = C_{CU}\frac{b}{2}\left(1-\frac{\sin\alpha}{\sin\beta}\right),\\
&m = C_{CL}\frac{b}{2}\left(1-\frac{\sin\alpha}{\sin\beta}\right),\\
&C_C = \frac{C_{CU}+C_{CL}}{2},\\
&\sigma = \frac{A_C}{A} = \frac{U}{U_c} = \left(1-\frac{\sin\alpha}{\sin\beta}\right)C_C.
\end{aligned} \tag{3}$$

The value of the contraction coefficient C_C depends on the angle of incidence α (e.g. $C_C \approx 0.74$ at $\alpha = 10^\circ$, $C_C \approx 0.67$ at $\alpha = 40^\circ$). The solution is found by conformal mappings and the application of the Schwartz–Christoffel transformation. Hassenpflug [8] has also studied this problem using series expansions in conjunction with the method of Frobenius.

By applying Bernoulli's equation between (1) and (2) and then the momentum equation between (2) and (3), it can be seen that

$$\bar{F}_{\mathrm{DRAG}} = \Delta p A = \frac{\rho_0 U_c^2}{2} A(1-\sigma)^2. \tag{4}$$

Finally, by replacing the mean drag force in Eq. (2) by Eq. (4), then introducing the result into Eq. (1), the two following expressions can be used to predict the noise radiated at the exit of a semi-infinite duct enclosing a butterfly valve:

$$\begin{aligned} W_{\mathrm{rad}} &= (1-R^2)\frac{\rho_0}{16c_0} A K^2(St)(1-\sigma)^4 U_c^4, & f < f_0, \\ W_{\mathrm{rad}} &= \frac{\pi\rho_0}{24c_0^3}\left[1+\frac{3\pi c_0}{4\omega}\frac{(a+b)}{A}\right]\left(\frac{A}{b}\right)^2 K^2(St)(1-\sigma)^2 St^2 U_c^6, & f > f_0, \end{aligned} \tag{5}$$

where the Strouhal number is based on the wake height and the contraction velocity:

$$St = \frac{fd}{U_c} = \frac{fb(1-\sigma)}{U_c}. \tag{6}$$

In Eq. (5), account is taken of the partial sound reflection at the duct exit below f_0 by introducing the transmission term $(1-R^2)$. The reflection effect is neglected above f_0.

The following hypothesises are also made:

(i) The contribution of the fluctuating lift to the sound generation is neglected. This is based on the study by Heller and Widnall [4] which shows that sound measured beyond the duct exit is almost only correlated at low frequency with the drag forces exerted on spoilers. However, this assumption is not valid above the cut-off frequency for which transversal waves can propagate.

(ii) The aerodynamic sound is assumed to be generated by axial dipoles distributed on a vertical section whose surface area is the projection of the butterfly valve upon a vertical plane.

(iii) The effect of mean flow on sound propagation and reflection is neglected because of the low values of the Mach number $M_C = U_c/c_0$ (max $M_C < 0.08$ at $\alpha = 10°$, max $M_C < 0.2$ at $\alpha = 40°$). For the same reason, the sound absorption which may take place in the shear layers developed behind the valve and at the exit should concern only the very low frequency range and therefore may be neglected [9].

Of these three assumptions the third is the least defensible; indeed, as will be seen, the results admit a small but significant variation with Mach number.

3.2. Determination of K(St)

Using the inverse method, Nelson and Morfey deduced values of $K(St)$ from the sound radiation outside the duct. The results obtained with various vertical spoilers and orifices collapse within ± 5 dB and seem to be in good agreement with Gordon's data obtained at low speed [2,3] on oblique plates. Although no single universal curve $K(St)$ exists, different works [5,10–13] lend support to the important idea that a range of components might produce a similar curve.

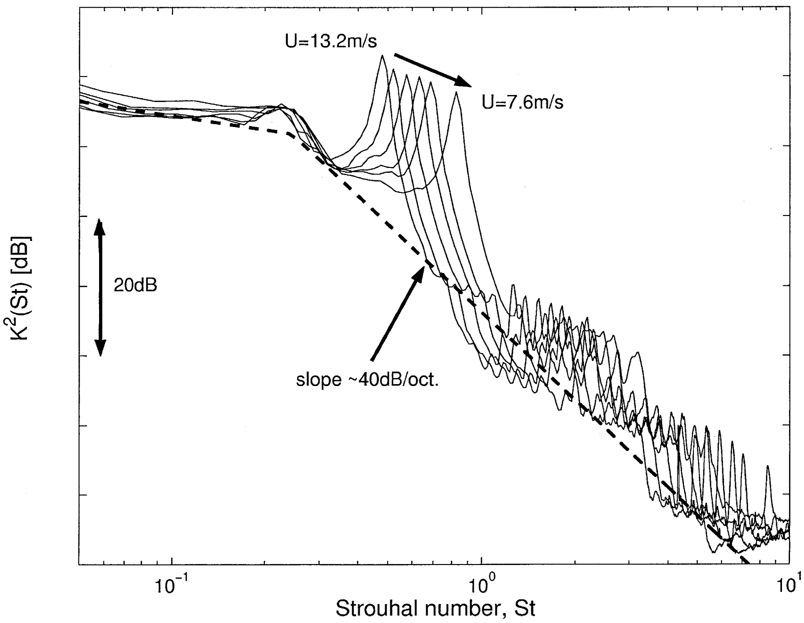

Fig. 5. $K(St)$ spectrum measured at $\alpha = 24°$.

In the present study, $K(St)$ is measured. For that purpose miniature piezoelectric force sensors PCB 209C01 with high sensitivity (500 mV N^{-1}) were mounted at each side of the butterfly valve which was "floating" and only held by two spindles which are constrained by screws and the force transducers. The screw applies a preloading charge to the sensor and blocks the longitudinal translation of the plate. The total fluctuating force experienced by the plate is the sum of the force measured by the two sensors. The results at the median angular position $\alpha = 24°$ are given in Fig. 5. The mean drag has been calculated with Eq. (4) and the following modified contraction coefficient has been introduced to take into account the plate thickness t,

$$C'_C(\alpha) = C_C(\alpha) - \frac{t}{b}\frac{\cos\alpha}{(1-\sin\alpha)}. \tag{7}$$

Each curve exhibits several peaks whose frequency remains constant with velocity variation. These peaks result from bending resonances of the plate and therefore can be ignored. The $K(St)$ spectrum can then be described by two segment lines. In its second part, $K(St)$ decays by 40 dB per decade (30 dB when the results are given by third octave band). These results are consistent with those of Nelson and Morfey.

3.3. Transmission loss

The sound reflection coefficient R at the exit was measured with no flow[1] by the two-microphone random-excitation technique [15]. The effect of the grille has also been investigated but will be discussed in Section 4.1. The recommendations given by Boden and Abom [16] to

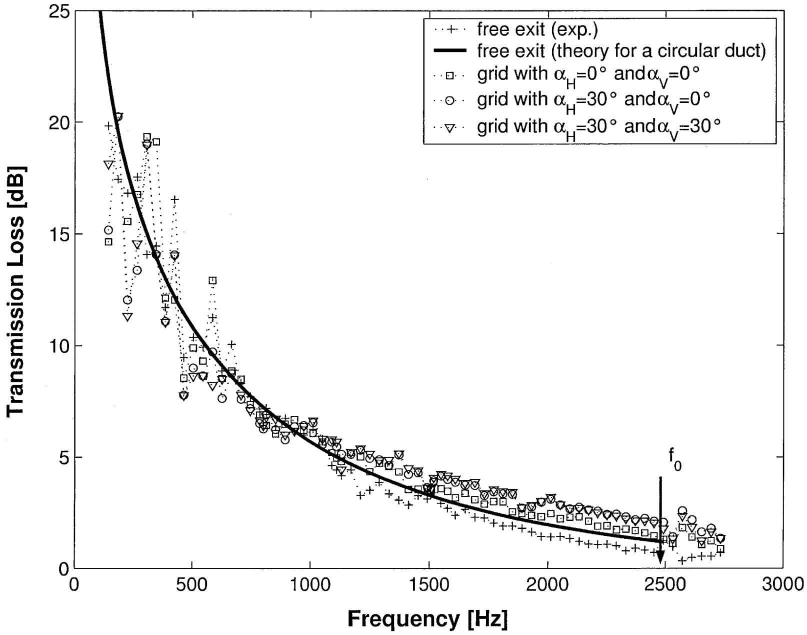

Fig. 6. Transmission loss (no flow).

minimize measurement errors were followed. The separation s between the two wall flush-mounted microphones was taken smaller than half the wavelength of the first transverse mode (i.e. $s<a$) so that the problem to solve was not ill conditioned. In order to improve the measurement accuracy, the separation was changed with the frequency ($s = 60$ mm at low frequency, $s = 30$ mm at high frequency). Moreover, the distance of the first microphone to the exit was fixed at one equivalent diameter; thus the microphones were placed far enough from the exit to minimize any near-field or high-mode effects, or the effect of evanescent waves, but close enough to minimize the dissipative losses between the termination and the microphones. For the relative calibration, the two microphones were flush-mounted on a rigid plate attached to the end of the duct where the acoustic pressure is identical and excited by random noise. Fig. 6 shows the result in terms of transmission loss ($=10\log(1-R^2)$). At low frequencies, for which the reflection coefficient R is close to 1, small errors in the measurement of the modulus of R induce large-amplitude variation of the transmission loss values. But at higher frequencies for which this artefact disappears, the discrepancy between the measurements and the theory for a circular duct of same cross-section area [17] is rather limited (e.g. about 1 dB at 2 kHz). Above $f_0 = 2430$ Hz the method is no longer valid since higher acoustic modes can propagate. Flow is known to modify sound transmission at free duct exits; on the one hand convective effects tend to raise the sound transmission; on the other, a portion of the sound transmitted may be absorbed by the fluctuating vorticity growing in

[1] An attempt was made to measure it with flow using the multi-microphone method described by Peters et al. [14] but this failed for technical reasons.

the shear layers. However, these effects are restricted at low Helmholtz number and are relatively small at low Mach number [9].

3.4. Comparison with measurements

The values of $K(St)$ and the transmission loss found experimentally may now be introduced into Eq. (5); the predictions obtained can be compared to the direct measurements of the sound power level in the reverberant chamber.

The sound pressure level measured at low frequency is amplified by longitudinal acoustic resonances which take place inside the duct. These resonances may occur because the lining mounted on the duct is not efficient at absorbing acoustic waves at low frequency. Apart from that, Fig. 7 shows that the predictions are in good agreement with the experimental results at $\alpha = 20^\circ$ and 30°, but clearly underestimate the level at $\alpha = 10^\circ$ and overestimate it at $\alpha = 40^\circ$. The fact that $K(St)$ has been measured at $\alpha = 24^\circ$ may first explain why the results are better at 20° and 30° than at 10° and 40°. Unfortunately, there were no subsequent opportunities to measure $K(St)$ for various values of α to check if this factor is more strongly dependent of α than assumed initially. The results may be also better around 20–30° because here the drag effects are important but at the same time the flow constriction is not too strong.

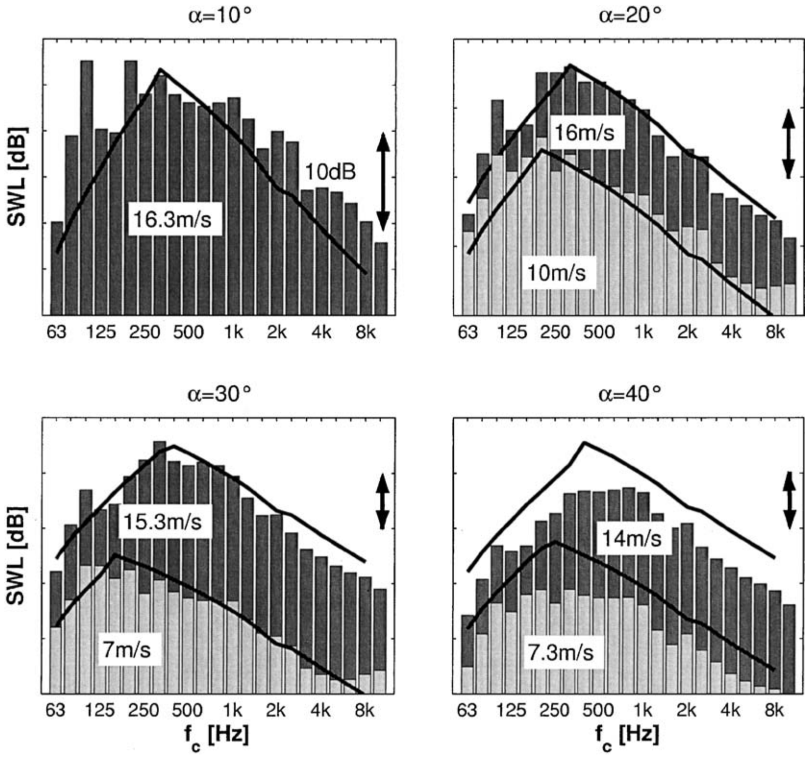

Fig. 7. Butterfly valve noise measured (bars) and predicted (lines).

4. Grille–valve interaction

4.1. Grille acoustics

The problem of sound generation by the grille is complex, combining as it does the difficulty of identifying the correct sound sources in such a complex flow with that of evaluating the effect of the different solid boundaries on the sound propagation (multi-reflection between the deflectors, etc.). It was not an objective of this study to unravel these issues; nevertheless, some measurements of the grille self-noise have been performed. The results may be summarized as follows. When all the deflectors are parallel to mean flow, the noise radiation is dominated by relatively strong tones at frequencies of duct transversal modes. These tones are the result of a coupling between a periodic shedding behind the plates and the transversal acoustic resonances. The velocity dependence of the sound level is higher above the cut-off frequency f_0 than below. Although the grille is located at the exit of the duct, the effect of the duct in cutting off higher modes is still apparent. Fig. 8 shows that the in-duct[2] sound power velocity dependence has values close to those predicted by the Nelson–Morfey theory for a vertical spoiler ($\gamma = 4$ above f_0 and $\gamma = 6$ below it) when the deflectors are angled at 30°.

The deflectors generate self-noise but have the practical advantage of reducing the transmission of the sound waves coming from upstream. In the absence of mean flow, the two-microphone technique was used to quantify this effect in terms of insertion loss.

Fig. 6 shows the values of the transmission loss measured in the three following configurations:

- all the deflectors are parallel to mean flow;
- the horizontal deflectors are turned upwards at an angle $\alpha_H = 30°$;
- both horizontal and vertical deflectors are inclined at 30°.

The insertion loss is calculated by subtracting the results found with no grille from each one of these new measurements. As expected, the grille has a beneficial effect on sound transmission at the exit. When the frequency approaches f_0, sound transmission is reduced by 1 dB when all the deflectors are parallel to mean flow, and by about 1.5 dB when the deflectors are angled at $\alpha_H = \alpha_V = 30°$. Since the wavelength is considerably longer than the characteristic obstruction length of the deflectors, the grille insertion has only a small impact on the plane wave reflection at the exit.

When flow is going through the grille, shear layers are created behind the deflectors. This may absorb sound coming from upstream by conversion into vorticity. The characteristics of these shear layers are likely to be the same as those created beyond the open end of the pipe because of the similar geometric dimensions and velocities. In this case, flow may have some additional sound-absorbing effect limited to very low frequency.

[2]The transmission loss at the exit is added to the sound power radiated outside.

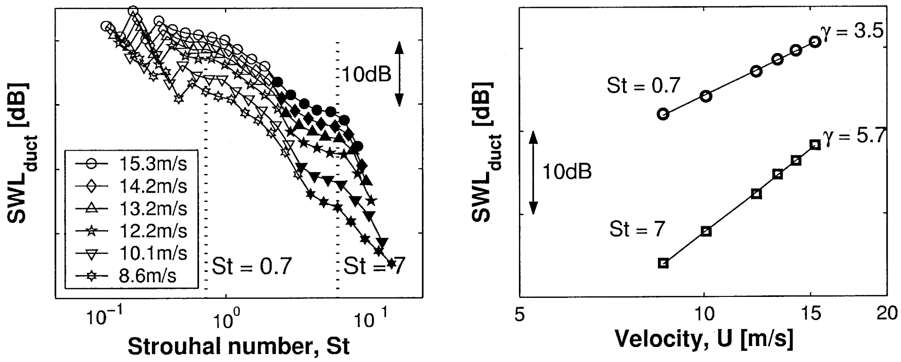

Fig. 8. Grille noise measured at $\alpha_H = 30°$ and $\alpha_V = 0°$.

4.2. Grille–wake interaction noise

When the valve and the grille are mounted in tandem, the impingement of the wake shed from the butterfly valve reinforces the flow unsteadiness around the grille. This increases the noise level as shown by the arrows in Fig. 9 where the total sound power radiated by the entire system is compared to the sum of the sound powers generated by the valve alone and by the grille alone ($\alpha_H = \alpha_V = 0°$). The levels have been added under the assumption that the sources are uncorrelated. A small correction is made on the radiation level from the valve in order to take into account the grille insertion loss.

When the butterfly valve is fully open ($\alpha = 0°$), it creates only weak noise levels and generates only small flow disturbances behind it. As a consequence, grille self-generated noise is the main source of sound. But when the butterfly valve is inclined, the mutual interaction between the valve and the grille (mainly the valve wake impingement on the deflectors) results in a very significant jump of the noise level radiated by the entire system attaining more than 10 dB. It seems, however, that the presence of the grille reduces the level of the longitudinal acoustical resonances occurring at low frequencies. The mechanism pertaining to this sound reduction may be linked to some sound absorption in the wake of the grille as discussed above.

Fig. 10 shows typical spectra measured in the horizontal plane at right angles to the axis of the duct in an anechoic room. The valve was inclined at 24°. The sharp peaks at low frequency are almost harmonically related. They come from the longitudinal acoustical resonances of the portion of duct located between the large muffler and the open exit. The two first peaks at $f_{1,0} = 2500$ Hz and $f_{0,1} = 3400$ Hz (corresponding to the two first higher modes for which half the wavelength is equal to the cross-sectional dimensions $a = 0.07$ m and $b = 0.05$ m) and the two following ones at $f_{2,0} = 4860$ Hz and $f_{2,1} = 5930$ Hz appear only when the grille is present. This last peak is related to the vertical deflectors, since its amplitude and frequency remain constant when the horizontal deflectors are moved but change when the incidence of the vertical deflectors is slightly changed.

4.3. Influence of the grille–valve separation on the total sound radiation

In a context where the wake shed behind the valve interacts with the grille, the spacing between these two elements is obviously a key parameter. The turbulence convected in the wake decays

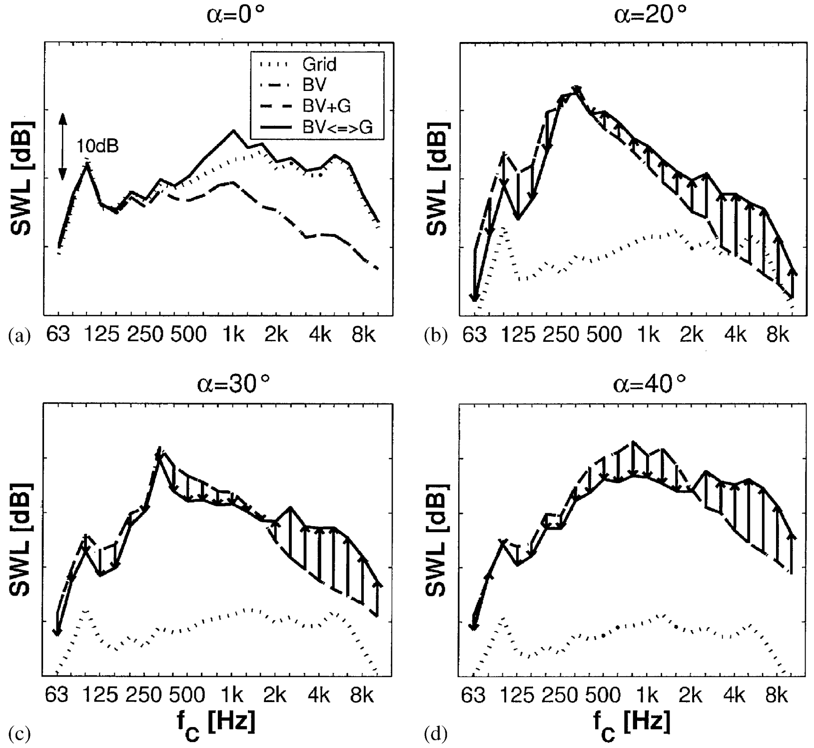

Fig. 9. Grille–valve interaction.

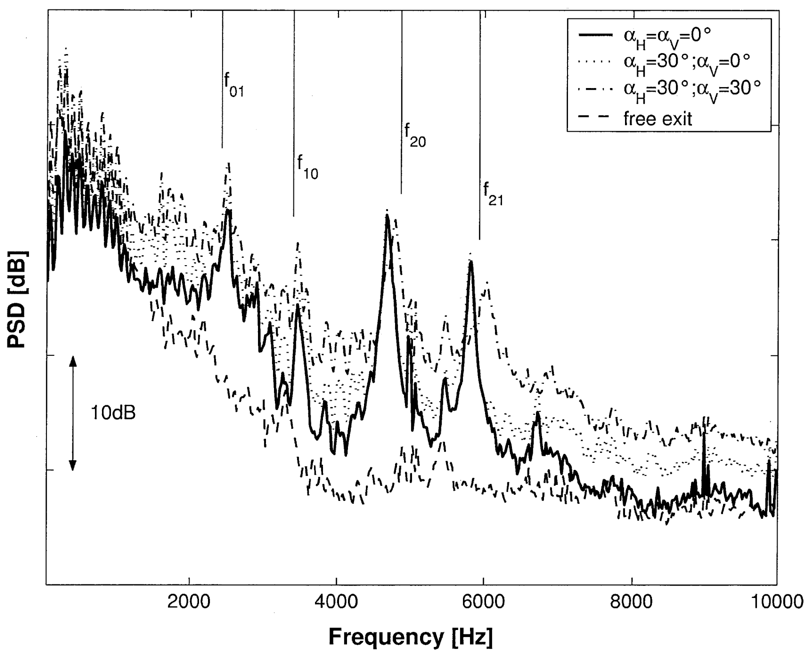

Fig. 10. Sound power spectra measured at right angle of the exit.

away from the valve. Consequently, the further the grille is from the valve, the less unsteady is the flow impinging on the grille and the lower will be the sound level produced by the vent. In the extreme case of a butterfly valve very far from the grille, the flow would have time to recover from being sheared by the valve before arriving at the deflectors. In the absence of flow-duct resonance coupling, these two elements would act as two uncorrelated sources of noise whose total sound level would be the decibel sum of the noise produced by each of them. Fig. 9 shows that 8 dB can potentially be gained by simply increasing the distance between the butterfly valve and the grille. In practice, of course, this separation cannot be infinite. However, flow recovers rapidly behind the valve and it is likely that a substantial sound reduction can be achieved within a few diameters.

Fig. 11 shows that several decibels of noise reduction can actually be attained just by increasing the distance between the valve and the grille by one diameter. Not surprisingly, there is almost no difference when the valve is fully open because the valve produces no intense disturbances in that position. However, at higher angles of incidence, there is a significant gain. An averaged reduction of about 4 dB is obtained at $\alpha = 30^\circ$ at the expense of a slight increase of the sound level at low frequency. Given the lower relative annoyance levels of sounds at these frequencies this solution appears worthwhile.

4.4. Sound directivity pattern

The directivity of the sound pressure has been measured when the butterfly valve is fixed at position ($x = -63.5$ mm, $\alpha = 24^\circ$) and the grille mounted at the exit. Fig. 12

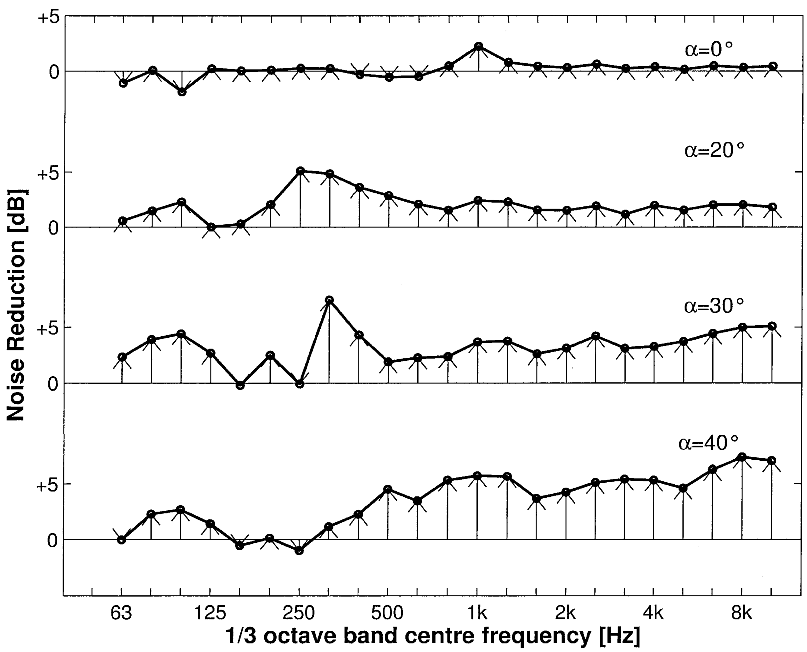

Fig. 11. Effect of increasing the valve–grille distance on the sound radiation ($U \approx 15$ ms).

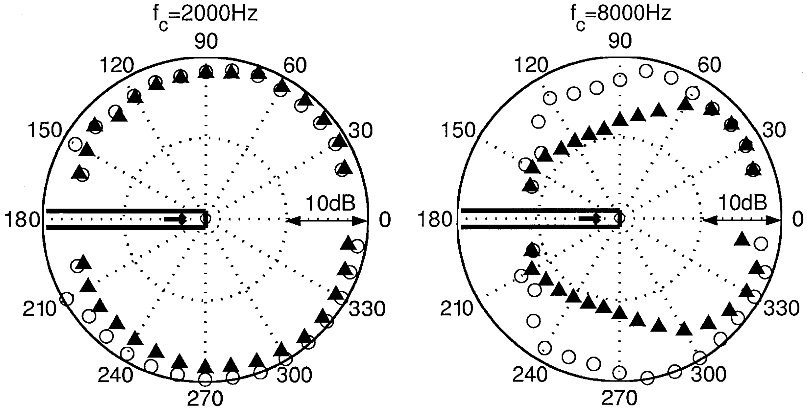

Fig. 12. Sound pressure directivity pattern measured in the horizontal (triangle) and vertical (circle) planes.

presents typical diagrams obtained for the third octave bands at 2000 and 8000 Hz when all the deflectors are parallel to mean flow. Each polar diagram shows the patterns measured in the horizontal and vertical planes every 10° within the range −160° to +160° along a circle of radius 1.5 m.

At $f_c = 2000$ Hz (below f_0) both directivity patterns are nearly identical. Further investigations including measurements with the deflectors angled have shown that the directivity is omnidirectional at low frequency but becomes stronger in the forward arc than in the backward arc as the frequency increases. The same behaviour is observed with no grille; it describes the transmission of plane waves through a duct exit. As shown in Fig. 12, the level measured at 8000 Hz (above f_0) at right angles to the exit is no longer identical in the vertical and horizontal planes when the grille is installed. With no grille, this lack of symmetry is not observed. One explanation for this is based on the assumption that each deflector generates noise perpendicularly to its chord (consistent with a dipole model of the source of sound). Since there is one more horizontal deflector than there are vertical ones, and furthermore they are longer and located just at the exit of the duct, this could explain the greater radiation in the vertical plane than in the horizontal one.

5. Conclusion

Some aspects of the sound generation by ventilation outlet systems have been studied. It has been shown that the Nelson–Morfey theory may be used to characterize, to a first approximation, the behaviour of the whole system. In particular, the duct effect seems to be well accounted for by this theory. An attempt to predict the noise generated by the valve alone has been made. However, more investigation would be needed to know how the ratio $K(St)$ varies with the angle of opening α.

An initial investigation has been made of the grille–valve interaction. Firstly the experiments have shown that the grille reduces sound transmission at the exit with no flow. But this phenomenon is limited in amplitude and there are still questions to be answered about the effect of the grille on sound conversion into vorticity when flow is present. Secondly, we have shown that there is an interaction between the wake shed behind the valve and the deflectors. We have seen that increasing the distance between the valve and the grille is an easy way to reduce the total sound level by several decibels. The investigation of the wake shed behind the butterfly valve and its effect on the noise generation by the grille could be the subject of future research.

Analysis in narrow frequency bands has revealed that the configuration for which the deflectors are parallel to mean flow is conducive to the generation of tones at resonance modes. Inclining the deflectors reduces these tones but at the expense of an increase in the broadband noise.

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