FLOW NOISE GENERATION IN EXPANSION CHAMBERS

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1 INTRODUCTION

One essential consideration[1] in the design of all automotive intake/exhaust systems is the prediction and control of flow generated noise. Each acceptable system must reduce engine breathing noise emission levels to comply with legislation, provide a sound quality that meets customers' expectations[2], while maintaining optimum fuel efficiency and vehicle performance. Similar considerations apply to the design and operational integrity of flow ducts for environmentally acceptable industrial installations, agricultural and construction plant. The diversity of functional requirements, coupled with operational, system layout and space allocation constraints, together with the benefits of rapid prototyping underline the practical advantage of a design methodology[1] that is firmly based on realistic predictive modelling of acoustic and operational performance[3]. Appropriate existing software[3-5] can adequately describe the passive acoustic and resonant behaviour of geometrically complex flow duct systems, such as intakes and exhausts, but does not yet include any influence of flow noise sources distributed along each system on predicted acoustic performance. This deficiency is of practical importance, since such flow generated noise is increasingly the major contributor[1-5], at the higher engine speeds, to exhaust orifice emission spectra above 200 Hz. Currently, predictions of the flow noise contributions[6] must rely on empirical methods, except in some special cases[7,8] where the relevant details of the coupled fluid and acoustic motion have been measured or derived.

Intakes and exhausts include the valves, throttles, branches and junctions essential for flow control and distribution, with filters, silencers, turbochargers and other components for flow conditioning, all connected in sequence by lengths of uniform pipe. Wave reflections at the junctions, combined with any local sound generation[1,3-9] produces an internal acoustic climate comprised of standing and progressive waves excited initially by the pulsating flow through the valves. Over the frequency range of interest, the fluid and wave motion in the pipes remains essentially one-dimensional[3]. However, with the other components, their more complex geometry ensures that the internal fluid motion with its fluctuating acoustic and velocity fields is also complex. The associated expansions, contractions, junctions and the like are often sites where flow noise is generated[3-9]. Such sources can be usefully represented as a flow driven generator, that is highly nonlinear, exciting a lightly damped resonant system, that has an effectively linear acoustic response. The appropriate nonlinear aeroacoustic source mechanism[10] can be represented by a fluctuating Coriolis force acting on the fluctuating flow. This generates flux levels[8] estimated from the Coriolis acceleration $(v\times\Omega)$ at each frequency f,

$$\nabla.H'\overline{(\rho v)'_f} = -\overline{(\rho v)'_f(\Omega\times v)'_f}\ , \quad H = h + 0.5v^2 \qquad (1)$$

where H' is the fluctuating stagnation enthalpy per unit mass, the primes represent fluctuating quantities, ρ is the mass density, p the pressure, while $h = E + p/\rho$ with E the internal energy. One might note that equation (1) is the correct form of equation (10) in reference 7.

Experimental confirmation of this model has been obtained in a few geometrically simple cases[7,8] where the details of the relevant coupled fluid and acoustic motion have been derived or measured. Alternative mechanisms, based on Lighthill's acoustic analogy, have been suggested[6,9] but this model ignores essential detail of the fluid motion in the source region and therefore remains more relevant to empirically based descriptions of the generation of jet, wake and boundary layer noise associated with high flow velocities. Current failure to produce numerical predictions for more complex cases follows from the inability of existing computational fluid dynamic (CFD) or aeroacoustic (CAA) numerical codes to quantify the associated vortical and turbulent fluid motion. Currently, therefore, one must rely on direct measurement of the aeroacoustic sources, as they relate to the associated geometry, local acoustic climate and unsteady fluid motion. However, existing experimentally validated predictive software[1,3-5] does exist that can adequately describe the passive linear acoustic response of complex intake and exhaust systems. The record also demonstrates[3-5] that observed orifice noise spectral levels are usually distributed in accordance with system resonances. These are normally adequately described with one-dimensional linear acoustic modelling, in spite of the fact that the internal fluctuating pressure amplitudes may be several times the acoustic limit.

2 FLOW NOISE MEASUREMENTS

Intake and exhaust orifice noise emission during controlled engine acceleration or deceleration on an engine or chassis dynamometer[3] is usually measured during system or vehicle development to assess their acoustic performance and quality.[2] These measurements also describe the contribution of such emission to the overall noise climate of the vehicle, but they cannot usually provide specific identification of the position and strength of the contributions by flow noise sources distributed along the flow path. These include both new sound generation and the reverberant amplification of sound[5] already travelling through the system. Such identification requires measurement of the acoustic power or intensity propagating along the system at the sequence of relevant sites. Normally this represents a small fraction of the total fluctuating acoustic energy that is trapped in the standing wave fields existing throughout such reverberant systems. One finds that the isentropic fluctuations of acoustic pressure, p_a, and velocity, v_a, that propagate at the speed of sound, c_0, are contaminated by significant rotational or solenoidal disturbances[5] embedded in the flow. These represent turbulent and vortical motion generated at duct boundaries or shed[8] with separating shear layers and wakes. They include fluctuating components of velocity, v_s, and entropy, s, with their associated pressure, p_s, and density, ρ_s. Thus the sound power measurements require appropriate experimental procedures[5] that isolate and identify the acoustic contribution, $p_a(t)$ to the observed fluctuating pressure time history, $p'(t) = p_a(t) + p_s(t)$. Furthermore, as well as such contamination, the systematic axial distribution of acoustic pressure and velocity amplitude associated with the standing waves may result in adverse signal-to-noise ratios for sequences of specific frequencies at corresponding axial positions along each pipe or system component. Direct measurement of either velocity or pressure distributions within flow ducts without introducing further disturbances can present severe problems, unless one takes full advantage of the one-dimensional wave motion present in the pipes. The amplitude of each spectral component of the acoustic pressure p_a and velocity, u_a, is then related to the corresponding amplitude of the incident $\hat{p}^+$ and reflected $\hat{p}^-$ component waves[1] at a mean flow Mach number M by

$$\hat{p}_a = \hat{p}^+ + \hat{p}^-, \; \rho_0 c_0 \hat{u}_a = \hat{p}^+ - \hat{p}^- \qquad \text{(2a,b)}$$

where equation (2a) is always true and equation (2b) remains a close approximation[1], so long as the influence of visco-thermal attenuation remains sufficiently small. The corresponding acoustic intensity, $I = I^+ - I^-$ is then expressed by

$$I = \left|\hat{p}^+\right|^2 (1+M)^2 - \left|\hat{p}^-\right|^2 (1-M)^2] / \rho_0 c_0, \qquad (3)$$

where I^+ and I^- are respectively the incident and reflected components of the spectral acoustic intensity *I.*

2.1 Acoustic Intensity Measurements in Exhausts

Robust procedures yielding reliable estimates of the sound power flux, or of acoustic intensity in such systems under strongly reactive conditions, have been developed at the ISVR[4,5,7,8,11]. These show that narrow band cross-power spectral analysis of the signals acquired simultaneously from a sequence of pairs of flush wall mounted pressure transducers can be processed to evaluate the spectral components $\hat{p}^+$ and $\hat{p}^-$ with the required high precision implied by equation (3). The associated cross-correlation in narrow frequency bands represents a high resolution space-time filter that distinguishes the isentropic acoustic disturbances travelling at the speed of sound, from those travelling with the flow. With selective averaging[11] of the transfer function spectra yielding high coherence, one can recover coherent acoustic spectral pressure components lying 10 dB or more below the corresponding overall pressure signal autospectral level. The methods were successfully applied to pilot laboratory tests of expansion chamber acoustic behaviour[5,7,11] at the ISVR and to similar bench and controlled acceleration tests at Bosal Africa[4,5,12]. The appropriate procedures differed in detail with each application, which included strong excitation by an acoustic driver[5,11] or aeroacoustic excitation by flow alone[5,7] and finally by the exhaust process of a petrol engine installed in a vehicle on a chassis dynamometer[4,5,12].

In all these examples, validation of the procedures adopted for the measurements was demonstrated by the close agreement of the observed orifice emission spectra in free field, with those calculated with the corresponding measurements of the tail pipe intensity. These predicted tail pipe emissions spectra were calculated at each frequency with the previously well validated[11] expression

$$W_r = \frac{|1-\hat{R}|^2}{K} \sigma \frac{[1+(KM)^2/3] \, |\hat{p}^+|^2 \, S}{[1-(KM)^2]^3 (\rho c)_{pipe}}, \qquad (4)$$

where $\hat{R}$ is the pressure reflection coefficient at the orifice[1], $K = c_{pipe}/c_0$, the ratio of sound speed in the pipe to the ambient value, σ is the radiation efficiency[11] of an open orifice with outflow, and S is the cross-sectional area of the pipe. These pilot experiments were performed on a number of simple expansion chambers since they are often found in practice[1-6,9] adjacent to the tailpipe orifice of exhausts. Some laboratory experiments[5,11] concerned strongly acoustically excited chambers with different flow velocities and geometry, producing cases with low and high flow noise. When flow noise remained low[5,11], close agreement was always found between the observed tailpipe and radiated sound power spectra, normalised by the incident power flux in the downpipe and the corresponding predictions with APEX[4,5]. Similar comparisons[5] with the high flow noise system showed clear increases in level in the tail pipe at the chamber and tail pipe resonances, with systematic dips between, indicating reverberant amplification producing both sources and sinks. Comparison (unpublished) with the observed and predicted power flux in the chamber revealed a corresponding behaviour, where the influence of any tail pipe resonances was now absent while that of chamber resonances was also clearly present. Similar comparisons with the engine acceleration measurements[4,5,12] also gave similar clear evidence of sources and sinks produced by reverberant amplification of sound incident on expansion chambers.

2.2 Flow Excited Expansion Chambers

Measurements were also made with the external excitation switched off [5,7], of both the total and coherent power flux spectra. Within the system, there were clear differences of some ten to fifteen dB between their spectral levels[5] except at the first chamber and tail pipe resonances, where they remained equal. There were no such differences in the emitted sound power[5] where the spectral levels corresponded to those calculated with the coherent sound in the tail pipe. Corresponding broad band tones were also clearly audible in the emitted sound. Presumably as a result of selective averaging, such systematic differences between coherent and total sound power spectral levels were not perceptible in the strongly acoustically excited system measurements, where all the corresponding levels were also some 25 dB higher. These observations all indicate that the coherent power flux or intensity spectra observed in the expansion chamber and tail pipe represented the acoustic excitation, while the total power included the contributions by non-acoustic, vortical or turbulent flow disturbances. The close similarity of the total and coherent power spectra with strongly excited systems, also indicated the effectiveness with which contamination of the results by flow disturbances was suppressed. It also adds further credence that any systematic differences between the predicted spectral levels and normalised power flux measurements did represent sound amplification or attenuation by aeroacoustic mechanisms in the strongly excited systems.

Other observations of sound emission spectra from flow excited expansion chambers[6,9] also show the clear presence of tail pipe resonances, and sometimes also include chamber resonances.[13] Parametric studies[6,9,13] of radiation into anechic semi-anechoic[6,9], or reverberant enclosures[13], show that the observed overall sound power was proportional to the mean flow Mach number M, raised to a power lying between 4 and 6 for $M < 0.4$. They also show[9,13] that the overall sound power was a rather complex nonlinear function of the flow path length x. This was either that of the chambers[9], or of the length of the free shear layer[13] between the chamber inlet and exit pipes with diameter, d. The results of a repeat analysis of the matrix of observations, first described in reference 13, are plotted in Figures 1 and 2. The first shows the observed sound power in the reverberation chamber plotted against flow Mach number for a sequence of flow path lengths x. The average value of the index relating power to velocity lies just below 5[11] with a range from 4.3 to 5.3. The results reproduced in Figure 2 are a cross plot from Figure 1, showing the observations as a function of x/d for a sequence of values of M. The results reported elsewhere[6,9] also exhibit a roughly similar behaviour to that described here.

Predictions have also been made with source filter models[3,8] of the excitation by flow of resonators in ducts, where the acoustic components of the fluid motion remain essentially one-dimensional. With a fluctuating velocity source u_s, that is equivalent to an acoustic monopole in free space, a model that adopts equality of acoustic pressure p_s across the source plane with conservation of mass separately on either side of it[3,8], defines an effective source impedance as p_s/u_s. A corresponding expression for the input mobility u_s/f_s associated with a pressure source can be derived[3,8] by representing the force f_s by a pressure differential acting across the source plane with continuity of velocity through it. Calculations of the observed sound emission spectra with calculated spectral distribution based on the calculated source mobility at the chamber tail pipe junction are presented in Figure 3. Here each observed tail pipe and chamber resonance has been identified with the corresponding predicted tail pipe resonances. The calculated levels included an appropriate scaling[7,8,13] to account for the estimated strength and space/time distribution of the pressure field associated with each travelling vortex indicated in the inset on the figure. The mobility spectrum was calculated with an appropriate version of APEX.

3 DISCUSSION

Pilot studies[3,5] have established and validated robust and sufficiently precise experimental technology for establishing the position, strength and spectral characteristics of flow noise sources and their orifice emissions in relation to the boundary geometry, the time averaged flow and the associated acoustic climate. This includes the influence of regenerative amplification[4,5] by acoustic feedback on source characteristics, with some basic examples[7,8] of source modelling. The record shows that problems arising from poor signal-to-noise ratio associated with the presence of standing waves and also from signal contamination by turbulent pressure fluctuations, can be overcome[4,5] by adopting swept sine or swept periodic excitation, combined with appropriate selective averaging. With uncontrolled excitation by flow alone, coherent power flux measurements[5,7] were closely identified with the acoustic components of the total power flux and with the corresponding orifice emissions. The measurements also demonstrated that in appropriate circumstances, data reduction based on one-dimensional linear acoustic models, can provide realistic descriptions of the acoustic characteristics of cyclically excited exhaust systems and their elements, with sound pressure levels well in excess of 170 dB. Acoustic characteristics calculated with the linear acoustic models, such as those adopted in APEX, remained always in good agreement with observations.

Current studies in predictive modelling of aeroacoustic sources are concerned with establishing, by appropriate measurements, the essential details of the coupled acoustic, vortical and turbulent motion associated with flow noise sources. A first step concerns representative sets of parametric studies of the combined influence of flow, geometry and acoustic behaviour and environment on flow induced noise, or on the reverberant amplification of sound propagating along flow ducts. These will then be combined to produce maps describing the measured or calculated distribution of the fluctuation potential and vortical components with the corresponding Coriolis accelerations. These can then be processed to quantify the associated aeroacoustic sources. To accomplish such measurements without introducing further contaminating sources also represents a current challenge, as does the subsequent development of appropriate numerical models for inclusion in future predictive codes.

4 REFERENCES

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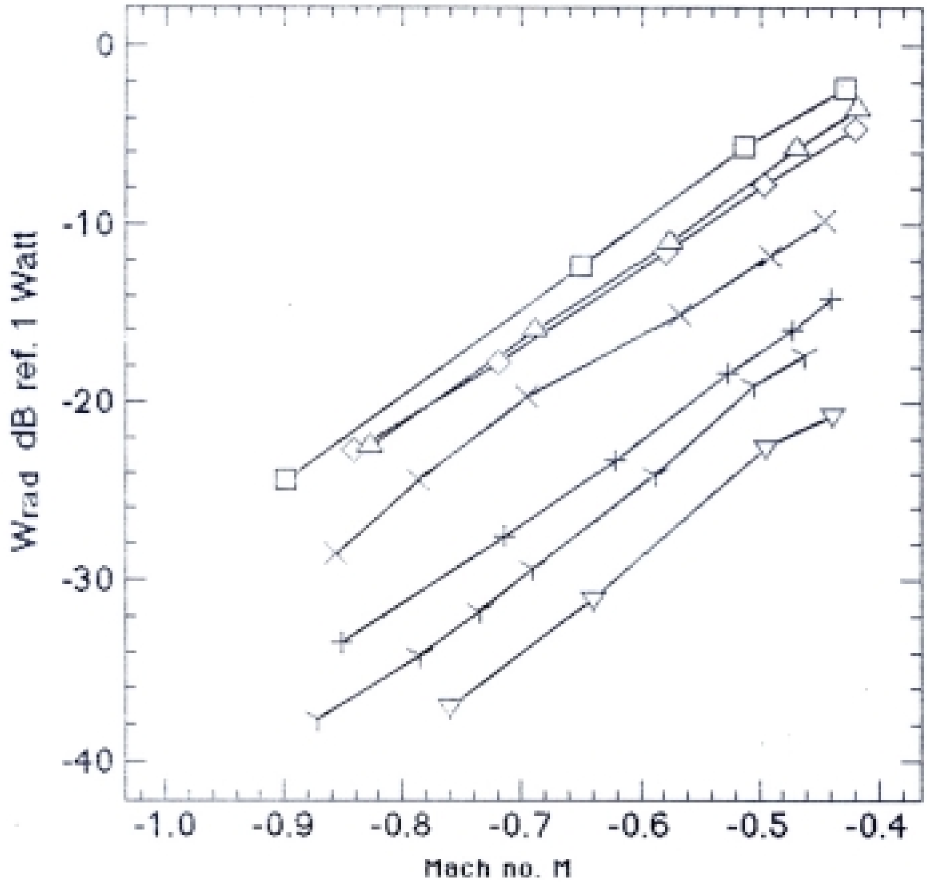

Figure 1 Sound power emitted by a flow excited simple expansion chamber.
Chamber / pipe diameter ratio D / d = 3, flow path length x.
x / d = 3, ; x / d = 5, Δ; x / d = 7, ◊; x / d = 8, $\times$; x / d = 10, +; x / d = 12, Y; x / d = 15, ∇

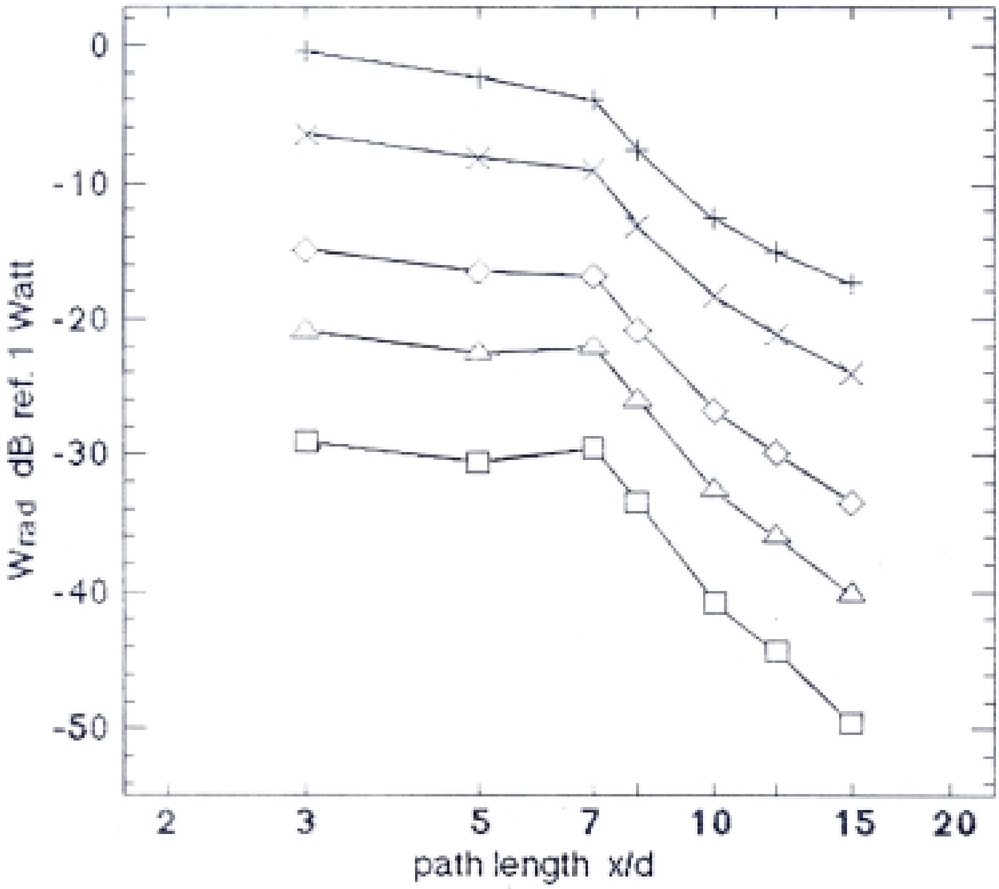

Figure 2 Sound power emitted by a flow excited simple expansion chamber.
Chamber / pipe diameter ratio D / d = 3, flow path length x.
M = 0.1, ; M = 0.15, Δ; M = 0.2, ◊; M = 0.3, ×; M = 0.4, +

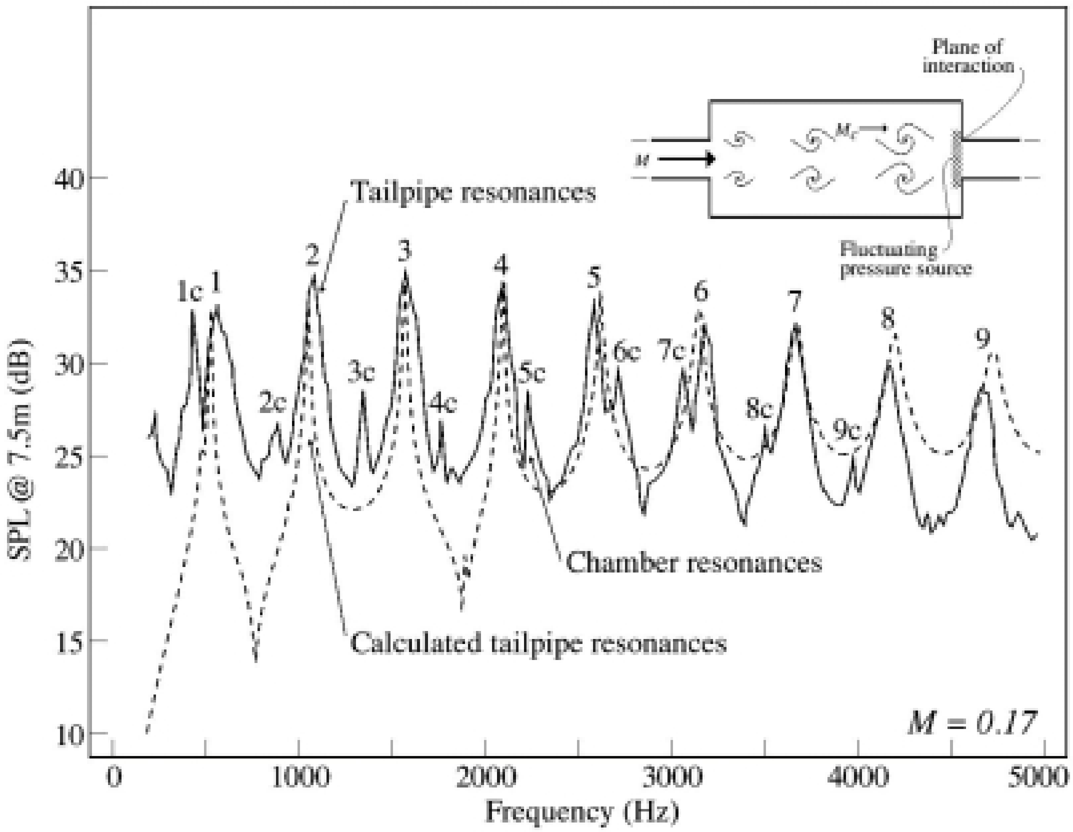

Figure 3 Sound emission from a flow excited expansion chamber —, measured; – –, calculated