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USE OF LINEAR MODELS FOR FLOW DUCT DESIGN

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1. Introduction

This contribution reviews progress in the application of linear acoustic models to the design of exhaust systems for internal combustion engines. Exhaust noise control, environmental factors and energy conservation all require consideration in developing an acceptable flow duct design.

2. Determination of acoustic energy transport in flow ducts

Sound propagation in flow ducts can be described by linear transmission line equations, which specify the variation of acoustic pressure and particle velocity associated with the wave motion, in terms of position in the duct. In general, one-dimensional descriptions suffice, though conservation of mass, energy and momentum at area discontinuities require the inclusion of higher order modes to match the boundary conditions (1,2).

With exhaust systems, evaluation of the source characteristics is difficult, so it is simpler to refer analysis to the exhaust outlet plane, where conditions can be measured and thus specified. The acoustic transfer characteristics of each element of the system can then be calculated working towards the source.

The total radiated power W_r can be measured using free field microphone measurements made at a distance r from the exhaust outlet. Acoustic conditions at the exhaust outlet plane, reference zero, can then be specified by equating the nett acoustic energy flux there W_o to W_r and W_o is obtained (1) from

$$W_o = \frac{S_o}{\rho_o c_o} \{(1+M_o)^2|p_o^+|^2-(1-M_o{}^2)|p_o^-|^2\}, \qquad (1)$$

where S_o is the outlet pipe area (πa_o^2 for a round pipe), $\rho_o c_o$ the specific impedance of the exhaust gas, M_o the mean flow Mach number, p_o^+ and p_o^- are respectively the wave components moving with and against the flow. The wave components are also (1) related by

$$p_o^- = Re^{i\phi}p_o^+ , \qquad (2)$$

where R is the reflection coefficient, a known function of both ka and M and ϕ the relative phase is a known function of k. The wavenumber $k = \omega/c_o$, ω being the radian frequency. The wavenumber associated with p^+ is $k/(1+M)$ and that with p^- is $k/(1-M)$.

A set of appropriate linear models for calculating the transfer characteristics of system elements can be found in references 1 and 2 so will not be repeated here. The transfer characteristics between any point x and the system outlet can be readily calculated by working systematically along the system from the outlet to x, to evaluate p_x^+ and p_x^-. The nett acoustic energy flux at x, W_x

provides one such characteristic value and can be evaluated from (1) with the subscript x substituted for the subscript zero. The local pressure p_x and velocity v_x can be evaluated (2) by

$$p_x = (1+M)p_x^+ + (1-M)p_x^- , \tag{3}$$

and

$$v_x = \{(1+M)p_x^+ - (1-M)p_x^-\}/\rho_x c_x . \tag{4}$$

The load impedance Z_x offered by the system at x is then simply

$$Z_x = p_x/v_x , \tag{5}$$

while the effective volume velocity at x, V_x is given by

$$V_x = W_x/\{Z_x\}_R , \tag{6}$$

where $\{Z_x\}_R$ is the real part of Z_x.

3. Design procedures

A systematic procedure can be usefully subdivided into a sequence of steps, set out below. Conditions in the exhaust duct include high fluctuating pressures with an amplitude around 0.5 bar, mean flow velocities as high as Mach 0.3, gas temperatures from 200 to 800 degrees centigrade, with substantial axial and radial temperature gradients.

3.1 Measurement and characterization of the acoustic source

Measurement of the engine noise signature should cover the complete load and speed range of the engine. A common practice is to measure the sound radiated close to the outlet of a standard exhaust system (usually a uniform pipe) with the engine running on a test bed. Any exhaust system presents an acoustic load, so its characteristics must be assessed when evaluating the measurements. The radiated sound field can possess directional properties, which should be identified. Exhaust gas temperatures, flow rates and gas composition must also be determined to define conditions through the exhaust system.

The measurements can be analysed to provide an exhaust driving signal at some reference plane, say the manifold outlet flange or turbocharger outlet flange. Repeating the measurements with two or more acoustic loads (pipes of different length say), provides information on the effective source impedance and on the sensitivity of the driving signal to changes in acoustic load.

3.2. Specification of design aim

A satisfactory design must control noise emission to within legislative limites, fit within the space available for the installation and satisfy engine or vehicle performance requirements. Acoustic performance can be specified in terms of an insertion loss, I.L., defined as

$$I.L. = 10 \log_{10}(W_s/W_u)_r, \quad (7)$$

where the subscript s refers to "silenced" and u to "unsilenced" total radiated noise power. The effect on engine performance can be specified in terms of mean system back pressure, which can be related to smoke emission, nett engine power output and fuel economy.

3.3 Design layout

The number, size and arrangement of silencing elements can be deduced from the overall unsilenced noise spectrum and the required insertion loss, taking into consideration the geometric constraints offered by the space available. In general terms acoustic performance increases with the system back pressure, while back pressure and acoustic energy transport, for a given geometry, both increase with mean flow Mach number, while its value depends on the cross-sectional area of the exhaust pipe. The high performance silencer design layout thus involves effecting some optimum compromise between acoustic performance and permitted or desired system back pressure.

3.4 Performance prediction

The performance of a given design is assessed by calculating the acoustic characteristics as outlined in Section 2, making detail changes to the design and then reassessing performance until the design aim is approached. Flow losses are assessed at the same time to determine the mean back pressure. This process is normally restricted to elements whose acoustic behaviour is adequately represented by linear models. Care is necessary, for example, to avoid flow separations at area discontinuities, which can provide resonant enhancement of 20 dB or more by edge tones or vortex shedding.

The consequences of interactions between the source (engine) and the system as they may affect performance must also be assessed. One can assume for example that the source maintains a high effective impedance so the source volume velocity remains constant. Though this is not always the case, appropriate indications for source modelling should have been obtained during the measurements under 3.1 above.

4. Conclusions

The use of linear modelling for acoustic performance predictions has been criticised on the grounds that the results are unreliable. The reasons for this, however, appear to lie more with neglect of the source characteristics and of the presence of the mean flow, also with neglect of temperature gradients and of flow generated noise, rather than with inadequacies in the modelling. Since flow conditions vary widely with changes of speed and load, an assessment of performance must be made for flow and temperature conditions covering the full range observed in the measurements. All this involves is a change to the flow details input to a computer programme which makes due allowance for temperature gradients and mean flow. Flow noise generation must be minimised by care in setting out details of the design.

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The variable gas conditions and the inadequacies in the specification of the source characteristics result in uncertainties in the acoustic performance predictions, which can only be resolved by testing a prototype. Experience has shown that two or three prototype systems, involving changes in design details may be necessary, before a satisfactory performance is eventually attained. This effort should be compared with 30 to 50 trial systems that are otherwise made and tested before a satisfactory performance is obtained using a purely empirical approach to design.

References

(1) P.O.A.L. DAVIES and R. ALFREDSON 1971 Proc. I. Mech. E. Symposium on Vibration and noise in vehicles, 17-23. Design of silencers for internal combustion engine exhaust systems.

(2) P.O.A.L. DAVIES 1978 Proc. of United States Environmental protection agency conference EPA 550/9-78-205, 5-48. Bench test procedures and exhaust system performance prediction.