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I.C. ENGINE INTAKE AND EXHAUST NOISE ASSESSMENT

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The results of current studies of intake and exhaust noise from piston engines are described in the context of intake and exhaust system acoustic design. The objectives combine the achievement of sound emission targets with the maintenance of optimum engine performance and fuel efficiency throughout a specified range of operating conditions. Practical experience demonstrates that the relevant technology includes a quantitative evaluation of those factors directly concerned with engine operation and breathing, those factors influencing the excitation and propagation of pressure waves through the relevant system and those factors controlling the emission of sound to the surrounding environment. A summary is provided of previous and current developments in appropriate existing numerical codes for performing the detailed calculations and associated performance assessments with increasing realism.

1. INTRODUCTION

Public concern for the environment, with the marketing advantage conferred by rapid development and manufacture of high quality products, has given a major impetus to continuous developments of design technology applied to the control of vehicle drive-by and interior noise. Intake and exhaust system design includes a systematic tuning of the components that control or influence sound emission, with an optimized matching of these to the engine operational and breathing characteristics that influence its pollutant emission, performance and fuel economy. This process takes place within a general vehicle development programme, being subject to other vehicle design and styling decisions that affect intake/exhaust system layout and space allocation. The diversity of function, coupled with such constraints underline the practical advantage of a design methodology [1] that is firmly based on realistic predictive numerical modelling [1–4] of the relevant operational, gas dynamic and acoustic characteristics of the intake and exhaust in relation to its geometry. Such software should offer substantial advantages in design office time and resources, facilitate rapid optimization of design detail and thus yield significant reductions in lead time to production. To this end it is clearly desirable that the results of each calculation

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should be open to clear interpretation during system design and development. The fact that individual elements normally interact dynamically means that their individual contributions to the overall system performance should be clearly identified [1, 2], rather than their behaviour in isolation. Furthermore, the software should provide sufficient interactive facilities to provide physical insight into the relations between gas dynamic or aeroacoustic behaviour and the geometry or other relevant features of the system, or of its individual elements, as well as the consequences of any changes in detail. This is essential [1, 2] both to guide the optimization process, and to build up experience and thus develop expertise. Finally, as well as objectively meeting legislative specifications, the resultant sound emissions will normally also be assessed subjectively, though this topic is not considered further here.

2. I.C. ENGINE BREATHING NOISE EMISSIONS

Sound associated with engine breathing is normally radiated from the open orifices of intake and exhaust systems and is then known as orifice or tailpipe noise. It may also be radiated by their external surfaces when these are excited by the internal fluctuating pressure field (shell noise), although this is not considered further here. Orifice noise forms one component of the total A weighted vehicle noise measured during drive-by testing to establish compliance with specific legislative limits. Orifice noise may also contribute to vehicle interior noise by exciting acoustic body space resonances.

Appropriate assessments of such contributions to the vehicle noise climate by orifice or tailpipe noise should include the influence of both engine load and speed, as well as that of the related gas temperature distribution and corresponding mass flow throughout the systems, since these last two factors have a controlling influence on both system acoustic performance and on the strength and spectral characteristics of the sources of acoustic excitation [1–4]. Thus, practical orifice noise evaluation measurements are currently made during controlled engine acceleration and run down on an appropriate test bed. Alternatively, the relative acoustic performance of different designs may also be assessed [1, 2] with acoustic excitation on a cold flow test bench, although appropriate scaling of the results to represent operational conditions may present problems [1, 4]. Test bed acceleration results may be recorded as a Campbell diagram, as in Figure 1(a), which is a waterfall plot of a sequence of narrow band spectra at regular rpm intervals. Here, the straight lines to the left can be identified as order noise, that is the narrow band components or tones that are harmonically related to the engine firing frequency, normally expressed as multiples of the rotational frequency. These are generated by the cyclicly pulsating flow through the valves and labelled as primary sources in the schematic Figure 1(b). The more extensive broadband spectral components in Figure 1(a) represent flow noise, and arise from turbulence and other flow induced aeroacoustic sources associated with shear layers, vortex generation and its coupling with resonant acoustic feedback. These are identified in Figure 1(b) as secondary sources distributed along each system. Flow noise, which varies

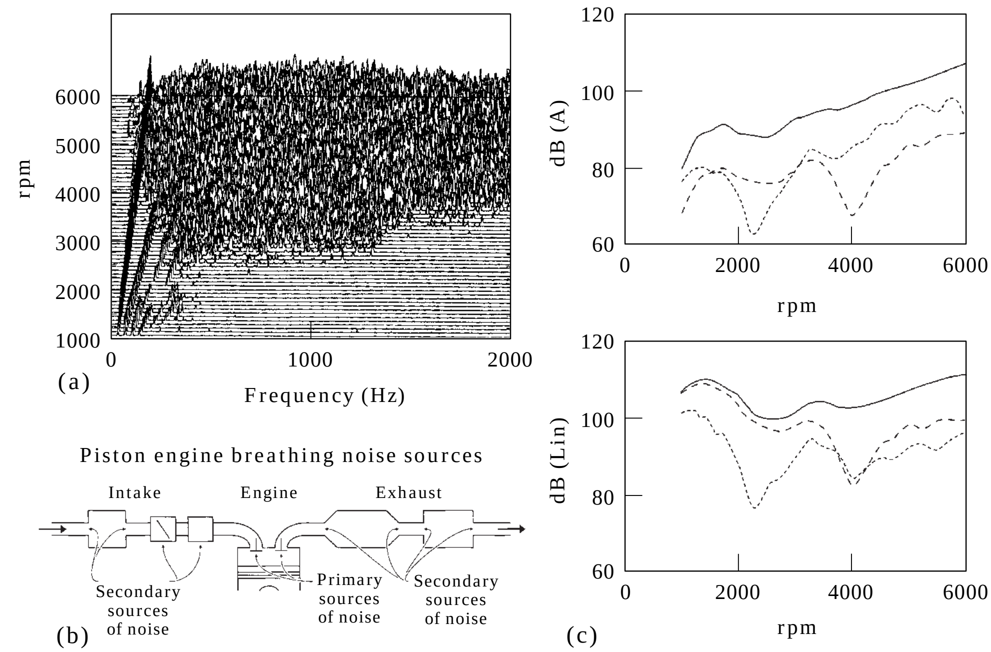

Figure 1. Exhaust tailpipe emissions from a 4-cylinder 4-stroke engine. (a) Campbell diagram; (b) system schematic; (c) components measured at 0·25 m and 45° from orifice; ————, overall; ---------, 2nd order; ...., 4th order.

roughly with the fourth power of the mass flow [1, 5], normally provides a major contribution to tailpipe sound power emissions above say 300 Hz.

The test bed data have been replotted in Figure 1(c) to separate the relative contributions by order noise and by flow noise to drive-by and interior noise. Thus, the upper figure presents measurements of A weighted orifice noise emissions, identifying the overall, 2nd and 4th order noise contributions. Since perception of loudness or noise nuisance correlates well with A weighted *SPL*, this presents the data in a form that is relevant for assessments of vehicle drive-by noise. These results reveal that the more significant contributions by orifice noise to engine drive-by levels are provided by flow noise in this particular example. Since the acoustic excitation of the body surfaces depends on the spectral distribution of the adjacent fluctuative pressure, the corresponding unweighted noise level plots in the lower figure suggest that 2nd order noise at the lower rpm may be of more significance for interior noise (body boom) by exciting internal acoustic modes (resonances) at the corresponding frequencies.

2.1. OVERALL ACOUSTIC, BEHAVIOUR OF INTAKE AND EXHAUST SYSTEMS

As indicated in Figure 1(b), intake and exhaust system geometry normally consists of a sequence of silencing elements (or mufflers), with further essential elements such as air filters, catalysts, manifold plenums, junctions and so on connected to each other by lengths of uniform pipe. In general terms [1, 2], the propagation of sound energy from the sources to the terminating orifices is governed by the system resonances which, in turn, depend on the coupled (cascaded) resonant behaviour of the individual elements. All such resonances can be linearly related to the effective lengths of the wave paths through each element as modified by the boundary conditions corresponding to transmission, reflection and dissipation of acoustic energy at the interfaces between them.

For example, in its simplest form, an exhaust system can be represented by a uniform pipe of length L closed at the valves and open at the other end. The acoustic effect of the pulsating flow through the valves can then be represented by an oscillating piston. With sound speed c_o the resonance frequencies f_r of this system are given by

$$f_r = nc_o/4L; \; n = 1, 3, 5, \text{etc.} \tag{1}$$

Such behaviour has been recognized from time immemorial as the basic physical mechanism controlling the pitch of primitive musical wind instruments, such as the Pan pipe or Peruvian flute where, in this case, the acoustic excitation is produced by blowing across the open end. In contrast, the transverse flute, recorder, whistle, or organ flue pipe representing pipes open at both ends, have resonance frequencies given by

$$f_r = mc_o/2L, m = 1, 2, 3, \text{etc.} \tag{2}$$

Alternatively, the first example may be recognized as a "quarter wave" acoustic resonator and the latter as a "half wave" one. Furthermore, anti-resonances exist between the resonances, while the efficiency of acoustic energy propagation is highest at the resonances and lowest at the anti-resonances. Such characteristic behaviour implies that the acoustic performance of intake and exhaust systems depends, among other factors, on the connecting pipe lengths with the relative positioning of their individual silencing and other elements along the system, as well as on the distinct acoustic properties of each component.

2.2. INTAKE AND EXHAUST SYSTEM MODELLING

The relevant design technology for intake and exhaust systems should take due account of all the functions performed by each. Thus, the intake system is designed to deliver an appropriately controlled supply of cool filtered air to the engine, while further functions include dynamic performance tuning as well as control of intake noise emissions. The exhaust system is designed to conduct exhaust gas away from the vehicle, to reduce polluting emissions, to minimize the influence of back pressure and cross scavenging on engine performance as well as controlling the orifice noise to within specified limits. Current specifications include a catalyst combined with exhaust gas recirculation for emissions control and may require a turbocharger to increase engine power output. Contemporary design technology incorporates appropriately realistic predictive modelling [1–4] of the relevant operational, gas dynamic and acoustic characteristics of the intake and exhaust system and their constituent elements in an integrated procedure that calculates the cyclic thermodynamic and gas dynamic processes in the cylinder with the corresponding flows through the valves, and then the associated wave motion throughout the manifolds and remainder of the intake and exhaust system, corresponding to all conditions of engine operation.

As indicated in Figure 1(b) the cylinder, pistons, valves and manifolds etc. represent the time varying part of the integrated system while the remainder of the intake and exhaust acts mainly as an acoustic or wave filter [1–5]. One notes that engine operations with the associated wave motion are cyclic, so that time histories are readily transformed to provide spectral descriptions in the frequency domain, or the converse. The analysis of the thermodynamic and non-linear gas dynamic processes concerned with the associated wave action in the time varying components of the running engine [1–5] is best performed in time, with appropriately representative boundary conditions imposed by the passive parts of the systems. The corresponding analysis describing wave energy propagation in the remaining "passive" system components that are responsible for the acoustic filter performance adopted for controlling orifice noise, can be performed either in the frequency domain as part of a hybrid approach [1, 4–6] or in time [2, 3], accompanied by acoustic bench test validations [2, 7]. The references cited, together with the further lists of references they present, provide a representative selection from the extensive literature describing well established and documented methods for evaluating the cyclic thermodynamic and gas dynamic processes associated with engine operation and providing the corresponding temporal or spectral descriptions of wave energy propagation in intake and exhaust.

2.3. SOURCE SYSTEM OR SOURCE FILTER MODELLING

Over the frequency range of most interest, the transverse dimensions of the system elements normally remain a small fraction of the wavelength. In such cases one finds [1–7] that one-dimensional descriptions of wave propagation, with those of the associated fluctuating pressure and velocity, remain in close agreement with observations. Similarly, acoustic theory also remains appropriate so long as the lengths of the transmission paths remain sufficiently short in terms of the wavelength and fluctuating pressure amplitude remains below some corresponding limiting value [5]. Although its amplitude in the intake and exhaust manifold normally exceeds the acoustic limit by at least an order of magnitude, acoustic models may still be adequate [1, 4] with interfering wave motions in pipes less than half a wavelength long.

Practical experience has established [1, 2, 4] that the transfer of acoustic energy from the source to the system is strongly influenced by the acoustic impedance or load that the system presents to the source. Realistic descriptions of the acoustic behaviour illustrated in Figure 1 can be provided by adopting source/system or source/acoustic filter models to describe this process and then the propagation of sound energy to the open orifice, and finally its radiation to the surroundings.

Normally the acoustic characteristics of the source and of the system it excites are both functions of frequency, so the discussion that follows applies to each propagating spectral component of the fluctuating pressure. Two representative acoustic circuit models are included in Figure 2. In practical applications [4] the

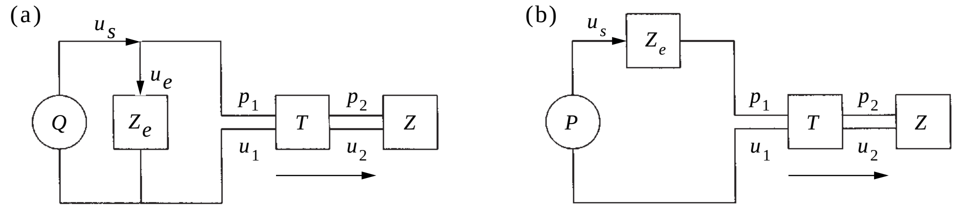

Figure 2. Acoustic circuits for intake/exhaust. Excitation by (a) fluctuating mass or volume velocity, (b) fluctuating pressure or aerodynamic force.

acoustic properties of the filter correspond to the transfer element T combined with the termination element Z, which represents the radiation impedance p_2/u_2 at the orifice. Appropriate expressions for evaluating Z and estimating the corresponding sound emission with hot gas outflow can be found in the Appendix to reference [1].

The source in Figure 2(a) represents one spectral component of the cyclicly fluctuating flow $Q(t)$ through the valves with an associated effective shunt impedance Z_e, while the corresponding input impedance of the filter at the source plane $Z_1 = p_1/u_1$. One notes that Z_e includes among other factors the influence of the local acoustic environment on sound emission at the source. One notes too that this current model represents continuity of acoustic pressure across the source/system interface with the source strength equated to the discontinuity in particle velocity $u_e = u_s - u_1 = p_1/Z_e$. Thus, it appears equivalent to a point acoustic monopole in free space. The acoustic power output of this source is expressed by

$$W_m = 0{\cdot}5\ \mathrm{Re}\{p_1^* Q\} = 0{\cdot}5\,|\,Q^2\,|\,\mathrm{Re}\{Z_1/(1 + Z_1/Z_e)\}/S_s, \qquad \text{(3a b)}$$

where Re{ } denotes the real part, the asterisk represents the complex conjugate value and S_s is the effective area at the source.

Alternatively, where the source is associated with fluctuating boundary stresses, the circuit model Figure 2(b) represents continuity of particle velocity u_s across the source plane with a source of strength $\partial f_s/\partial x$ equated to the discontinuity of pressure $(P - p_1)$ acting over the area S_s, or $(P - p_1)\ S_s$, which appears equivalent to a dipole in free space. The acoustic power output of this source is expressed by

$$W_d = 0{\cdot}5\ \mathrm{Re}\{f_s^* u_s\} = 0{\cdot}5\,|f_s^2\,|\,\mathrm{Re}\{M_I\}, \qquad \text{(4a, b)}$$

where the input mobility $M_I = u_s/f_s = 1/Z_e S_s$, with $Z_e = (P - p_1)/u_s = P/u_s - Z_1$. Here again, one notes that the series impedance Z_e includes the influence of the local acoustic environment. Analogous models [1, 5] can be derived for other relevant source types, combinations, or distributions of fluctuating mass injection or shear stresses. One notes too that the transfer of acoustic power from the source to the system is, among other factors, directly influenced by the system impedance Z_1 at the source plane, which depends on the acoustic properties of the filter represented by the transfer element T with its load impedance Z.

3. PREDICTIVE CALCULATION OF ORIFICE ORDER NOISE

Currently the propagation of wave or sound energy through the intake and exhaust system is sufficiently well understood to ensure that the influence of intake and exhaust system tuning on engine performance can usually be predicted with sufficient realism [1–4, 6]. However, practical experience suggests that both the optimization of acoustic filter performance and similarly effective predictions of the resulting control on orifice noise emissions currently relies [1] on some form of acoustic modelling. Corresponding predictive models giving the relevant quantitative descriptions of the sources of excitation, or wave generation, have not yet reached a similar stage of development [5]. These have been classified in Figure 1 as primary sources, associated with the cyclic flow through the valves, and as secondary sources associated with flow separation and vortex generation at junctions, expansions and the like. In both cases they represent transfers of mean or cyclically pulsating flow energy to wave energy. The local dynamic characteristics of the relevant parts of the duct systems also have a direct influence on both the efficiency and extent of such energy transfer [1], including the forced dynamic response to the pulsating flow and wave amplification by flow excited resonance and acoustic feedback [5, 8], or flow noise. Thus, one finds that the associated noise emissions from intake and exhaust may be controlled or modified [1–4] by adopting appropriate geometric or acoustic design features, either to reduce the net energy transfer from the sources, or to attenuate selectively those wave components that make significant contributions to the overall sound emission. On the other hand, intake and exhaust design strategies that simply adopt measured source characteristics [9] combined with predictive acoustic modelling, seem less likely [1] either to maintain engine performance and fuel efficiency, or to produce reliable orifice noise predictions as a consequence of their neglect of source system interactions expressed by equations (3) and (4).

Contemporary intake/exhaust system technology adopts suites of predictive software that first evaluates the cyclic thermodynamic and gas dynamic processes in the engine cylinder with the associated valve flow and then produce estimates of the associated cyclic wave motion in the manifolds, intake and exhaust systems, and finally predict the orifice noise emissions that are harmonically related to the engine firing frequency (order noise). The calculations are repeated at a sequence of engine speeds and loads to correspond to the measurements of noise emission included in Figure 1.

3.1. THE PROPAGATION OF WAVE ENERGY

In principle, fully representative modelling of the wave motion requires a complete analytic description of the corresponding mean and fluctuating fluid motion throughout the system: that is, the equations of mass, momentum and energy transport, with those defining the axial and radial distributions of relevant fluid properties, the geometry of each component element or pipe and the boundary conditions over each connecting interface. At each area or other discontinuity along the flow path a proportion of the incoming wave energy is reflected, a fraction is dissipated, or perhaps enhanced by resonant feedback, while the resulting balance is transmitted. The relative proportions may be calculated by matching the associated fluid motion over appropriate surfaces on either side of the discontinuity that, with any relevant fixed boundaries, form a control volume. A similar procedure may be adopted for each of the uniform pipes, or in an appropriate sequence for each successive element or component of the intake or exhaust. The results then define a systematic set of representative boundary conditions for each component along the flow path from the sources of excitation to the open termination. Thus the wave motion consists of sets of progressive and interfering waves. One notes too that at any position and instant of time, the incident wave has a propagation speed of $c + u$ and the reflected one a speed of $c - u$, where c and u are respectively the corresponding acoustic and flow velocities that are normally functions of both time and axial position x. The values of the controlling physical variables and fluid properties such as density ρ, pressure p, velocity u, kinematic viscosity ν, thermal conductivity k_T, ratio of specific heats γ and so on all depend, among other factors, on the local absolute temperature T of the flow. In particular, so does the speed of propagation of the waves relative to the gas, according to $c = (\gamma RT)^{0\cdot5}$, where R is the specific gas constant.

The generic systems of equations is multi-dimensional and non-linear [10] so that, for components with complex geometry in particular, finite element methods seem ideally suited to their solution. However, the human and computing resources necessary to create and run finite element models preclude their use in design procedures requiring a rapid and systematic modification of detail in an iterative approach to an optimum compromise solution. At sufficiently low frequencies, where only plane waves can freely propagate along uniform pipes, the simplifying restriction to one-dimensional axial motion [1–7] normally provides realistic solutions. This restriction corresponds to an upper frequency limit of around 1·5 kHz for large commercial vehicles and 3 kHz for passenger cars. In most practical applications this is more than adequate, since the record shows that discrepancies between observed and predicted behaviour increase with frequency, more significantly so above 1 kHz, or three to four times the firing frequency. These discrepancies can be attributed to such factors as flow noise, or deficiencies in the specifications of flow temperature, source characteristics and so on, that tend to degrade the realism of the predictions as the frequency increases.

Acoustic theory, where all waves are assumed to travel at a constant velocity c_o, dependent on local time averaged temperature T_o in each element [10], remains appropriate so long as the spectral distribution of fluctuating pressure or velocity in the incident and reflected component waves does not change significantly along each uniform section of pipe. With the cyclic flows considered here, the limiting value of the pressure fluctuation amplitude for reliable predictions with acoustic theory depends on a number of factors. Thus, it increases with mean flow velocity and the ratio of wavelength to pipelength. However, the fluctuating pressure amplitude in the intake and exhaust manifold normally exceeds the acoustic limit by at least an order of magnitude.

Nevertheless, with interfering wave motions in pipes less than half a wavelength long, acoustic models may still be adequate.

3.2. NON-LINEAR WAVE PROPAGATION ALGORITHMS

There is an extensive and rapidly expanding literature describing non-linear fluid dynamic modelling of the wave motion in intake and exhaust systems [2, 3]. Similarly there are several commercially available software packages offering corresponding facilities. They now adopt numerical techniques based on the MacCormack (MAC) and two step Lax-Wendroff (LW2) shock-capturing, finite difference schemes and more recently the discontinuous finite element Galerkin method [11], yielding more realistic results than the mesh method of characteristics (MOC). They have proved most successful in predicting the influence of intake/exhaust system tuning [3, 12] on engine breathing and performance. They are apparently significantly less successful in calculating valid predictions of intake [12] or tailpipe orifice [3] noise emissions above 200 Hz, or the second harmonic of the firing frequency (3rd order), as illustrated in Figure 3, adapted from reference [3].

This result is typical of the rather limited number of similar comparisons available in the literature [2, 3, 11–14], the discrepancy with measurement normally increasing with system geometric complexity, coupled with a failure to identify with sufficient clarity the acoustic behaviour with corresponding features of the system geometry.

The failure of shock capturing numerical techniques to predict orifice noise emitted by a running engine with sufficient precision above 200 Hz, even at low engine rpm when flow noise [1, 5] is an unlikely cause, arises from a number of other factors. One such factor is the adoption of time averaged [2] rather than more realistic time varying coefficients [1] in the modelling of unsteady flow through the valves, of wave transfer across the interfaces between elements [1, 13] or of sound radiation [1] from the orifice, or tailpipe. Furthermore, both

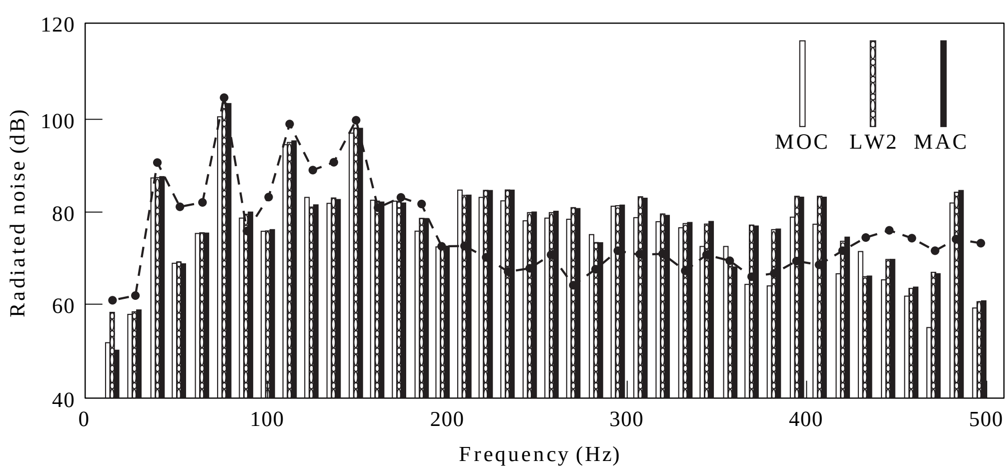

Figure 3. Exhaust emissions from a 6-cylinder engine at 1500 rpm measured at 0·5 m and 45° from the orifice. ·−−−·, measured.

the authors and contributors to the discussion [14] noted that the observed behaviour with periodic waves differs significantly from that with single shocks. Another factor [15] is that the local geometry also has a profound influence on the subsequent time history of wave reflection and transmission, even with single shocks. In particular at open terminations [16] as well as at other discontinuities [2, 7, 11] one must take due account of the time delays that accompany both wave reflection and propagation, as well as the influence of the local time history of the fluctuating flow velocity [17]. Finally, the influence of the moving boundaries on the modelling of the primary sources [18] with the presence of secondary flow noise sources [5, 8] can have a significant influence on the spectral characteristics of the emitted sound.

3.3. ACOUSTIC PLANE WAVE PROPAGATION ALGORITHMS

The restriction to linear plane wave motion (1-D) affords considerable simplification in the algebra [19], with the corresponding advantage that the influence on wave propagation of physical factors such as flow, temperature gradients and so on can be more clearly demonstrated. Furthermore, since the acoustic pressure and particle velocity in a plane wave remains invariant over any cross-section they can be defined by measurements at the duct boundaries without disturbing the field, facilitating experimental validation [4, 5, 19, 20], while the theory applies to cross-sections of arbitrary shape. It also implies that any acoustic modes of higher order generated by the sources or at discontinuities, do not propagate, but are evanescent and their amplitude suffers rapid exponential decay. The first of these, a circumferential acoustic mode with wavenumber $k = 2\pi f/c_o = 2\pi/\lambda$, will propagate in a cylindrical duct of radius a, once the Helmholtz number ka exceeds the value $1{\cdot}84\ (1 - M^2)^{0{\cdot}5}$ [21], where the mean flow Mach number $M = u_o/c_o$.

The acoustic wave motion between discontinuities in either intake or exhaust will include waves travelling out from the sources with waves reflected from each discontinuity. It is physically realistic, as well as convenient to describe the wave motion in terms of the complex amplitude of each spectral component p^+ of the incident, or positively travelling and p^- of the reflected, or negatively travelling component waves. Thus the fluctuating pressure $p(x, t)$ and velocity $u(x, t)$ at any point along the acoustic path will be expressed by the simple summations

$$p(x, t) = p^+(x, t) + p^-(x, t), \quad u(x, t) = u^+(x, t) + u^-(x, t). \tag{5a, b}$$

With appropriate substitution for the component waves in the Euler equation representing dynamic equilibrium for plane waves, namely

$$\rho_o[\partial/\partial t + u_o(\partial/\partial x)]u + \partial p/\partial x = 0,$$

one can show [19] that, to first order

$$\rho_o c_o u = p^+ - p^-. \tag{6}$$

In practice, the positively and negatively travelling component waves are normally subject to some decay in amplitude with x, due to viscothermal, or other losses at the duct boundaries [19, 21]. In such cases equation (6) is strictly valid only so long as such losses remain negligible [19, 22]. Further useful analytic models exist describing sound propagation in terms of incident and-reflected waves in any flow duct with analytically described boundary geometry [23–25]. One notes that the magnitude of the fluctuating pressure $|p_x|$, or of the velocity $|u_x|$, will vary continuously with x. This is illustrated by simultaneous measurements of cyclic fluctuating pressure–time histories at a sequence of axial stations towards the end of the exhaust downpipe in Figure 4, where the changes between them arise from interference. On the other hand, the magnitudes of the components $|p^+|$ and $|p^-|$, subject perhaps to a modest decay [19], will remain effectively invariant between discontinuities, while the relative phase of each will vary in an organised manner along the acoustic path. Appropriate and well validated signal processing techniques [19, 20] are available for the experimental evaluation of the wave components p^+ and p^- corresponding to a set of simultaneous time histories, such as those in Figure 4. Finally, the wave energy flux has two components, each one corresponding to either the incident or reflected wave motion, the net energy flux being their difference. These facts provide one possible explanation for the poor correlation often found between single in-duct measurements and the radiated sound. The wave energy is also convected across any reference plane along the intake/exhaust system by the mean flow. The corresponding acoustic intensity I or net energy flux per unit area is [19] expressed by

$$\rho_o c_o I = (1+M)^2 |p^+|^2 - (1-M)^2 |p^-|^2, \tag{7}$$

for plane acoustic waves.

The source system models in Figure 2 represent wave propagation through intake or exhaust by the transfer element T with its acoustic load Z, expressed in terms of the spectral components of the fluctuating acoustic pressure and particle velocity p_1 and u_1 at the source plane and p_2 and u_2 at the terminating orifice.

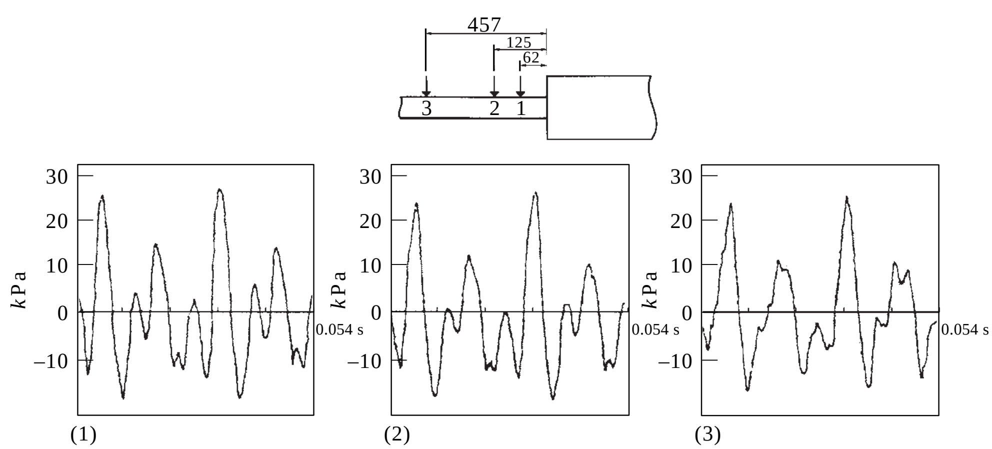

Figure 4. Fluctuating pressure in the downpipe of a 4-cylinder 4-stroke engine at 2232 rpm: (1) at 62 mm, (2) at 125 mm and (3) at 457 mm upstream of the expansion chamber.

Alternatively, and often more conveniently perhaps, the transfer can be described in terms of the spectral components of p^+ and p^-. The transfer function T is normally complex and may be represented by the four elements of a transfer matrix $[T_{ij}]$ or by the transmission coefficients

$$T_i = p_1^+/p_2^+, \quad T_r = p_1^-/p_2^-, \tag{8a, b}$$

together with a reflection coefficient r, where

$$r = p_2^-/p_2^+ = (\zeta - 1)/(\zeta + 1), \tag{9a, b}$$

and $\zeta = Z/\rho_o c_o = p_2/\rho_o c_o u_2$. Alternatively, one finds

$$p_1^+ = T_{11}p_2^+ + T_{12}p_2^-, \quad p_1^- = T_{21}p_2^+p_2^+ + T_{22}p_2^-. \tag{10a, b}$$

The four complex values of T_{ij} are functions of geometry, ρ_o, c_o, f and mean flow Mach number $M = u_o/c_o$ but remain independent of the values of Z. Finally, one also has

$$T_i = T_{11} + rT_{12}, \quad T_r = T_{21}/r) + T_{22}, \tag{11a, b}$$

showing that T_i and T_r are both functions of the load Z as well. However, the elements of $[T]$ and similarly T_i and T_r all remain independent of the source of acoustic excitation, provided it is external to the corresponding system components. Thus, it seems logical to calculate the filter or transfer characteristics of the system starting at their open terminations and proceeding towards the primary sources at the valves.

Despite the relatively high fluctuating pressure amplitude as indicated in Figure 4, linear acoustic models often provide sufficiently precise descriptions of the system's acoustic transfer characteristics to represent adequately any relative changes in system acoustic behaviour that accompany systematic modifications to their geometry. Thus, they can give positive guidance for the adoption of appropriate detailed changes to the design during development. The current APEX software provides realistic predictions of the acoustic transfer characteristics, even when their geometry is extremely complex [20], as the representative comparison between predicted and observed transfer function modulus 20 $\log_{10} p_1/p_2$ in Figure 5 demonstrates.

Other examples can be found for example in references [1, 4–7, 19]. Note that the excitation amplitudes in Figure 5 are about an order of magnitude below those in Figure 4, the limit being set by microphone overload, while the frequency resolution of the measurements was 1 Hz. With flow at representative Mach numbers the acoustic pressure field is contaminated by shear layer turbulence in the boundary layers and with aeroacoustically generated (flow) noise. This represents a severe challenge in making validation measurements that provide realistic assessments of the acoustic behaviour.

3.4. HYBRID APPROACH TO PREDICTIVE MODELLING

This approach [1, 4–6, 13] is based on the assumption that the spectral distributions of orifice noise are strongly controlled by the acoustic filter

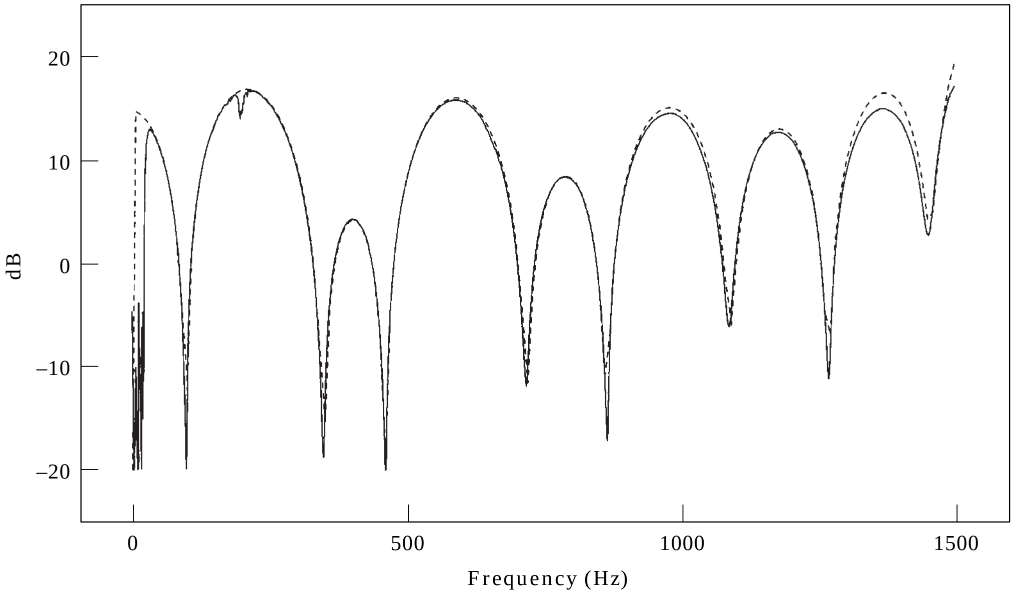

Figure 5. System transfer function when acoustically excited at 160 dB. ————, Measured; - - - - - -, predicted.

characteristics of the intake/exhaust system and scaled in amplitude by the cyclic mass flow through the valve ports. The former are calculated in the frequency domain, while the latter are calculated in time. The cyclic behaviour readily facilitates transformation between the two domains, with appropriate matching of all the conditions at the interface between them. Comparisons between measured and predicted orifice order noise [1] showed that satisfactory agreement has now been extended to 4th or even 6th order, compared with non-linear gas dynamic modelling, though the results suggested further inadequacies existed in the representation of the higher harmonics associated with the primary sources at the valve ports.

The interfacing at the time/frequency boundary involves careful housekeeping, but is otherwise fairly straightforward. This requires constancy or matching in the time domain of the cyclically fluctuating velocity $v(t)$ and pressure $P(t)$ across the interface. On the frequency domain side the corresponding values of $v(t)$ and $P(t)$ can be found by making an appropriate summation of the periodic harmonic acoustic components $u_o + u(f)$ and $p_o + p(f)$. To this end it is necessary that, for each cycle period T, the frequency resolution $\delta f = N/T$ corresponds to the time resolution $\delta t = T/N$, where N is a binary multiple of 256 say. The pressure reflection coefficient $r(f)$ can then be Fourier transformed to give the corresponding temporal reflection coefficient $R(t)$ at the appropriate times. Note too that the time references in both domains should correspond! Upon recalling that $v(t) = u_o + u(t)$ and noting that $P(t) = p_o + p(t)$, after making use of equation (6), the relation between their fluctuating parts can be expressed as

$$p(t) = \rho_o c_o v(t)(1+R)/(1-R), \tag{12}$$

since effectively that flow conditions across the interface can be regarded as, isentropic. As well as introducing time varying boundary conditions, care must be taken with the choice of initial conditions at the start of the calculations [26], since inappropriate ones can introduce spurious waves that can persist and may then lead to unstable solutions.

An alternative approach [13] is similar in principle to that described in references [1] and [26], in that the wave motion throughout the system is first separated into incident and reflected component waves. It differs however in adopting non-linear (gas dynamic) wave propagation algorithms in all the connecting pipes and adopts measured frequency domain, acoustic characteristics, appropriately transformed to the time domain, to represent wave propagation across all the other elements. Thus, the single interface for each system in references [1, 26] was now replaced by a sequence of interfaces. Also a somewhat different procedure was adopted for the matching across interfaces. Furthermore, constant pressure rather than time varying models were again adopted for describing wave transmission and reflection at discontinuities. This assumption lacks sufficient realism for acoustic calculations [14] at the higher harmonic frequencies, a fact the authors [13] also emphasized. Comparison between predicted and measured exhaust orifice emissions by a silenced four-cylinder spark ignition engine under full load [6, 13] revealed good agreement at the firing frequency and in one case for its 4th harmonic at 200 Hz, but the agreement deteriorated rapidly above this frequency. Other evidence [6] from a sequence of measurements in the exhaust system downpipe showed that this was due to inadequacies in the description of the higher harmonics of the primary sources.

4. AEROACOUSTICS AND FLOW NOISE

The aeroacoustic generation of sound and its prediction has been a topic of active research, at least since the production of jet propelled aircraft, if not prior to this in the context of industrial and other transport activities. Although many of the basic physical mechanisms have been understood almost from the beginning, realistic quantitative predictions [5] have always relied on observed sound emissions either at model or full scale. Recently, Doak [27] has presented a rigorous general formulation that describes the propagation and aerodynamic generation of sound. Although it identifies the basic factors concerned, many of the necessary fundamental controlling features of the flow are currently difficult to quantify explicitly for a given practical situation. Nevertheless it provides a generic framework for guiding future analysis and experimental research.

The application of a simplified version of this approach combined with source/system or source/acoustic filter modelling [1, 4] of I.C. engine sound emissions, with some further representative practical examples [5], has already produced predictions that are in close agreement with observations. With reference to Figure 1, the fluctuating mass flux or volume velocity through the valves combined with enthalpy fluctuations is represented by the primary sources, while the fluid dynamic and acoustic characteristics of the remainder of the system that control the propagation of wave energy complete the model. Otherwise vortex shedding at abrupt area expansions with the associated boundary stresses or mixing noise associated with shear layers, both possibly being combined with regenerative feedback, is represented there by the secondary sources. The factors controlling the propagating wave energy or fluctuating enthalpy are again represented by the relevant parts of the system acting as an acoustic or wave filter.

Acoustic fields are generally described in terms of the spatial and spectral distributions of the sound pressure level. In free space, each of its spectral components at any observer position are dependent on the source power spectrum, the associated directional properties of the source and the distance from it. However, in an enclosed or reverberant space, such as an intake or exhaust system, this is no longer the case. It then also depends on the influence of the acoustic properties of the enclosure, and of its boundaries both on the sound power emission at the source and its propagation along the transmission path. It is also well established, both by observation and analysis, that the sound power or strength of aeroacoustic sources normally scales as a function of some characteristic flow velocity U as well as depending on a number of other factors. In free space this is expressed by the familiar expressions $W \propto U^4 c^{-1}$ for sources associated with fluctuating injections of mass, $W \propto U^6 c^{-3}$ for those associated with fluctuating forces and $W \propto U^8 c^{-5}$ for those associated with fluctuating momentum exchanges. In contrast to this, the evidence from measurements in ducts [1, 5, 28, 29] and with exhaust systems [30], showed that with $M < 0{\cdot}3$, which is typical for intake and exhaust flows, the observed sound power scaled according to $W \propto U^4 c^{-1}$ at frequencies where acoustic propagation is restricted to plane waves, changing to $W \propto U^6 c^{-3}$, for excitation by fluctuating forces, etc. [28, 30] at frequencies where multi-modal propagation can exist. The physical reason for this behavior is not properly understood, but might be associated [5] with reactive system behavior.

The influence of reverberation [8] with other geometrical and flow factors on the power of both primary and secondary sources is not yet understood to the extent that it can be reliably quantified in general, but with some examples of rather simple geometry the sound sources associated with the non-linear interaction between vortices and acoustic resonators have been modelled [5, 8], with some success. The sound generated by the vortices excites the acoustic resonator and the resulting acoustic signal then gives rise to the generation of further vortices that are in synchronism or phase locked to the acoustic signal. The non-linear feedback mechanism provides an amplifying mechanism that causes the acoustic signal to increase in amplitude until a limit cycle is reached. Describing function theory provides a technique for modelling such a feedback mechanism, which was demonstrated [8] by application to a flow excited Helmholtz resonator [29].

A further example is a flow excited expansion chamber with its tailpipe by combining the source/filter modelling [5] as illustrated in Figure 2 with quasilinear modelling of developing vortices [28]. Parametric experimental studies [28] suggested that the dominant acoustic resonator was associated with the tailpipe. Recent source/filter modelling studies by the author demonstrated that a fluctuating pressure source representing the non-linear vortex dynamics, located at the chamber/tailpipe junction, gave the close match to the observed radiated sound spectrum as the comparisons in Figure 6 illustrate.

5. SOUND RADIATION FROM OPEN TERMINATIONS

The prediction of the efficiency and directivity of sound radiation from the intake or exhaust orifice is of direct relevance to the prediction of drive-by and interior noise. Also, one notes that the reflection of sound travelling down the intake snorkel or exhaust tailpipe has a direct influence on the wave motion in the corresponding system. Although the radiation and reflection of sound from the intake or exhaust [31] into free space is well documented, as summarised in the Appendix to reference [1], the same is not true for sound emission into reverberant or semi-reverberant spaces, where standing waves are present. For example, the presence of reflecting surfaces including the ground plane and vehicle structure in the vicinity of the exhaust/intake open termination give rise to a corresponding standing wave field. It seems physically plausible to expect that the presence of external standing wave fields have a significant influence on

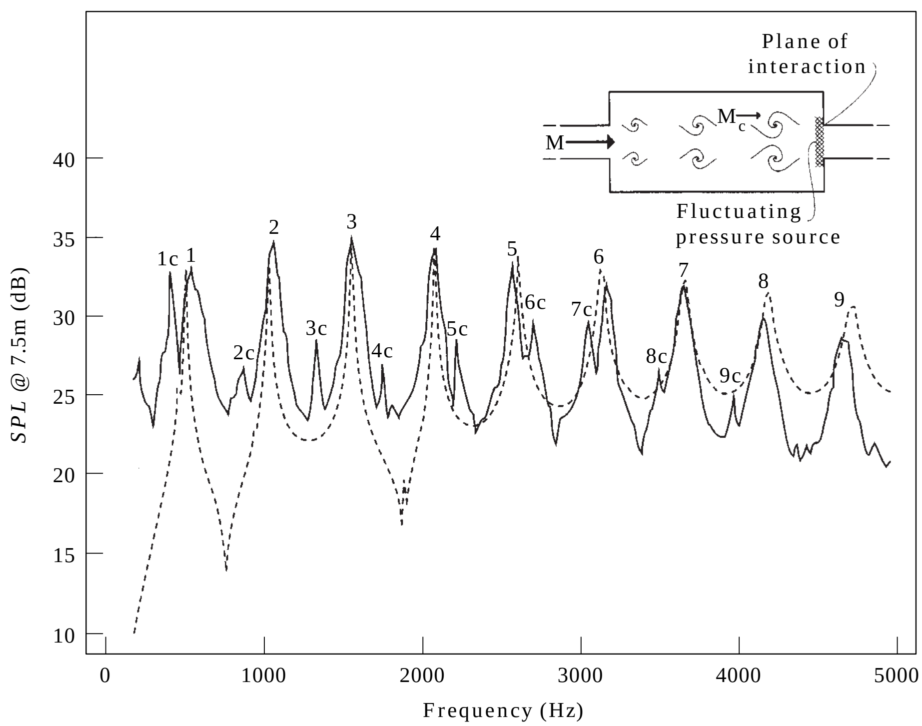

Figure 6. Sound radiation from a flow excited resonator at $M = 0{\cdot}17$. 1c, 2c, 3c, ... 9c, chamber resonances; 1, 2, 3, ... 9, tailpipe resonances, ————, Measured; ------, calculated.

wave reflection at pipe open terminations, and presumably there is a corresponding influence on the extent and directivity of sound power emission.

Since the acoustic performance of exhaust systems is often assessed by using measurements in a semi-anechoic room, this must present similar uncertainties in the correlation of such test cell measurements with predictions of the breathing noise contributions to drive-by or internal noise. Ideally such measurements should be performed under more representative conditions. Appropriate measurements of the pressure transfer functions between an appropriate position in the vicinity of the open termination and the points of observation with a stationary vehicle in an open site under load, are used at the ISVR to reduce the extent of this uncertainty. Clearly, similar effective measurements with a moving vehicle could reduce this still further. Otherwise, any realistic evaluation of test bed or other measurements in terms of the design aims for driving conditions will rely heavily on experience, and thus include subjective as well as objective judgements.

6. DISCUSSION

Sophisticated software packages for engine simulation and integrated performance prediction are commercially available in increasing numbers. They model the cyclic thermodynamic, thermochemical and fluid dynamic processes in the cylinder, combined with the wave motion in manifolds, intake and exhaust with sufficient realism to design for an optimum compromise between engine performance, fuel efficiency and pollutant emissions. The record to date shows, however, they perform this task more effectively than they predict intake or exhaust orifice noise emissions at frequencies above 200 Hz say, or above twice the firing frequency, with non-linear propagation models [2, 3], extending to the 4th or 5th harmonic with hybrid models [1, 13, 26]. Experience suggests that among the likely reasons for this shortfall on predicted noise emissions one should include inadequate modelling of the higher frequency components of the cyclic valve flow time histories, [1, 6, 14, 26], the failure to adopt sufficiently realistic models, [13, 14] either for calculating wave reflection and transfer at discontinuities or for describing the system boundary conditions [1] at the inlet or exhaust orifice by, for example, the adoption of inappropriate time averaged modelling. Similar deficiencies may also exist in the modelling of the more geometrically complex silencing elements [6, 13]. In appropriate circumstances, some of the discrepancies between the predicted and observed orifice noise emissions that are directly related to intake and exhaust system acoustic performance can be significantly reduced [1] by adopting a hybrid approach.

Software packages that currently employ plane wave linear models to predict acoustic propagation in intake and exhaust are also commercially available. They have reached a high stage of development and within the limitations imposed by the acoustic approximation [4] provide realistic predictions of sound propagation along intake and exhaust systems up to 1500 Hz or more [20], however complex their geometry. They can quantify the relative changes in acoustic behaviour that accompany changes in system layout or in the structural detail of their component elements [1]. Otherwise, they rely on measurement to assess the acoustic performance of the integrated system. Thus their effective role [1] is normally restricted to system refinement rather than the prediction of orifice noise emissions at a concept stage of engine or vehicle development, unless they are incorporated [1] into a hybrid scheme. Experience to date indicates that the adoption of acoustic modelling combined with measured source characteristics [9] has not produced consistently reliable estimates of orifice noise emissions, presumably due to inadequate representation of the dynamic interaction between source and system elements.

These software packages differ substantially in the facilities and guidance they provide for rapid refinement of intake and exhaust system design detail for controlling orifice noise as well as maintaining engine performance. Ideally they should allow the design engineer appropriate opportunities to exercise fully his skills, experience and creative imagination in developing effective solutions for each problem as it arises. For example, the software should include appropriate provision [1, 26] to ensure one can identify the significant features of the predicted or observed installed acoustic behaviour with the geometric properties of individual elements in the integrated system. Such correlations seems to be naturally more direct in the frequency domain. The packages also differ widely in the design office time and resources they require for their successful implementation. Computer running times provide one useful index of comparison in this context. Examples for similar representative system components and integrated systems in the literature [2, 7, 32] indicated that finite element models take 10 h [2] or more for each engine running speed and load, compared with 10 to 15 min with non-linear gas dynamic models [2, 32]. In contrast similar calculations with hybrid models [1, 26] take around 10 s or less on a contemporary desk top computer.

None of the predictive software currently available includes quantitative estimates of the contributions by flow noise to the orifice sound emissions. It seems that a sufficiently quantitative understanding of the influence of resonant feedback and separating shear layers [4, 5, 8, 27–29] is not yet available for this purpose. The increases in exhaust flow velocity that accompany continuous development in engine performance, with the necessity of complying with ever more restrictive noise legislation, provide a strong incentive for urgent research to improve our knowledge both of the aeroacoustic and of the other relevant factors that influence orifice noise emission. The subsequent incorporation of such knowledge into the relevant predictive software will surely play an essential role in future developments of optimized intake and exhaust system design technology.

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