Development of Techniques for the Reduction of Radiated Noise of Air Intake Manifolds using ANSYS and COMET

R. J. Tyrrell, J. Blanco

Automated Analysis Ltd.

Summary:

Advanced analytic methods are now in use to develop the designs of plastic inlet manifolds for automotive powertrains for low noise. A requirement is identified to develop the methodology of analysis to optimise the use of the computational tools. The first part of the work reported here involved determining loading data that could be applied to finite element models to represent typical vibration characteristics of actual running engines, without the need to measure actual engine vibration data for each manifold application.

Work is also reported on the identification of general guidelines for the reduction of noise from plastic intake manifolds.

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Keywords:

Noise, plastic, manifolds, engine vibrations, generic loads, design guidelines

Introduction

This paper discusses recent work to improve the turn-around time of work to achieve noise reductions of plastic air intake manifolds, and is also relevant to the design of other plastic ancillary parts, such as timing drive and valve covers.

Developments in computer hardware and software in the last 5 years have made it possible to predict the vibrations and radiated noise levels from many automotive components. These predictions are now reliably accurate, and may be achieved in acceptable elapsed times. However, all predictive analysis work is only as good as the data used as input. The quality of any analysis project is dependent upon :

- geometric accuracy : do the models represent the true geometry?
- material accuracy : are the relevant materials properties known, for the true operating conditions?
- input boundary conditions : are they valid?
- mathematical solution : are the correct mathematics being used for the problem in hand?
- results interpretation : is the analysis answering the correct questions?

With modern CAD and finite element modelling, geometric accuracy is primarily a function of the choices the analyst makes to simplify geometry. Solution times are now so short, for example, as to make it feasible to make solid-element models of sufficient mesh density to accurately represent the geometry and performance of even thin-walled parts.

Close collaboration with material suppliers, supported by material tests when appropriate, means that material property data is understood to an acceptable level.

Input boundary conditions are the main subject of this paper. They are therefore discussed in the following sections.

Modern finite element software (such as ANSYS) includes the necessary mathematical tools to reliably predict vibrational behaviour. The accuracy of the solution is generally extremely high, especially when compared with the magnitude of uncertainty in material properties and boundary conditions.

Results interpretation is a very broad subject. The first requirement of any analysis project is to determine "What is the question?". Only when this is answered may the results interpretation methods be usefully defined. This question is also addressed in the sections below.

Overall, the main problem facing the analyst who must reduce noise from plastic manifolds is to design an analysis which will lead to a low-noise part in the minimum possible elapsed time. This reduces to a decision on whether to answer the question :

A) "Will this manifold produce less than x dBA?"
OR
B) "Will this manifold be as quiet as possible?"

This decision influences the type of boundary condition to select.

Boundary Conditions : Approach

There are two approaches to selection of boundary conditions for vibration prediction work. The analyst may either aim at an 'absolute' prediction of vibration and noise levels (question (A) above), OR may try to design a part which is inherently quiet, no matter what the excitations applied to it (question (B)). The first approach may satisfy the purist, but it entails a great deal of time-consuming data collection and processing (e.g. measurement of running engine vibration data). The second approach is a practical alternative, with reduced elapsed times and effort, but which needs some confidence in results such that, for example, a plastic manifold will be 'at least as quiet' as an aluminium equivalent.

The authors company has performed work to assess the radiated noise from the external skin of plastic manifolds, as excited by structural vibrations (transmitted through the component mountings) and by internal pressure fluctuations (due to engine operation and air flows). Detailed discussion of the results of this work is beyond the scope of this paper, but it is noted that the total noise radiated from plastic manifolds is dominated by the noise due to structural excitations. The work described in this paper thus only considers this excitation mechanism.

Work was performed with three different 'types' of input vibration data, for two different engine types. The objective of this work was to determine whether the design solutions found using the 'absolute' approach would be the same as those found when using simplified methods.

The input data sets used were :

	V8 Engine	**I4 engine**
Measured running engine data	MeasA	MeasB
Constant acceleration with freq.	Const	Const
Derived 'generic' vibration spectra	GenA	GenB

Analysis Frequencies

The vibration and noise prediction work was all based on the assumption of steady-state conditions. Thus, the work was performed in the frequency domain, with analyses performed at a fixed frequency interval. One decision the analyst must make is what frequency interval to use. A smaller interval means greater accuracy, but at the expense of requiring increased computer resource. A larger interval carries the danger of 'missing' the response at frequencies close to resonance, and these near-resonance responses tend to dominate the overall response for lightly damped assemblies. The author generally selects the frequency interval to use after inspection of the results of a free-modal analysis of the component : the frequency interval is selected to achieve a balance of computational time vs. accuracy in terms of 'hitting' the resonant modes. A frequency interval of 10 to 25Hz is usually used.

The maximum frequency for analysis also influences the computer resource requirements. In general, accuracy of vibration prediction becomes questionable above approximately 3,000Hz for practical analyses. However, some specialised work (e.g. studies of 'whistling' noise transmission, as excited by air flow through partially closed throttle butterflies) may demand upper frequency values as high as 7,000Hz.

Measured Engine Data

The measured engine vibration data sets contained data at the manifold mounting flanges, in three orthogonal directions. Measurements were made for three points along each flange.

One major problem with the use of measured data is that it is very rarely obtained using the same narrow-band frequency interval as selected for the analysis work. The measurement frequency interval is usually either very fine (typically data is obtained at 1Hz steps), or it is linked to an order of engine rotational speed. In either case, the data must be converted to equivalent excitation data at the frequency interval to be used in the analysis.

This frequency conversion is not simple. It is NOT sufficient to simply use the data at the points in the spectrum which coincide with the analysis frequency step. The important point to bear in mind is that the input data to the analysis should represent the same energy over a given frequency range as was obtained in the measured data. Figure 1 illustrates typical conversion of measured engine vibration velocity data from a measurement frequency interval (of 1Hz) to the

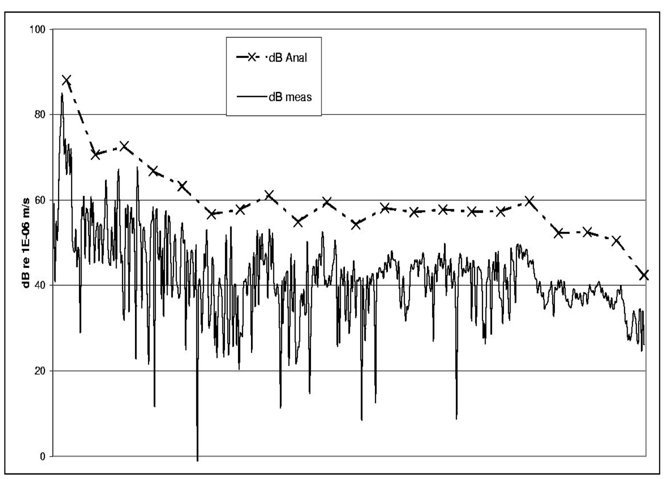


Figure 1 : Frequency Conversion

selected analysis frequency interval (25Hz). The converted data has a level typically 14dB higher that of the original data in this case, as each point in the converted spectrum has the energy from 25 points of the original, and 10Log(25) ~ 14dB. This conversion always includes various assumptions :

- the conversion is based on the signal energy content, i.e. on the square of the individual vibration velocity levels.
- assumptions must be made concerning how to sum data at different frequencies, which will therefore have varying phases relative to each other. The mechanism for doing this is complex but is essential : for example, the excitations illustrated in Figures 2a and 2b may produce significantly different forced responses, even though the energy content of the two cases is the same. The author's company has developed software to achieve the frequency interval conversion, which essentially re-generates the time history of the measured signal and then re-samples it at the required time interval to obtain the wanted frequency step.

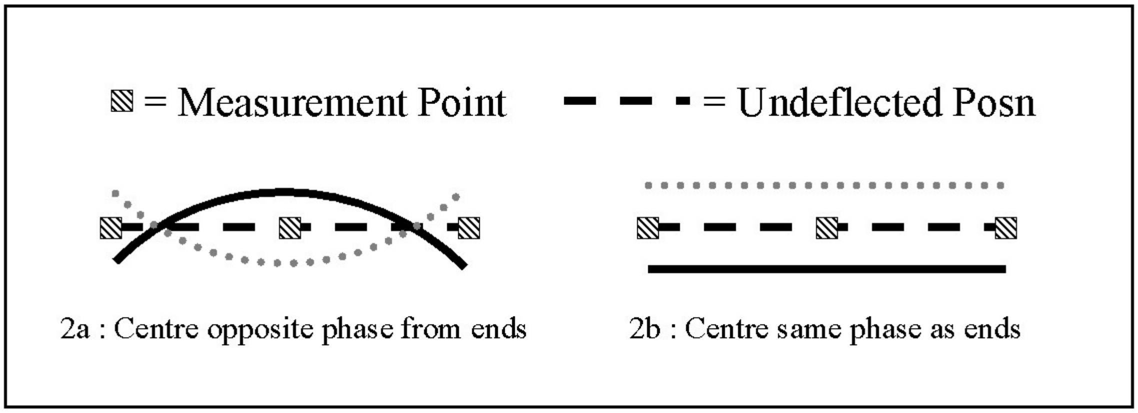


Figure 2 : Alternate Deflection Forms

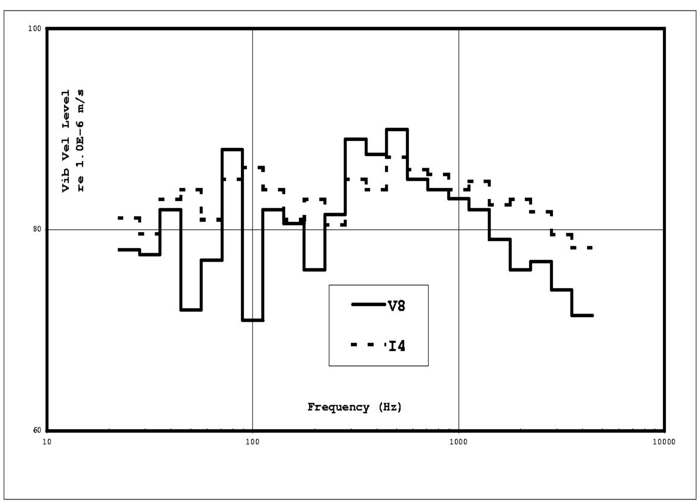


Figure 3 : Example Vibration Spectra

Figure 3 presents a comparison of the measured data from an I4 engine and a V8 unit. It may be seen that the trend of variation in level with frequency is similar for the two engines, but that the two spectra have (as expected) different absolute levels. Both these spectra were converted to a narrow band frequency step of 25Hz, but are presented in 1/3rd octave forms.

Constant Acceleration Data

Figure 4 shows one of the spectra from Figure 3, but expressed in terms of displacement, velocity and acceleration. From this, it may be seen that the data in acceleration form is approximately constant.

The simplest form of vibration loading which may thus be used for noise reduction work is thus selected to be a constant acceleration value, applied across the frequency range. In addition to this gross simplification, it is assumed that the phase of the applied accelerations is unchanging with location, and that the accelerations are equal in three orthogonal directions.

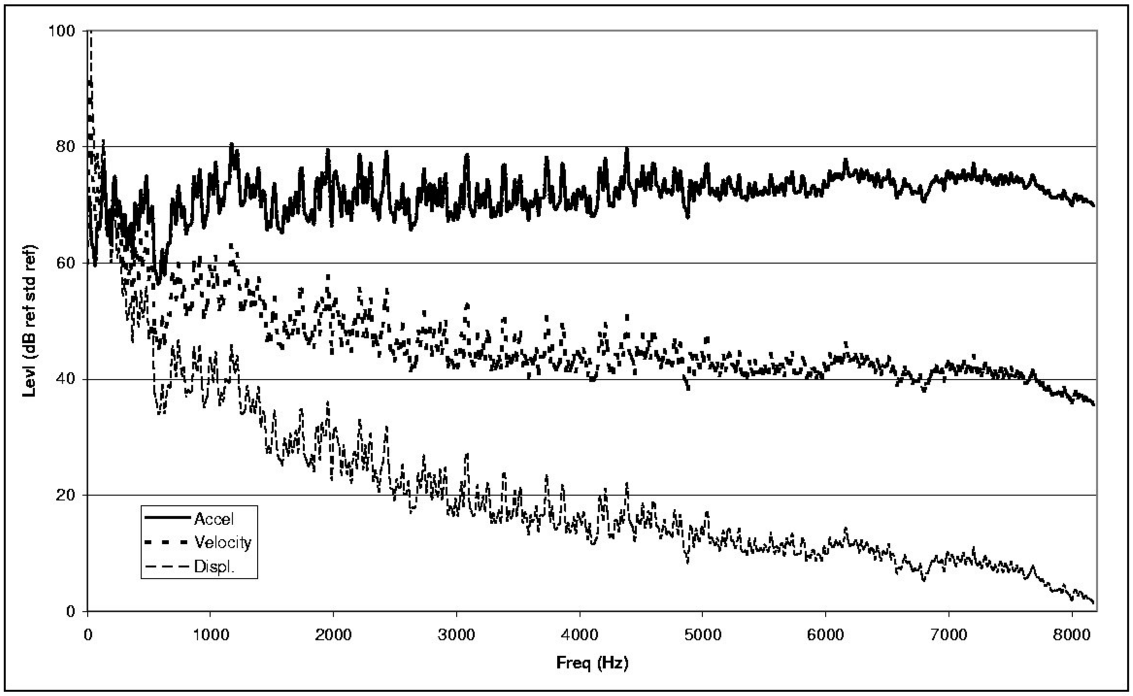


Figure 4 : Alternate Presentation Forms

The gross assumptions inherent in the use of constant acceleration loadings may mask many of the vibration characteristics of a plastic intake manifold, such as modes involving bending or rotations of the mounting flange.

Generic Engine Loads

The objective of the work described here was to see whether a low number of generic loading data sets would be sufficient to obtain reliable reductions in manifold noise levels, but avoid the necessity of obtaining specific measured vibration data for each engine installation.

Thus, a series of methods of deriving a generic load set were developed, based on results of comparative analyses of alternate manifold designs. The frequency and phase content of these load sets were derived from data from a minimum of three engines of each basic configuration.

Prediction of Manifold Vibrations

Finite element methods are used to predict the vibrations of the intake manifolds. For plastic air intake manifolds, experienced analysts are able to generate the models using shell type elements. These have the advantage of reduced computational requirements, but do require experience in their generation to obtain reliable results. In particular, local features such as vibration weld joints and flanges must be modelled with care. Modern computer software plus increased hardware powers mean that it now more cost effective to use high-order tetrahedral meshes, which may be generated at least partly automatically, provided that good quality CAD data is available.

The basic approach to prediction of frequency-domain vibration levels is well understood and documented, and will not be described here. Modal synthesis methods were used, i.e. prediction of free vibrations followed by modal synthesis to obtain predictions of the forced response. Experience has shown that the vibrations of plastic air intake manifolds are essentially de-coupled from the surrounding fluid (air), for all but exceptional cases where there are very large internal air pressure fluctuations (e.g. large, single cylinder engines). Thus, predictions of vibration levels are made for the manifold *in vacuo*.

Depending on the analysis code used, the excitation vibrations may either be applied directly (e.g. ANSYS), or by the use of local, large added masses, to which sufficient forces are applied such that the point masses vibrate with the required magnitude, and the attached manifold (which has a much smaller mass) follows these vibration magnitudes. Some codes (such as NASTRAN and ABAQUS) have features whereby this procedure is implemented automatically.

Prediction of Radiated Noise

The COMET / Acoustics boundary element software was used for all the noise prediction work described in this paper. Compared with finite element models, boundary element meshes for manifold noise prediction work usually appear very coarse. This is because the effectiveness of noise radiation, and hence the requirement for mesh refinement, is controlled by the wavelength of the sound in air. At a frequency of 3,000Hz, this is approximately 110mm. With five elements per wavelength (a good guide for external radiation problems), this means a maximum element size as high as 22mm. Thus, a typical boundary element model for a manifold for an I4 engine may contain around 2,000 elements.

The predicted surface vibration levels are used as input to the noise prediction, and the calculated noise is expressed both as noise radiation distributions, and also as a total radiated sound power spectrum.

Test Case

Figure 5 shows a preliminary manifold design used for the assessment of the different loading types. Figure 6 shows a grossly modified version of the same manifold. A variety of design modifications were used, ranging from wall thickness changes to the addition of stiffeners.

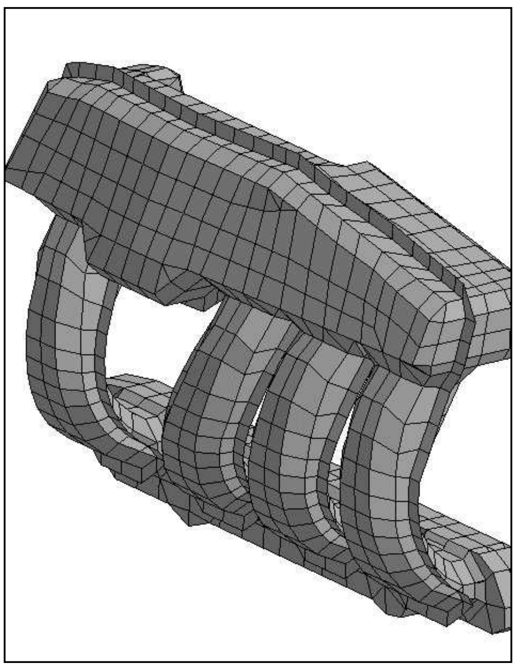


Figure 5 : Baseline Manifold Design

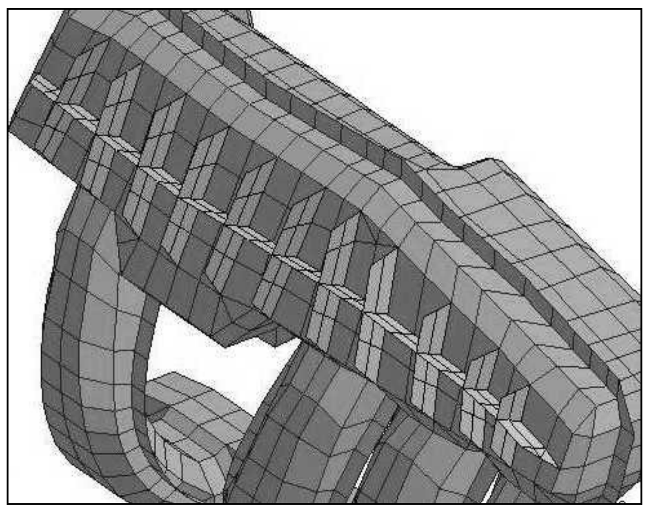


Figure 6 : Modified Manifold Design

Figure 7 shows the predicted total radiated sound power spectra for the original design as shown in Figure 5, when excited using three different types of vibration spectra for an I4 engine. The three types used were :

- Measured ("MeasB")
- Constant Acceleration ("Const")
- A generic in-line engine vibration set, derived from measured data from three different in-line engines ("GenB")

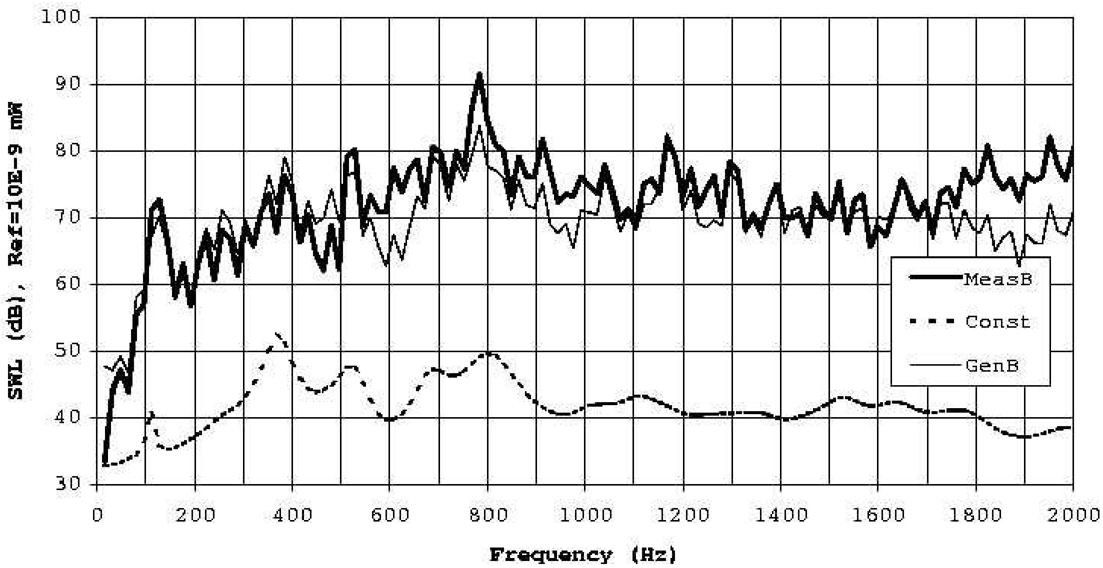


Figure 5 : Total Predicted Sound Power, Original Design

Figure 8 shows a similar comparison, but for the modified design as shown in Figure 6. Figure 9 (overleaf) shows the predicted sound power change obtained using the different vibration excitations. It can be seen that the generic data set produces results very similar to those obtained with measured, i.e. "true" data. Similar results have been obtained with V form engines. Based upon these results, the 'generic' loading data is now used for most manifold noise prediction projects within the author's company.

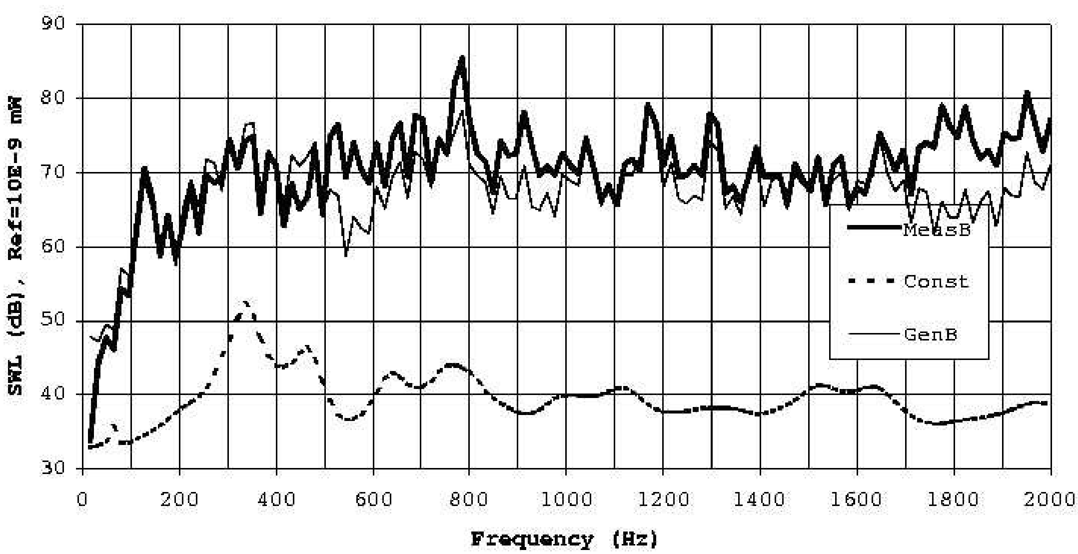


Figure 6 : Total predicted Sound Power, Modified design

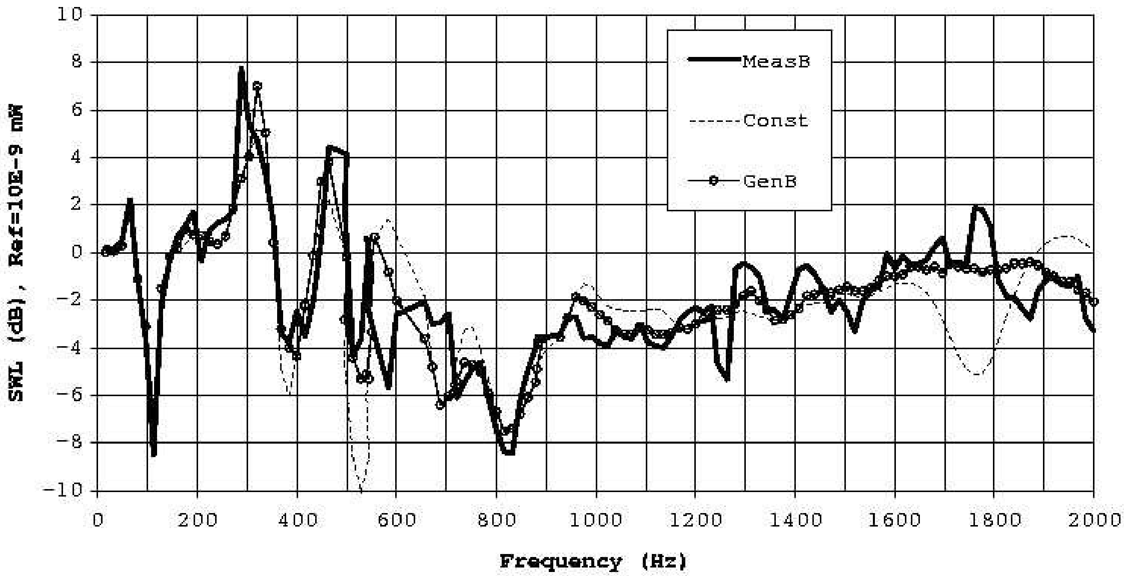
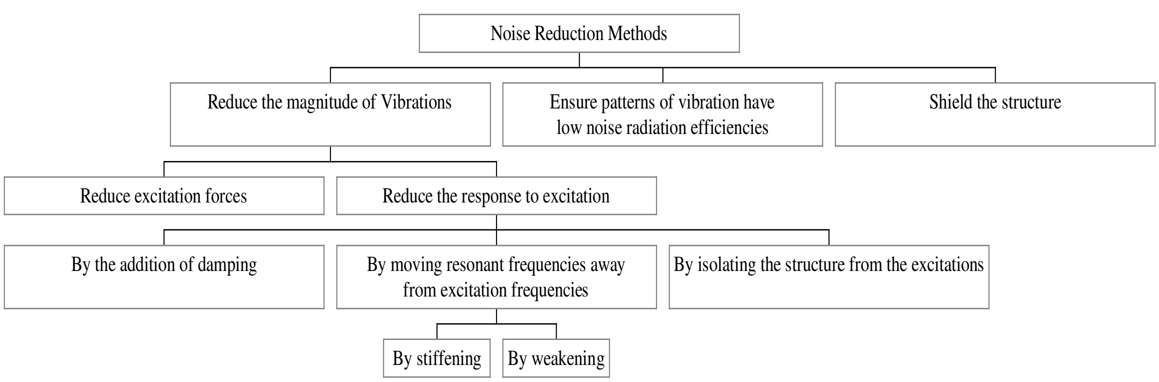


Figure 7 : Comparison of Noise Change for Different Excitation Methods

Design Change Guidelines

There are generic approaches to the reduction of noise for any component. These may be summarised according to the flow chart below:



Unfortunately, the designer of plastic intake manifolds does not have the option of reducing excitation forces : these are fixed by the base engine design. As automotive engines must operate over a wide range of operating conditions, it is not possible to tune a design such that individual resonant modes are detuned from specific excitations, although this is possible for some applications (such as power generation, or when the excitation is due to high response of a particular mode of engine vibration such as longitudinal bending).

The damping within a plastic manifold either comes from the material (which is generally determined by other factors, such as durability and production feasibility and cost), or from the addition of local mass damping. This latter option is of limited application : mass dampers may only be used to shift specific resonant modes, and they add unwanted mass and cost to the component.

It is rarely feasible to reduce radiation efficiencies. Whereas modern boundary element (and other) software can give the necessary data on what the radiation efficiencies of different shapes are, and can indicate which regions of the surface produce active intensity (i.e. noise which 'reaches' to the far field), the reduction of radiation efficiency usually involves significant design changes. As the necessary changes for given deflection shapes are likely to be different from each other, it is impractical to create a design with no possible deflected shape having high radiation efficiency. However, the data obtained for local active intensity is a very useful guide to regions of the design which should be modified. For example, if a particular local region is shown to produce active intensity for the majority of the high noise level frequencies, then it is useful to concentrate efforts at reduction of the vibrational response at that location. Conversely, if a region is shown to *absorb* noise, i.e. to have a negative intensity, then reduction in the forced response in that region will actually increase the noise radiated.

Shielding can be very effective. The use of shields is generally unacceptable due to increased complexity, cost and packaging of the component. It may also be unacceptable because of servicing requirements, or the need to mount ancillary components.

Isolation systems are often used, when the added cost is acceptable. In general, isolation of an intake manifold comprises use of soft rubber mounts in which care is taken to avoid direct transfer of bolting loads to the manifold. The main problems to be overcome in the design of an isolation system are durability of the mounts, maintenance of the seal between the manifold and the cylinder head, and the need to ensure that the isolated mount does not result in large, low frequency deformations which can lead to durability problems. This is especially important for any mounted ancillary, such as ECU systems, which generally have low tolerance to peak accelerations.

Thus, the main opportunity for the designer to reduce the noise of a plastic intake manifold is to modify the component stiffness. In the world of metallic component design, the conventional approach is to increase stiffness globally or locally, such that resonant modes are above frequencies of excitation. Experience has shown that this approach is not usually applicable for plastic intake manifolds. The vibrations of these manifolds are largely controlled by transmission of excitations from the mounting points (as opposed to response to forces internal to the manifold), and increase in manifold stiffness usually means that the complete manifold more closely 'follows' the vibrations of, for example, the inlet port mounting face. As the manifold assembly is light compared with the base engine, it effectively magnifies the magnitude of vibrations in the form of a cantilevered noise radiator.

The challenge to the designer is therefore to reduce manifold stiffness in the areas indicated by active noise intensity results, whilst meeting requirements of durability and production feasibility. It is also important to ensure that large regions of the manifold (such as the plenum) are not so weakened as to cause them to vibrate significantly due to internal pressure fluctuations.

Conclusions

This paper summarises work and methodologies developed over many years of work in the area of plastic component noise reduction. The conclusions are:

- Generic engine vibration loadings have been developed which provide reliable results, without the need to measure excitations from individual engines.

- Over-simplified vibration loadings may yield invalid results.

- The most effective general approach to adopt when designing plastic air intake manifolds for low noise is to design a part with as low a stiffness as possible, compatible with production feasibility and durability requirements.