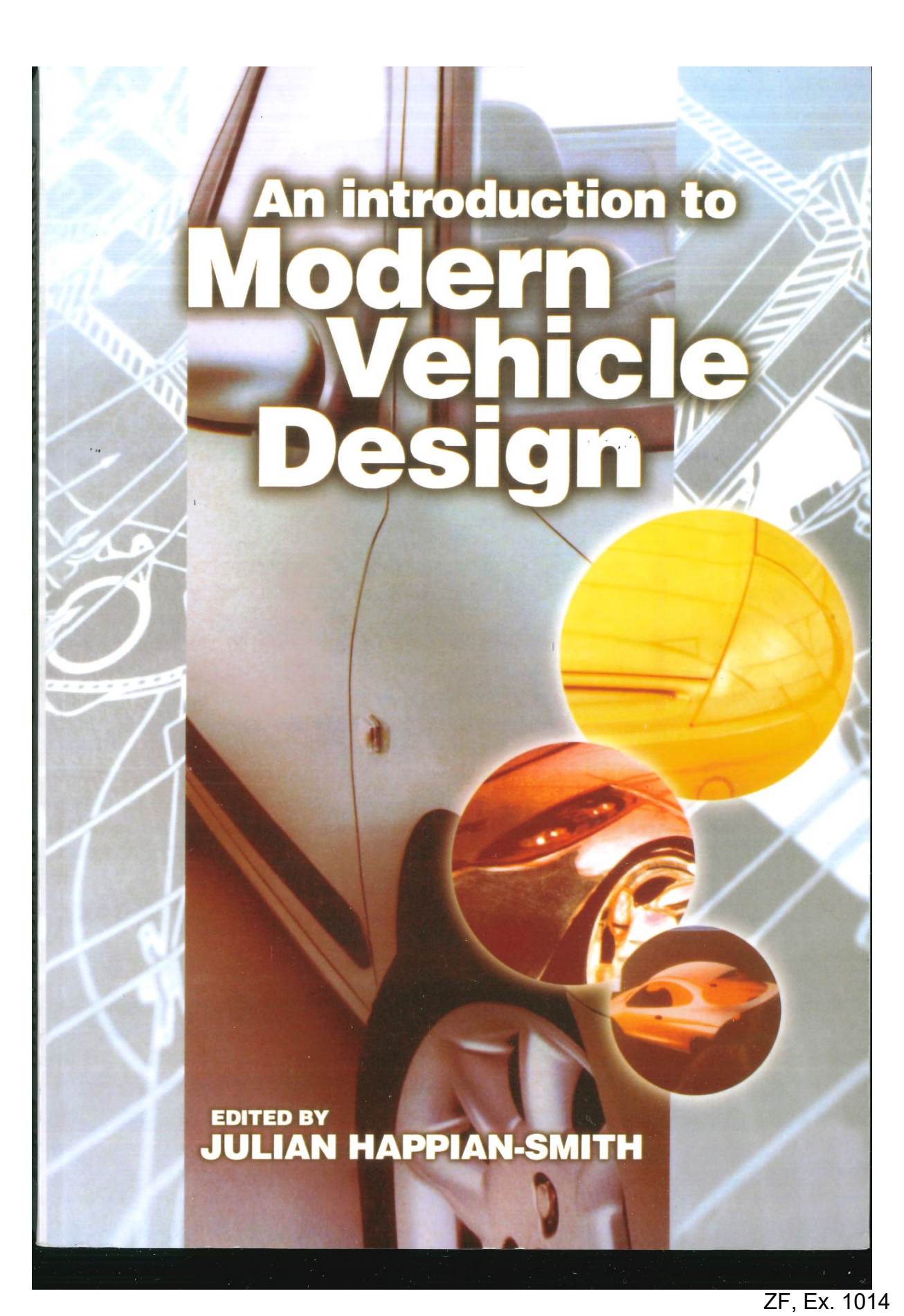


The book cover features a central image of a car's body panel, possibly a door, with technical line drawings overlaid. Three circular cutouts are positioned on the right side, showing internal mechanical components. The title is prominently displayed in the upper half.

An introduction to Modern Vehicle Design

**EDITED BY
JULIAN HAPPIAN-SMITH**

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Julian Happian-Smith
PhD, MSc, BTech, Cert Ed HE, MSAE

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PLANT A
TREE



FOR EVERY TITLE THAT WE PUBLISH, BUTTERWORTH-HEINEMANN
WILL PAY FOR BTCV TO PLANT AND CARE FOR A TREE.

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8. Noise vibration and harshness

Brian Hall, PhD, BScEng, CEng, MIMechE

The aim of this chapter is to:

- Introduce the basic concepts and importance of vibration theory to vehicle design;
- Consider the role of the designer in vibration control;
- Demonstrate methods for the control of vibration to help the elimination of noise and harshness;
- Indicate methods by which the designer can control vibration and noise to create an equitable driving environment.

8.1 Introduction

Noise, vibration and harshness (NVH) have become increasingly important factors in vehicle design as a result of the quest for increased refinement. Vibration has always been an important issue closely related to reliability and quality, while noise is of increasing importance to vehicle users and environmentalists. Harshness, which is related to the quality and transient nature of vibration and noise, is also strongly linked to vehicle refinement.

Controlling vibration and noise in vehicles poses a severe challenge to the designer because unlike many machine systems, motor vehicles have several sources of vibration and noise which are interrelated and speed dependent. In recent years, the trend has been towards lighter vehicle constructions and higher engine speeds to meet the requirements for improved fuel consumption and engine performance. This has tended to increase the potential for noise and vibration, posing many new problems for automotive engineers. These developments have also coincided with a reduction in the time to market for new vehicles and created an increased dependency on computer-aided design and analysis with less time spent on prototype testing.

This accelerated development of new and highly refined vehicles is dependent on accurate dynamic analysis of vehicles and their subsystems and calls for refined mathematical modelling and analytical techniques. While NVH analysis has in recent years been aided by developments in finite element and multi-body systems analysis software, there is still an underlying need to apply basic vibration and noise principles in vehicle design.

There are many excellent texts dealing with vibration and noise, but few are devoted to automotive applications. It is therefore the objective of this chapter to address some important noise and vibration issues arising in vehicle design. It is assumed that the reader has some previous knowledge of noise and vibration theory, since space only permits a brief review of the fundamentals.

8.2 Review of vibration fundamentals

We begin this section by reviewing some vibration fundamentals in an automotive context and proceed to summarize the characteristics and response behaviours of vibrating systems. For further details of the fundamental principles the reader is recommended to read some of the well established texts such as Timoshenko *et al.* (1974), Meirovitz (1986), Rao (1995) and Dimaragonas *et al.* (1992).

8.2.1 Basic concepts

Vibration arises from a disturbance applied to a flexible structure or component. Common sources of vibration in vehicles are road and off-road inputs to suspensions, rotating and reciprocating unbalance in engines, fluctuating gas loads on crankshafts, gear manufacturing errors and tooth loading effects in transmissions, generation of fluctuating dynamic forces in constant velocity joints and inertia and elasto-dynamic effects in engine valve trains.

Vibration sources are characterized by their time and frequency domain characteristics. In automotive engineering, most vibration sources produce continuous disturbances as distinct from shocks and short duration transients encountered in some machine systems. They can therefore be categorized principally as either periodic or random disturbances. The former are the easiest to define and originate from the power unit, ancillaries or transmission, while random disturbances arise from terrain inputs to wheels.

The simplest form of periodic disturbance is harmonic and might typically be produced by rotor unbalance. In the time domain this is represented by a sinusoid and in the frequency domain by a single line spectrum. It might be noted that a full representation in the frequency domain requires both amplitude and phase information. This is important when the disturbance includes several frequency components each of which may be phased differently to one another. Typical of these are the general periodic disturbances produced by reciprocating unbalance and crankshaft torque.

In the case of random disturbances, it is not possible to predict the precise level of the disturbance at any given time and hence it is not possible to express such disturbances as continuous functions in the time domain – only statistical representations are possible. From the vibration point of view, the frequency content of a random signal is very important. For example the frequency spectrum of a road input to a vehicle is a function of the spatial random profile of the road and the speed of the vehicle (see Chapter 10). For a given set of conditions this results in a large (theoretical infinite) number of frequency components distributed over a wide band of frequencies and is commonly represented by its power spectrum (Newlands, 1975).

All mass-elastic systems have natural frequencies, i.e. frequencies at which the system naturally wants to vibrate. For a given (linear) system these frequencies are constant and are related only to the mass and stiffness distribution. They are not dependent on excitation applied to the system provided that the system can be classified as linear. Non-linear effects (which often arise in automotive systems) are beyond the scope of this chapter. The interested reader should consult one of the specialised texts such as Thomsen (1997).

An arbitrary short duration disturbance applied to the system tends to excite all the system's natural frequencies simultaneously. Most systems have a very large number of natural frequencies,

but normally only a few of the lower order ones are of interest because the higher ones are more highly damped. At each natural frequency a system vibrates in a particular way, depicted by the relative amplitude and phase at various locations. This is called the *mode of vibration*.

Lightly damped structures can produce high levels of vibration from low level sources if frequency components in the disturbance are close to one of the system's natural frequencies. This means that well designed and manufactured sub-systems, which produce low level disturbing forces, can still create problems when assembled on a vehicle. In order to avoid these problems at the design stage it is necessary to model the system accurately and analyse its response to anticipated disturbances.

The general approach to vibration analysis is to:

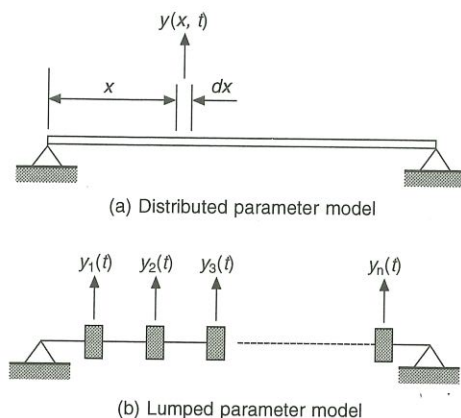
- (a) develop a mathematical model of the system and formulate the equations of motion
- (b) analyse the free vibration characteristics (natural frequencies and modes)
- (c) analyse the forced vibration response to prescribed disturbances and
- (d) investigate methods for controlling undesirable vibration levels if they arise.

8.2.2 Mathematical models

These provide the basis of all vibration studies at the design stage. The aim is to represent the dynamics of a system by one or more differential equations. It is possible to represent the distributed mass and elasticity of some very simple components such as uniform shafts and plates by partial differential equations. This is called the *distributed-parameter* approach. However, it is not generally possible to represent typical engineering systems (which tend to be more complicated) in this way. Hence the approach normally adopted is to model a system by a set of discrete mass, elastic and damping elements, resulting in one or more ordinary differential equations. This is called the *lumped-parameter* approach. Masses are concentrated at discrete points and are connected together by massless elastic and damping elements. The number of elements used dictates the accuracy of the model – the aim being to have just sufficient elements to ensure that an adequate number of natural vibration modes and frequencies can be determined while avoiding unnecessary computing effort.

Figures 8.1(a) and (b) compares the distributed and lumped parameter approaches for modelling a uniform beam undergoing free lateral vibration. In Figure 8.1(a) the displacement at a particular point is described in terms of a variable which is a continuous function of position and time, e.g. $y(x, t)$ and leads to a second order partial differential equation (Meirovitz, 1986). In Figure 8.1(b) the mass of the beam is broken up into a series of equal masses connected together by massless beam elements. The beam stiffnesses are calculated from a knowledge of the flexural rigidity of the beam. The displacements of the masses are represented by a finite number n of generalized coordinates (also called degrees of freedom) which are a function of time only, e.g. $y_1(t), y_2(t), \dots, y_n(t)$. Hence the number of generalized coordinates is equal to the number of degrees of freedom (DOF) of the model. Thus in general an n -DOF system will (a) be described by n -second order differential equations and (b) have n -natural frequencies and modes.

Thus it is apparent that the building blocks of discrete models are mass, spring and damping elements. In basic mathematical models these elements are two dimensional, representing translational and rotational motion. The dynamics of each element can be represented by a constituent equation (see Rao, 1995 for a table of elements and equations) and the dynamics of

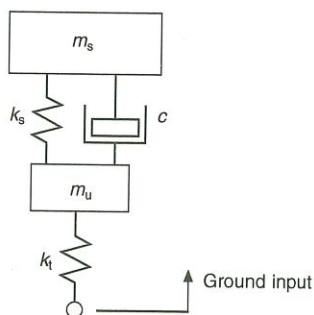
**Figure 8.1** Types of vibration

the interconnected set of elements can be found either by applying Newton's second law to each of the mass elements or by use of an energy method e.g. Lagrange equations (Meirovitz, 1986). In general the topology of the assembled model bears close similarity to the real system.

The system model

The following suspension model illustrates the use of modelling elements. A more detailed analysis of this model is developed later in this chapter.

Suspension systems have a variety of geometries and can be modelled in many different ways depending on the objectives of the analysis. One of the simplest forms of suspension model which is used for conceptual studies is the quarter vehicle model associated with one wheel-station (Figure 8.2). This type of model normally has two degrees of freedom comprising a sprung mass m_s (a proportion of body mass) and an unsprung mass m_u (incorporating a proportion of suspension components plus wheel, axle and brake). The suspension stiffness k_s and damping c are represented by linear elements in this simplified model. In reality suspension motion together with the characteristics of the spring and damper are all non-linear (see Chapter

**Figure 8.2** Quarter vehicle suspension model

10), however for small suspension motions linear representation is generally acceptable. The model is completed by including the tyre stiffness k_t . The input to the model is determined by the ground profile characteristics and the speed of the vehicle. It is convenient to measure horizontal motion of the ground relative to the vehicle, in which case the model is constrained to move within a vertical plane while the ground moves vertically at the bottom of the tyre spring.

8.2.3 Formulating the equations of motion

The first step in the formulation of the equations of motion is to assign a set of generalized coordinates (a minimum set of independent coordinates) to the model which describes the general motion. For the relatively simple multi-body systems discussed in this chapter, the equations of motion can then be determined from a set of free-body diagrams (FBDs) of the masses. The equation of motion can then be determined by applying Newton's second law to each free-body. For cases where the geometry of the model is complicated, the equations can be formulated more elegantly by energy methods (Ginsberg, 1988).

As an example of the FBD approach, consider the suspension model in the previous section. This 2DOF example (which leads to two second order differential equations) can be expressed in matrix form and is typical of linear multi-degree of freedom systems in general. In this case two generalized coordinates y_2 and y_1 are required to represent the displacements of the sprung and unsprung masses respectively and y_0 represents the ground input. If it is assumed that there is no separation between tyre and ground, the annotated model and FBDs are then as shown in Figure 8.3. Note that if deflections are measured from the static mean positions, there is no need to show the gravity force acting on the mass and the mean force in the spring – they are equal and opposite. Defining the displacements positive upwards results in the corresponding velocities and accelerations being positive upwards.

Applying Newton's second law to each of the two masses gives:

$$k_t(y_0 - y_1) - k_s(y_1 - y_2) - c(\dot{y}_1 - \dot{y}_2) = m_u \ddot{y}_1 \quad (8.1a)$$

$$k_s(y_1 - y_2) + c(\dot{y}_1 - \dot{y}_2) = m_s \ddot{y}_2 \quad (8.1b)$$

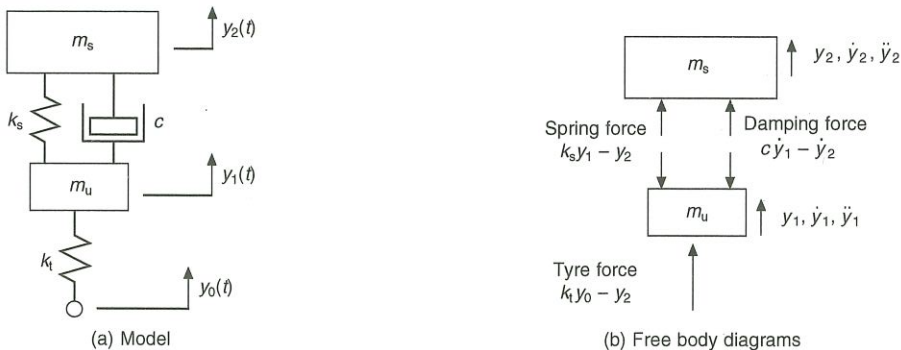


Figure 8.3 The quarter vehicle model and free-body diagrams

These equations can be re-arranged and written in matrix form as:

$$\begin{bmatrix} m_u & 0 \\ 0 & m_s \end{bmatrix} \begin{Bmatrix} \ddot{y}_1 \\ \ddot{y}_2 \end{Bmatrix} + \begin{bmatrix} c & -c \\ -c & c \end{bmatrix} \begin{Bmatrix} \dot{y}_1 \\ \dot{y}_2 \end{Bmatrix} + \begin{bmatrix} (k_1 + k_s) & -k_s \\ -k_s & k_s \end{bmatrix} \begin{Bmatrix} y_1 \\ y_2 \end{Bmatrix} = \begin{Bmatrix} k_t y \\ 0 \end{Bmatrix} \quad (8.2)$$

Equation 8.2 is of a form typical of linear multi-degree of freedom (MDOF) vibrating systems. Such equations can be written more concisely as:

$$[M]\{\ddot{x}\} + [C]\{\dot{x}\} + [K]\{x\} = \{F(t)\} \quad (8.3)$$

where $[M]$, $[C]$ and $[K]$ are the mass/inertia, damping and stiffness matrices, $\{x\}$, $\{\dot{x}\}$ and $\{\ddot{x}\}$ are displacement, velocity and acceleration vectors, $\{F(t)\}$ is the excitation vector.

This general form is also applicable to rotational systems and will be used in a later application.

8.2.4 System characteristics and response

Single degree of freedom systems

Despite its limitations for accurately modelling most automotive systems, a knowledge of SDOF behaviour provides a basic understanding of more complex systems. The important features related to the classic SDOF model shown in Figure 8.4 are:

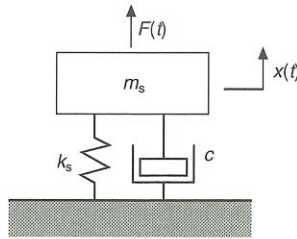


Figure 8.4 Classic SDOF model

1. The equation of motion is given by:

$$m\ddot{x} + c\dot{x} + kx = F(t) \quad (8.4)$$

2. The characteristics of the system are obtained from the free-vibration behaviour, i.e. when $F(t) = 0$.
3. If $F(t) = 0$ and $c = 0$, we have the equation for simple harmonic motion which has the solution:

$$x = X \cos (\omega_n t - \phi), \quad (8.5)$$

where X and ϕ are arbitrary constants (determined from the conditions at $t = 0$) and ω_n is the undamped natural frequency given by:

$$\omega_n = \sqrt{\frac{k}{m}} \quad (8.6)$$

i.e. the system vibrates at this frequency with amplitude X .

4. If damping is included there are two possible free vibration characteristics. If disturbed, the mass either returns to its equilibrium position with or without oscillation (termed underdamped and overdamped respectively). In the former case the oscillations are of progressively reducing amplitude. The characteristics are determined by the relative magnitude of c in relation to m and k . When c is such as to be on the boundary between the two characteristics it is said to critically damped. Then $c = c_c = 2\sqrt{mk}$.

The level of damping in a SDOF system is often described in terms of the damping ratio ζ defined as:

$$\zeta = \frac{c}{c_c} \quad (8.7)$$

The values of ζ for underdamped, critically damped and overdamped are thus < 1 , 1 and > 1 respectively.

5. For an underdamped system ($\zeta < 1$) the solution for free vibration is:

$$x = Xe^{-\zeta\omega_n t} \cos(\omega_d t - \phi) \quad (8.8)$$

where X and ϕ are arbitrary constants and ω_d is the damped natural frequency given by:

$$\omega_d = \omega_n \sqrt{1 - \zeta^2} \quad (8.9)$$

6. If $F(t) \neq 0$ the solution to the equation of motion is made up from two components: the Complementary Function (CF) and the Particular Integral (PI). The CF is identical to the free vibration solution (i.e. equation 8.8 for $\zeta < 1$) and quickly dies away for realistic levels of damping to leave $x = \text{PI}$.
7. If $F(t) = F_0 \sin \omega t$, the steady-state response of the mass (after the CF has become zero) is given by:

$$x = A(\omega) \sin [\omega t - \alpha(\omega)] \quad (8.10)$$

where $A(\omega)$ is the steady-state amplitude and $\alpha(\omega)$ is the phase lag – both are dependent on ω . Note: the steady-state response is at the excitation frequency ω .

8. It may be shown that

$$A(\omega) = F_0 |H(\omega)| = F_0 \left| \frac{1}{(k - m\omega^2) + (c\omega)i} \right| = \frac{F_0}{\sqrt{(k - m\omega^2)^2 + (c\omega)^2}} \quad (8.11)$$

$H(\omega)$ which is complex, is called the frequency response function. It relates the input (excitation) and output (response) in the frequency domain.

9. The amplitude response can be presented conveniently in dimensionless form in terms of the dynamic magnifier $D = kA/F_0$ and frequency ratio $r = \omega/\omega_n$. It may be shown (Rao, 1995) that

$$D = \frac{1}{\sqrt{1 - \left(\frac{\omega}{\omega_n}\right)^2 + \left(2\zeta \frac{\omega}{\omega_n}\right)^2}} \quad (8.12)$$

Figure 8.5 shows D plotted for two different values of ζ .

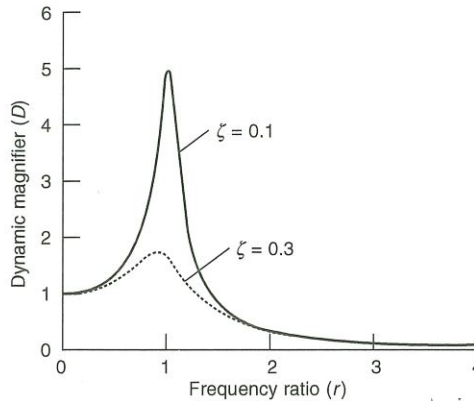


Figure 8.5 Amplitude response of a SDOF system

10. It follows from Figure 8.5 that:

- (a) maximum response amplitude occurs at resonance when $\omega \approx \omega_n$.
- (b) the amplitude is strongly influenced by the level of damping in the system when $\omega \approx \omega_n$. When ω and ω_n are appreciably dissimilar, damping has very little effect on response amplitude. This is an important point when considering the use of damping to control vibration levels.

Multi-degree of freedom systems

Realistic modelling of most forms of automotive vibration requires the use of MDOF models. These have two or more degrees of freedom and lead to two or more equations of motion which can be written in matrix form as shown in equation 8.3. The characteristics of such systems can be determined by considering the free vibration behaviour. This requires the excitation vector in equation 8.3 to be set to zero giving

$$[M]\{\ddot{x}\} + [C]\{\dot{x}\} + [K]\{x\} = \{0\} \quad (8.13)$$

Generally equation 8.13 represents a set of coupled differential equations.

(a) Negligible damping

One can begin to understand the characteristics of MDOF systems best by neglecting damping, i.e. setting $[C] = [0]$. This gives

$$[M]\{\ddot{x}\} + [K]\{x\} = \{0\} \quad (8.14)$$

Assuming solutions of the form $\{x\} = \{A\} e^{st}$ leads to a set of homogeneous equations

$$([M]s^2 + [K])\{A\} = \{0\} \quad (8.15)$$

The non-trivial solution of these is the characteristic equation (or frequency determinant)

$$|[M]s^2 + [K]| = 0 \quad (8.16)$$

This leads to a set of real roots, typically

$$s_i^2 = -\lambda_i = -\omega_i^2 \quad (8.17)$$

where λ_i is the i -th eigenvalue and ω_i the i -th natural frequency. For a system having more than two degrees of freedom it is necessary to find these by numerical methods or by using mathematical software (e.g. MathCAD, 1995). This is most easily accomplished by forming the eigenvalue equation (from equation 8.17).

$$\lambda[M]\{u\} = [K]\{u\} \quad (8.18)$$

Corresponding to each eigenvalue λ_i , there is an eigenvector $\{u\}_i$, which relates the relative amplitudes at each of the degrees of freedom, i.e. they describe a normal (or natural) mode of vibration. The eigenvectors can be found by substituting each eigenvalue in turn back into equation 8.15 or more directly using mathematical software. Vibration in the i -th mode then can be described by

$$\{x\}_i = \{u\}_i A_i \sin(\omega_i t + \alpha_i) \quad (8.19)$$

If the system is given a short duration disturbance at some arbitrary position all modes of vibration will tend to be excited and the ensuing motion will be a combination of these, i.e.

$$\{x\} = \sum_{i=1}^n \{x\}_i = [u]\{q(t)\} \quad (8.20)$$

where:

$$[u] = [\{u\}_1 \{u\}_2 \dots \{u\}_n]$$

is called the modal matrix and

$$\{q(t)\} = \begin{Bmatrix} A_1 \sin(\omega_1 t + \alpha_1) \\ \vdots \\ A_n \sin(\omega_n t + \alpha_n) \end{Bmatrix}$$

is a vector of *modal (principal) coordinates*.

Equation 8.20 represents a linear transformation between generalized coordinates $\{x\}$ and modal coordinates $\{q\}$.

The eigenvectors have a special property called orthogonality, such that when the product of eigenvectors $\{u\}_i$ and $\{u\}_j$ is formed with either the mass or stiffness matrices the result is zero, i.e. $\{u\}_i^T [M] \{u\}_j = 0$ and $\{u\}_i^T [K] \{u\}_j = 0$, provided that $i \neq j$,

If $i = j$ the result is $\{u\}_i^T [M] \{u\}_i = M_{ii}$ and $\{u\}_i^T [K] \{u\}_i = K_{ii}$ where M_{ii} and K_{ii} are called the *modal mass* and *modal stiffness* respectively.

The orthogonality property can be used to uncouple the equations of motion and express the equations in modal coordinates. This facilitates the solution of the general forced vibration problem and highlights the contribution which the excitation makes to each of the modes.

Replacing $\{x\}$ with $[u]\{q\}$ in equation 8.3, premultiplying by $[u]^T$ and applying the orthogonality condition gives:

$$\text{diag}[M]\{\ddot{q}\} + \text{diag}[K]\{q\} = [u]^T \{F(t)\} \quad (8.21)$$

where $\text{diag}[M]$ and $\text{diag}[K]$ are diagonal matrices containing modal mass and stiffness elements respectively. The i -th equation (of the set of n) is of the form

$$M_{ii}\ddot{q}_i + K_{ii}q_i = \{u\}_i^T \{F(t)\} \quad (8.22)$$

Hence a set of uncoupled equations is formed, each of which is similar to that for forced vibration of a SDOF system. When the set of solutions $\{q\}$ have been obtained they can be transposed back into generalized coordinates using the transformation $\{x\} = [u]\{q\}$.

(b) Viscous damping

In (stable) lightly damped systems the frequency determinant is

$$| [M]s^2 + [C]s + [K] | = 0 \quad (8.23)$$

For an n -DOF system this produces n complex conjugate roots having negative real parts providing information about the frequency and damping associated with each mode of vibration. There is a possibility that some of the roots will be equal or have zero real and/or imaginary parts. The latter being the case if rigid body motion is possible.

Except in those cases where damping has been deliberately added, damping in automotive systems is so low as to have a negligible effect on the natural frequencies and modes of vibration. However, damping must be considered in the analysis if the response of the system is required for a relatively short period of time in comparison to the natural periods of the system or when one or more components of a periodic excitation is at or near to one of the system's natural frequencies.

(c) Forced-damped vibration (harmonic excitation)

Since most of the excitations in automotive systems are of a periodic nature, this aspect of vibration analysis is of great importance to us. In general, it is possible (using Fourier series) to decompose any periodic signal into a set of harmonic components having differing amplitudes and frequencies. After determining the response to each component, it is then possible to determine the overall response by adding the individual responses together using the principle of superposition provided the system is linear. This approach allows us to develop the solution in the frequency domain relating the response at location i to the excitation at location j via a frequency response function.

Many of the features of harmonic response analysis of SDOF systems extend to MDOF ones, e.g.:

- (1) When subjected to harmonic excitation an MDOF system vibrates at the same frequency as the excitation.
- (2) The displacement amplitudes at each of the degrees of freedom are dependent on the frequency of excitation and
- (3) The dynamic displacement at each DOF lags behind the excitation.

In MDOF systems the excitation can be applied simultaneously at any of the DOFs. For a linear system the response at any of the DOFs is the sum of the responses due to each excitation force.

Consider a MDOF system with viscous damping subjected to a set of harmonic excitations $\{F(t)\} = \{F\}f(t) = \{F\} \sin(\omega t)$ applied at the DOFs. Equation 8.3 can be written

$$[M]\{\ddot{x}\} + [C]\{\dot{x}\} + [K]\{x\} = \{F\}f(t) \quad (8.24)$$

By taking Laplace transforms of both sides with zero initial conditions, replacing s with $i\omega$ (Schwarzenbach *et al.*, 1984), and pre-multiply both sides by $(-\omega^2[M] + i\omega[C] + K)^{-1}$ results in

$$\{H_x(\omega)\} = [H(\omega)] \{F\} \quad (8.25)$$

$\{H_x(\omega)\}$ is a vector of frequency responses at the DOFs and $[H(\omega)]$ is a matrix of frequency response functions such that H_{ij} = frequency response at i due to unit amplitude excitation at j .

$$\{H(\omega)\} = \begin{bmatrix} H_{11} & H_{12} & \cdots & H_{1n} \\ H_{21} & H_{22} & \cdots & H_{2n} \\ \vdots & \vdots & \ddots & \vdots \\ H_{n1} & \cdots & \cdots & H_{nn} \end{bmatrix}, \quad (8.26)$$

Thus the frequency response at DOF i is

$$H_{xi}(\omega) = H_{i1}F_1 + H_{i2}F_2 + \dots + H_{in}F_i = \sum_{j=1}^n H_{ij}F_j \quad (8.27)$$

i.e. is made up of contributions from excitations at the various DOFs in the systems.

The frequency response functions (and hence the frequency responses) are complex if damping is included in the analysis. Amplitude and phase response at each DOF is given by taking the modulus and argument of the elements of $\{H_x(\omega)\}$ respectively.

(d) Forced damped vibration (random excitation)

Random excitation arises particularly from terrain inputs and is important in the analysis and design of vehicle suspensions. This form of excitation is non-deterministic in that its instantaneous value cannot be predicted at some time in the future. There are, however, some properties of random functions which can be described statistically. The mean or mean square value can be determined by averaging and the frequency content can be determined from methods based on the Fourier transform (Newlands, 1975).

The effect of random excitation on suspension design and analysis will be discussed in more detail in the chapter on Suspension Systems and Components in Chapter 10.

8.3 Vibration control

While it is recognized that the ideal form of vibration control is 'control at source', there is a limit to which this can be performed. For example the most dominant on-board source of vibration in motor vehicles is the engine. Here, engine firing and reciprocating unbalance combine to produce a complex source of vibration which varies with engine operating conditions. Reciprocating unbalance arises at each cylinder because of the fluctuating inertia force associated with the mass at each piston. This force acts along the cylinder axis and in multi-cylinder engines gives rise to a shaking force and moment acting on the engine block. By carefully arranging the relative crank positions on the crank shaft it is possible to reduce these forces and moments significantly, but because the forces contain a number of higher harmonic components the unbalance effect can never be completely removed. This topic will be discussed in more detail later in Section 8.3.5.

Another important source of on-board vibration is due to the unbalance of rotating parts.

While these may meet balancing standards it should be appreciated that there is no such thing as 'perfect balance'. Hence, small amounts of allowable residual unbalance are there to cause unwanted levels of vibration.

It follows that even when the best practices are followed there will always be some unwanted sources of vibration present. It is then necessary to minimize the effect of these on driver and passengers. In this section we review some of the ways in which this can be achieved.

8.3.1 Vibration isolation

This is a way of localizing the vibration to the vicinity of the source, thereby preventing its transmission to other parts of a (vehicle body) structure where it may result in the generation of noise. It can be achieved by the use of either passive or controllable vibration isolators. Passive isolators range from simple rubber in shear or combinations of shear and compression components to quite sophisticated hydro-elastic elements. The simpler forms of isolator are a cost-effective solution for sources with a limited range of operating conditions (amplitudes and frequencies). For situations where the source of vibration produces a range of operating conditions (as in i.c. engines) it is necessary to consider the use of hydro-elastic or controllable isolators. Some of these devices are discussed below. In all cases it is essential to understand the basic principles to achieve the best results.

The basic principles for selecting the appropriate isolator can be illustrated with reference to the SDOF model shown in Figure 8.6 (a more detailed treatment can be found in Snowdon, 1968). This represents a machine of mass m , subjected to a harmonic excitation arising from rotating unbalance $m_e r$, supported on elastomeric mounts having a complex stiffness k^* described by

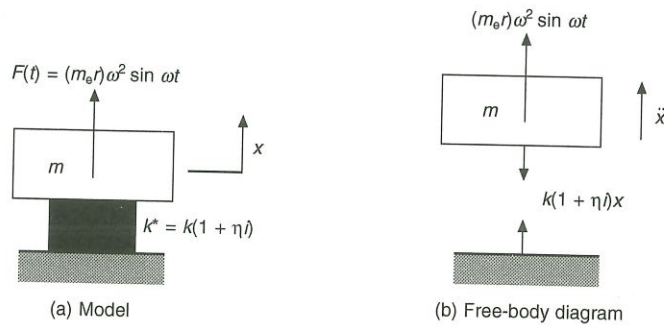


Figure 8.6 SDOF vibration isolation model and free-body diagram

$k^* = k(1 + \eta i)$, where k is the dynamic stiffness and η the loss factor.

The effectiveness of the isolation (a function of frequency ω) can be defined by the transmissibility

$$T(\omega) = \frac{P}{F_0} \quad (8.28)$$

where P = the amplitude of the force transmitted to the foundation and F_0 is the amplitude of the excitation force due to unbalance. Applying Newton's second law to the FBD gives:

$$m\ddot{x} + k(1 + \eta i)x = m_e r \omega^2 \sin \omega t = F_0 f(t) \quad (8.29)$$

and the force transmitted to the foundation is

$$P(t) = k(1 + \eta i) x(t) \quad (8.30)$$

Employing the approach outlined in Section 8.2.4 for harmonic excitation it can be shown that:

$$T(\omega) = k \sqrt{\frac{1 + \eta^2}{(k - m\omega^2)^2 + (k\eta)^2}} \quad (8.31)$$

To appreciate how $T(\omega)$ varies with frequency it is helpful to show equation 8.31 in dimensionless form by dividing numerator and denominator by k we then have

$$T(\omega) = \sqrt{\frac{1 + \eta^2}{\left[1 - \left(\frac{\omega}{\omega_n}\right)^2\right]^2 + \eta^2}} \quad (8.32)$$

where $\omega_n = \sqrt{\frac{k}{m}}$ is the natural frequency of the mass on its isolators.

For low damping elastomers η is of the order of 0.05 (Snowdon, 1968). The variation of transmissibility with frequency ratio $r = \frac{\omega}{\omega_n}$ for $\eta = 0.05$ is shown in Figure 8.7.

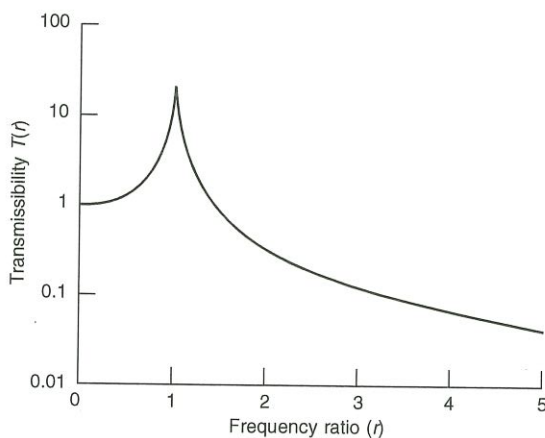


Figure 8.7 Transmissibility of an elastomeric isolator

For the isolators to be effective the transmissibility must be less than unity, i.e. P must be less than F_0 . From Figure 8.7 it is clear that frequency ratios from 0 to approximately 1.4, P is greater than F_0 and therefore the isolators **magnify** the force to the foundation in this range.

Importantly around resonance ($r = 1$) the isolators magnify the force to the foundation approximately 20 times emphasizing the danger of not doing some simple analysis. As the value of r increases beyond 1.4 the isolators become increasingly effective. However, for very large values of r it is possible to induce wave-effects in isolators (Snowdon, 1968). These are due to local resonances in the distributed mass and elasticity of the isolator material and produce additional resonance peaks in the transmissibility curve (and reduced isolator performance) at certain frequencies of excitation.

An alternative way of describing isolator effectiveness is to use the term isolator efficiency. This is defined as

$$E_{\text{iso}} = [1 - T(\omega)] 100 \quad (8.33)$$

and expressed as a percentage.

8.3.2 Tuned absorbers

Vibration absorbers are useful for reducing vibration levels in those systems in which an excitation frequency is close to or coincides with a natural frequency of the system. Such absorbers consist of a spring-mass sub-system which is added to the original system. In effect, energy is transferred from the original system to the absorber mass which can vibrate with significant amplitude depending on the amount of damping contained in the absorber sub-system. These devices are particularly effective for reducing large amplitude oscillations in the original system but add another degree of freedom to the overall system, producing new natural frequencies above and below the original natural frequency. These result in resonant amplitudes (which can be controlled with an appropriate choice of damping in the absorber), permitting them to be used for variable speed applications.

The principles of undamped and damped tuned absorbers can be understood by outlining first the analysis of the damped absorber and then treating the undamped absorber as a special case of this. Assume the problem system has a SDOF with harmonic excitation as shown in Figure 8.8(a). By adding an absorber to the original system, the two-DOF system shown in Figure 8.8(b) is formed.

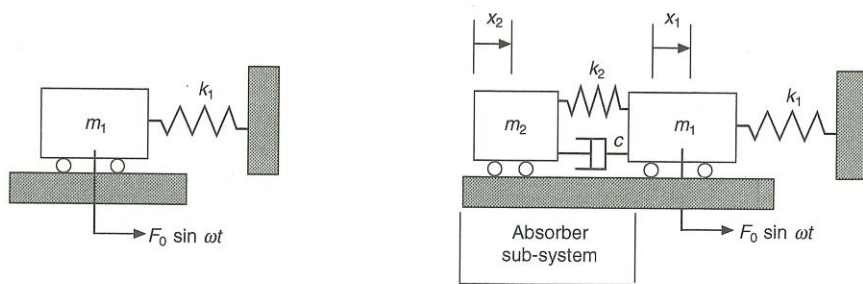


Figure 8.8 Models for analysing the tuned absorber. (a) Original system; (b) original system with absorber

The natural frequency of the original system is $\omega_1 = \sqrt{\frac{k_1}{m_1}}$ and at resonance $\omega = \omega_1$. By drawing the FBDs for the two mass system, it can be shown that the equations of motion in matrix form are:

$$\begin{bmatrix} m_1 & 0 \\ 0 & m_2 \end{bmatrix} \begin{Bmatrix} \ddot{x}_1 \\ \ddot{x}_2 \end{Bmatrix} + \begin{bmatrix} c & -c \\ -c & c \end{bmatrix} \begin{Bmatrix} \dot{x}_1 \\ \dot{x}_2 \end{Bmatrix} + \begin{bmatrix} (k_1 + k_2) & -k_2 \\ -k_2 & k_2 \end{bmatrix} \begin{Bmatrix} x_1 \\ x_2 \end{Bmatrix} = \begin{Bmatrix} F_0 \\ 0 \end{Bmatrix} \sin \omega t \quad (8.34)$$

Following the method outlined in Section 8.2.4 it can be shown that the (complex) steady state responses $\bar{X}_1$ and $\bar{X}_2$ of the two masses are given by:

$$\begin{Bmatrix} \bar{X}_1 \\ \bar{X}_2 \end{Bmatrix} = \begin{bmatrix} ((k_1 + k_2 - m_1\omega^2) + c\omega i) & -k_2 \\ -k_2 & ((k_2 - m_2\omega^2) + c\omega i) \end{bmatrix}^{-1} \begin{Bmatrix} F_0 \\ 0 \end{Bmatrix}$$

This produces the amplitudes of vibration

$$X_1 = F_0 \sqrt{\frac{(k_2 - m_2\omega^2)^2 + (c\omega)^2}{[(k_2 - m_2\omega^2)(k_1 + k_2 - m_1\omega^2) - k_2^2]^2 + (c\omega)^2(k_1 - m_1\omega^2 - m_2\omega^2)^2}} \quad (8.35)$$

and

$$X_2 = \frac{F_0 k_2}{\sqrt{[(k_2 - m_2\omega^2)(k_1 + k_2 - m_1\omega^2) - k_2^2]^2 + (c\omega)^2(k_1 - m_1\omega^2 - m_2\omega^2)^2}} \quad (8.36)$$

The undamped tuned absorber ($c = 0$)

In this case the amplitudes are given by:

$$X_1 = \frac{F_0(k_2 - m_2\omega^2)}{\Delta(\omega)} \quad (8.37, 8.38)$$

$$X_2 = \frac{F_0 k_2}{\Delta(\omega)}$$

where

$$\Delta(\omega) = (k_2 - m_2\omega^2)(k_1 + k_2 - m_1\omega^2) - k_2^2 \quad (8.39)$$

In order to make X_1 zero, $k_2 - m_2\omega^2$ must be zero and hence $\omega = \sqrt{\frac{k_2}{m_2}} = \omega_2$ the natural frequency of the absorber sub-system. It follows that for this case $\omega_1^2 = \omega_2^2 = \frac{k_1}{m_1} = \frac{k_2}{m_2}$ and hence the natural frequencies of the two sub-systems must be the same.

Resonance of the complete 2-DOF system (i.e. when X_1 and X_2 tend to infinity) occurs when ω coincides with the system's natural frequencies Ω_1 and Ω_2 . This occurs when $\Delta(\omega) = 0$, representing the characteristic equation for the 2-DOF system.

In designing an untuned absorber it is necessary to consider the magnitude of the absorber mass in relation the original system mass m_1 . In general the larger the mass ratio $\mu = m_2/m_1$, the more widely separated are the frequencies Ω_1 and Ω_2 and the wider is the range of frequencies at which the system can operate without exciting resonance.

The general response of the complete system is best described in terms of dimensionless amplitudes and frequencies ratios. Denoting the dimensionless amplitudes of m_1 and m_2 as $A_1 = \frac{k_1}{F_0} X_1$ and $A_2 = \frac{k_1}{F_0} X_2$ together with the frequency ratio as $r = \frac{\omega}{\omega_1}$ enables equations 8.37 and 8.38 to be written as:

$$A_1 = \frac{1 - r^2}{(1 - r^2)(1 + \mu - r^2) - \mu} \quad (7.40)$$

$$A_2 = \frac{1}{(1 - r^2)(1 + \mu - r^2) - \mu} \quad (7.41)$$

This enables the amplitude responses to be plotted for various values of μ . Figure 8.9 shows plots for $\mu = 0.2$. In Figure 8.9(a) A_{10} is the amplitude response of the original system.

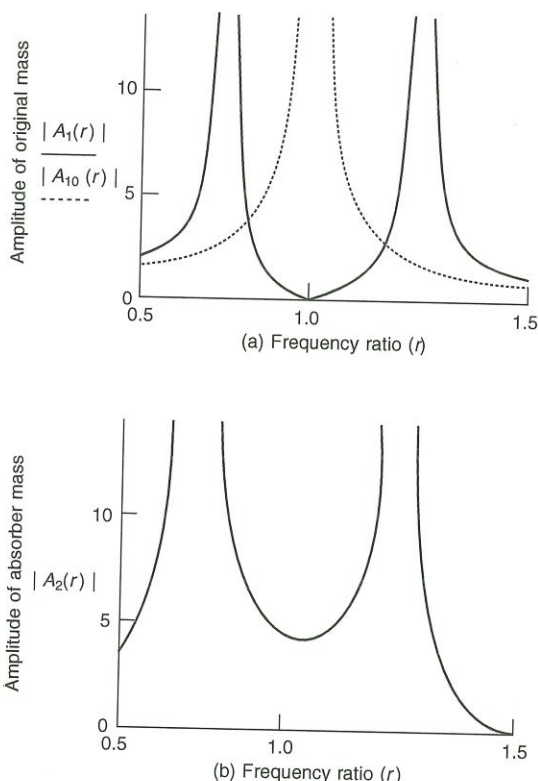


Figure 8.9 Frequency responses of a system fitted with an untuned absorber ($m = 0.2$)

The damped tuned absorber

The undamped absorber transfers energy from the original system to the absorber sub-system resulting in large amplitudes of vibration of the absorber mass. This can lead to the possibility of fatigue failure in the absorber spring. To overcome this problem it is necessary in practice to add some damping to the absorber. This also allows a wider operating range and limits the

resonant amplitudes in the region of the two natural frequencies. In this case the system is represented by the model in Figure 8.8(b) and the amplitude responses are given by equations 8.35 and 8.36. When $c = \infty$ the two masses in the system are effectively locked together resulting in a new undamped SDOF system having a natural frequency $\Omega_n = \sqrt{k_1/(m_1 + m_2)}$. When the response for this case is superimposed on that for $c = 0$, the response curves intersect at two points P and Q . This is shown in Figure 8.10. It can be shown (Dimaragonas *et al.*, 1992) that when $0 \leq c < \infty$ the amplitude response A_1 passes through P and Q as shown in Figure 8.10 for the case $\omega_1 = \omega_2$ and $\mu = 0.2$, illustrating how the resonant response amplitudes are limited. By careful optimization of the parameters (Snowdon, 1968) it is possible to minimize the resonant amplitudes. Unlike the case of the undamped absorber, it is not possible to reduce the amplitude of mass m_1 to zero at the original natural frequency of the system. Thus some of the effectiveness of an absorber is lost at this frequency when damping is introduced.

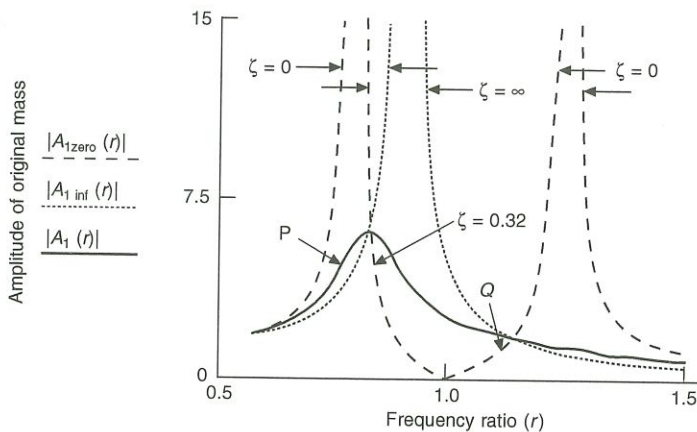


Figure 8.10 Response resulting from a damped absorber ($m = 0.2$)

8.3.3 Untuned viscous dampers

While tuned absorbers are tuned to a particular system resonance, an untuned viscous damper is a device designed to generally increase damping in a system and thereby reduce resonant amplitudes across a wide range of frequencies. These devices consist of an inertia (seismic) mass which is coupled to the original system via some form of damping medium, usually silicone fluid.

These devices are commonly used to limit torsional oscillations in crankshafts which have a number of natural frequencies and are subjected to a wide range of excitation frequencies. The torsional damper consists of a free rotating disc mounted on bearings inside the casing of the damper which is filled with silicone fluid. The casing of the damper is attached to the crankshaft at the opposite end to the flywheel. The principles of vibration control can be studied by assuming that the mass-elastic model of the crank shaft can be simplified down to a single rotating mass supported on a shaft fixed at one end with the mass subjected to a harmonic

excitation torque $T_0 \sin \omega t$. An application of this will be discussed in more detail in Section 8.3.5. The resulting model is shown in Figure 8.11(a) where I_1 is the inertia of the mass about the shaft axis and K is the torsional stiffness of the shaft. Adding the damper of inertia I_2 and damping coefficient C results in the 2-DOF system shown in Figure 8.11(b). It is assumed here that the masses of the fluid and damper casing are negligible. θ_1 and θ_2 are the angular position of the two masses.

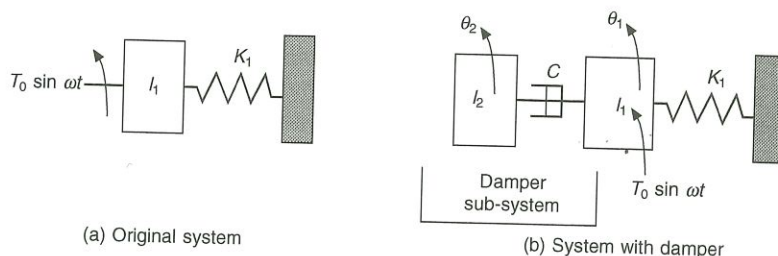


Figure 8.11 Models for analysing the untuned viscous damper

The equations of motion are

$$I_1 \ddot{\theta}_1 + C \dot{\theta}_1 + K \theta_1 - C \dot{\theta}_2 = T_0 \sin \omega t \quad (8.42, 8.43)$$

$$I_2 \ddot{\theta}_2 - C \dot{\theta}_1 + C \dot{\theta}_2 = 0$$

Comparing with equations 8.34 and 8.35 it follows that the amplitude of vibration of the mass in the original system is

$$\theta_1 = T_0 \sqrt{\frac{(I_2 \omega^2)^2 + (C \omega)^2}{[(I_2 \omega^2)(K_1 - I_1 \omega^2)]^2 + (C \omega)^2 (K_1 - I_1 \omega^2 - I_2 \omega^2)^2}} \quad (8.44)$$

Using the following notation:

undamped natural frequency of the original system, $\omega_n = \sqrt{\frac{K}{I_1}}$, damping ratio, $\zeta = \frac{C}{2\sqrt{I_1 K}}$, inertia ratio, $\mu = \frac{I_2}{I_1}$, dimensionless amplitude of I_1 , $A_1 = \frac{K \theta_1}{T_0}$ and frequency ratio $r = \frac{\omega}{\omega_n}$, it can be shown that

$$A_1 = \sqrt{\frac{(\mu r)^2 + 4\zeta^2}{(\mu r)^2 (1 - r^2)^2 + 4\zeta^2 [\mu r^2 - (1 - r^2)]^2}} \quad (8.45)$$

A_1 is thus a function r , μ and ζ . For a given value of ζ the response will exhibit a single peak similar to that for a damped SDOF system. The extreme values of damping are $\zeta = 0$ and ∞ .

When $\zeta = 0$ the system response is that of the original SDOF system having a natural frequency ω_0 , and when $\zeta = \infty$, both masses move together as one and the undamped natural frequency is $\sqrt{K/(I_1 + I_2)}$. When A_1 is plotted on the same axes for these two extreme cases and for a given μ , the curves intersect at a point P . It can be shown (Thomson, 1989) that the curves for other values of damping also pass through P . These features are illustrated in Figure 8.12.

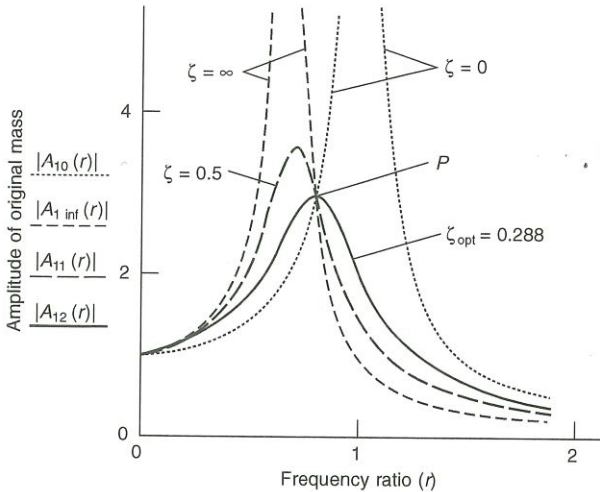


Figure 8.12 Response of an untuned viscous damper ($m = 1.0$)

Clearly the optimum value of damping is the one which has its maximum value at P . It can be shown (Thomson, 1989) that this is given by:

$$\zeta_{\text{opt}} = \frac{\mu}{\sqrt{2(1 + \mu)(2 + \mu)}} \quad (8.46)$$

In general the minimum peak amplitude decreases as μ and ζ increases up to the value given by equation 8.46.

8.3.4 Damping treatments

Damping treatments (in the form of high damping polymers) can be used to limit resonant response amplitudes in structures and are particularly effective for flexural vibration of panels and beams. In an automotive context they are used extensively to limit the resonant responses of body panels and bulkheads.

Because of the poor structural strength of high damping polymers it is necessary to either bond them to the surface of load-bearing elements or to incorporate them into load-bearing

elements by sandwich construction. These forms of damping are termed *unconstrained-* and *constrained-layer damping* respectively, with the latter being by far the most effect way of deploying this type of structural damping treatment. Flexing of the load-bearing element produces shearing effects in the damping layer and thus vibrational energy is converted into heat and dissipated. Shear properties of polymer materials are generally temperature and frequency dependent. Furthermore their use in the form of constrained layer damping poses problems with bending and forming in manufacturing operations (Beards, 1996).

8.3.5 Applications

Two practical examples will be presented, both are of considerable importance in automotive vibration control. The first of these is concerned with isolation of the engine from the vehicle structure and the other relates to the control of torsional oscillation amplitudes in engine crankshafts.

In order to understand how the above principles relate to these two examples it is first of all necessary to understand the processes by which the engine vibration is generated. The first part of this sub-section therefore outlines how the excitation forces and moments arise in single cylinder and multi-cylinder i.c. engines.

Dynamic forces generated by i.c. engines

(a) Single cylinder engines (Shigley *et al.*, 1980; Norton, 1992).

The dynamic forces exerted on single cylinder engines arise from the reaction torque at the crankshaft and a shaking force acting along the line of stroke. Both are cyclic and related to engine speed. The torque on the crankshaft consists of a component generated by the gas force in the engine cylinder and on the inertia force component, associated with the acceleration of the connecting rod and piston assembly. The shaking force is attributed solely to the inertia forces. The component of torque arising from the gas forces has a fundamental frequency equal to running speed for a two-stroke engine and at half engine speed for a four-stroke engine, with higher order harmonics (called engine orders). The kinematics of reciprocating engine mechanisms results in inertia torque and shaking force components which have fundamental components at engine speed with higher order harmonics. For four-stroke engines this leads to half order components since an engine cycle is half the engine speed.

The fluctuating reaction torque and shaking force produce translational and rotational vibration of the engine and calls for careful mounting to minimize transmission of vibration to the vehicle body.

(b) Multi-cylinder engines

Since the gas torque from each cylinder is basically a pulse occurring during the expansion stroke it is essential that for a multi-cylinder engine the torque contributions from each cylinder are evenly spaced to give a regular train of pulses. This dictates that the crank throws in the order of firing must have a regular angular spacing. For two-stroke and four-stroke engines this is $360/n$ and $720/n$ respectively, where n is the number of cylinders.

Also, in a multi-cylinder engine, the shaking forces act along the line of each cylinder

(Figure 8.13) producing a shaking moment on the engine about an axis perpendicular to the plane containing the cylinders. For the example shown this is given by:

$$\vec{M}_{\text{Total}} = \sum_{i=1}^n \vec{z}_i \otimes \vec{F}_i \quad (8.47)$$

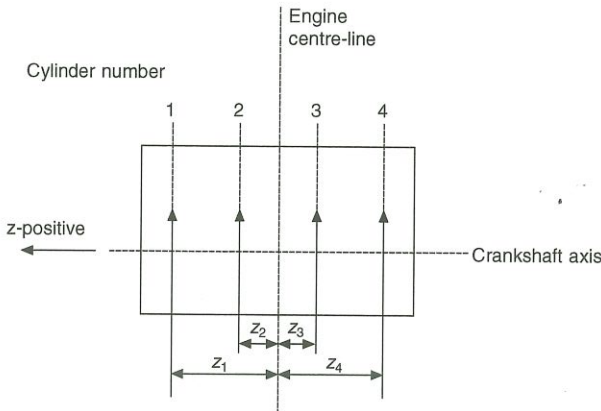


Figure 8.13 Axial location of cylinders relative to centre-line

By careful arrangement of the firing order and relative angular position of the cranks it is possible to cancel out some of the shaking force and shaking moment components. For example for a four cylinder four-stroke engine with equally spaced cylinders numbered from 1 at one end to 4 at the other, the relative crank spacings are at 180 degree intervals and possible firing orders are 1-3-4-2 and 1-2-4-3. It can be shown (Norton, 1992) with the latter firing order, that the first and third order inertia torques are balanced, there are 2nd, 4th and 6th order shaking force components and a second order shaking moment component (the 1st, 4th and 6th being balanced). Norton (1992) gives some general equations which can be used to check the state of balance of multi-cylinder engines.

Engine isolation

The variable operating conditions of automotive engines presents a complex problem in vibration isolation for the engineer. In addition to the fluctuating torque at the crankshaft and the shaking forces and moments identified in the previous section there are additional dynamic inertial loads arising from vehicle manoeuvring and terrain inputs to the wheels.

The primary components of engine vibration at idling are integer multiples of engine speed and the dominant component in four cylinder engines occurs at twice the engine speed and results from the combustion pulses. Idle speeds for four cylinder engines are typically in the range from 8–20 Hz producing dominant frequency components in the range from 16–40 Hz. Since the primary bending mode of passenger cars can be less than 20 Hz it is obvious that it is easy to excite body resonance at idle if engine isolation is not carefully designed.

When supported on its vibration isolators the powertrain (engine and gearbox) can be treated as a rigid body having six degrees of freedom (three translational and three rotational). In order

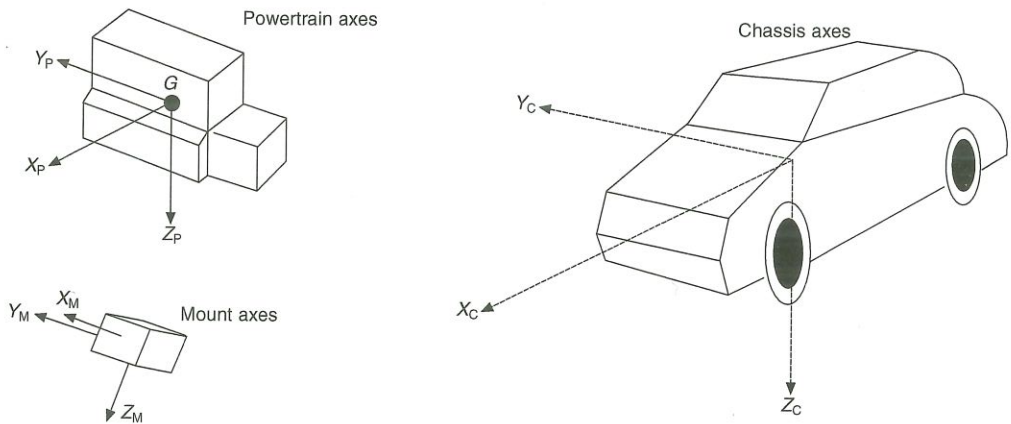


Figure 8.14 Reference axes for analysing the dynamics of a powertrain-mount system

to uncouple these motions the isolators should lie in planes aligned with the principal inertial axes passing through the centre of gravity of the unit. The problem facing the chassis mounting engineer is to select a set of appropriate mounts and position them in such a way as to isolate the chassis from the above excitations and restrain the engine against excessive movement due to the engine torque. In general, engine mounts require complex characteristics to meet the above demands. Some of the important features of these will be discussed below.

(a) Formulation of the vibration equations

In order to formulate the problem for dynamic analysis one must consider the position and orientation of the engine relative to a set of chassis-based coordinates together with the orthogonal characteristics of each mount, its location and orientation. Typical sets of axes are shown in Figure 8.15. In the static equilibrium position the mass centre G of the powertrain coincides with the origin of the chassis fixed axis system. In the following development of the equations of motion it will be assumed that the displacements of the powertrain are small.

Denoting the position of the i -th mount attachment point relative to the powertrain in the equilibrium position as $\vec{r}_{i0}$ (in chassis coordinates) and the translation and rotation of the powertrain relative to the chassis axes as $\vec{r}_G$ and $\vec{\Theta}$, the deflection of the mount is given by

$$\vec{d}_{ci} = \vec{r}_i - \vec{r}_{i0}, \quad (8.48)$$

where

$$\vec{r}_i = \vec{r}_G + \vec{\Theta} \otimes \vec{\rho}_i \quad (8.49)$$

and $\vec{\rho}_i$ is the location of the mount attachment point relative to the powertrain axes.

If the coordinate transformation relating the orientation of the mount axes to those of the chassis is $[T_{mc}]$, then the mount deflection in terms of mount coordinates is

$$\{d_m\}_i = [T_{mc}]_i \{d_c\}_i \quad (8.50)$$

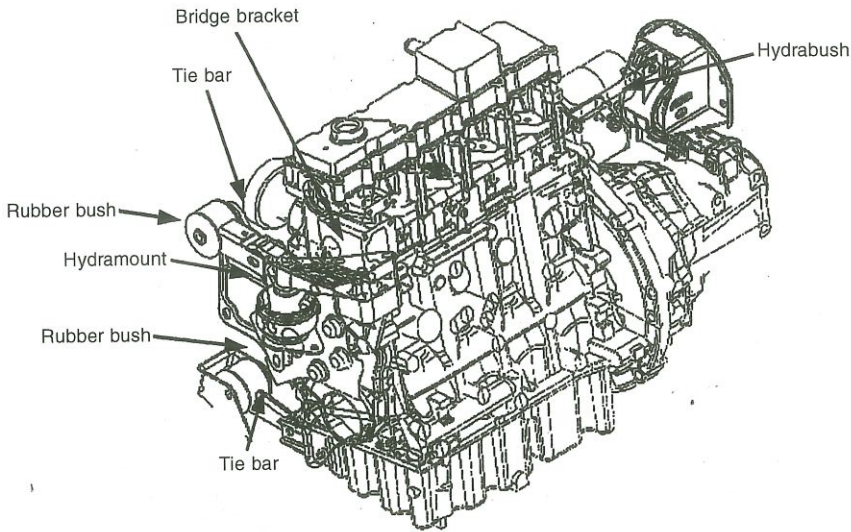


Figure 8.15 A torque axis engine mounting system (courtesy of Rover Group Ltd)

The 'elastic' potential energy for a set of mounts typically having orthogonal complex stiffness components represented by the diagonal stiffness matrix $[k_m]_i$ is

$$V = \frac{1}{2} \sum_{i=1}^n \{d_m\}_i^T [k_m]_i \{d_m\}_i \quad (8.51)$$

in terms of mount displacements. Using the transformation in equation 8.50 the equivalent equation in terms of chassis coordinates is

$$V = \frac{1}{2} \sum_{i=1}^n \{d_c\}_i^T [k_c]_i \{d_c\}_i \quad (8.52)$$

where $[k_c]_i = [T_{mc}]_i^T [k_m]_i [T_{mc}]_i$ is a non-diagonal matrix.

The kinetic energy of the system is given by

$$T = \frac{1}{2} m \dot{\vec{r}}_G \cdot \dot{\vec{r}}_G + \frac{1}{2} \dot{\vec{\Theta}} \cdot \vec{h}_G \quad (8.53)$$

where $\vec{h}_G$ is the angular momentum of the powertrain about G , the components of which are given by $\{h_G\} = [M]\{\dot{\Theta}\}$.

$[M]$, the symmetrical mass/inertia matrix, is of the form:

$$[M] = \begin{bmatrix} m & 0 & 0 & 0 & 0 & 0 \\ 0 & m & 0 & 0 & 0 & 0 \\ 0 & 0 & m & 0 & 0 & 0 \\ 0 & 0 & 0 & I_{xx} & -I_{xy} & -I_{xz} \\ 0 & 0 & 0 & -I_{yx} & I_{yy} & -I_{yz} \\ 0 & 0 & 0 & -I_{zx} & -I_{zy} & I_{zz} \end{bmatrix} \quad (8.54)$$

where m is the mass of the powertrain, elements I_{xx} , I_{yy} , I_{zz} are moments of inertia and I_{xy} etc are products of inertia about the powertrain axes. The equations of motion can then be derived from the linearized form of Lagrange's equations (Meirovitz, 1986):

$$\frac{d}{dt} \left(\frac{\partial T}{\partial \dot{q}_k} \right) + \frac{\partial V}{\partial q_k} = Q_k, \quad k = 1 \dots 6 \quad (8.55)$$

where q_k are the generalized coordinates (components of $\vec{r}_G$ and $\vec{\Theta}$ and Q_k are a set of generalized forces derived from the excitations imposed on the powertrain). The result is a set of six equations in matrix form:

$$[M]\{\ddot{q}\} + [K]\{q\} = \{Q\} \quad (8.56)$$

These are a set of equations coupled in both the mass and stiffness matrices, the elements of the latter being complex (containing dynamic stiffness and loss factor values). The equations can be solved for harmonic inputs using the techniques described earlier.

The dynamic forces transmitted to the chassis can be determined for a particular mount configuration by solving the above equations and translating the dynamic displacements $\{q\}$ on the powertrain into mount deflections $\{d_c\}$ in chassis coordinates and transposing into mount coordinates with equations 8.50. The dynamic force components transmitted to the chassis at the i -th isolator is then given by

$$\{F_m\}_i = [k_m]_i \{d_m\}_i. \quad (8.57)$$

(b) Mount requirements and types

The requirements for engine mounts are:

- (i) a low spring rate and high damping during idling and
- (ii) a high spring rate and low damping for high speeds, manoeuvring and when traversing rough terrains.

The following types of mount attempt to meet these conflicting requirements:

- (i) Simple rubber engine mounts
These are the least costly and least effective forms of mount and clearly do not meet all the conflicting requirements listed above. They do not provide the high levels of damping required at idling speeds.
- (ii) Hydro-elastic mounts
These generally contain two elastic reservoirs filled with a hydraulic fluid. Some also contain a gas filled reservoir. This type of mount exploits the feature of mass-augmented dynamic damping which is a form of tuned vibration absorber. In operation there is relative motion across the damper which produces flexure of the rubber component and transfer of fluid between chambers, thereby inducing a change in mount transmissibility. This type of mount has become common in recent years and some examples are given in the literature (Kim *et al.*, 1992; Muller *et al.*, 1996).
- (iii) Semi-active mounts
The operation of these mounts is dependent on modifying the magnitude of the forces

transmitted through coupling devices. They may be implemented via low-bandwidth low-power actuators which are suited to open-loop control. Some forms of hydraulic semi-active (adaptive) mount use low powered actuators to induce changes in mount properties by modifying the hydraulic parameters within the mount. The actuators may then be on-off (adaptive) or continuously variable (semi-active) types. Considerable effort has been devoted to this type of technology in recent years. Some examples are given in Morishita *et al.* (1992) and Kim *et al.* (1993).

(iv) Active mounts

This type of mount requires control of both the magnitude and direction of the actuator force used to adjust the coupling device. High-speed actuators and sensors require having an operating bandwidth to match the frequency spectrum of the disturbance. Power consumption is generally high in order to satisfy the response criteria. Active vibration control is typically implemented by closed-loop control. An example of active engine mount modelling and performance is given by Miller *et al.* (1995).

(c) A typical example

A practical implementation of an engine mounting system for a four-cylinder diesel engine is shown in Figure 8.16. It comprises two mass carrying mounts (one a hydramount, the other a hydrabush, both passive mounts) and two torque reacting tie bars. The hydramount is linked to the power unit by an aluminium bridge bracket. Both tie bars have a small bush at the power unit end and a large bush at the body end. The lower tie bar has its power unit end carried by a bracket attached to the sump and its body end attached to a subframe which also carries the vehicle suspension. The vertical stiffnesses of the mass carriers have very little effect on the torque performance of the system and can therefore be tuned for ride. The function of the hydramounts is of course to improve ride. The tie bar fore and aft rates do not affect ride and can be tuned for the torque loading.

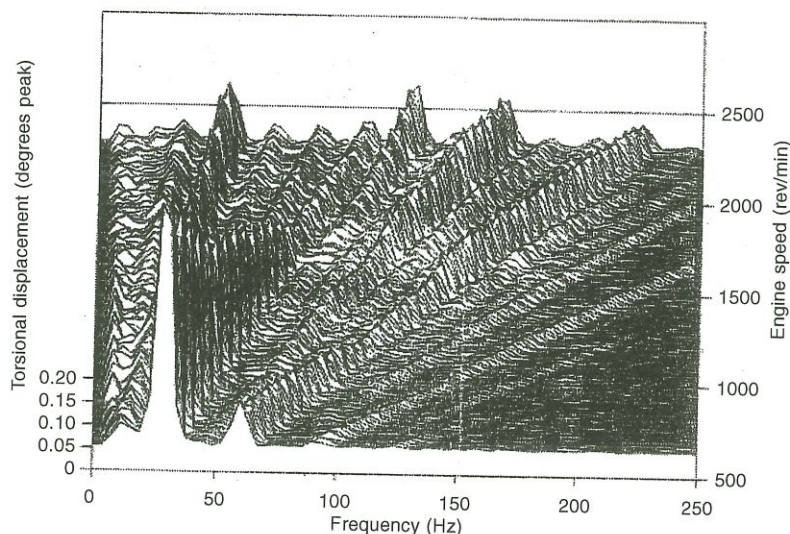


Figure 8.16 Waterfall plot for a multi-cylinder engine (courtesy of Simpson International (UK) Ltd)

Crankshaft damping

The torsional dynamics of crankshafts are dependent on the distribution of their mass and elasticity (defining the modal characteristics) and the excitations arising from the torque/cylinder discussed earlier in this section. Because the torque contains a number of harmonic components and engine speed is variable there is a tendency to excite a large number of torsional resonances as illustrated by the waterfall plot (torsional amplitude plotted as a function frequency for a range of engine speeds) in Figure 8.17.

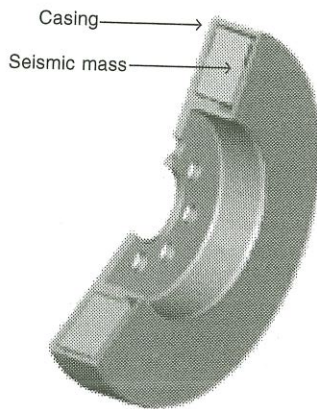


Figure 8.17 Typical crankshaft damper (courtesy Simpson International (UK) Ltd)

Because crankshafts are lightly damped in torsion, the resulting resonant amplitudes can be large, resulting in high cyclic stresses which tend to cause fatigue failure. One solution to this problem is to introduce some torsional damping into the system, with emphasis on damping the fundamental mode of vibration because this tends to be subjected to the highest torsional amplitudes. A typical crankshaft damper is shown in Figure 8.17 and a simplified selection procedure is outlined as follows.

(a) Determination of the mass-elastic model

The case of a 6-cylinder in-line diesel engine is taken as an example. The objective is to determine the torsional stiffnesses and disc inertias in the system model shown in Figure 8.18(a).

The disc inertias I_1 to I_6 are each made up from the moments of inertias of the crankshaft elements (webs and pins etc) about the crankshaft axis and the inertia equivalents for the connecting rod and piston components associated with each cylinder (Shigley *et al.* 1980). The moments of inertia of the crankshaft elements can be determined from a draughting package and torsional stiffnesses between cylinders K_1 to K_5 from a finite element model.

(b) Modal analysis of the mass-elastic model

Having determined the numerical data for the mass-elastic model in Figure 8.18(a), it is then possible to formulate the mass and stiffness matrices, $[M]$ and $[K]$ respectively – in our example

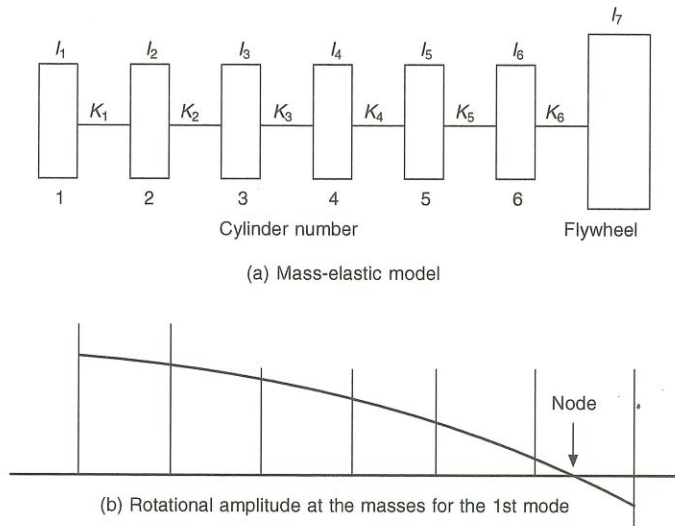


Figure 8.18 Mass-elastic model and first torsion mode of a six cylinder engine

these are 7×7 matrices. It should be noted that the mass-elastic model in this case is a so-called *free-free* system since it is not anchored to ground. This will result in a rigid-body mode in the eigenvalue solution identified as a vector of seven equal numerical values. The corresponding eigenvalue will be zero. Of particular interest is the first torsional mode which may be denoted by its frequency ω_1 and mode shape $\{u\}_1$, shown in Figure 8.18(b).

Biasing the selection of the damper towards the first mode of vibration, it is possible to replace the mass-elastic model in Figure 8.18(a) with a single DOF equivalent and when the damper is added to this model the two-DOF model in Figure 8.19 results. The ground fixing in this model coincides with the node position determined from the first mode of vibration and the damper casing mass is assumed negligible. The inertia of the damper casing can be incorporated into the eigenvalue analysis if data is available. The SDOF equivalent inertia of the crankshaft I_e , can be determined by equating the maximum kinetic energy of the SDOF system to that in the first mode of the mass-elastic system to the left of the node. This results in

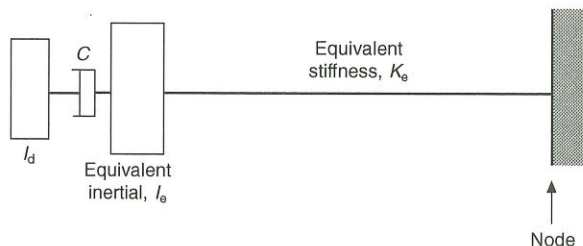


Figure 8.19 Equivalent model based on first mode of the mass-elastic model

$$I_e = \sum_{i=1}^6 \left(\frac{I_i u_i}{u_1} \right)^2 \quad (8.58)$$

The equivalent torsional stiffness of the SDOF model K_e , is then determined from $\omega_1 = \sqrt{\frac{K_e}{I_e}}$ or $K_e = I_e \omega_1^2$. The theory in Section 8.3.3 can then be applied to determine an optimum damping rate C_{opt} for a given damper inertia I_d .

8.4 Fundamentals of acoustics

An understanding of acoustic fundamentals is essential in controlling noise and interpreting noise criteria. This section outlines some of the basic principles in sound propagation.

8.4.1 General sound propagation

Sound is transmitted from the source to the receiver by an elastic medium called the path. In an automotive context this is the surrounding air or the vehicle body structure, giving rise to the term *structure-borne sound*.

The simplest form of sound propagation occurs when a small sphere pulsates harmonically in free space (away from any bounding surfaces). The vibrating surface of the sphere causes the air molecules in contact with it to vibrate and this vibration is transmitted radially outwards to adjoining air molecules. This produces a propagating (travelling) wave which has a characteristic velocity c , the velocity of sound in air. At some arbitrary point on the path, the air undergoes pressure fluctuations which are superimposed on the ambient pressure. A sound source vibrating at a frequency f , produces sound at this frequency. Taking a snapshot of the instantaneous pressure and traversing away from the source, the variation of pressure with distance is also sinusoidal. The distance between pressure peaks is constant and known as the *wavelength* λ . This is related to c and f by the equation:

$$\lambda = \frac{c}{f} \quad (8.59)$$

From this equation it is seen that as f increases λ decreases. In the audible range from 20 Hz to 20 kHz; the wavelength correspondingly varies from 17 m to 17 mm.

8.4.2 Plane wave propagation

The fundamentals of wave motion are most easily understood by considering the propagation of a plane wave (having a flat wavefront perpendicular to the direction of propagation). Denoting the elastic deformation as ξ at some distance x from a fixed datum and combining the continuity and momentum equations for the element with the gas law leads to (Kim *et al.*, 1993) the one dimensional wave equation

$$\frac{\partial^2 \xi}{\partial t^2} = c^2 \frac{\partial^2 \xi}{\partial x^2} \quad (8.60)$$

where the propagation velocity

$$c = \sqrt{\frac{p\gamma}{\rho}} = \sqrt{\gamma RT} \quad (8.61)$$

and the notation is as follows: p = the ambient pressure, ρ = corresponding density of the medium, γ = ratio of specific heats for air, R = universal gas constant and T = absolute temperature. For air at 20°C the magnitude of c is 343 m/s.

The general solution to equation 8.60 for harmonic waves is:

$$\zeta = Ae^{(\omega t - kx)} + Be^{(\omega t + kx)} \quad (8.62)$$

The first term on the right hand side represents the incident wave (travelling away from the source) while the second term represents the reflected wave (travelling in the opposite direction).

The wavenumber $k = \frac{\omega}{c}$ or $k = \frac{2\pi f}{c} = \frac{2\pi}{\lambda}$ and hence is defined as the number of acoustic wavelengths in 2π . c varies considerably for fluid and solid materials. Some typical values are shown in Table 8.1.

Table 8.1 Velocity of wave propagation in various media

Medium	c , m/s
Air at 1 bar and 20°C	343
Mild steel	5050
Aluminium	5000
Vulcanized rubber	1269
Water at 15°C	1440

Specific acoustic impedance, z

The impedance which a propagating medium offers to the flow of acoustic energy is called the *acoustic impedance*. It is defined as the ratio of acoustic pressure p to the velocity of propagation u . It can be shown (Reynolds, 1981) that

$$z = \frac{P}{u} = \rho c \quad (8.63)$$

and for normal temperature and pressure (101.3 kPa and 20°C) is equal to 415 rayls (Ns/m³).

Acoustic intensity, I

This is defined as the time averaged rate of transport of acoustic energy by a wave per unit area normal to the wavefront. It is given by Reynolds (1981);

$$I = \frac{P_{\text{rms}}^2}{\rho c} \quad (8.64)$$

where p_{rms} is the rms pressure fluctuation. For a harmonic wave $p_{\text{rms}} = \frac{\hat{p}}{\sqrt{2}}$ and then

$$I = \frac{\hat{p}^2}{2\rho c} \quad (8.65)$$

where $\hat{p}$ is the peak pressure.

8.4.3 Spherical wave propagation; acoustic near and far fields

Spherical waves more closely approximate true source waves, but approximate to plane waves at large distances from a source. It may be shown (Reynolds, 1981) that the wave equation in spherical coordinates is

$$\frac{\partial^2(rp)}{\partial t^2} = c^2 \frac{\partial^2(rp)}{\partial r^2} \quad (8.66)$$

The general solution for an incident wave only (no reflection) is

$$p = \frac{1}{r} A e^{i(\omega t - kr)} \quad (8.67)$$

when equation 8.67 is used in conjunction with the definition for acoustic impedance it may be shown that

$$z = \rho c \frac{(kr)^2}{1 + (kr)^2} + i(\rho c) \frac{kr}{1 + (kr)^2} \quad (8.68)$$

At large distances from the source ($kr \gg 1$ or $r \gg \lambda/2\pi$), $z \rightarrow \rho c$. Then pressure and particle velocity are in phase and we are in what is called the *acoustic far field* where spherical wavefronts approximate to those of plane waves and pressure and velocity are in phase.

At distances close to the source ($kr \ll 1$ or $r \ll \lambda/2\pi$), $z \rightarrow i\rho ckr$. Then pressure and velocity are 90° out of phase and we are in what is called the *acoustic near field*.

The transition from near to far field is in reality a gradual one, but is normally assumed to take place in the vicinity of $\lambda/2\pi$. For a harmonic wave at 1 kHz ($\lambda \approx 1$ kHz), $r = 50$ mm; while for at 20 Hz; $r = 2.5$ m. The far field/near field transition has important implications for microphone positioning in sound level measurements.

8.4.4 Reference quantities

Certain reference quantities are used for sound emission measurements. For sound transmission in air the reference rms pressure is taken to be $p_{\text{ref}} = 20 \mu\text{Pa}$ corresponding approximately to the threshold of hearing at the reference frequency of 1 kHz. With a reference impedance $z_{\text{ref}} = (\rho c)_{\text{ref}} = 400$ rayls; the reference intensity $I_{\text{ref}} = 10^{-12} \text{ W/m}^2$ from equation 7.64. Since the acoustic power is the intensity times the spherical area, for a reference area $A_{\text{ref}} = 1 \text{ m}^2$ the reference sound power is 10^{-12} W .

8.4.5 Acoustic quantities expressed in decibel form

The human ear is capable of detecting acoustical quantities over a very wide range, e.g. pressure variations from $20 \mu\text{Pa}$ to 100 Pa . Hence there is a need to represent acoustical data

in a convenient form. This is achieved by using the decibel scale. The quantity of interest x is expressed in the form $10 \log_{10} (x/x_{\text{ref}})$, where both x and x_{ref} have units of power. Using the above reference quantities the sound power level (L_W), the sound intensity level (L_I) and the sound pressure level (L_p) are as follows:

$$L_W = 10 \log_{10} \left(\frac{W}{W_{\text{ref}}} \right), \text{ dB} \quad (8.69)$$

$$L_I = 10 \log_{10} \left(\frac{I}{I_{\text{ref}}} \right), \text{ dB} \quad (8.70)$$

$$L_p = 10 \log_{10} \left(\frac{p}{p_{\text{ref}}} \right)^2 = 20 \log_{10} \left(\frac{p}{p_{\text{ref}}} \right), \text{ dB} \quad (8.71)$$

When it is noted that the threshold of hearing corresponds to a sound pressure level of 0 dB it may be shown (Reynolds, 1981) that for normal temperature and pressure (101.3 kPa and 20 °C). L_W , L_I and L_p are related as follows:

$$L_I = L_p - 0.16 \text{ dB} \quad (8.72)$$

$$L_W = L_p + 20 \log_{10} \left(\frac{r}{r_{\text{ref}}} \right) - 0.16 \text{ dB} \quad (8.73)$$

where $r_{\text{ref}} = 0.282 \text{ m}$.

From equation 8.72 it is seen that L_p and L_I are approximately equal numerically, while equation 8.73 is useful for determining the sound power level of an acoustic source from sound pressure level measurements in free field conditions. For these whole field radiation conditions the reduction in sound pressure level with doubling of distance from the source is 6 dB.

8.4.6 Combined effects of sound sources

It is often necessary to determine the sound pressure level of two or more uncorrelated sound sources when the level for each source is known. This can be achieved by using the equation.

$$L_{p,\text{total}} \approx L_{I,\text{total}} = 10 \log_{10} (\sum 10^{0.1 L_{p_i}}) \text{ dB} \quad (8.74)$$

8.4.7 Effects of reflecting surfaces on sound propagation

When an incident wave strikes a reflecting surface the wave is reflected backwards towards the source. In the vicinity of the reflecting surface the incident and reflected waves interact to produce what is known as a *reverberant field*. The extent of the depth of this field towards the sound source is dependent on the absorptive properties of the reflecting surface. A typical interaction between incident and reflected waves is shown in Figure 8.20 and sound pressure level variations as function of distance r from the sound source are as shown in Figure 8.21.

Practical situations arise where the sound source is positioned close to a hard reflecting surface. Four idealized cases are (a) whole space radiation – when there are no reflecting surfaces, i.e. the source is in free space, (b) half-space radiation – when the source is positioned

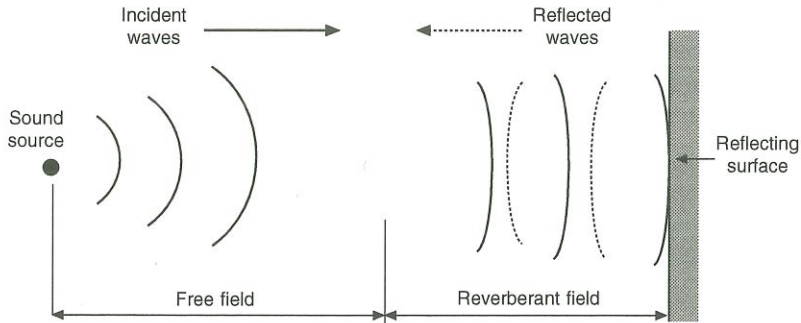


Figure 8.20 Interaction of incident and reflected waves

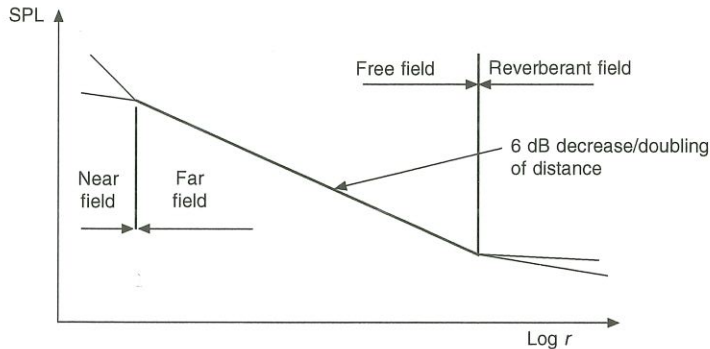


Figure 8.21 Sound pressure level as a function of distance from a simple spherical source

at the centre of a flat hard (reflecting) surface, (c) quarter space radiation – when the source is positioned at the intersection of two flat hard surfaces which are perpendicular to one another and (d) eighth space radiation – when the source is positioned at the intersection of three flat perpendicular hard surfaces. In each case there is an increase in acoustic intensity. The effect can be described by the *directivity index* DI in terms of the *directivity factor* Q .

$$DI = 10 \log_{10} Q \text{ dB} \quad (8.75)$$

where $Q = 1$ ($DI = 0$, dB) for a whole space, $Q = 2$ ($DI = 3$ dB) for a half space, $Q = 4$ ($DI = 6$ dB) for a quarter space and $Q = 8$ ($DI = 9$ dB) for an eighth space.

When evaluating sound power level (PWL) of a source from sound pressure level (SPL) measurements with the above source locations, equation 8.73 can be modified to

$$L_W = L_p + 20 \log_{10} \left(\frac{r}{r_{\text{ref}}} \right) - DI - 0.16 \text{ dB} \quad (8.76)$$

8.5 Human response to sound

The human ear is a delicate and sophisticated device for detecting and amplifying sound (Reynolds, 1981). It consists of an outer ear, a middle ear containing an amplifying device (the ossicles) and an inner ear containing the cochlea. This small snail-shaped element contains lymph and a coiled membrane to which are connected thousands of very sensitive hair endings of varying thickness. These respond to different frequencies, converting the sound stimulus into nerve impulses which are transmitted to the brain. A certain threshold level is required to stimulate the nerve cells, while over-stimulation can lead to temporary or permanent deafness. This latter effect was recognized in the early 70s as a cause of industrial deafness resulting in a number of regulations to protect workers.

The audible range for a healthy young person lies within the envelope shown in Figure 8.22. The frequency range extends from 20 Hz to 20 kHz and the SPL extends from the threshold of hearing at the lowest boundary to the threshold of feeling (pain) at the highest. It is observed that the sound pressure level at the upper and lower boundaries vary markedly with frequency. Typically at 1 kHz the range of sound pressure levels is from 0 to 130 dB. The shape of the curves for sounds of increasing loudness are generally similar to that for the threshold of hearing. It follows therefore that the human ear is most sensitive between 500 Hz and 5 kHz and is insensitive to sounds below 100 Hz.

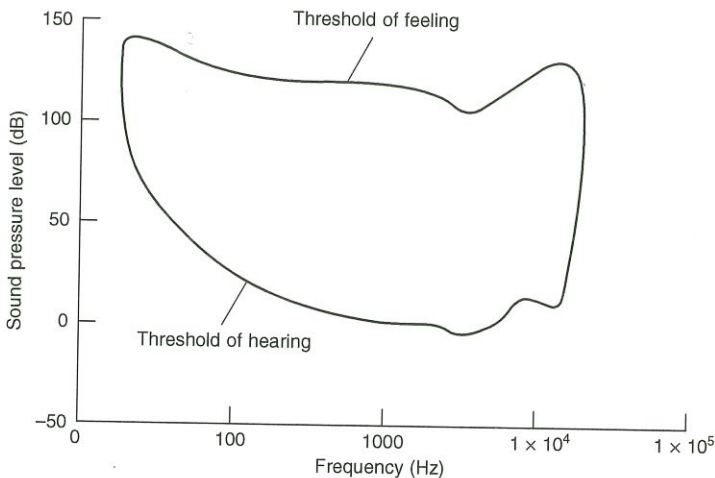


Figure 8.22 The audible range

8.6 Sound measurement

Automotive noise measurement is required for a variety of purposes dictating the need for a range of measuring equipment. In development work there is a requirement for measuring continuous noise levels such as that from drivetrains and their ancillaries, there are requirements

for component noise testing for sound power, frequency analysis and source identification. For type approval there are requirements for assessing whole vehicle noise. Controlled test environments are also required to ensure that tests are repeatable and not weather dependent. This calls for special acoustical test chambers such as the anechoic chambers used to simulate free-field environments.

8.6.1 Instrumentation requirements

Sound level meters

The most basic instrument for sound measurement is a sound level meter comprising a microphone, r.m.s. detector with fast and slow time constants and an A-weighting network to enable measurements to be made which relate to human response to noise, leading to so called A-weighted noise levels L_{pA} , expressed in dB(A). Because of the frequency sensitivity of the human ear, the A-weighting network has the form shown in Figure 8.23. This emphasises the frequencies in 500 Hz to 5 kHz range and produces increasing attenuation below 100 Hz. (B.S. 5969, 1981) specifies four quality grades of sound level meter ranging Type 0 laboratory reference standards to Type 3 industrial grade meters. For development work Type 1 instruments are generally recommended. If the instrument is required for measurement of transient noise it should also be equipped with a peak hold facility.

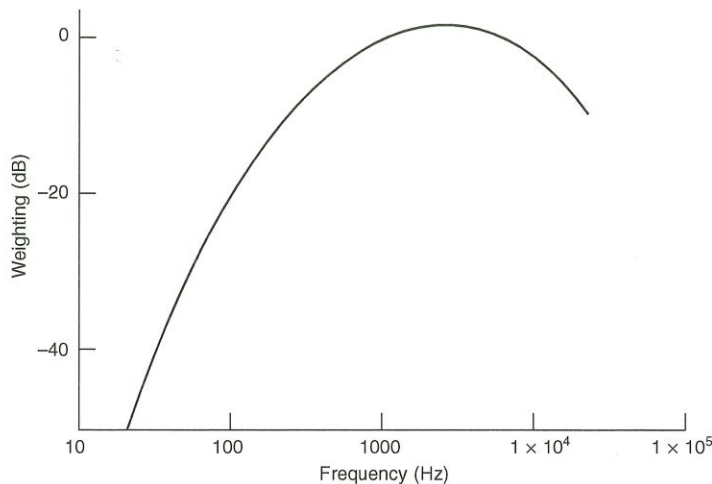


Figure 8.23 The A-weighting curve

Because sound levels are rarely constant (e.g. noise resulting from changes in engine speed), there is a need to average levels over prescribed intervals of time. This type of measurement leads to an equivalent noise level such as $L_{Aeq,T}$, which is related to noise deafness and annoyance criteria. This indicates the A-weighted noise level has been averaged over a measurement period T to give a level having the same energy content as a constant sound of the same numerical value. Mathematically this can be written:

$$L_{Aeq,T} = 10 \log_{10} \left(\frac{1}{T} \int_0^T \left(\frac{p_A(t)}{p_{ref}} \right)^2 dt \right) \quad (8.77)$$

An integrating type of sound level meter is required for this type of measurement.

Another requirement of sound level meters is to determine the level which has been exceeded for a prescribed portion of the measurement time, L_N , e.g. L_{90} represents the A-weighted level exceeded for 90% of the measurement period and is used in environmental background noise measurements associated with traffic noise.

Frequency analysers

Since the frequency spectrum of noise is closely related to the origins of its production, frequency analysis is a powerful tool for identifying noise sources and enables the effectiveness of noise control measures to be assessed.

The simplest frequency analysers split the frequency range into a set of octave bands having the following standardized centre frequencies: 31.5, 63, 125, 250, 500, 1k, 2k, 4k, 8k and 16k; Hz. These filters have a constant percentage bandwidth implying that the bandwidth increases with centre frequency giving increasingly poor discrimination at high frequencies. This can be improved with the use of third octave analysis. A number of product noise and environmental standards require the use of octave and third octave analysis and much of the performance data for noise control products is expressed in terms of octave band centre frequencies.

For serious noise control investigations, narrow band frequency analysers are a necessity. Instead of switching sequentially through a set of filters, the signal in a narrow band analyser is presented simultaneously to the inputs of all filters in the analysis range. The signal processing is done digitally. The outputs are updated many times per second and are fed to continuous display devices such as VDUs and/or downloaded to computers.

Sound intensity analysers

Sound intensity analysers allow sound power measurements to be made in-situ in the presence of background noise, i.e. they do not require special noise testing installations. They also allow noise source identification from sound intensity mapping.

A typical sound intensity probe consists of two closely spaced pressure microphones which measure the sound pressure and pressure gradient between the two microphones. Signal processing converts these measurements into sound intensity values in a sound intensity analyser.

8.7 Automotive noise criteria

As a result of the ever increasing numbers of vehicles on the roads of developed countries, the level of road traffic noise has continued to grow alarmingly. This is in spite of regulations imposed by governments and the significant reductions in noise levels which have already been achieved by new vehicles. The quest for quieter vehicles coupled with good design of new roads is set to go on in an attempt to drive overall noise levels down. Vehicle manufacturers are being faced with increasingly stringent noise regulations for new vehicles. The current limits for drive-by noise of new vehicles are being harmonized within the EEC and are intended to be progressively reduced into the foreseeable future.

8.7.1 Drive-by noise tests (ISO 362, 1981)

The procedure is to accelerate the vehicle in a prescribed way and in a prescribed gear past a microphone set up at a height of 1.2 m above a hard reflecting surface and 7.5 m from the path of the vehicle. The test area is required to be flat, have a low background noise level and not be influenced by reflecting obstructions, bystanders, tyre noise and wind noise. The test site should be as shown in Figure 8.24 and the vehicle should follow path A–B. When the vehicle reaches A the throttle should be opened fully and maintained in this position until the rear of the vehicle reaches B. A minimum of two measurements should be made on each side of the vehicle. In addition to the results of the measurements, vehicle details such as loading, rating, capacity and engine speeds should be reported.

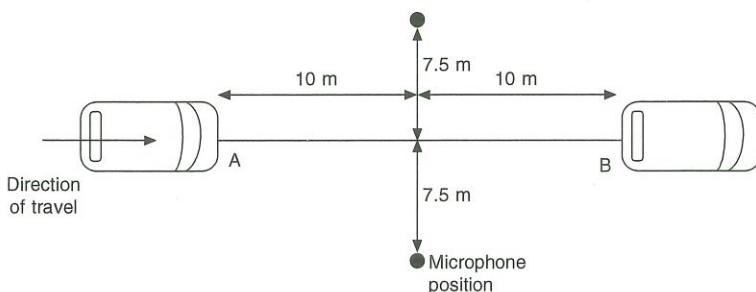


Figure 8.24 Drive-by test site and measurement locations

8.7.2 Noise from stationary vehicles

Since exhaust noise is one of the major sources of vehicle noise and vehicles spend a significant amount of time stationary in traffic queues, noise measurements are often taken from stationary vehicles in the vicinity of the exhaust silencer (ISO 5130, 1978). Measurements are carried out with the engine running at 75% of the speed at which it develops maximum power. When the engine speed has become constant the throttle is quickly released to the idling position while the A-weighted SPL is measured. For these measurements, the exhaust outlet and microphone are in the same horizontal plane with the microphone 500 mm from the exhaust outlet and at an axis of 45° to it. The background noise level is also measured and the maximum difference between the vehicle noise and the background noise is then compared with vehicle's specified noise level.

This noise test is currently being adopted by a number of European countries to check exhaust silencer performance as a part of routine vehicle testing.

8.7.3 Interior noise in vehicles

There are no legal requirements for assessing interior noise in vehicles. Because of the necessity for subjective assessment it has long been the practice to have this work performed by a team of experienced assessors. This has its disadvantages in a development programme where there

is a need to quantify the essential characteristics and relate these to noise sources and transmission paths. To aid this process a number of ad hoc criteria have been developed by different manufacturers for specific types of noise. For example a modified form of Articulation Index, AI (Greaves *et al.* 1988) which is designed to quantify the intelligibility of conversation has been used. The audible range between 200 Hz and 16 kHz is split into sixteen third octave bands. The SPL is measured in each band and the articulation index A_i for the i -th band is determined from the equation:

$$A_i = \frac{W_f (A_0 - \text{SPL})}{A_0 - A_{100}} \quad (8.78)$$

where A_0 = SPL for zero intelligibility

A_{100} = SPL for 100% intelligibility

W_f = weighting factor for each third octave band

The overall (single number) AI is then determined by adding together the sixteen individual indices.

Diesel powered vehicles pose very difficult problems for the NVH engineer. In particular the noise resulting from cold idling conditions contains periodic high frequency large amplitude sound pressure variations. These pulses of sound vary from cylinder to cylinder and from firing stroke to firing stroke leading to an irregularity which is subjectively very annoying.

An analyser has been developed by one manufacturer (Russell *et al.*, 1988) which is designed specifically to assess this problem. The analyser is capable of measuring the impulsive content (based on the kurtosis, the 4th statistical moment) of the sound pressure variation and irregularity of the sound pressure variation by measuring the standard deviation of the low pass filtered amplitude of the diesel knock pulses.

8.8 Automotive noise sources and control techniques

The frequency composition of a sound is one of its most identifiable features. When the sound occurs at a single frequency it is called a *pure tone*. However, the great majority of sounds are far more complicated than this, having frequency components distributed across the audible range. Because there are numerous sources of automotive noise, most of which are cyclic and related to engine speed, the result is what is called *broad band noise* containing a number of dominant frequency components related to engine speed. The frequency characteristics of this sound are represented by its frequency spectrum in a similar way to those for vibration sources.

8.8.1 Engine noise

Engine noise originates from both the combustion process and mechanical forces associated with engine dynamics. The combustion process produces large pressure fluctuations in each cylinder giving rise to high dynamic gas loadings and other mechanical forces such as piston slap. These forces combined with the dynamic forces from inertia and unbalance effects (which are generally dependent on engine configuration and speed) produce the excitations applied to

the engine structure. The resulting vibration produces noise radiation from the various surfaces of the engine.

Noise control at source therefore has to be concerned with controlling the extent of cylinder pressure variations (combustion noise) and choice of engine configuration (dynamic effects). Both of these options tend to conflict with the need for small high-speed fuel efficient engines.

In the case of diesel engines there is evidence (Pettitt *et al.*, 1988) that combustion force reduction can be achieved by controlling the rate of pressure rise in cylinders. This requires careful attention to the design of combustion bowls and selection of turbocharger and fuel injection options. The mechanical noise associated with piston slap can be reduced by careful selection of gudgeon pin offset and minimizing piston mass.

Ranking of engine noise components indicates (Pettitt *et al.*, 1988) that most engine noise is radiated from the larger more flexible surfaces such as sumps, timing case covers, crankshaft pulleys and induction manifolds. It makes sense therefore to isolate these components from the vibration generated in engine blocks using specially designed seals and isolating studs. This enables the isolation of components such as sumps and intake manifolds from high frequency excitation components. Noise shields can also be effective in attenuating radiated noise from components such as timing case covers and the side walls of engine blocks. The shields are generally made from laminated steel (see Section 8.3.4) or thermoset plastic material designed to cover the radiating surface and are isolated from it by flexible spacers. The high internal damping of laminated steel can also be used to produce other effective noise reduced components such as cylinder head covers. Noise from crankshaft pulleys can be reduced either by using spoked pulleys or fitting a torsional vibration damped pulley.

8.8.2 Transmission noise

Gear noise rises with speed at a rate of 6 to 8 dB with a doubling of speed, while measurements have shown that gear noise increases at a rate of 2.5 to 4 dB for a doubling of power transmitted (Hand, 1982).

In an ideal pair of gears running at constant speed, power will be transmitted smoothly without vibration and noise. In practice, however, tooth errors occur (both in profile and spacing) and in some cases shaft eccentricities exist. If a single tooth is damaged or incorrectly cut a fundamental component of vibration is generated at shaft speed f_{ss} . If the shaft is misaligned or a gear or bearing is not concentric vibration (and noise) is generated at tooth meshing frequency f_{tm} with sidebands f_{s1} and f_{s2} given by:

$$f_{s1}, f_{s2} = f_{tm} \pm f_{ss} \quad (8.79)$$

For a wheel having N teeth rotating at n rev/min; the tooth meshing frequency f_{mf} is:

$$f_{tm} = \frac{nN}{60}, \text{ Hz} \quad (8.80)$$

Furthermore gear teeth are elastic and bend slightly under load. This results in the unloaded teeth on the *driving* gear being slightly ahead of their theoretical rigid-body positions and the unloaded teeth on the *driven* gear being slightly behind their theoretical positions. Thus when contact is made between the teeth on driving and driven wheels there is an abrupt transfer of load momentarily accelerating the driven gear and decelerating the driving gear. This leads to

noise generation at tooth meshing frequency. Considerable effort has been devoted in recent years to correcting standard tooth profiles to account for tooth elasticity effects, but because gear teeth are subjected to variable loading, it is impossible to correct for all eventualities.

8.8.3 Intake and exhaust noise

Intake noise is generated by the periodic interruption of airflow through the inlet valves in an engine, thus creating pressure pulsations in the inlet manifold. This noise is transmitted via the air cleaner and radiates from the intake duct. This form of noise is sensitive to increases in engine load and can result in noise level increases of 10 to 15 dB from no-load to full load operation. When a turbocharger is fitted, noise from its compressor is also radiated from the intake duct. Turbocharger noise is characterized by a pure tone at blade passing frequency together with higher harmonics. Typical frequencies are from 2 to 4 kHz.

Exhaust noise is produced by the periodic and sudden release of gases as exhaust valves open and close. Its magnitude and characteristics vary considerably with engine types, valve configurations and timing. The fundamental frequency components are related to the engine firing frequency, which for a four-stroke engine is given (in Hz) by:

$$f = \frac{\text{enginespeed (rpm)}}{60} \times \frac{\text{number of cylinders}}{2} \quad (8.81)$$

Levels of exhaust noise vary significantly with engine loading. From no-load to full-load operation these are typically 15 dB. Turbocharging not only reduces engine radiated noise by smoothing combustion, but also reduces exhaust noise.

Attenuation of noise at engine intakes and exhausts calls for devices which minimize the flow of sound waves while allowing gases to flow through them relatively unimpeded. Such devices are effectively acoustic filters. The operational principles of intake and exhaust silencers ('mufflers' as they are called in the USA) can be divided into two types, dissipative and reactive. In practice, silencers are often a combination of both types.

Dissipative silencers contain absorptive material which physically absorbs acoustic energy from the gas flow. In construction, this type of silencer is a single chamber device through which passes a perforated pipe carrying the gas flow. The chamber surrounding the pipe is filled with sound absorbing material (normally long-fibre mineral wool) which produces attenuation across a very broad band of frequencies above approximately 500 Hz. The degree of attenuation is generally dependent on the thickness and grade of the absorbing material, the length of the silencer and its wall thickness. Figure 8.25 shows the cross-section through a typical dissipative exhaust silencer with venturi tube (see later paragraph).

Reactive silencers operate on the principle that when the sound in a pipe or duct encounters a discontinuity in the cross-section, some of the acoustic energy is reflected back towards the sound source thereby creating destructive interference. This is an effective means of attenuating low frequency noise over a limited range of frequencies. The effectiveness of this technique can be extended by having several expansion chambers within the same casing connected together by pipes of varying lengths and diameters (Figure 8.26). Silencers of this type increase the exhaust back-pressure and result in some power loss.

Intake noise attenuation is generally incorporated into the air filter and is achieved by designing the filter to act as a reactive silencer based on the Helmholtz resonator principle. For

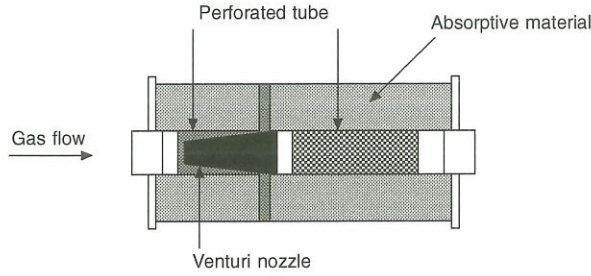


Figure 8.25 Two chamber dissipative silencer with venturi

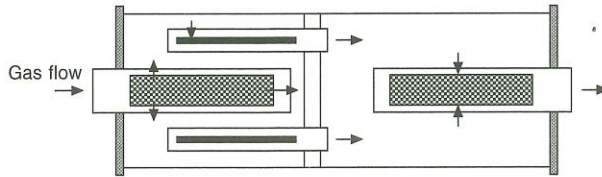


Figure 8.26 Two-chamber reactive silencer

an intake system (Figure 8.27) comprising an intake venturi pipe of mean cross-sectional area A and length L together with a filter volume V , the resonance frequency is given by:

$$f = \frac{c}{2\pi} \sqrt{\frac{A}{LV}} \quad (8.82)$$

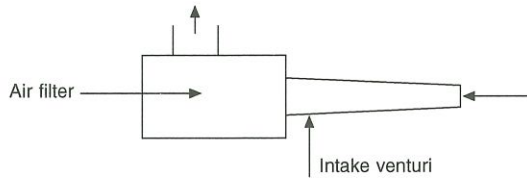


Figure 8.27 Air cleaner and venturi tube

where c is the velocity of sound in air. This type of design produces a low frequency resonance (a negative attenuation) but an increasing attenuation at higher frequencies. This can, however, be offset by high frequency resonances within the intake venturi (Peat *et al.*, 1990).

The exhaust systems of present-day vehicles are required to perform the dual task of reducing both the exhaust gas pollutants and exhaust noise. Catalytic converters are fitted immediately downstream from exhaust manifolds to ensure that they quickly achieve operating temperature and thus become quickly effective in urban driving. In addition to acting as exhaust gas scrubbers, catalytic converters also have an acoustic attenuation effect resulting from the gas flow through narrow ceramic pipes. This attenuation is produced by both interference and dissipation.

Additionally, silencers are positioned in the exhaust system downstream from the catalytic converter. These are required specifically to smooth exhaust gas pulsations and make them as inaudible as possible. The silencers and their pipework form an acoustically resonant system which is tuned to avoid exciting bodywork resonances which would aid transmission of structure borne noise. For this latter reason, it is common for silencers to have a double skin and insulating layer which also provides thermal insulation. Exhaust systems need to be isolated from vehicle bodywork to avoid transmission of structure-borne sound and for this reason are suspended from the underbody of the vehicle by flexible suspension elements. There is also a risk that the noise emitted from the tailpipe can cause body resonances if the exhaust is not properly tuned.

The following are some of the devices used to overcome specific silencer tuning problems.

- (a) The Helmholtz resonator – a through-flow resonator which amplifies sound at its resonant frequency, but attenuates it outside this range.
- (b) Circumferential pipe perforations – create many small sound sources resulting in a broadband filtering effect due to increased local turbulence.
- (c) Venturi nozzles – designed to have flow velocities below the speed of sound they are used to attenuate low frequency sound.

Computer software is being increasingly used in the design and analysis of intake and exhaust silencer systems. One example of this is LAMPS (Loughborough and MIRA program for Silencers) (Peat *et al.*, 1990).

8.8.4 Aerodynamic noise (Barnard, 1996)

Aerodynamic noise is due principally to pressure fluctuations associated with turbulence and vorticity. For road vehicles this can be broken down into three noise generating components: the boundary layer distributed over the vehicle body, edge effects and vortex shedding at various locations on the vehicle body and also at cooling fans.

Boundary layer noise tends to be random in character and is spread over a broad band of frequencies. Noise levels due to boundary layer effects are not normally troublesome and the higher frequency components in the spectrum can easily be attenuated with absorbent materials inside body panels.

Edge noise is produced by flow separating from sharp corners and edges on the body structure. As the flow separates from an edge it rolls up into large vortices which also break up into smaller vortices. It is the intermittent formation and collapse of these vortices which leads to the narrow band characteristics associated with edge separation. The noise level associated with edge noise is generally higher than boundary layer noise and has a more defined band of frequencies. This band of frequencies is a function of vehicle speed such that changes in speed can be observed by the change in noise signature. It is possible to reduce edge noise by minimising protrusions from the body surface, making the body surface smooth and continuous and ensuring that gaps around apertures such as doors are well sealed. There is also a strong tendency for vortices to be produced at the pillars supporting the front windscreen. These vortices extend rearwards and envelope the front sidelights which tend to have a low resistance to sound transmission. There is generally very little which can be done to improve this problem

since a change of profile to a well rounded contour while improving the aerodynamics is generally unacceptable from a vision point of view. Edge noise also arises at protuberances such as wingmirrors and at wheel trims. In these cases there is generally scope for improvements to the profiles without impairing their function.

Vortex shedding occurs when an airflow strikes a bluff-body producing a periodic stream of vortices downstream. This results in the production of pure tones – subjectively the most annoying of all sounds. The frequency f of the vortices are related to the air speed U and depth d of the bluff body by the equation

$$f = \frac{SU}{d}, \quad (8.83)$$

where S is the Strouhal number. Typically $S = 0.2$ for a long thin rod, so that for a vehicle fitted with a roof-rack having 10 mm diameter bars (facing the air flow) and travelling at 113 km/h (70 mph), the vortices shed a frequency of 640 Hz, i.e. in the frequency range where the human ear is most sensitive. A much quieter form of roof carrier is the enclosed pod shaped design which tends to avoid the production of vortices.

The cooling fan is also a source of noise. In this case the fan blades shed helical trailing vortices which result in periodic pressure fluctuations when they strike downstream obstacles. To overcome this problem fan rotors are made with unevenly spaced blades and with an odd numbers of blades. The use of thermostatically controlled electrically driven fans now ensures that fan noise does not increase with engine speed, as was the case for earlier belt-driven designs.

Noise from internal air flows designed for ventilation and occupant comfort is becoming increasingly important as overall cabin noise is reduced. It is essential that in modern vehicles inlet and outlet apertures are carefully sited and designed to ensure that they do not themselves generate noise and that noise from the engine compartment is not carried into the cabin space by the ventilating air.

8.8.5 Tyre noise

As engine noise is progressively reduced and with the advent of electric vehicles, tyre noise is emerging as a serious problem. Tests have shown that tyre noise can be broken down into two components (Walker *et al.*, 1988) tread pattern excited noise and road surface excited noise. While the problem of road surface excited noise is the province of highway engineers, that of tread pattern excited noise clearly belongs to the automotive engineer. Tyre designers are concerned with reducing tyre noise at source while chassis engineers are concerned with reducing the transmission of noise from the tyre contact patch to the vehicle interior. The mechanism of tyre noise generation is due to an energy release when a small block of tread is released from the trailing edge of the tyre footprint and returns to its undeformed position. There is also a contribution from the opposite effect at the leading edge of the footprint.

With a uniform tyre block pattern, tonal noise (at a single frequency with harmonics when the wheel rotates at a constant speed) is generated. To overcome this problem tyre designers have produced block pitch sequences which re-distribute the acoustic energy over a wider band of frequencies. When tread patterns are taken into account there is a need to analyse the effect of the individual impulses produced across the width of the tyre. Computer software has been

developed (Membretti, 1988) to aid this aspect of tread evaluation at the design stage. Models of tyres which take account of their structural dynamic characteristics and the air contained within them are also used at the design stage.

8.8.6 Brake noise

Despite sustained theoretical and experimental efforts over many years, the mechanism of noise generation in disc and drum brakes is still not fully understood. The problem of brake noise is one of the most common reasons for warranty claims on new vehicles with market research evidence suggesting that as many as 26% of owners of one year old medium sized cars complain of brake noise problems.

The intractable nature of the problem arises from the complex assemblage of components in which shoes or pads are held in contact with either a drum or disc under hydraulic and friction loading. A dynamically unstable brake system results in vibration of the brake components and noise is generated by components of significant surface area such as brake drums and discs. Progress towards a better understanding of the noise generating mechanisms has been aided by experimental investigations (e.g. Fieldhouse *et al.*, 1996) and mathematical models (e.g. Nishiwaki, 1991) have been put forward to aid the design of quieter brakes.

In the absence of comprehensive brake models, a number of noise 'fixes' have been implemented to cure specific problems. For low frequency drum brake noise these have included the addition of either a single mass or a combined mass and a visco-elastic layer applied at anti-nodes of the drum backplate (Fieldhouse *et al.*, 1996). At higher frequencies other solutions are necessary such as a redistribution of drum mass to eliminate some of the specific backplate diametral modes.

8.9 General noise control principles

8.9.1 Sound in enclosures (vehicle interiors)

For small enclosures of regular shape (e.g. cubes, cylinders) it is possible to determine the sound field in precise mathematical terms. There are in theory an infinite number of natural frequencies and modes and the situation is analogous to the vibration of homogeneous elastic solids of similar shapes. Broadband sources appropriately positioned in such enclosures are capable of exciting standing wave patterns resulting in SPLs which are very position sensitive.

The complex shapes encountered in vehicle interiors means that such analytical techniques are not applicable and the sound field tends to be diffuse. The sound pressure levels tend to vary much less throughout the vehicle interior compared to the standing wave behaviour described in the previous paragraph. In general there are a number of sound sources emitting noise into vehicle interiors and these can produce discrete components which are superimposed on a lower level of broadband noise.

8.9.2 Sound energy absorption

Absorption is one of the most important factors affecting the acoustic environment in enclosures.

Increasing the average absorption of internal surfaces is a relatively inexpensive way of reducing sound levels in enclosures and is effective in vehicle interiors.

The *absorption coefficient* α , is defined as the ratio of sound energy absorbed by a surface to the sound energy incident on it. Its value is dependent on the angle of incidence and since in an enclosure all angles of incidence are possible, the values of α are averaged for a wide range of angles. Absorption is also dependent on frequency and published data normally quotes values at the standard octave band centre frequencies.

For an enclosure having a number of different internal surface materials, the average absorption coefficient α can be determined (for n surfaces) from:

$$\alpha = \frac{\sum_{i=1}^n S_i \alpha_i}{\sum_{i=1}^n S_i}, \quad (8.84)$$

where S = surface area. In general, the materials having high levels of absorption are porous materials in which movement of air molecules is restricted by the flow resistance of the material. Typical absorption coefficients can be found in (Beranek, 1971).

8.9.3 Sound transmission through barriers

One of the principal noise transmission paths in a vehicle is through the bulkhead separating the cabin space from the engine compartment. The bulkhead can be considered to be a sound barrier. The effectiveness of sound barriers is normally quoted in terms of *transmission loss* TL. This is the ratio of the incident sound energy to that of the transmitted sound energy, expressed in dB. For a thin homogeneous barrier and random incidence (in the range from 0 to 72 degrees) it can be shown (Wilson, 1989) that the field incidence transmission loss is given by:

$$TL = 20 \log_{10} (fm) - 47 \text{ dB} \quad (8.85)$$

where f is the frequency of the sound in Hz and m is the mass per unit area of the barrier in kg/m^2 .

Equation 8.85 applies to what is called the mass-controlled frequency region in which the transmission loss increases by 6 dB per octave increase in frequency, while doubling the barrier thickness or density increases transmission loss by 6 dB at a given frequency. It is evident from this that an effective means of increasing transmission loss is to use a high density material for acoustic barriers.

Sound transmission through barriers is governed at low frequencies by panel bending stiffness and panel resonances; these tend to reduce their low frequency effectiveness. Barriers can be considered to be mass-controlled above twice their lowest natural frequency, but below a critical frequency f_c . This frequency is related to the ability of sound in barriers to be transmitted as bending waves and occurs when the wavelength of the incident wave coincides with the bending wavelength λ_b . The lowest frequency at which this can occur is when the incident sound grazes the surface of the barrier and is given by:

$$f_c = \frac{c}{\lambda_b} \quad (8.86)$$

In practice the range of incidence angles varies from zero to something less than 90 degrees

which means that the decrease in transmission loss associated with coincidence occurs at a frequency somewhat higher than the value given by equation 8.86. The effectiveness of barriers is also sharply reduced by even the smallest apertures and can pose problems in noise isolation when electrical trunking and pipework is required to run between the engine to cabin compartments.

In practice layered materials consisting of a dense core and surface layers of absorptive material can perform the dual role of providing sound absorption with a high transmission loss.

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