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A DIAGNOSTIC VIBRATION AND ACOUSTIC PERFORMANCE ANALYSIS OF MONOCOQUE VEHICLE STRUCTURES

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1. Introduction

In the dynamic analysis of vehicle body structures, the test and analysis techniques must be capable of relating the total structural dynamics profile to specific performance standards for both specific vibration energy input points and total structural dynamics performance. This provides a basis for a decision making process on the acceptability or non-acceptability of a body structure as a vibro-acoustic package, a fact of vital importance at the engineering prototype stage of vehicle development.

A computer based measurement and analysis technique, developed for this purpose is described, applied to the analysis of the low frequency structural dynamic and acoustic performance of monocoque vehicle structures, typically saloon cars or truck cabs.

2. Representation of Vehicle Body Structural Dynamic Behaviour

Natural frequencies and associated mode shapes of vehicle structures can now be predicted theoretically, using finite element methods (1,2), but due to the difficulties of damping representation, absolute amplitudes of vibration with respect to specified input force levels have not been predicted. Experimental measurement of the structural dynamics profile is therefore necessary. The need to relate total structural dynamics profile to specific performance standards necessitates a shift from the traditional presentation of results in terms of resonant mode shape diagrams to the use of standardised dynamic performance parameters which can be readily interpreted in an absolute sense.

Such dynamic performance parameters are described here and have been developed from the two basic forms of mechanical mobility, i.e. point mobility and transfer mobility, to cope with the structural dynamic performance standardisation concepts (3). These have the vital added advantage of being applicable to both resonant and off-resonant modes.

3. Vehicle Body Structural Dynamics Performance Parameters

Point mobility describes quantitatively the ability of a point in a structure to admit vibrational inputs at that point and is therefore a measure of the performance of discrete attachment points at which vibration is transmitted to the vehicle structure, principally from suspension and engine sources.

Transfer mobilities at points all over a structure describe the vibration profile of the structure in response to a vibration input. For any vibration mode it is possible to measure an average transfer mobility vector for the whole structure, or parts of it, by averaging the transfer mobility vectors in the given mode over a number of elemental areas into which the structure is divided, taking relative phase into consideration. This average transfer mobility vector is referred to as the Modal Mobility (3), and is given by:-

$$\eta(\omega) = \frac{\sum_{i=m}^{n}\left[R_i(\omega) + jI_i(\omega)\right]S_i}{\sum_{m}^{n} S_i} \qquad (1)$$

where $R_i(\omega)$ and $I_i(\omega)$ are the real and imaginary components of the modal mobility vector of the ith element, of area S_i, at frequency ω, and m and n represent the first and final elements appropriate to the structural system or subsystem under test.

As a total motion vector, modal mobility satisfies the necessary requirement, from a dynamics standards point of view, of being a single parameter capable of describing the total dynamic response of a whole structure, or definitive parts of it, with respect to a specific input point.

Point mobility and modal mobility are normally measured at discrete frequencies, but average point mobility or modal mobility levels can be measured over any desired frequency bandwidth. A bandwidth average mobility taken over a frequency bandwidth from ω_1 to ω_2 is given by:

$$\bar{M} = \frac{1}{\omega_2 - \omega_1}\int_{\omega_1}^{\omega_2} |M| \, d\omega \qquad (2)$$

where $|M|$ is the mobility (point or modal) vector modulus at a given frequency. Average levels are normally measured over the entire test frequency, (10-200Hz), to describe the broadband performance, and narrow bandwidths of 10Hz to facilitate the diagnostic application of the technique. Examples of Point and Modal Mobility responses with bandwidth average levels are shown in figures 1 and 2.

4. Interior Noise Generation

The acceptance or rejection of the total vehicle system as a vibro-acoustic package must be judged against the interior noise level generated in service. Hence, the relationships between the generated noise and the structural dynamics performance of the vehicle body, the terminal element in the path of vibration input to the vehicle system, is vital in predicting eventual noise levels at the prototype stage. A theoretical relationship has been established between the modal mobility parameter and the generated interior noise, and confirmed by comparing measured and predicted levels in the passenger space (4). This has been applied to the prediction of on-the-road noise levels (5). The average sound pressure P in the passenger space is given by:-

$$P = \frac{\rho CKFMS}{2\pi r} \qquad (3)$$

where:- M is the overall structural modal mobility modulus
S is the total structural area of noise generating panels
r is half the average dimension of the passenger compartment
ρ is density of air
C is velocity of sound in air
K is the acoustic wave number (ω/c)
F is the input force at the structural input point

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5. Experimental Method

The experimental method is designed to measure point and modal mobilities of vehicle structures over the typical engine and road noise frequency bandwidth (10 -200Hz). This consists of measuring the input mobility and transfer mobilities for elemental areas, into which the structural surface area is divided, typically between 50 and 60, at discrete incremented frequencies over the test frequency bandwidth. Modal mobilities for the whole structure and component panels are computed from the measured data and bandwidth average parameters are computed from discrete frequency results. A computer controlled automatic vibration excitation and response measuring system has been developed for this work based on the SOLARTRON 1191 computer based Frequency Response Measuring System and 60 channels of measuring accelerometers (3).

6. Diagnostic Application and Experimental Data

Structural dynamics performance standards have been established from tests on a wide variety of vehicle body structures. The dynamic performance of a vehicle structure can thus be assessed against the dynamic standards for its own class, examples of which are shown in Table 1, and a decision made against these as to the acceptability or otherwise of the structure. This is particularly valuable at the prototype stage before commitment to tooling is made.

Table 1. Dynamic Performance Standards for Monocoque Saloon Car and Truck Cab Structures

	Point Mobility, Ref 10^{-3}Nm/s		Modal Mobility, Ref. 10^{-3}Nm/s	
	Saloon Car	Truck Cab	Saloon Car	Truck Cab
10Hz Bandwidth Average	-10 dB	-20 dB	-22 dB	-35 dB
Broad Band Average	-15 dB	-25 dB	-30 dB	-43 dB

In the event of unsatisfactory performance, the diagnostic strength of this technique is such that the poor performance frequency bandwidths are exposed from the narrow band mobility levels and further, the relative contribution of individual component panels, such as engine bulkhead, roof, floor section, etc, to the total structural dynamic profile can be isolated and assessed 'at a glance'. Vibrational energy mismatch between separate panels is exposed and appropriate structural modification or redesign can be implemented.

7. Conclusions

(1) A computer based measurement and analysis technique for assessing vehicle body structural dynamics performance is outlined.

(2) The structural dynamics target performance parameters and their application are described.

(3) The relationship between the modal mobility parameter and the generated interior noise is given, highlighting the vibro-acoustic strength of this analysis technique.

(4) Dynamic performance standards for two classes of vehicle structures are presented and their diagnostic application in problem solving exposed.

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8. References

(1) B. MILLS and J. SAYER. 1974. Inst. of Physics Annual Conference, "Stress, Vibration and Noise Analysis in Vehicles", Aston University, Birmingham. Static and Dynamic Analysis of a light body using the finite element method.

(2) G.E. TOWNLEY and J.W. KLAHS, 1978. SAE paper no.780364. Dynamic simulation of an Automobile Body utilising finite element and modal analysis techniques.

(3) J.W. DUNN, O.A. OLATUNBOSUN and B. MILLS. 1978. Inst. Meas. and Control Symposium, "Dynamic Analysis of Vehicle Ride and Manoeuvring Characteristics" London. Standardisation Techniques for the Dynamic Performance of Monocoque Vehicle Structures.

(4) J.W. DUNN, O.A. OLATUNBOSUN and B. MILLS. Prediction of Low Frequency Sound Pressure Level Distribution inside a Vehicle Passenger Compartment from its Structural Dynamic Response. (To be published).

(5) J.W. DUNN, O.A. OLATUNBOSUN and B. MILLS. ASME Vibration Conference 1979 St. Louis, MO, USA. Realistic Prediction and Control of Vehicle noise resulting from road inputs.

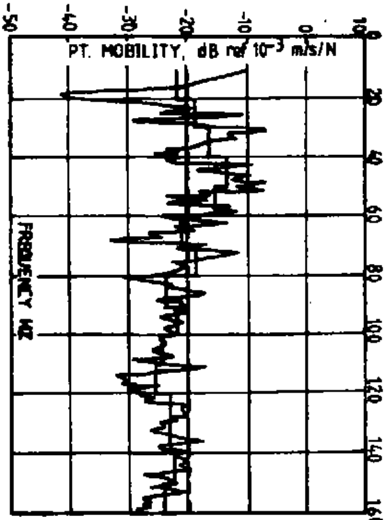

Figure 1. Point Mobility Response

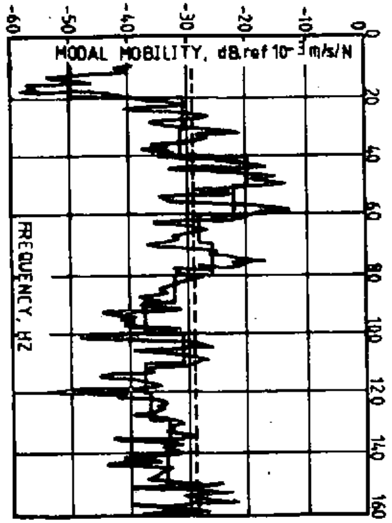

Figure 2. Modal Mobility Response