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Letter to the Editor

A note on the subdivision of a volume of air in a vehicle enclosure into sea subsystems

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1. Introduction

The practice of subdividing enclosed air volumes in passenger vehicles into statistical energy analysis (SEA) subsystems has attracted considerable criticism on the basis that it violates certain conditions that are considered necessary for SEA solutions to be reliable. This note presents a brief qualitative discussion of the physics of such enclosed sound fields, with emphasis on the passenger saloon car, and suggests that the practice of subdivision may be justified in a limited range of frequency. It is not the purpose of this note to propose that the conventional SEA model is the best, or most appropriate, representation of the sound fields in vehicles; it is simply to suggest that subdivision of the air space into SEA subsystems is acceptable in case where the sound field may be reasonably considered to approximate to the ideal diffuse model.

2. Substance of the criticism

In the application of SEA to vehicle enclosures it is common practice to subdivide the enclosed volume of air in a passenger compartment into a number of SEA subsystems. This practice has been subject to criticism on a number of grounds. The principal objection is that conventional SEA is generally considered to give reliable predictions of energy distributions among the subsystems only if the subsystems are 'weakly coupled'. The insertion of subsystem interfaces that separate contiguous volumes of air seems incompatible with the concept of weak coupling, since such an interface represents no geometric or material discontinuity, and sound waves pass unhindered across it.

The second objection refers to the idea that SEA models are unreliable unless the modal densities of chosen subsystems are sufficiently high for there to be enough natural frequencies (or resonant modes) within an analysis band to allow the subsystem wave fields to approximate to

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the ideal diffuse state upon the assumption of which the ‘wave transmission’ estimation of coupling loss factors (CLFs) is conventionally based. A related condition that allows damping-independent CLFs to be assumed is that the modal overlap factors of uncoupled subsystems are sufficiently large to give adequate probability, in an ensemble sense, that the resonant modes of one interact effectively with those of the other in the coupled state. The practice of dividing up an air volume into arbitrarily small subsystems, without regard to the above considerations, is a matter of debate.

The term ‘weakly coupled’ has been interpreted in various ways. In qualitative terms it is taken to indicate that the vibration/acoustic fields in any one subsystem are so weakly coupled to the fields in all other subsystems that its response to broad band excitation may be well represented in terms of its natural modes in an isolated state: that is, when it is uncoupled from the rest of the system by the imposition of suitable boundary conditions, such as ‘free’ or ‘blocked’. An alternative statement is that each subsystem exhibits ‘local’ mode behaviour such that, when it is directly excited, minimal coherent modal motion occurs in other subsystems. Hence, its vibrational/acoustic energy density will substantially exceed that in contiguous (and all other) subsystems.

In contrast, the condition of ‘strong’ coupling is taken to indicate that wave transmission across the interfaces between contiguous subsystems is so ‘easy’ that part, or all, of the complete system exhibits ‘global’ mode behaviour in which the energy of response to excitation of one subsystem results in comparable energy densities in some (at least) of the other subsystems. SEA is considered not to be reliable under these circumstances since it is implicit in SEA theory that the vibration fields in contiguous subsystems are statistically independent, at least in an ensemble sense: vibration in a global mode clearly violates this condition. The crucial implication of weak coupling is that the net rate of transmission of wave energy across an interface may be assumed to be proportional to the difference of two terms, each proportional to the time-average vibrational energy stored in one of the subsystems sharing this interface; the so-called ‘power flow-energy difference’ relation. This is also implicit in the employment of CLFs based upon wave transmission between infinitely extended (non-modal) subsystems, or between a reverberant subsystem and a heavily damped neighbour.

3. The acoustic field in a vehicle enclosure

The acoustic field in a passenger vehicle is complicated by the geometric irregularity of its boundaries, particularly the ‘intrusion’ into the enclosed volume of seats, head rests and passengers, together with the spatial variation of the acoustic properties of those boundaries. It is largely driven by the boundary vibration field generated, in a road vehicle, by such sources as the engine, gearbox, tyres, wind and exhaust system. In the low frequency range associated with ‘boom’ and low order engine vibration (say 40–150 Hz), the air volume in a passenger car exhibits well-separated (discrete) acoustic resonance frequencies. The associated modes are undoubtedly global in character, exhibiting strong interference patterns and highly non-uniform distributions of energy density (or sound pressure level). The sound generated by body vibrations depends strongly on the degree of spatial and natural frequency ‘matching’ of the body modes and acoustic modes. The Finite Element Method is clearly better suited than SEA to the analysis of sound generation in this frequency range.

However, above about 170 Hz, the modal density of the air volume of a typical medium size family saloon car begins to increase at a rate closely proportional to the square of frequency. The acoustic absorption of the non-glazed surfaces and the contained objects increases approximately linearly with frequency: the associated loss factor is therefore more or less independent of frequency. These dependencies yield an approximately third power dependence of modal overlap factor on frequency. The boundaries of the enclosure are excited by a number of mutually uncorrelated external sources. The high modal density of the enclosure shell structures further decreases the extent of finite-frequency-band spatial correlation of the boundary vibration field.

Each acoustic mode of a three-dimensional, reverberant, rectangular enclosure may be decomposed into six constituent travelling waves, the direction of propagation of these waves depending upon the mode orders in the three Cartesian co-ordinate directions. In irregularly shaped enclosures the modes cannot be analytically decomposed, but the concept of decomposition is qualitatively the same. The simultaneous, uncorrelated, excitation of many modes in a not-too-disproportionate enclosure generates a sound field that approximates to an ideal model called a ‘diffuse field’. The ideal diffuse sound field model has been defined in a number of ways. For the present purpose we shall define it as a field in which a large number of mutually uncorrelated plane waves pass through every point in directions that have uniform probability in a solid angle 4π. This does not imply that the mean square sound pressure (or sound pressure level) is spatially uniform. Diffuseness of a field is promoted by high modal overlap together with excitation by a number of uncorrelated sources located at different positions.

Hence, it is reasonable to imagine that there is a rapid transition of acoustic response of a car compartment from discrete modal domination to a fair approximation to a diffuse field in the so-called ‘mid-frequency’ range, say 200–800 Hz. Even though the assumption of a diffuse field in the region above the tops of the seats appears to be reasonable in the mid-frequency range, it is less likely to be applicable to the constricted lower regions of the enclosure below seat level. At higher frequencies, where the absorption of the space becomes considerable, the direct fields of the various boundary sources will tend to dominate the regions of the enclosure in their close proximity and the diffuse model becomes less appropriate.

4. SEA and the diffuse field model

The sound intensity incident from *one* side upon any plane surface within a diffuse sound field is given by $I = \langle p^2 \rangle / 4\rho_0 c$, where $\langle p^2 \rangle$ is the space-average mean square pressure of the incident field. The *net* sound intensity passing through the plane is the arithmetic difference between the magnitudes of the diffuse field intensities incident from each side. In terms of the SEA formalism the time-average *net* power exchange between subsystems (1) and (2) is expressed by

$$P_{12} = \eta_{12} n_1 \omega [E_1/n_1 - E_2/n_2], \tag{1}$$

where η_{12} is the CLF, n is modal density and E represents the time-average acoustic energy stored in a subsystem. Substituting expressions for the asymptotic acoustic modal density of an enclosure together with the conventional expression for energy density in terms of mean square pressure yields the CLF as

$$\eta_{12} = cS_{12}/4V_1\omega \tag{2}$$

in which S is the interface area and V is subsystem volume. If the sound is transmitted through a subsystem interface in the form of a physical partition, such as a seat, that has a sound power transmission coefficient τ, the right-hand side of Eq. (2) is multiplied by τ. Energy is drained from those subsystems that are in contact with absorbing boundaries. In Eq. (1) the volumes of the subsystems disappear from the SEA power flow expression because the CLF η_{12} is inversely proportional to V_1, the modal densities are proportional to the respective volumes and the subsystem energies are proportional to the products of the respective space-average mean square pressures and volumes.

The preceding qualitative description of the transition of the acoustic field in an enclosure from a discrete modal field to an approximation to an ideal diffuse field relates to the whole volume of the air in, at least, the upper regions of the compartment. However, since we assume that diffuse waves pass through every point in the region where it applies, we may select any volume, however small, that lies wholly within such a field. Paradoxically, the assumption of an all-pervading diffuse field fortuitously allows us to disregard concerns about strong coupling due to perfect transmission through interfaces and low modal densities of small volumes.

It is possible that the SEA model using the diffuse field CLF gives reasonable estimates even for the fields in the lower regions of the compartment. The smallness of the volumes together with the associated absorption due to carpets, trim, legs, etc. suggests that the energy densities will be such that little 'feedback' of energy into the upper, more diffuse, region will occur, therefore allowing an assumption of an incident diffuse field intensity and negligible 'return' intensity. However, the direct injection of sound power into the lower regions by vibration of the local structure (e.g., the floor) will cause power transfer into the upper regions, sometimes via the direct field; the diffuse field SEA model may be less accurate in such cases.

Interestingly, we may consider trim panels to act both as vibrational sound sources, driven by structural vibration or airborne sound, and absorbers of diffuse incident sound. The absorbed power may simply be subtracted from the radiated power because the surface pressure fields generated by local structural or acoustic inputs on the one hand, and by the incidence of waves from the reverberant field in the enclosure on the other, may reasonably be assumed to be uncorrelated.

5. Caveat

Although the above analysis appears to justify the subdivision of a compartment into smaller and smaller subsystems, at least above seat level, it is likely that SEA solutions will be dependent on model definition. It is therefore recommended that numerical experiments should be made on various models of a given system in order to check the robustness of the solution. However, the SEA balance between total injected power and total absorbed power seems likely to constrain the solutions to within reasonable limits, especially in compact systems such as car compartments. Application to extended volumes such as train and aircraft compartments may be more problematic. It must also be kept in mind that passengers' heads are often close to radiating surfaces; the importance of sound waves directly radiated by vibration of the enclosure boundaries, particularly on sound quality, should not be underestimated.

6. Conclusion

The assumption that the sound field within a vehicle compartment approximates to a diffuse field in the 'mid-frequency' range appears to allow one to set aside concerns associated with the division of the enclosed air volume into arbitrarily small subsystems. Whether or not such a model is valid can only be established by comparison with data generated by measurements in operating vehicles. Evidence from those having access to such data would be welcomed.