AN APPLICATION OF THE STATISTICAL ENERGY ANALYSIS (SEA) TO THE INTERNAL NOISE PREDICTION IN TRAINS

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ABSTRACT

Noise has become an important attribute in the transport industry because of the competitive market and increasing customer awareness. Over recent years much work have been done to develop simulation tools for vehicle NVH. NVH analysis has become relatively easy and quicker with the emergence of new CAE methodologies like the boundary element method (BEM) or the finite element method (FEM) for vibro-acoustic analysis. These deterministic tools are useful to predict low and medium frequency noise levels but in the high frequency range other methods, as the Statistical Energy Analysis, must be used.

This paper shows the application in a real case of a methodology based on the SEA method to predict interior and exterior noise in trains as well as its correlation with experimental measurements. This procedure can be used in early design phases and allows the engineer to identify sources' contribution to the total noise and the noise transmission paths.

INTRODUCTION

Low noise emission becomes a synonymous of comfort. Therefore, noise requirements in trains' specifications are becoming stricter. Trains not fulfilling noise requirements are rejected by the client or not accepted by the customers, originating high loses in terms of money and prestige. Solving an acoustical problem once the train is built is expensive an often non-viable. Therefore, prediction methods are essential in order to detect future acoustic problems during the design phase and apply corrective actions.

PREDICTION METHODS

Deterministic vibro-acoustic prediction methods (as FEM or BEM) could be very effective in the lower frequency range. However, in the high frequency range these methods are useless for several reasons [1]:

- The vast model sizes required avoiding 'spatial aliasing'.
- The inefficient design optimization due to the long computing time required for a single run.

- The lots of attributes influencing high-frequency performance, such as manufacturing variability, which cannot be defined in a deterministic way [2].

On the contrary, the *Statistical Energy Analysis* involves a statistically based model that is easier to use, simpler to understand, and well suited to high-frequency applications. SEA predicts the flow of energy between the various components of a multi-assembly system. It handles the complex interactions between the structure and the fluid in a straightforward way. And, because it is computationally efficient, it is ideal for interactive feasibility studies, sensitivity analysis and design optimization. [3]

SEA divides the model in coupled sub-system. Each vibration mode of the first sub-system is coupled to all of the relevant modes of the other sub-systems as shown in **figure 1**.

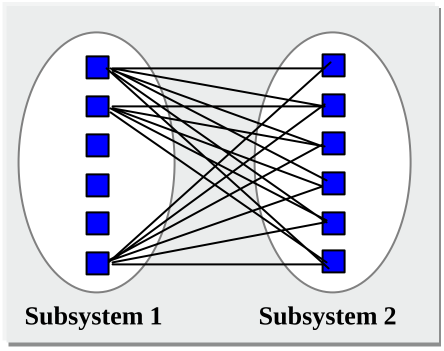

Figure 1: Coupling of sub-system modes

The total energy flow is the sum of the individual mode-to-mode energy flows. If the significant coupling is assumed to occur in a limited frequency band Δf, then the net energy flow is given by:

$$P_{12} = N_1 N_2 \left(\frac{E_1}{N_1} - \frac{E_2}{N_2} \right)$$

Where E_1 and E_2 give the time-averaged total energy of the sub-systems 1 and 2 respectively. N_i means the total number of modes within the frequency band. Hence, the energy flow is proportional to the difference in modal energy:

$$P_{12} = \omega(\eta_{12} E_1 - \eta_{21} E_2) \qquad \text{and} \qquad n_1 \eta_{12} = n_2 \eta_{21}$$

Where ω: band centre frequency; $n_i(\omega) = N_i/\Delta\omega$: modal density of sub-system *i*; η_{12} and η_{21}: SEA coupling loss factors. The expression $\omega\eta_{12}E_1$ represents the energy lost by sub-system 1 due to coupling with sub-system 2, whilst the quantity $\omega\eta_1 E_1$ represents the energy lost by sub-system 1 due to internal damping.

In many situations, the system comprises multi-modal sub-systems physically coupled to more than one contiguous sub-system. SEA assumes that the energy flow between two single sub-systems is given by the energy flow equation, which holds for a two-sub-system model [4]:

$$P_{ij} = \omega(\eta_{ij} E_i - \eta_{ji} E_j) \qquad n_i \eta_{ij} = n_j \eta_{ji}$$

The time-averaged energy dissipated in a sub-system i is given by:

$$P_{i,diss} = \omega \eta_i E_i$$

The global SEA equations of a system can be obtained through an energy balance of each individual sub-system (**figure 2**) [5]:

$$P_i = P_{i,diss} + \sum_{j \neq i}^{n} P_{i \to j} - \sum_{j \neq i}^{n} P_{j \to i}$$

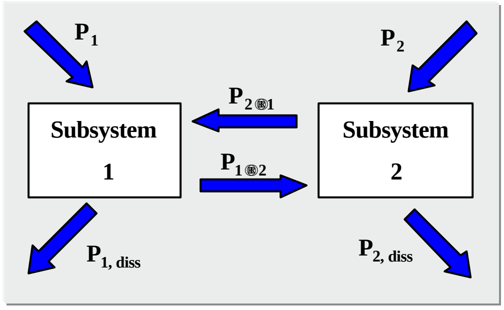

Figure 2: Energy balance of a two-sub-system model

Substituting equations concerning energy flow and energy dissipation into last equation:

$$P_i = \omega \eta_{ii} E_i + \sum_{j \neq i}^{n} \omega \left(\eta_{ij} E_i - \eta_{ji} E_j \right)$$

And in the reciprocity equation:

$$n_i \eta_{ij} = n_j \eta_{ji}$$

VEHICLE VIBRO-ACOUSTIC DESCRIPTION

The methodology has been applied to a real train (with known measured data) in order to validate it. It is a regional train with a diesel engine and a maximum velocity of 120 km/h. The general arrangement of the train set is shown in **figure 3**.

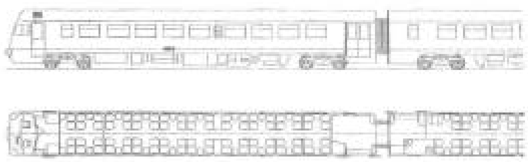

Figure 3: Train set studied

The noise and vibration sources are described by means of its sound power level and the vibration level transmitted to the car body. These data are obtained from noise and vibration SENER database, collecting previous experiences, from sub-suppliers information or even predicting them. To describe in a vibro-acoustical way the car structure elements (roof, floor, doors, stairs, windows, walls...) next parameters are evaluated [6]:

- Sound Transmission Loss (TL)
- Vibration Transmission Loss (TL_v)
- Radiation efficiency (σ)

As for sources, these data are obtained from SENER database, from previous experiences, from sub-suppliers data or predicting them (see **figure 4**). Predicting the TL and the TL_v of all the structure parts using the SEA method is an essential way because in most cases experimental data is not yet available during the design phase. The SEA models are defined by the coarse geometry, damping and physical properties as stiffness, density and Poisson coefficient [7]. Good enough precision figures can be achieved [8]. For instance, **figure 5** shows the correlation between the predicted and the tested TL at the floor.

Figure 4: Experimental measurements of the TLv, TL and s at the floor configuration

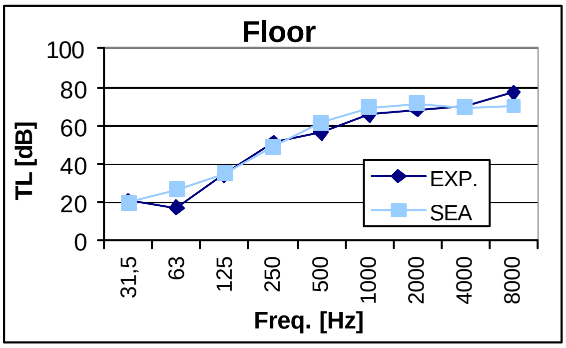

Figure 5: Floor TL correlation

INTERNAL NOISE

To analyze the total internal noise two SEA models have been developed: one for the structure-borne noise and another one for the air-borne noise.

<u>Structure-Borne Noise</u> [9, 10]

The procedure used to predict the structure-borne noise is:

- Obtaining the vibration velocity (*v*) for each of the interior panels using a SEA model.
- Calculating the radiated sound power for the interior panels. That depends on the panel surface, the space-time average mean-square vibration velocity and its radiation efficiency [11]:

$$W_{\Delta S} = Srcs_{\Delta S} \left\langle v^2 \right\rangle_{St\Delta}$$

- Applying the radiated sound power (W_{DS}) as a source in the air-borne model.

Most important vibration sources like the engine or the bogies must be taken into account. The train has been divided into cross sections to model it and the sections have to be chosen depending on the situation of the sources and the different structural elements.

The modeling criterion is avoid to add every structural part, but just the ones influent to the vibrating way of the car structure. For example, the fiberglass into the wall hasn't been modeled because of its low stiffness. See the sketch of the wall in **figure 6.** The use of FEM (in the middle frequency range) and SEA methods has been combined in the case of the floor because it is a complex floating structure.

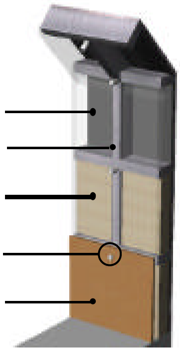

Figure 6: Wall sketch

Air-borne noise

In order to predict the air-borne noise, the interior and exterior air of the train have been modeled using the SEA method. To build the air-borne model the train has been organized in the same transversal sections as in the structure-borne model.

This model includes all the air-borne sources and sources from the structure-borne noise. The interior air sub-systems definition includes absorption elements (seats, perforated roof...). The connections between the interior and exterior sub-systems are defined by the sound transmission loss (TL) of the bodywork elements. **Figure 7** shows the experimental operation.

Figure 7: Interior noise measurements

The differences between the predicted total sound pressure levels and the experimental data were always lower than 3 dB. **Figure 8** shows the predicted sound pressure levels distribution inside the train.

Figure 8: Interior Noise at 120 km/h

This methodology allows the engineer to predict not only the total interior noise in each transversal section, but the sources' contribution as well. This way makes possible to detect the critic sources for each situation. It is also possible to determine the sound and vibration transmission paths, getting which a bodywork element needs to be reinforced.

EXTERNAL NOISE

External predictions have been performed under the following conditions:

- Standstill, idling - 7,5 m
- Maximum speed, 120 km/h - 25 m

A combined methodology named SOEXT was developed based in the SEA method and in the free-field classical equations. The equations of outdoor sound propagation explicitly contains the effects of wave divergence, source directivity, and large surfaces near the source [12]:

$$L_P(r) = L_W - 20\log r + DI_{revr} - 10\log\frac{\Omega}{4p} - 11 - A_{combined,revr}$$

Where DI_{revr} is the source directivity index in the receiver direction, W is the solid angle at the source that is available for sound propagation, and $A_{combined,revr}$ is the combined attenuation from all significant propagation mechanisms between source and receiver.

In most cases, sources are located close to the ground (bogies, engine...), so sound energy originally headed downward is reflected upward by the ground, thereby increasing the sound energy radiated upward.

Figure 9 shows the predicted total sound pressure level in the noisiest external point (7,5 m) for the train at standstill idling. This graphic shows the predicted sources contribution to the total noise for this point as well.

The precision of this method proved excellent. **Figure 10** shows the comparison between the SOEXT predicted sound pressure levels and experimental data at two different points: one in front of the intercommunication gangway and another one in front of the engine.

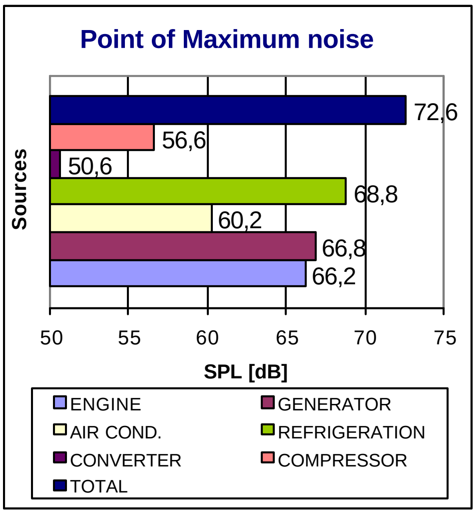

Figure 9: Sources contribution for the noisiest point (standstill idling)

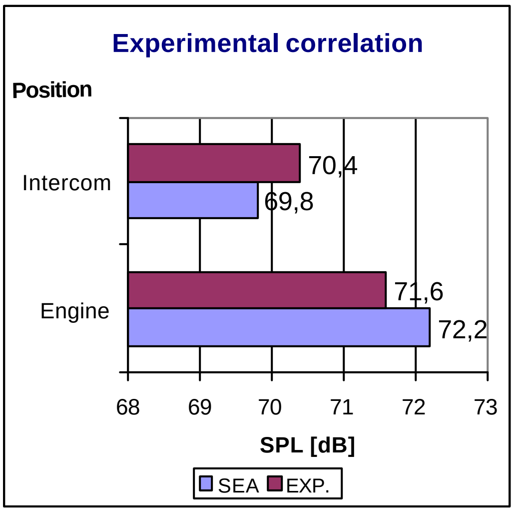

Figure 10: Correlation between predicted and experimental data

CONCLUSIONS

The external noise prediction method is simple and considerably exact (errors lower than 1 dB). It allows detecting problematic sources and evaluating correcting actions effect.

The internal noise prediction method is more complex and requires a larger amount of knowledge and longer developing time. However, it is really useful to carry out vibro-acoustic analysis and to diagnose acoustic problems allowing the detection of most problematical sources and the noise transmission paths. The results are good enough (approximately 1-3 dB of deviation). This is basically due to imprecision in the physical parameters definition and construction variability. However, this doesn't invalidate the method. Even an excess noise of 3 dB once the train is built can be solved using correcting actions. Otherwise, excesses of 10 dB are commonly impossible to solve.

The additional great advantage of the method is that can be used in the initial design phases due to the low vehicle definition grade required for developing a model.

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