inter.noise 2000
The 29th International Congress and Exhibition on Noise Control Engineering
27-30 August 2000, Nice, FRANCE

I-INCE Classification: 7.6

AN INNOVATIVE APPROACH FOR THE VIBRO-ACOUSTIC MODELLING OF A COMPLEX INDUSTRIAL INSTALLATION USING BOTH STATISTICAL ENERGY ANALYSIS AND FINITE ELEMENT METHODS

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Keywords:
SEA, FINITE ELEMENT, TRANSMISSION OF VIBRATIONS, SUBMARINES

ABSTRACT
In order to improve the Acoustic Stealthiness of nuclear submarines, it is necessary to predict the transmitted vibrations from the nuclear reactor to the hull. This implies to have a reliable method at one's disposal for vibration transfer modelling within the reactor, for a large frequency range. The latter is a complex welded structure with fluid-structure coupling, and a classical FE model is not workable, due to the calculation heaviness and high frequency limitation. On the opposite, the SEA method is an alternative one, allowing determining the energy flows at low numerical cost and with few parameters. Complementarily to SEA, FE method is used for studying virtually uncoupled subsystems from the whole structure and which are not reachable by measurement, synthesising coupling loss factors (CLF) and modal densities for complex non-standard subsystems for which an analytical model does not exist.

1 - INTRODUCTION
The purpose of this paper consists in validating the SEA method coupled with FE calculation and experimental results, for prediction of the transmitted vibrations from a complex vibrating structure to the supporting installation, in a wide frequency range − say low, medium and high frequency domain. The whole process is evaluated in order to predict the transmitted vibrations from a nuclear reactor to the hull, on which some "classical" FE models have already been achieved for the very low frequency domain (on the first resonant modes of the whole structure, below 100 Hz).
The structure chosen for the development of this methodology is a reduced scale model of the installation, on which any dynamic characterisation measurement is easier to make, so that comparison between modelling and experiment is possible. The parts (subsystems) that make up the whole are complex welded structures; some of them are filled in with water (figure 1).

2 - EXPERIMENTAL PART
The measurement used for comparison with SEA transfer functions and parameters are accelerometric response / force-hammer excitation γ/F transfer functions between the different subsystems of the whole structure, and local inertances γ/F on each subsystem, in the frequency band 0-10 kHz. Post-treatments are processed in order to synthesise SEA-like transfer functions γ/F, local mobility v/F, power input Π_i, modal densities and CLFs: integration, quadratic summation on bandwidths identical to the ones of the model, spatial quadratic mean on each element in case of several points measurement, complex arithmetic.

3 - METHODOLOGY
The process is described in figure 2. The software used for SEA calculation is *AutoSEA* (*Vibro-Acoustic Sciences*) versions 1.5 and 2.0.

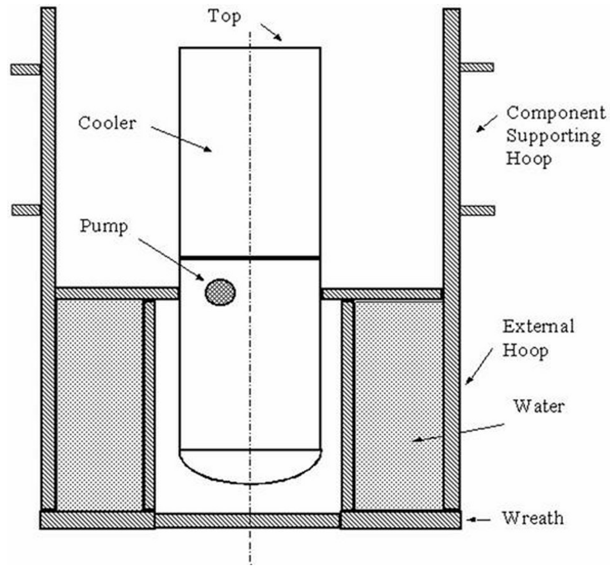


Figure 1: Schema of the installation.

4 - VALIDATION STAGE

Examining the modal densities on the main subsystems is one of the first ways for validating a SEA model. Indeed, the theoretical background of SEA is mainly built on both assumptions, for each subsystem located on the main energy paths, that the number N of mode per band is greater than about 5, and that the energy per band is equally distributed on each mode. The latter is probably the most important one, in that SEA calculation on the first resonant modes of a well-dotted subsystem will give worst results than on higher modes of one with few modes.

The number of modes per band $(f_c, \Delta f)$ can be obtained from the real part of local mobility $M{=}v/F$ at the point of excitation as following (m being the subsystem mass):

$$N\left(f_c\right) = \int_{\Delta f} 4m\mathrm{Re}\left[M\left(f\right)\right] df \tag{1}$$

This number of mode must be issued theoretically from measurement on subsystem isolated from the rest of the whole structure; this may consequently differ from measurement made on the assembled structure. In particular, evaluating the mass m implied may be difficult. This problem is particularly encountered in the low frequency domain of the first resonant modes of the whole. For higher frequencies, another problem is posed by proper measurement of the real part of local mobility: the phase of M depends on the distance between excitation and response points, especially at high frequency, where Re(M) may become negative.

Ensuring the power input Π_i is well taken into account provides another guaranty for a representative SEA modelling of the problem, still using the measurement of Re(M):

$$\Pi_i\left(f_c\right) = \int_{\Delta f} \left|F\right|^2 \mathrm{Re}\left[M\left(f\right)\right] df \tag{2}$$

Finally, experimental and SEA-computed transfer functions are to be compared, in order to determine the main energy flows within the installation, and estimate the representativeness of the model.

A "State 0" model have been achieved, as shown on figure 3. It is constituted with plates, beams, cylinders, and acoustical cavity, taking into account flexural waves essentially. The damping loss factors (DLF) are assumed to take values between 1% and 5%, and decrease logarithmically for high frequencies (usual welded structures).

Many of the comparisons described above have been made for that model. A transfer function between the external hoop and the wreath is shown on figure 4, in the range 0-10 kHz (bandwidth 1000 Hz). At this stage, the fully analytical SEA model is not considered to be satisfying. In particular, experimental modal densities of important subsystems like the pump or the wreath are not corresponding to those of the standard-modelled SEA elements, because of their geometrical and physical complexity.

5 - SPECIFIC FINITE ELEMENT COMPUTATION

Following the methodology presented above, further investigation is necessary to build an acceptable model of the installation. Some FE modelling has been achieved on one specific subsystem, in order to determine or validate the modal densities (pump, wreath), and CLF values (between wreath and external hoop, cooler and top). This approach allows to "replace" isolated experiment on those uncoupled subsystems, which is not possible on the assembled installation, and precisely model them complexity and evaluate its influence on the corresponding SEA parameters. The number of modes is computed by two ways:

- using equation (1), by computing an average mobility M for several points of virtual "measurement" on the FE model,
- by counting one by one the FE-computed modes.

The CLF η_{ij} are computed by "inverse-SEA", i.e. η_{ij} become the unknown variables, and the energies E_i are estimated by computation. A simple method consists in exciting one subsystem j at time, with the input power P'_j, and the corresponding energy responses with notation E_{ij} on each subsystem i. Thus, for a two-subsystems model, one obtains explicitly:

$$\begin{bmatrix} \eta_{11}n_1 & -\eta_{21}n_1 \\ \eta_{12}n_2 & \eta_{22}n_2 \end{bmatrix} = \frac{1}{\omega}\frac{n_1 n_2}{E_{11}E_{22} - E_{12}E_{21}} \begin{bmatrix} P'_1 & 0 \\ 0 & P'_2 \end{bmatrix} \begin{bmatrix} E_{22}/n_2 & -E_{12}/n_1 \\ -E_{21}/n_2 & E_{11}/n_1 \end{bmatrix}$$

The input power is computed using the mobility $\langle M \rangle_{\Delta f}$, following equation (2), and the vibratory energies by computing average transfer functions h_{ij}, so that

$$\eta_{12} = \frac{\mathrm{Re}\,(M_2)}{\omega m_1} \frac{|h_{21}|^2}{|h_{11}|^2\,|h_{22}|^2 - |h_{21}|^2\,|h_{12}|^2} \tag{3}$$

where m_1 is the mass of subsystem 1, and

$$h_{ij} = \sqrt{\sum_k \left(\frac{v_{k,i}}{F_j}\right)^2}$$

Figure 5 shows the FE-computed modes compared to SEA-analytical modes on the wreath subsystem, in the frequency range 0-10 kHz. Due to added thickness and stiffness, the FE-computed modal density n_1 is greater than that obtained with SEA. η_{12} relative to the wreath-to-hoop connection has been also FE-computed, in the only range 0-3 kHz because of high-frequency limitation and heaviness of the FE method, and then extrapolated for higher frequencies following a SEA-like $1/\sqrt{f}$ law. η_{12} is synthesised by the reciprocal condition $n_2\eta_{21} = n_1\eta_{12}$. The results are presented on figure 5.

6 - CONCLUSION

The model has been modified following the methodology described above, after FE calculations of CLF on the wreath subsystem, and modal densities of wreath and pump. Resulting transfer functions within the "Final State" model of the installation, presented on figure 6, prove complemented FE calculations on specific subsystem help building a more representative SEA model adapted to the problem given.

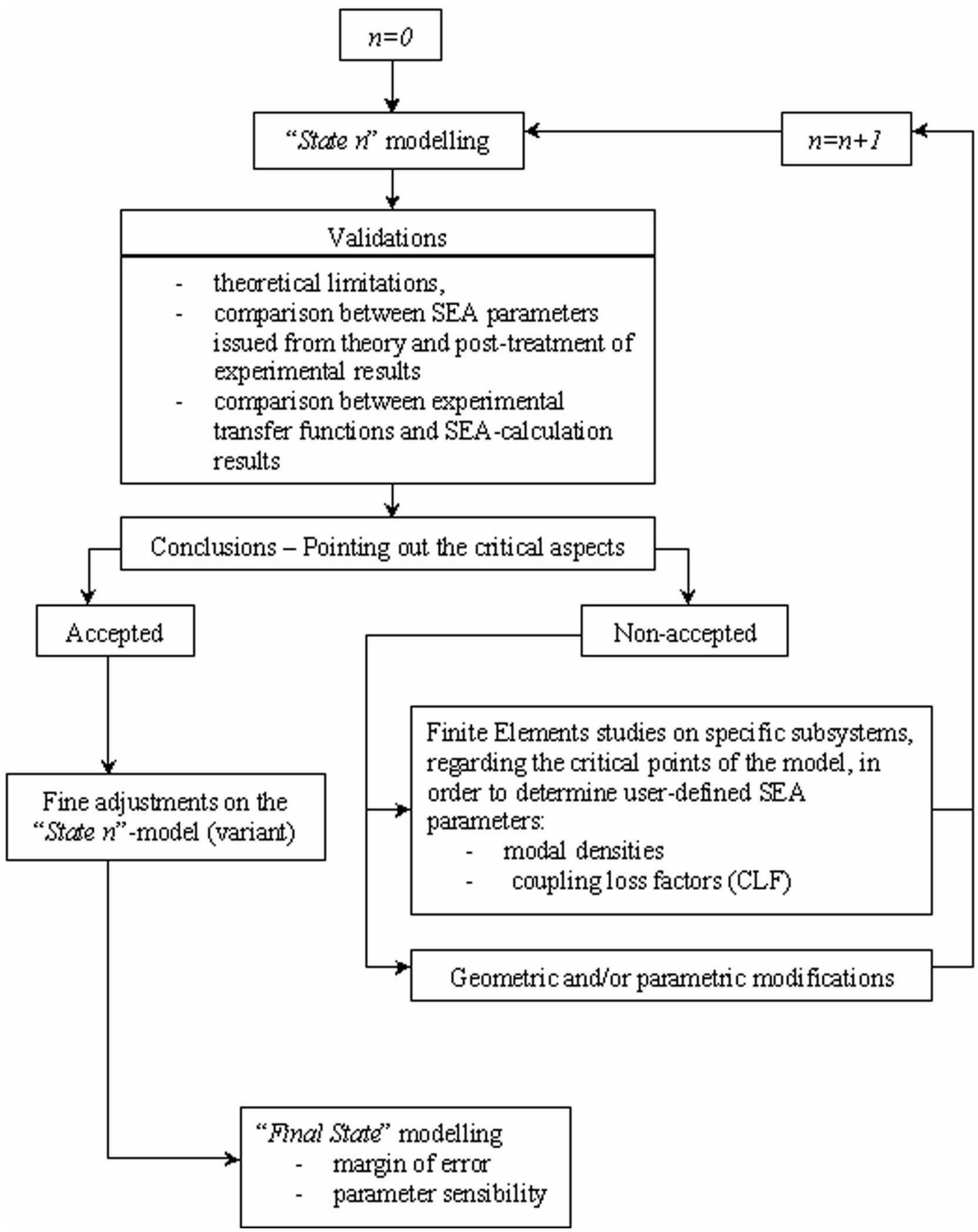


Figure 2: Methodology developed for studying the vibration transfers within the installation.

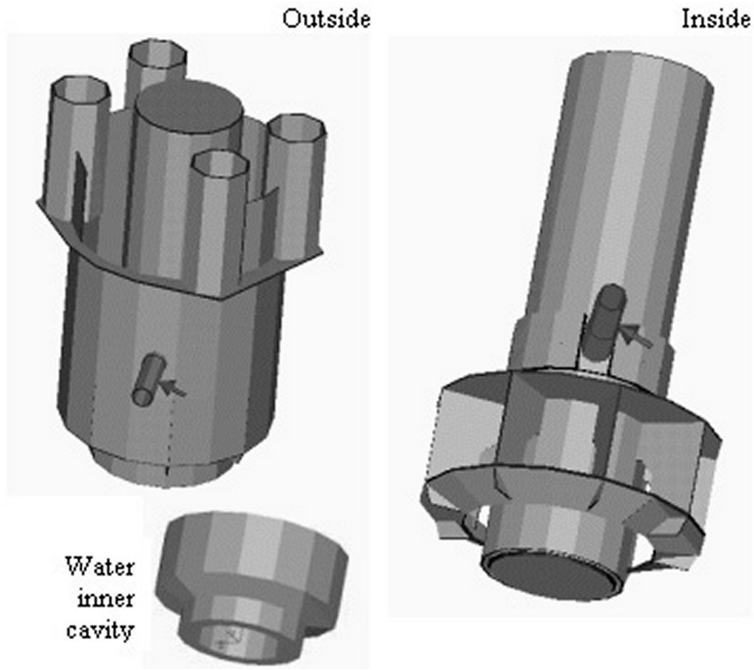


Figure 3: AutoSEA2 "state 0" model.

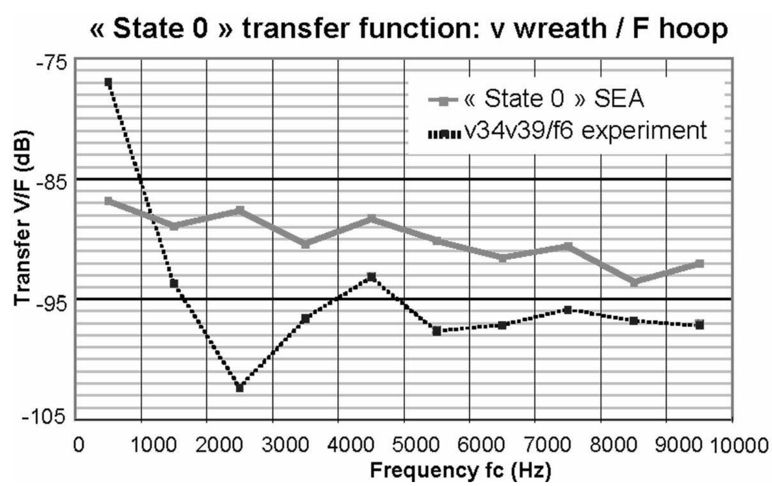


Figure 4: "State 0" model: comparison between experimental and SEA-computed transfers.

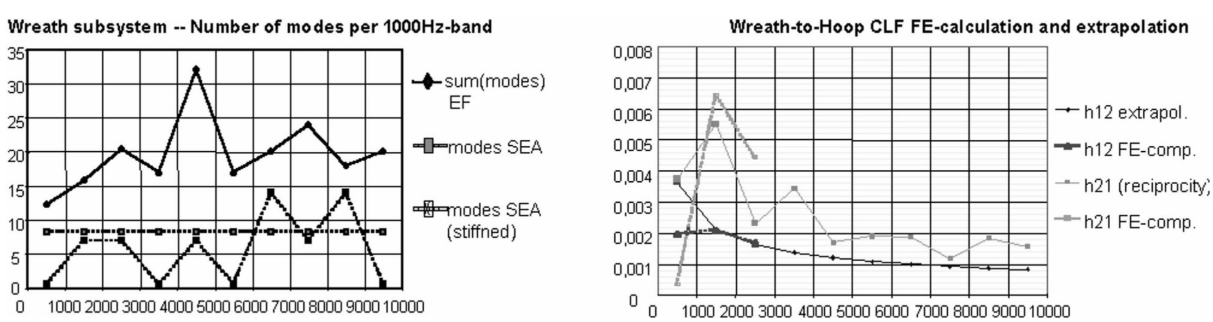
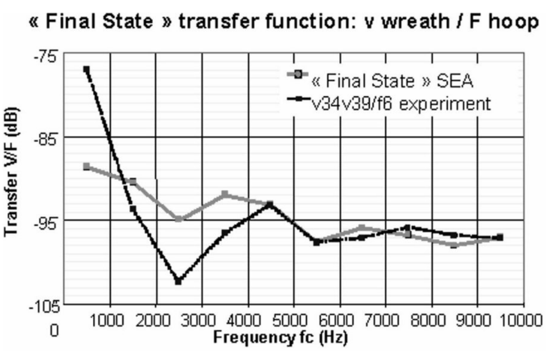


Figure 5: FE-computation of modal densities and CLF for the wreath subsystem.



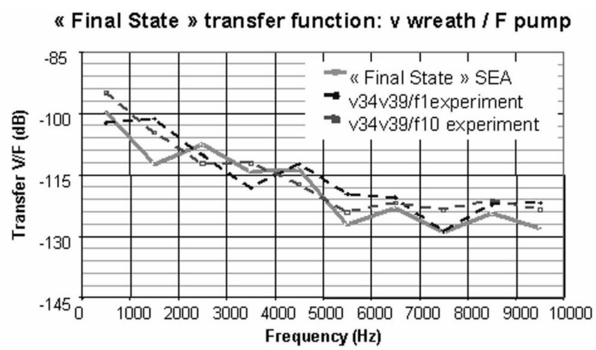


Figure 6: "Final state": comparison between experimental and SEA-computed transfers.