inter•noise 96
25th Anniversary Congress – Liverpool

INCE: 40

SOUND RADIATION FROM STRUCTURES: THE FREQUENCY AVERAGED QUADRATIC PRESSURE APPROACH

J L Guyader (1) & T Loyau (2)

(1) Laboratoire Vibrations-Acoustique, INSA de Lyon, Bat. 303, 69261 Villeurbanne Cedex, France,
(2) INRS, Avenue de Bourgogne, 54501, Vandoeuvre Cedex, France

1. INTRODUCTION

Prediction of noise radiated from machinery is a very difficult problem because of the complexity of the vibration fields producing noise. The classical Green's function formulation solve the problem using pure tone signals wich necessitate a very good knowledge of vibrations, including modulus and phase, this is quite impossible in machinery due to structural complexity. The necessity of finding appropriate method to predict radiation from machinery is then obvious. Our idea is to predict the acoustical behaviour using a quantity not so sensitive as the pure tone pressure radiated. As in S.E.A. we will introduce frequency band averaged quadratic quantity, in order to get a method (The FAQP method) that reduces the information necessary for calculations and also the computing time. The theory will be presented on the case of a baffled plate mechanically excited radiating in a semi infinite acoustic medium. Experiments have been done to check the validity of the method (number of mechanical excitations, far field condition, average frequency bandwidth, number of vibrating velocity points).

2. THEORY

The methods of prediction generally calculate the pressure at a given frequency in modulus and phase, on the contrary the experimental analysis of radiated noise is mainly made on frequency averaged quadratic pressure. Compared to experiments, the prediction gives non necessary information, that necessitates heavy calculations. To better fit with experimental reality and also to simplify calculations, the direct calculations of frequency band quadratic pressure is very convenient. The theory yet presented [1] is recalled on the case of a baffled plate radiating in a semi infinite acoustic medium.

The pressure can be calculated with the Rayleigh Integral [2]

$$p(M) = -\rho \int_s \frac{1}{2\pi} \frac{e^{-jkR}}{R} A(Q)dQ \tag{1}$$

Where R=|QM| distance between the two points M and Q, A(Q) is the plate acceleration at point Q, ρ is the fluid density and S the plate area. To simplify the prediction let us first calculate the square of the pressure modulus

(2) $$|p(M)|^2 = \rho^2 \int_S \int_S \frac{1}{4\pi^2} \frac{e^{-jk(R-R')}}{R.R'} A(Q)\, A^*(Q')\, dQ\, dQ'$$

This quantity is real and positive instead of complex like the pressure. This is a first regularization, however the pressure modulus can vary rapidly with frequency and space and is still difficult to predict. To obtain a smooth quantity one has to make an average of $|p(M)|^2$. We have made a frequency average over a band Δ of center angular frequency Ω

(3) $$\left\langle |p(M)|^2 \right\rangle = \frac{\rho^2}{4\pi^2} \int_S \int_S \left\langle \frac{e^{-jk(R-R')}}{R.R'} A(Q)\, A^*(Q') \right\rangle dQ\, dQ' \ ;\text{with } \langle\ \rangle = \frac{1}{\Delta} \int_{\Omega-\Delta/2}^{\Omega+\Delta/2} d\omega$$

Let us remark that the term $e^{-jk(R-R')}$ is generally slowly variable compared to the exponential term e^{-jkR} appearing in Rayleigh integral (1), the difference between R and R' can be small whatever the point of the plate, when the pressure is calculated in the far field of the plate.
This property suggests to use an asymptotic expansion of the exponential in (3)

(4) $$e^{-j\frac{\omega}{c}(R-R')} = e^{-j\frac{\Omega}{c}(R-R')} [\, 1 + \sum_{p=1}^{\infty} \frac{\left(-j\frac{\varepsilon}{c}(R-R')\right)^p}{p!}] \ ; \ \text{ with } \omega = \Omega + \varepsilon$$

In making use of (4) and (3) one obtains

(5) $$\left\langle |p(M)|^2 \right\rangle = \frac{\rho^2}{4\pi^2} \int_S \int_S \frac{e^{-j\frac{\Omega}{c}(R-R')}}{R.R'} \left[\left\langle A(Q)\, A^*(Q') \right\rangle + \sum_{p=1}^{\infty} G_p \right] dQ\, dQ'$$

with

(6) $$G_p = < \frac{-j\frac{\varepsilon}{c}(R-R')}{p!} A(Q)\, A(Q') >$$

The question is now : what is the number of terms G_p necessary for convergence. This is related to the value of the term ε/c(R-R') .It is not related to the central frequency (Ω) but to the band of averaging, the maximum value of ε being Δ/2. For practical application the approximation at zero order is generally sufficient, equation (5) reduces to

(7) $$\left\langle |p(M)|^2 \right\rangle = \frac{\rho^2}{4\pi^2} \int_S \int_S \frac{e^{-j\frac{\Omega}{c}(R-R')}}{R.R'} \left\langle A(Q)\, A^*(Q') \right\rangle dQ\, dQ'$$

The pressure square modulus field is directly related to the term $\langle A(Q)\, A^*(Q')\rangle$; it is the frequency average of the product of structural acceleration at each point Q and Q'.The classical problem (eq. (1)) needs the knowledge of the acceleration at each point at a given frequency. When people is interested in calculating the pressure radiated from measured plate vibrations it appears generally a problem with the phase, that vary more or less randomly, except at resonance frequencies. Thus it is quite impossible to apply (1) in this way, particularly for reverberant structures submitted to complicated excitations.
On the contrary, expression (7) seems particularly adapted to such cases as the term $\langle A(Q)\, A^*Q'\rangle$ is no more sensitive to large fluctuation because of averaging over frequency. Let us also notice that non coherent vibrations at points Q and Q' leads to $\langle A(Q)\, A^*Q'\rangle=0$, giving no contribution to the sound radiated. The expresssion (7) gives also the possiblity of separating radiation of independent zones of the structure where the averaged product is small.

3. VALIDATION OF THE METHOD

Several measurements and calculations have been done in order to show the influence of all parameters included in the formulation (7). For all the cases presented here, the structure was a steel baffled plate (1m x 1 m x 0.004 m) radiating in a semi-anechoic room. Vibrating velocity has been measured with a LASER vibrometer (POLYTEC OFV 3000). The optical fiber was automatically located at the points of a regular mesh. The pressure has been measured by a ½" B&K 2671.

a) Number of vibrating points

The figure (1) shows the measured averaged pressure and the calculated one (FAQP) for two regular meshes, 121 and 36 points. The frequency bandwidth was 200 Hz. The location of the acoustic pressure point was (x=0.53, y=0.8 and z=3.8 m). The plate was excited with an electrodynamic shaker. As in the narrow frequency analysis, it appears a cut off frequency depending on the size of the mesh [3,4].

b) Distance between the vibrating structure and the acoustic pressure point (R)

Several cases have been investigated in order to test the condition of far acoustic field induced by one of the FAQP approximation. These tests show that the calculated pressure stays close to the measured one for R going from 1 to 4 m.

c) Number of mechanical excitation sources

The figure (2) shows the measured averaged pressure and the calculated one (FAQP) in the cases of one, two and three real mechanical excitation respectively (fig.2 a,b,c). Each source was an electrical motor with 4 fixation points and with approximately the same electrical power than the others. The velocity was measured on a regular mesh of 121 points (11 x 11). The location of the acoustic pressure point was (x=0.53, y=0.8 and z=3.8 m). The frequency bandwidth was 200 Hz. As expected, the increase of sources decrease the phase relations between each vibrating point and so increase the quality of FAQP's results.

d) Frequency bandwidth

The figure (3) shows the measured averaged pressure and the calculated one (FAQP) for three frequency bandwidth 100, 200 and 400 Hz respectively (fig.3 a,b,c). The velocity was measured on a regular mesh of 121 points (11 x 11). The location of the acoustic pressure point was (x=0.53, y=0.8 and z=3.8 m). The plate was excited with one artificial source. The curves show that the discrepancies between calculated and measured results do not increase a lot for these three frequency bandwidth.

4.CONCLUSIONS

The prediction of the vibroacoustic behaviour of structures is generally done with the integral equation method. It gives the radiated pressure at a given frequency in phase and modulus. The method presented in this paper (FAQP) proposes a calculation more realistic in the case of machinery noise. The curves show that the cut off frequency depending on the mesh is present whatever the bandwith is. Below the cut off frequency, increasing the averaging frequency band permits to reduce the difference between measured and predicted pressure.

References

[1] J.-L. Guyader, T. Loyau, "Prediction of the frequency averaged quadratic pressure radiated from a plate", 15 th I.C.A., Trondheim, Norway, 26-30 June 1995, p. 117-120.

[2] Junger and D. Feit, "Sound, structures and their interaction", The MIT press, second edition, 1986.

[3] D. Blacodon, D. Brenot, Estimation du champ acoustique lointain d'une plaque plane bafflée à partir de la mesure de son champ d'accélération , Journal de Physique IV, colloque C1, supplément au journal III, Vol. 2, avril 1992.

[4] G. Lovat, J.-L. Barbry and T. Loyau, Calculation of the acoustic radiation of a parallelepipedic structure by using acceleration measurements. Part 1 : Evaluation of inaccuracies attached to the simplifications of the model in the case of a single vibrating side , Noise Control'95, Warsaw, Poland, 20-22 June 1995, p. 147-152.

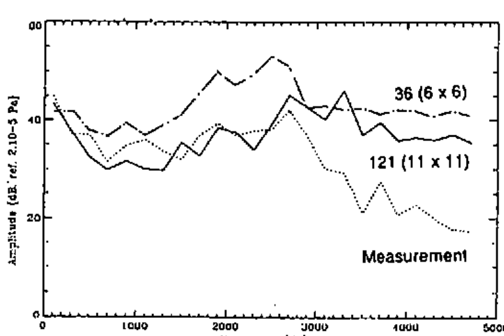

Figure 1 : Measured and calculated (FAQP) pressure radiated in the far field of a steel baffled plate for several number of vibrating velocity points.

(a)

(b)

(c)

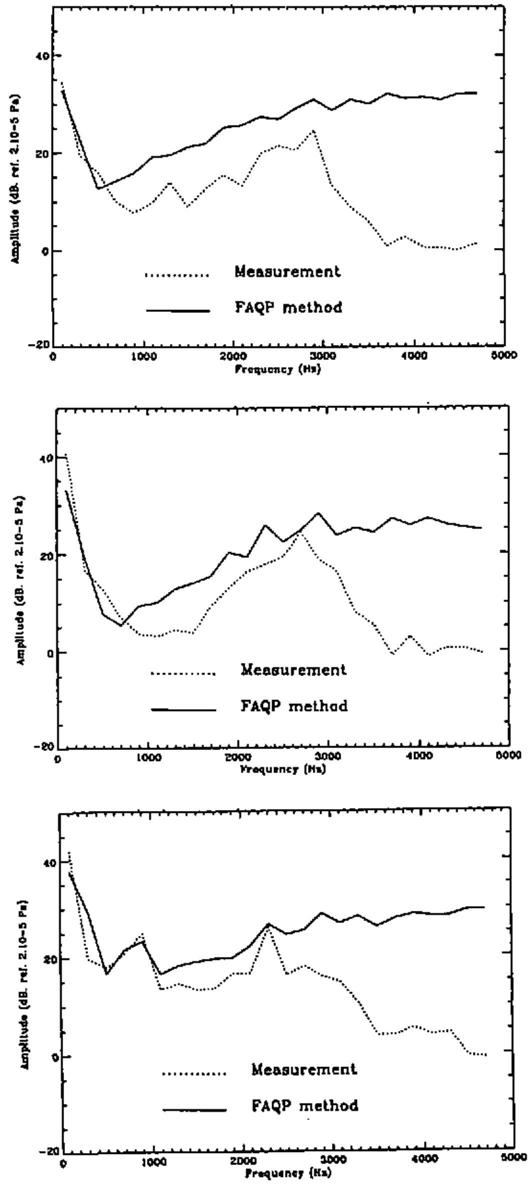

Figure 2 : Measured and calculated (FAQP) pressure radiated in the far field of a steel baffled plate ; Number of mechanical excitation sources : (a) 1 motor, (b) 2 motors and (c) 3 motors.

(a)

(b)

(c)

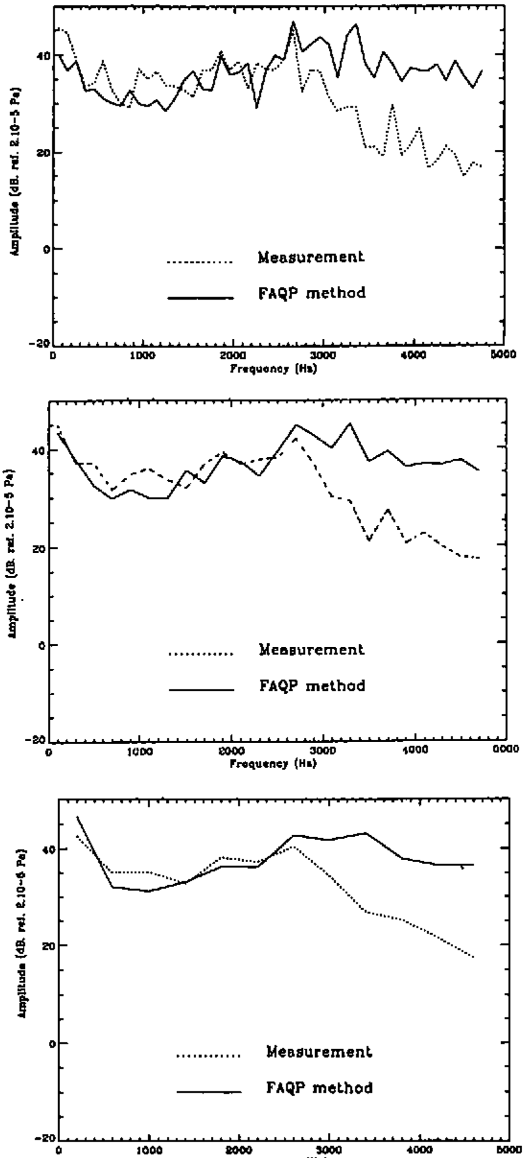

Figure 3 : Measured and calculated (FAQP) pressure radiated in the far field of a steel baffled plate ; frequency bandwidth : (a) 100 Hz, (b) 200 Hz and (c) 400 Hz.