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ACOUSTIC RADIATION FROM STRUCTURES : THE FREQUENCY AVERAGED QUADRATIC PRESSURE PREDICTION

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ABSTRACT

We propose a method (FAQP) for the prediction of the acoustic radiation of a structure using a quantity not so sensitive as the pure tone pressure radiated. As in S.E.A., we introduce frequency band quadratic averaging. The theory for a simply supported baffled plate, previously presented in the past, is briefly recalled. The main assumption is discussed and predictions are compared with experimental data in the case of a baffled steel plate. The present paper extends the method to the case of the hood of an internally excited machine. Comparison of theoretical and experimental data on a circular saw is presented in order to show the limitation of the method when applied to non-planar structures. The contribution of each side of the hood to the total acoustic radiation is also given.

1. INTRODUCTION

The prediction of noise radiated from machinery is a very difficult problem because of the complexity of the vibrational fields that produce it. It is not possible to use the classical Green's function formulation for the solution of this type of problem using pure tone signals because this approach requires good definition of the vibrations (including modulus and phase) which is impossible in the case of real machinery due to its inherent structural complexity. Other, more appropriate methods must be found to predict the acoustic radiation emitting from such structures. The idea that we propose here is based on the use of a quantity that is much less sensitive than the pure tonal radiated pressure. As in S.E.A., we introduce

frequency band quadratic averaging in order to obtain a method that reduces both the amount of information necessary for calculation and the computing time.
The use of this theory for the case of a simply supported baffled plate was previously presented [1], [2]. The main assumption was discussed, and predictions were compared with classical calculations and experimental data. The present paper extends the method to the case of the hood of internally excited machinery.
Comparison of theoretical and experimental data for a circular saw will be shown in order to explore the limitations of the method when it is applied to non-planar structures.

2. THEORY

Prediction methods generally calculate the pressure at a given frequency in terms of modulus and phase. On the contrary, the experimental analysis of radiated noise is generally done on the frequency averaged quadratic pressure. Predictions thus provide extraneous information when compared to experimental data, and often require heavy calculations. Frequency band quadratic pressure is therefore quite convenient since it provides information that is more directly verifiable by experiments, and is much simpler than other methods. The theory of the FAQP approach already presented [1], [2] is briefly reviewed for the case of a baffled plate radiating in a semi-infinite acoustic medium.

The pressure can be calculated with the Rayleigh Integral [3]

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

Where R=|QM| is the distance between the two points M and Q, A(Q) is the acceleration of the plate at point Q, ρ is the fluid density and S the surface area of the plate. To simplify the prediction let us first calculate the square of the pressure modulus and the average of |p(M)|2. We thus define a frequency average over a band Δ of center angular frequency Ω

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

with $\langle\ \rangle = \frac{1}{\Delta} \int_{\Omega-\Delta/2}^{\Omega+\Delta/2} d\omega$

After several assumptions
a) the pressure is calculated in the far field of the plate,
b) an asymptotic expansion of the exponential $e^{-j\frac{\omega}{c}(R-R')}$ is used.

The equation **(2)** is then reduced to :

$$\text{(3)} \qquad \left\langle |p(M)|^2 \right\rangle = \frac{\rho^2}{4\pi^2} \int_S \int_S \frac{e^{-j\frac{\Omega}{c}(R-R')}}{R.R'} \left\langle A(Q)\, A^*(Q') \right\rangle dQ\, dQ'$$

The square modulus of the pressure field is directly related to the term <A(Q) A*(Q')>, and is the frequency average of the product of structural acceleration at each point Q and Q'. The classical problem (equation **(1)**) requires knowledge of the acceleration at each point at a given frequency. When trying to calculate the pressure radiated from a plate where the vibrations are measured, one generally encounters problems with the phase of the vibrations, which except at the resonance frequencies, varies more or less randomly. Thus it is quite impossible to apply **(1)** in this case, particularly for reverberant structures submitted to complicated excitations.

On the contrary, expression **(3)** seems particularly adapted to such cases since the term <A(Q) A*Q'> is not sensitive to large fluctuation because of averaging over a range of frequency. One should also notice that non-coherent vibrations at points Q and Q' lead to <A(Q) A*Q'>=0, giving no contribution to the sound radiated. Expresssion **(3)** also provides the possiblity of separating radiation of independent zones of the structure where the averaged product is small.

3. EXPERIMENTAL RESULTS OBTAINED IN THE CASE OF A BAFFLED PLATE

The first structure considered was a rigid, baffled plate, radiated in a half domain (Figure 1). The steel plate used (0.6 m by 0.4 m by 5 mm) was simply supported in a rigid baffle (2 m by 2 m) and radiated in an anechoic room. The plate was excited by an electrodynamic actuator supplied by a random signal (normal point force applied to the point x=0.25 m, y=0.12 m, z=0 m). A force transducer was located between the plate and the electrodynamic actuator. The force signal was used as a reference phase signal. Velocity measurements were made at the center of each element of a 13 x 9 regular mesh. A LASER vibrometer was used. It was automatically displaced by a robot.
A comparison of the measured pressure radiated (average frequency band) by the rectangular plate *(0.25,0.12 and 1.6 m)* in the far field, and the FAQP (equation **(3)**) is shown in Figure 2. The agreement is excellent for far field prediction and frequency averaging on bands of 200 Hz.

4. RESULTS OBTAINED IN THE CASE OF THE HOOD OF A CIRCULAR SAW

Even though the formulation **(3)** is mathematically correct only in the case of a rigid baffled plate, the FAQP method is employed in the case of the hood of an internally excited machine.

4.1. Description of the circular saw

The study was carried out on the body of a 1.95 m x 0.75 m x 0.85 m circular saw (see Figure 3), located on the reflecting floor of a semi-anechoic room. The parallelepipedic shaped body of the machine is comprised of five plates of varying thickness.
The blade (Ø 400 mm) is indirectly driven by a belt drive electrical motor. During cutting operation, the blade is moved up and down by means of a mechanical system. In this study the blade was left in the lowered position inside the body. The excitation spectrum of the body includes the rotation frequency of the electrical motor (30 Hz), the rotation frequency of the blade (39 Hz) and all of their harmonics.
To ensure that the body of the machine was the only vibrating source, all holes and slits were acoustically treated. The temperature became stable after one hour.
A microphone located in the nearfield of the machine is used to obtain a phase reference signal.

4.2 Description of the measurements

Two types of measurement were carried out :

- vibration measurements on the body of the machine in order to calculate its acoustic radiation. Vibrating velocity was measured on the body of the machine at 946 points of a regular mesh using a LASER vibrometer (195 points on sides 1 and 3, 224 points on sides 2 and 4, 108 points on side 5).
- acoustic pressure measurements, which were used to validate the FAQP approach.

4.3 Results obtained

The calculated (FAQP, equation **(3)**) and the measured far field acoustic pressure of the machine *(0.26, -1.4 and 2.5 m)* are compared in Figure 4. The agreement is very good for far field prediction and frequency averaging on bands of 50 Hz.

Figure (5) shows a comparison in the far field of the machine *(0.26, 0.37, 3.2 m)* at another point, with a 1/3 octave frequency band. The agreement between measured and predicted values is again quite good.

For this last point, we calculated the acoustic pressure radiated by each of the five sides of the hood using the FAQP method with a 1/3 octave frequency band. The superimposition of the total acoustic pressure radiated by the hood and the acoustic pressure radiated by each side is given Figure 6 (Figure 6a for the sides 1 and 2 and Figure 6b for the sides 3, 4 and 5). As can be seen from this figure, side 5 is the main contributor to the acoustic radiation over all frequency band at the point considered here. The FAQP method can also be used to obtain contribution of each side of the hood to the total acoustic radiation.

5 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 more realistic calculation in the case of machinery noise. The results obtained in the case of a hood of a circular saw show that FAQP method gives good results for closed non-planar structures constituted of plates. The contribution of each side of the hood to the total acoustic radiation can also be obtained and FAQP can be so considered as an interesting vibroacoustic diagnostic tool.

REFERENCES

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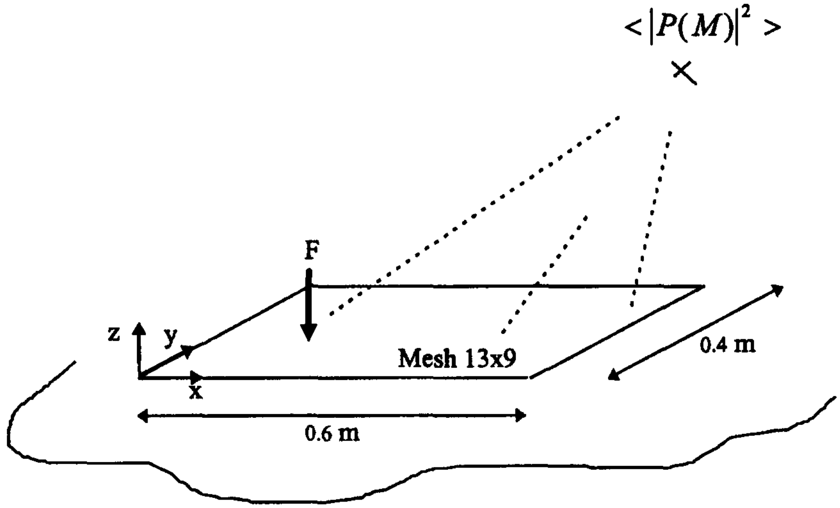


Figure 1 : First experiment. The case of a rigid baffled plate (thickness : 5 mm).

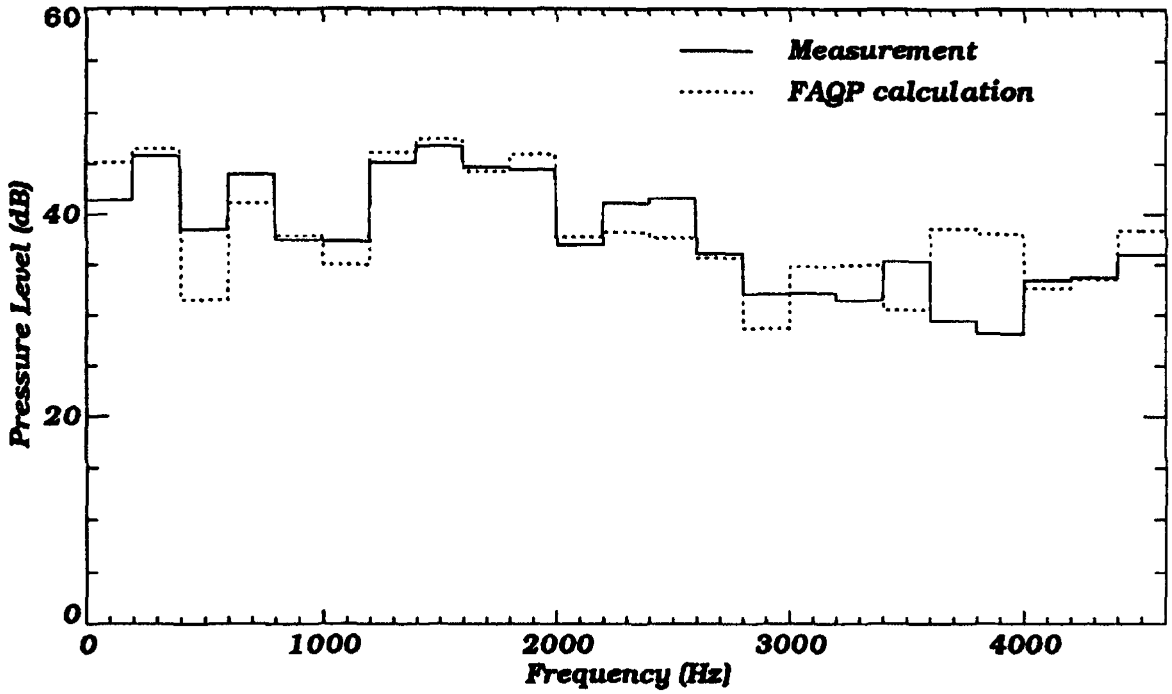


Figure 2 : Measured and calculated (FAQP) pressure radiated in the far field of a baffled steel plate. Plate dimensions (0.6, 0.4, 0.005 m), point of pressure (0.25, 0.12, 1.6 m).

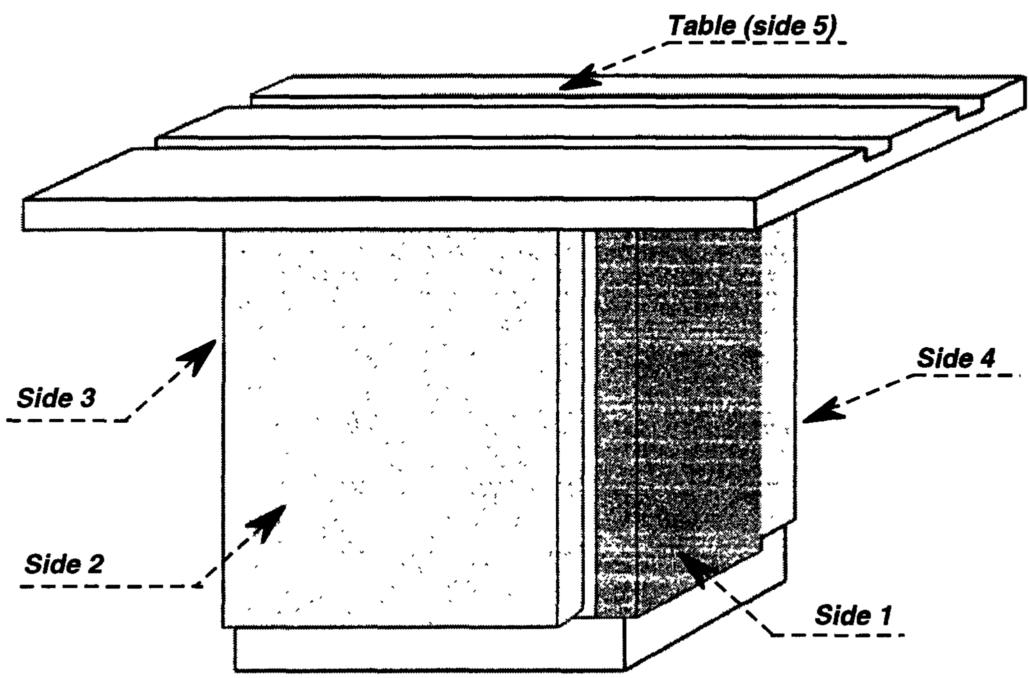


Figure 3 : Second experiment. The case of a circular saw.

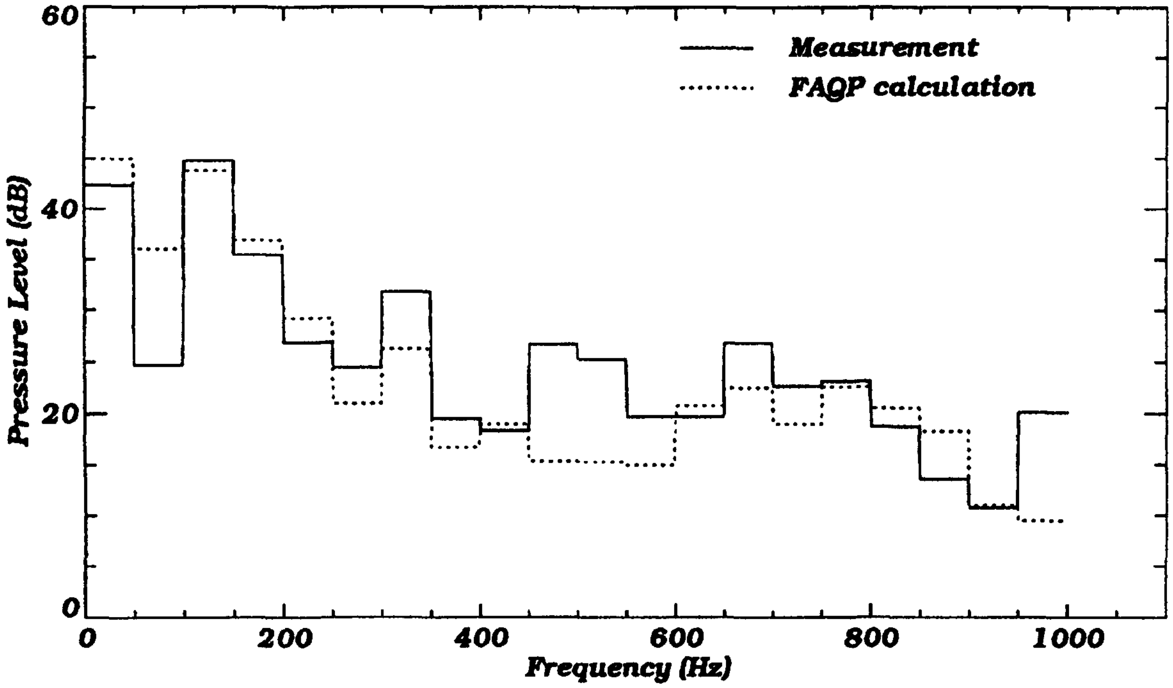


Figure 4 : *Measured and calculated (FAQP) pressure radiated in the far field (Point M1) of the circular saw. The frequency bandwidth* Δ *is 50 Hz.*

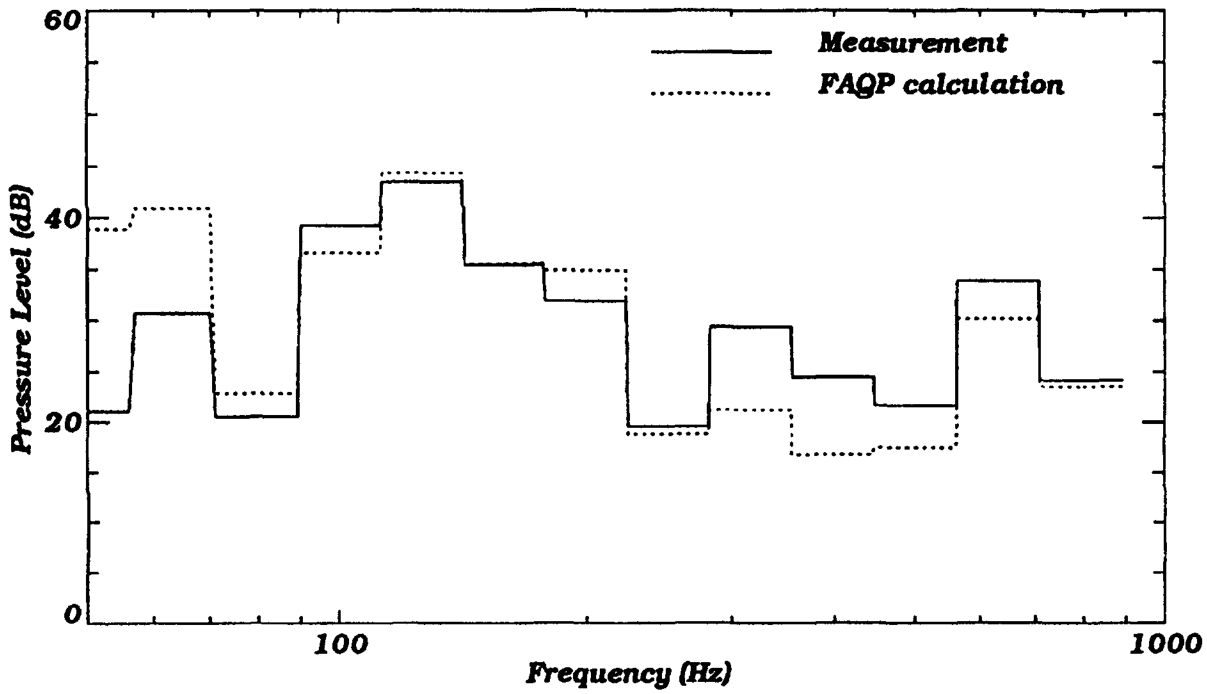
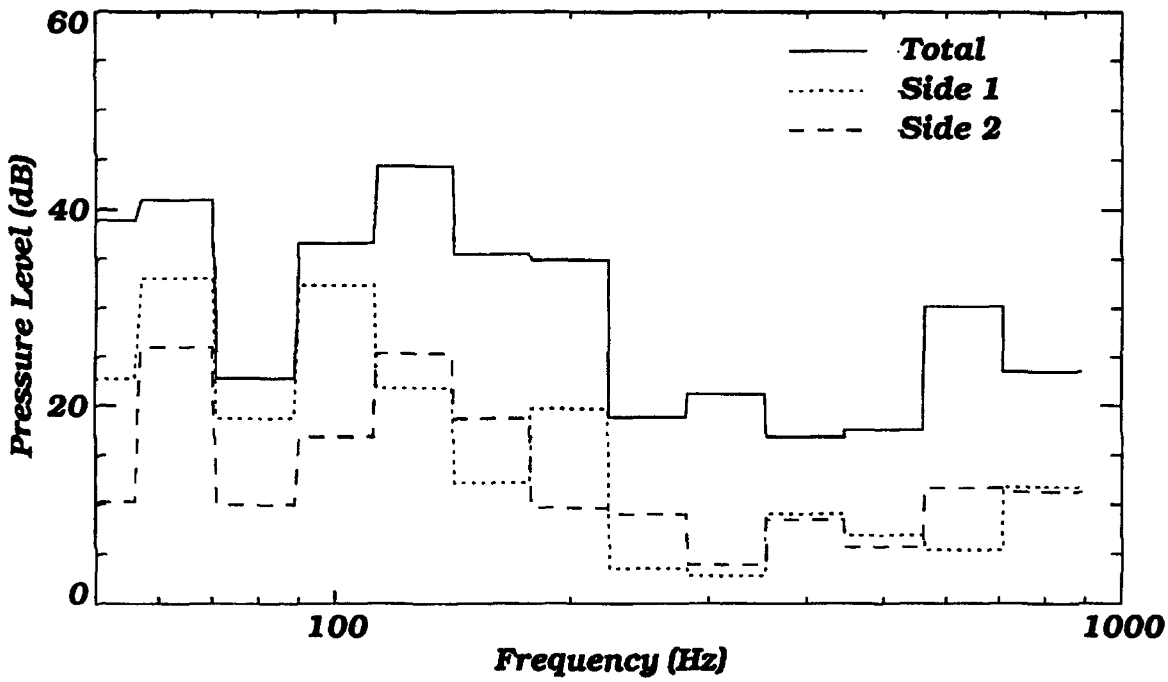


Figure 5 : *Measured and calculated (FAQP) pressure radiated in the far field (Point M2) of the circular saw.* Δ *is a 1/3 octave frequency band.*

(a)



(b)

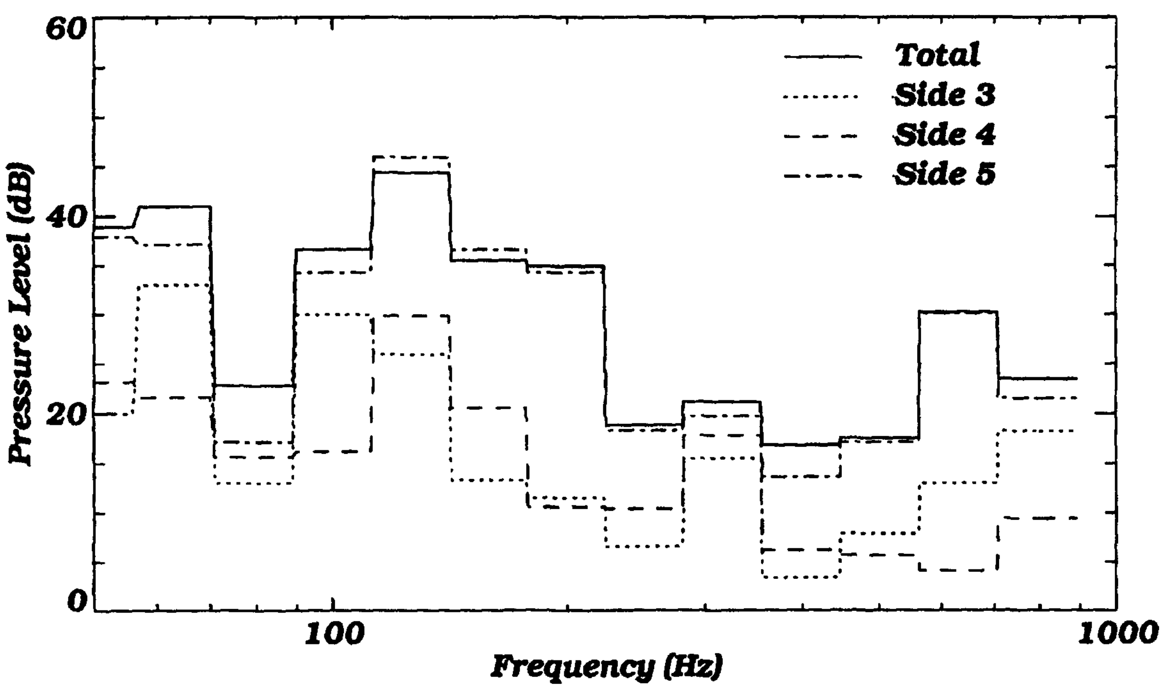


Figure 6 : *Acoustic pressure calculated by the FAQP method. Contribution of sides 1 and 2 (a) and sides 3, 4 and 5 (b) to the total acoustic pressure radiated (point M2).* Δ *is a 1/3 octave frequency band.*