# SCATTERING AND TRANSMISSION BY PLATE-LIKE BAFFLES

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## 1. INTRODUCTION

This paper considers the problem of modelling a plate-like baffle diffracting a plane wave. Experimental measurements of pressure near panels in this configuration can be used to determine material acoustic properties. Only very simple geometry and materials are analysed here, so as to permit comparison with currently available experimental results. However the technique used is applicable to situations of arbitrary geometry and material, thus allowing for example the design of panels made of viscoelastic materials and containing voids.

The baffle is considered to be elastic. Vibrations of elastic structures in vacuo can be predicted using the finite element method [1]. The fluid-structure interaction is modelled using a boundary element technique.

## 2. FLUID EQUATIONS

For small amplitude oscillations in an inviscid, irrotational, compressible fluid with no mean flow the pressure distribution satisfies the wave equation,

$$\nabla^2 p - \frac{1}{c^2}\frac{\partial^2 p}{\partial t^2} = 0, \tag{1}$$

where, $c$, is the acoustic wavespeed. For steady state vibrations at circular frequency, $\omega$, this reduces to the Helmholtz equation,

$$\nabla^2 p - k^2 p = 0, \tag{2}$$

where $k = \omega/c$ , is the wavenumber. The continuity of the normal velocity, $v$, on the interface between the fluid and the structural regions is enforced by the boundary condition,

$$\frac{\partial p}{\partial n} = -i\omega\rho v \, . \tag{3}$$

The total pressure field, $p$, can be decomposed into the sum of the incident pressure field, $p_i$, which would solely exist in the absence of the scattering object and the scattered pressure field, $p_s$, as,

$$p = p_i + p_s. \tag{4}$$

The scattered pressure field must satisfy the Sommerfeld radiation condition,

$$\lim_{r \to \infty} \left| \frac{\partial p_s}{\partial n} - ikp_s \right| = 0, \tag{5}$$

to ensure that it consists only of outgoing waves.

## 3. NUMERICAL DISCRETISATION

The PAFEC finite element analysis system was used to model both the fluid and the structure. The helmholtz equation was discretised using a direct formulation of the boundary element method as described in ref. [2]. The method of Burton and Miller [3], sometimes nicknamed CONDOR, was used to overcome the problem of characteristic frequencies. The baffle was idealised with a mesh of standard isoparametric 3D elements [4].

The combination of these two modelling techniques results in a set of linear equations,

$$\begin{bmatrix} \{ [S] - \omega^2 [M] \} & [T]^T \\ \omega^2 \rho [G] [E]^T & [H] \end{bmatrix} \begin{bmatrix} [u] \\ [p] \end{bmatrix} = \begin{bmatrix} [0] \\ [p_i] \end{bmatrix}, \tag{6}$$

to be solved, where, $[u]$, is a vector of displacements on the structural mesh, $[p]$, is a vector of pressures on the acoustic mesh and $[p_i]$ is a vector representing the incident pressures. $[S]$ and $[M]$ are structural stiffness and mass matrices respectively. $[H]$ and $[G]$ are complex frequency dependent boundary element matrices. $[T]$ and $[E]$ are coupling matrices. Further details of these are given in ref. [2].

## 4. TEST PROBLEM

A mild steel rectangular plate baffle in seawater scattering an incident plane wave was taken as a test problem for the above methods. The dimensions of the baffle were 0.9m x 0.9m x 0.03m. The plane wave was taken to be normally incident on the plate, see figure 1. The properties of steel were taken to be Young's modulus = $2.09 \times 10^{11}$ $Nm^{-2}$, Poisson's ratio = 0.3 and density = 7800 $kgm^{-3}$. The properties of seawater were taken to be density = 1040 $kgm^{-3}$ and wavespeed = 1487 $ms^{-1}$. In the experimental setup the panel was held in place using a small clamp at the top. The test were carried out in a large anechoic tank. The measured results [5], were available on a grid of 100 points at the centre of squares covering the area of the plate and 0.04m in front and behind. One quarter of these locations have been labelled as in figure 2, for the purposes of comparing the experimental and theoretical values. Pressures were measured and computed for three frequencies; 2000, 5000 and 10000 Hertz.

## 5. FINITE/BOUNDARY ELEMENT MODEL

Ignoring the effect of the clamp, the problem considered is symmetric and hence it is sufficient to model a quarter of the plate. the finite element mesh was a 20 x 20 x 1 block of 20-noded isoparametric elements and is shown in figure 3. The boundary element mesh consisted of 840 constant pressure patches coupled face to face with the finite element mesh. the effects of the planes of symmetry was included by using local restraints on the finite elements and using a symmetric Green's function in the boundary element method. The structural mesh contained 8803 degrees of freedom. In addition to calculating the scattering by a flexible plate a second computation was done to evaluate the scattering by a rigid baffle.

## 6. COMPARISON OF RESULTS

The experimental results have been scaled to agree with the theoretical values at point 1. Figures 4 and 5 show the comparison between the pressure amplitude computed by the theory and experiment at 2000 Hz on the transmitted and reflected sides of the baffle. Figures 5 and 6 give the same comparisons at 5000 hz and figures 8 and 9 are for 10000 Hz. Figures 10 and 11 show the comparison between the flexible and rigid cases. Figures 12 and 13 show the deformed shape of the baffle at 2000 Hz and 10000 Hz.

## 7. DISCUSSION OF RESULTS

The experimental and predicted results clearly show the same trends. The level of agreement is always better for the reflected side pressures than for the transmitted side pressures. This may be due to the effect of the clamp causing modification to the structural vibration, which has a greater influence on the transmitted field, as can be seen from looking at the comparison of the flexible and rigid calculations. The degree of accord between the calculated and measured pressure fields deteriorates at the highest frequencies. This may be due to the physical presence of the clamp, the size of which is of the order of a wavelength at 10000 Hz.

Generating a plane wave within the test tank is a difficult experimental task. The level of agreement with the theoretical results suggests that this has been achieved to a high degree of accuracy.

## 8. CONCLUSIONS

The finite/boundary element method can be used to predict the reflection and transmission characteristics of a baffle. To model accurately a particular experimental configuration, it seems necessary to take account of any restraints holding the panel in place.

## 9. FURTHER WORK

The work described in this paper is continuing. Oblique angles of incidence are being analysed for the same steel baffle. This is a more demanding task computationally, requiring greater computer time and memory. Results are also available for more complex baffle constructed of PVC tubes embedded in polyurethane. These will be discussed in due course.

## 10. REFERENCES

[1] Zienkiewicz O.C. "The Finite Element Method", McGraw Hill (1979)

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[3] Burton A.J. and Miller G.F. "The Application of Integral Equation Methods to the Numerical Solution of some Exterior Boundary Value Problems", Proc. Roy. Soc. London A323 (1971) pp 202-210.

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Fig 1

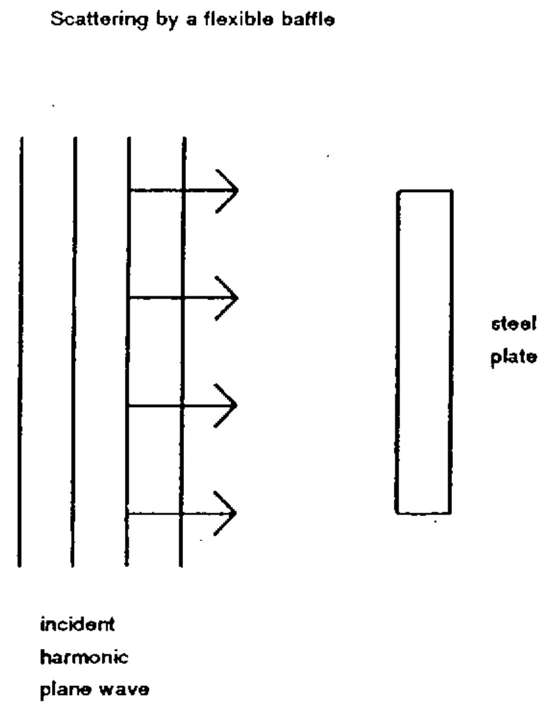


Fig 2

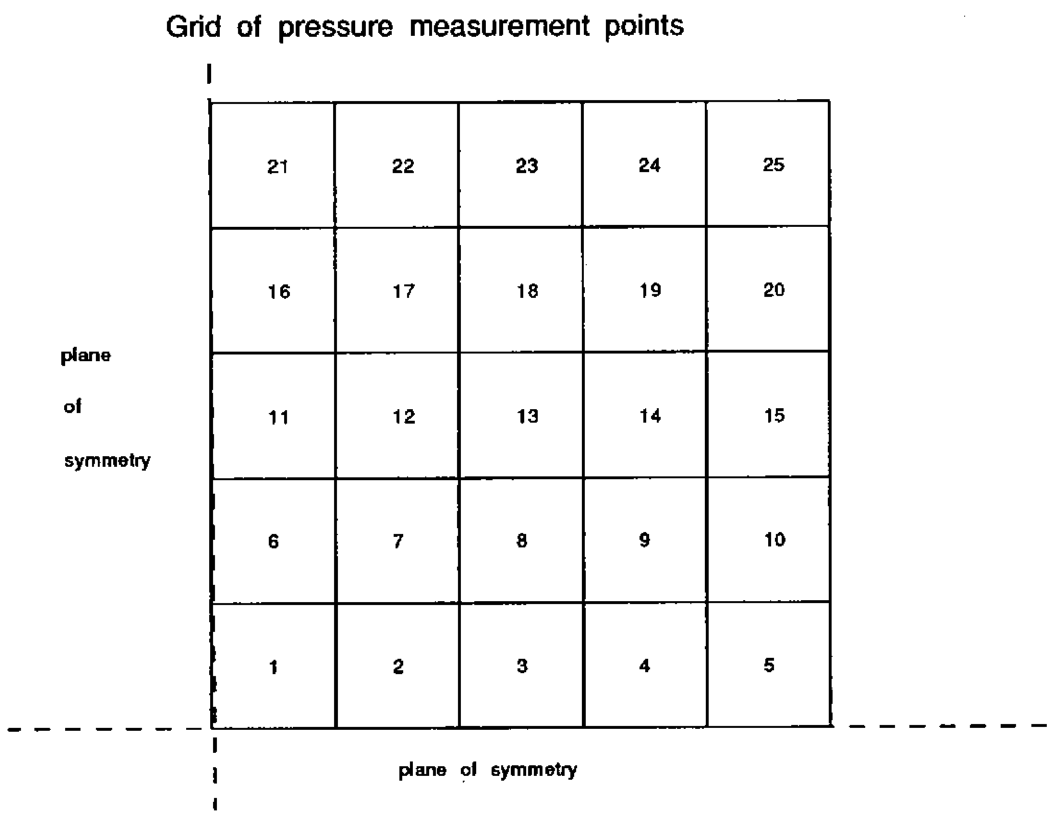

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Fig 3

Finite element mesh

used to model baffle

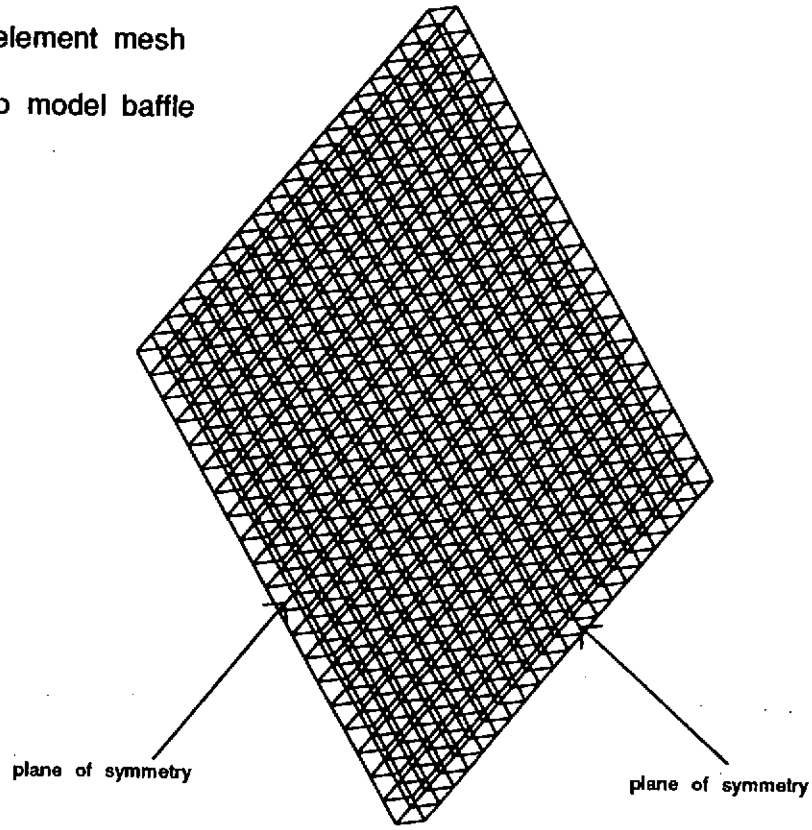


Fig 4

Comparison of measured and predicted pressures
on the tranmitted side of the steel baffle
at 2000 Hz

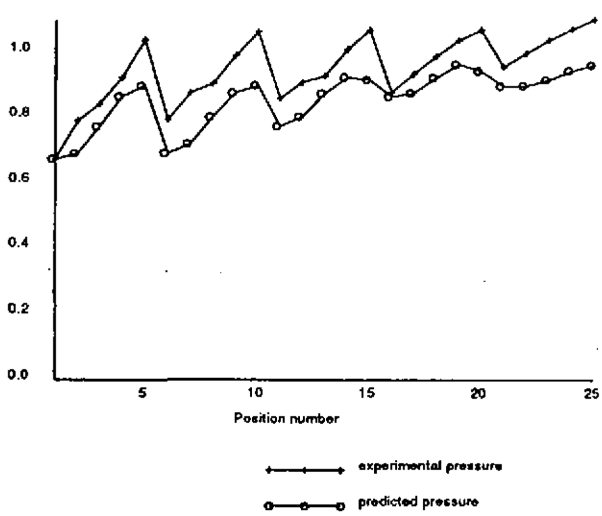


Fig 5

Comparison of measured and predicted pressures
on the reflected side of the steel baffle
at 2000 Hz

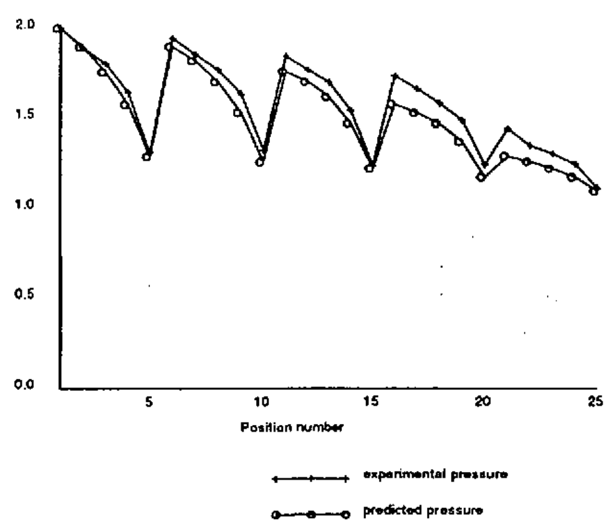

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Fig 6

Comparison of measured and predicted pressures
on the transmitted side of the steel baffle
at 5000 Hz

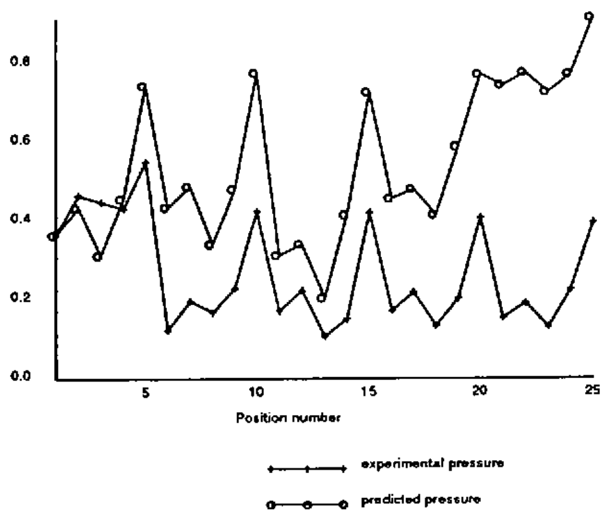


Fig 7

Comparison of measured and predicted pressures
on the reflected side of the steel baffle
at 5000 Hz

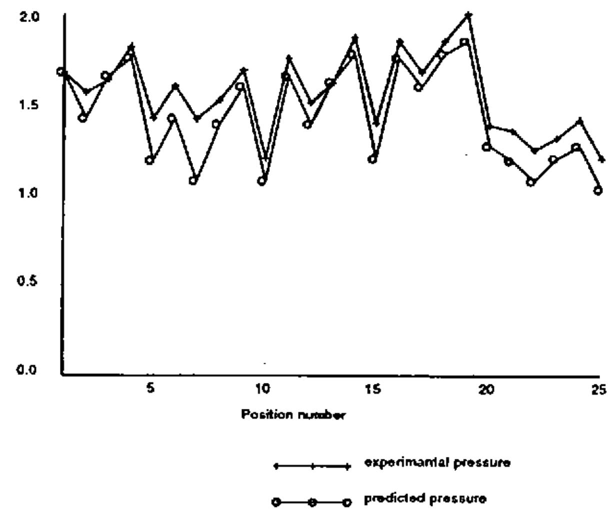


Fig 8

Comparison of measured and predicted pressures
on the transmitted side of the steel baffle
at 10000 Hz

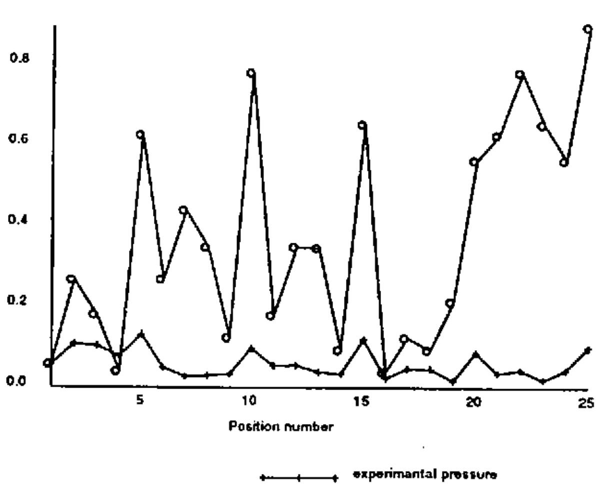


Fig 9

Comparison of measured and predicted pressure
on the reflected side of the steel baffle
at 10000 Hz

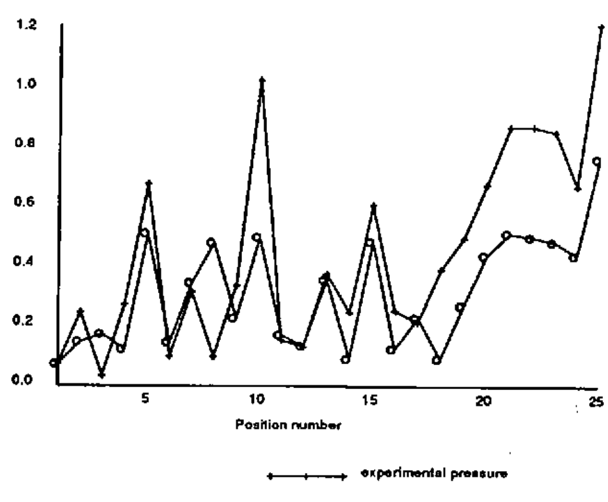

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Fig 10

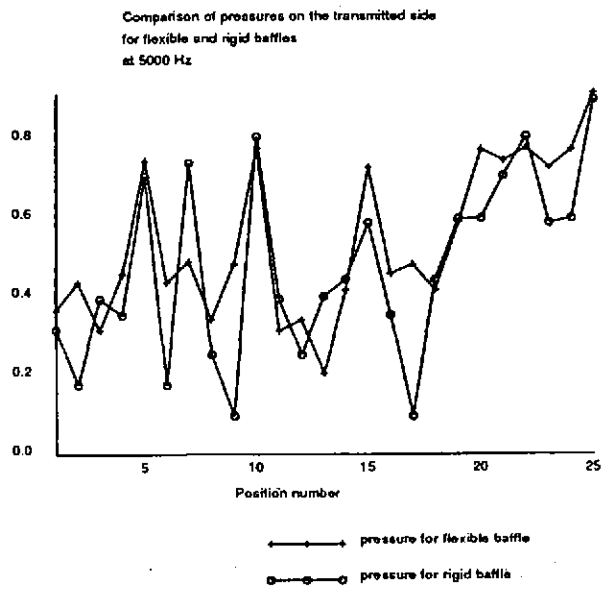


Fig 11

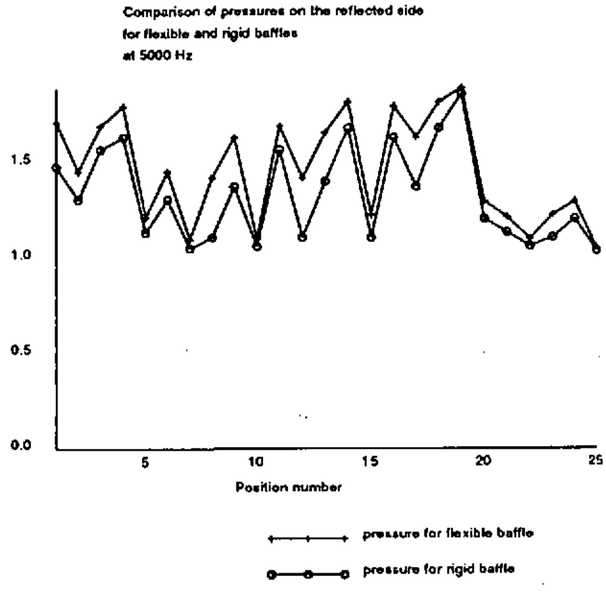

Fig 12

*Deformed baffle at 2000 Hertz*

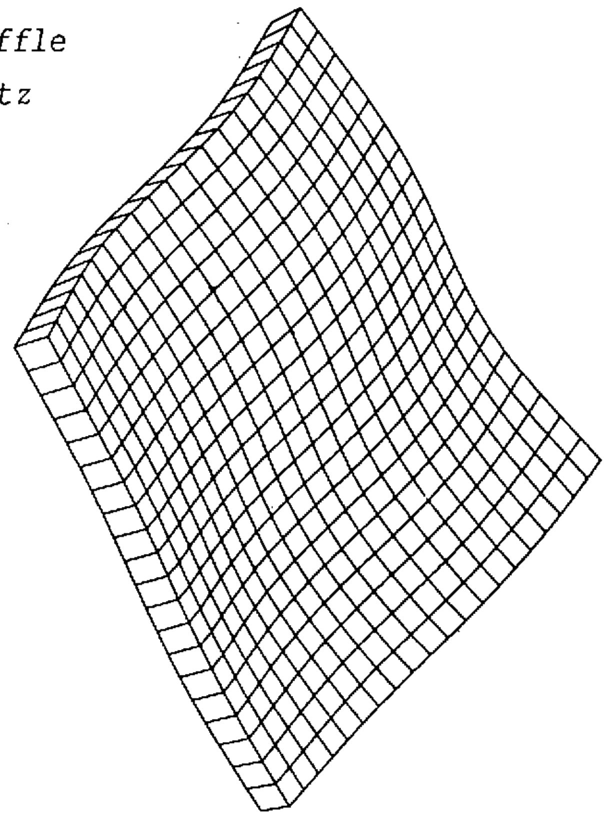

Fig 13

*Deformed baffle at 10000 Hertz*