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THE PRACTICAL REALISATION OF A TRANSDUCER WITH AN OCTAVE BANDWIDTH

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1.0 Introduction

This paper describes further progress in the development of the transducer elements for a multi-element array which has the form of a spherical cap and having a constant beamwidth over a wide range of frequencies. A major requirement is for the minimum variation in the sensitivities of the array on transmission and reception over the frequency range 27 to 54 kHz; the individual elements must therefore each have a uniform response over this range. The element is based on the traditional "Tonpilz" design, comprising a head mass radiating into water, a compliant stack of piezo-electric rings and a tail mass acting as a counter-mass, with more sections added to this basic structure, a mechanical section between the head mass and the radiation load and an electrical filter section at the electrical terminals of the ceramic stack. The original concept was described many years ago (1), and a similar design has been described recently (2) in which one or two quarter-wave coupling sections are used between the basic head and the load. The present arrangement uses on the mechanical side a "lumped element" equivalent of the latter arrangement. The theoretical design has been described in detail previously (3,4), only a summary being presented here, and this paper concentrates on the problems arising in turning theory into practice.

2.0 Outline of the theoretical design

This was based on analysis of a 'lumped' equivalent circuit for a single element as shown in fig.1 and consideration of the variation of the input admittance, the element being loaded by the average radiation load on each element in a closely packed array (the area to be actually occupied by active pistons is 85% of the total face area of the array). This complex load is modelled by third order polynomial functions of frequency for the real and imaginary components. The analysis is carried out using ordinary electrical network theory, but with mechanical analogues for the electrical components, i.e. inductance replaced by mass and capacitance by compliance; the electrical components to be used with the transducer, i.e. the clamped capacitance and the added tuning components are modelled by their mechanical equivalents on the mechanical side of the electromechanical transformer so that this can be omitted from the analysis. The aim is for the smallest variation in conductance over the bandwidth of 27 to 54 kHz, so that on transmission from a constant voltage source the radiated acoustic power is most nearly constant. The susceptance should be kept low compared with the conductance, to reduce the reactive power demanded from the power amplifier, and this aim can be partially achieved by the addition of a parallel

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tuned circuit at the input. The design started from a conventional element, designed for a modest bandwidth and assuming that the effective coupling coefficient of the piezo-electric stack would be significantly less than that of the ceramic alone. The diameter of the head was chosen to be around three quarters of a wavelength at the highest frequency, this being the largest size which it was thought would be free of major problems with 'flapping' mode resonances. The parameters indicated in fig.1 which were varied in the search for optimum performance were the outer and inner head masses Mr and Mh, the tail-to-head mass ratio Mt/Mh and the coupling coefficient represented by the ratio C1/Cm. The remaining components were determined by various internal resonances, including series tuning with the equivalent mass Ms (in practice of course this is realised by a series inductance on the electrical side). The head and tail masses of the basic element were initially kept to a minimum with the aim of maximising the bandwidth without the coupling section, but the minimum mass is dictated by practical limitations on the dimensions of the piezo-electric stack. Usually the tail/head mass ratio is much larger than unity, but in this design a low ratio of 1.2 to 1.3 gives better results. The assumed ratio of the clamped to motional capacitances, to which the effective coupling coefficient is related, is 4.0, but large changes, e.g. from 3.3 to 5.0, have only small effects on the bandwidth. The design of the coupling section has the greatest effect on the overall performance, effecting both the bandwidth (to a small degree) and the "peakiness" of the response (to a large degree). The mechanical component values finally selected are shown in table 1. The calculated admittance, referred to the mechanical side of the electro-mechanical transformer and for the array radiation load, is shown in fig.2 as a plot of susceptance vs. conductance with frequency as a parameter; it includes the parallel tuned circuit for correction of the susceptance, optimised for the minimum phase angle over the working bandwidth. The design has the required octave bandwidth between the frequencies at which the conductance falls to half the outer peak values, and the variation in conductance within the passband is within a range of 2:1, equivalent to a 3 dB. change in sensitivity. However for a single isolated element the radiation load is completely different from the average on each element in the full array, and this point is discussed in more detail below. Therefore it will be difficult to prove the design without building a number of elements and testing them in an array representing at least part of the final array (the original design has about 300 elements).

3.0 Effect of varying radiation loads

The element has been designed for a particular radiation load, that valid for the intended use in a multi-element array, but initial testing and development would be done on a single isolated element. Therefore the mechanical admittance has also been calculated for a single element in an infinite rigid baffle, and this is shown in fig.3 for comparison with that with the array radiation load shown in fig.2. The significant point to be noted is that the response is concentrated closely round two very narrow band peaks, at 31.13 kHz with a Q of 48.8 and at 50.17 kHz with a Q of 219; these frequencies are close to the unloaded resonances at 31.998 and 53.698 kHz. Calculations have also been done

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for an element loaded with a purely resistive load having a magnitude equal to the head area multiplied by the acoustic impedance of water, the results, shown in fig.4, indicate a double peaked response with nearly equal peaks at 31.7 and 51.3 kHz and a minimum at 43.2 kHz having a loss of about 4.4 dB. relative to the average peak. The conclusion to be drawn from these calculations is that the performance is critically dependent on the radiation load actually having the characteristics assumed for the design. This is perhaps only to be expected by analogy with electrical filter design, in which critical designs, e.g with small pass-band ripple and fast transition between pass and stop bands, also have narrow constraints on terminating impedances.

4.0 Mechanical Constraints

There are two important features to be considered for a practical design. One is the desirability of having a prestressing bolt along the centre of the element and the other is the question of sealing the element in its housing so that the water is kept out. The bolt is to provide a mechanical bias on the bonds between the various parts so that no nett tensile stresses occur during normal operation, but it should be decoupled as far as possible from the head and the tail so that neither additional stiffness nor mass is added to the ceramic stack. This can be achieved by the use of disc spring ('Belleville') washers at the ends of the bolt, but owing to the small size of this element it is not possible to include the outer head and coupling section within the prestressing; it remains to be seen whether the bonding of these parts can be made sufficiently robust and reliable. There are two possible ways of sealing the element into the housing, either by an 'O' ring round the head or by a bonded layer of rubber or a similar material covering the heads and the face of the housing. The latter method, apart from being semi-permanent, has the disadvantage of adding another component to the equivalent circuit because of its stiffness in shear, and also its acoustic properties, not being matched perfectly to those of water, would need properly to be taken into account in the design; on the other hand it could perhaps be used as a quarter-wave matching layer if a material with suitable properties is available (this type of design has not been investigated in detail). The difficulty with the 'O' ring is that in the usual design the ring is located within the thickness of the head mass, but in this element both heads are too thin. However it is possible to use the space between them by making the outer diameter of the coupling stub equal to the inner diameter of the 'O' ring groove, as long as the length of the stub is sufficient.

5.0 Practical realisation within constraints

The experimental assembly is shown in cross-section in fig.4. The ceramic stack was designed on the basis of an even number of 3 mm. thick rings of PZT-4 having inner and outer diameters such that the calculated compliance of this stack was slightly less than the original design figure in order to make some allowance for the compliance of the bonds between the rings, including the metal foil electrical contacts; six rings, each 3 mm. thick and with outer and inner diameters of 8.0 and 4.0 mm., were used. For reasons dicussed later it would be better to have rings of a larger diameter, but then for the same stiffness and length the annular width would have been uncomfortably narrow, e.g. less than 1 mm. Furthermore the length of the stack could not be increased (it is already nearly a quarter of a wavelength at the top frequency of 54 kHz), since its weight must be minimised; as it stands its weight is 5.6 gm., and corrections of half this have to be subtracted from the design values of the head and tail masses. Conversely the heads, which are made of aluminium as having favourable stiffness in relation to density, have a significant compliance, for which it is more difficult to estimate corrections, and the principal effect is that the heads bend instead of remaining flat like ideal rigid pistons. This bending can be minimised if the head is driven over an annulus which is concentrated around two thirds of the outer diameter (5); with the constraints on the PZT rings and on the coupling section, this optimum arrangement unfortunately cannot be achieved here. The compliant stub between the heads is made of cast epoxy resin, having low stiffness and low density.

6.0 Finite Element Analysis

This was used to confirm the practical mechanical design in terms of the resonant frequencies of the unloaded structure and to investigate the bending mode vibrations of the inner and outer heads. A purely mechanical approach was used without piezo-electric coupling but using the short-circuit parameters for the piezo-ceramic used (PZT-4), on the basis that with a constant-voltage drive to a divided stack the electrical terminating impedance is equivalent to a short-circuit. The resonances were excited by equal and opposite sinusoidal forces applied to the opposite ends of the stack. In order to make some allowance for the increase in compliance due to the bonds and contacts between the rings in the stack without detailed modelling the length of the stack was increased by about 0.25 mm. for each complete bond, 1.8 mm. overall (this approximate correction has been used successfully in the past). The extent of the head bending is indicated in fig.5, which shows the relative nodal displacements for a frequency close to the upper resonance; also shown is the extent of the radial strain in the coupling stub, which may be contributing to the bending. Theoretical investigations are continuing into the possible advantages of using a composite glassfibre/epoxy material for the stub, for which the circumferential stiffness would be substantially increased while retaining low longitudinal stiffness, in the hopes that this would lead to a reduction in the bending of the heads.

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7.0 Results for the practical realisation

At the time of writing only one element has been assembled and tested in the unmounted state and without the intended electrical networks. The resonant fequency of the basic element (without the coupling section) was 38.131 kHz compared with the design target of 36.224 kHz and the Finite Element estimate of 36.445 kHz, an error of 5% only, and the ratio of clamped to motional capacitance was 3.5 compared with a target value of 4.0; this ratio however can always be increased by the addition of a parallel capacitor, but it is not easy to estimate its value accurately since the clamped capacitance for the unloaded transducer is measured by a relatively small difference between two large values. For the complete element the resonances were at 27.560 and 53.437 kHz, compared with the design predictions of 31.998 and 53.698 kHz and the FE etimates of 27.641 and 48.459 kHz. Granted the uncertainties in material properties and the effects of the bonds, together with dimensional tolerances, there is reasonably good agreement between theory and practice for a first attempt.

8.0 References

(1) G.C. Rodrigo, Analysis and design of piezo-electric sonar transducers. Ph.D. thesis, London, 1970 (unpublished).

(2) M. Van Crombrugge, W. Thompson, Optimisation of the transmitting characteristics of a Tonpilz-type transducer by proper choice of impedance matching layers. J.A.S.A. 77, 747-752, 1985.

(3) J.R. Dunn, B.V. Smith, Problems in the realisation of transducers with octave bandwidths. Proc. Inst. Acoust. vol.9 pt. 2, 58-69, 1987.

(4) J.R. Dunn, Underwater transducers with an octave bandwidth. Proc. 13th. Internat. Congress on Acoustics, Yugoslavia 1989, vol. 4, 473-476.

(5) J.R. Dunn, Some aspects of transducer design by finite element analysis, in Progress in underwater acoustics, ed. H.M. Merklinger, Plenum Press, New York, 1987, 639-646.

Table 1 : Component Values

Outer head mass	Mr	1.6183 gm
Coupling compliance	Cr	8.0504E-9 m/N
Inner head mass	Mh	4.8 gm
Tail mass	Mt	5.4 gm
Stack compliance	Cm	7.5964E-9 m/N
Clamped compliance	C1	3.03856E-8 m/N
Series tuning mass	Ms	0.4900 gm
Shunt " "	Mp	0.2240 gm
" compliance	Cp	8.343E-8 m/N

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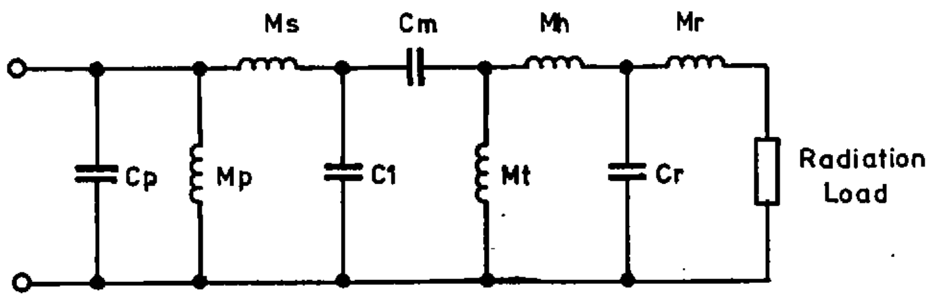

Fig.1 Mechanical equivalent circuit for analysis

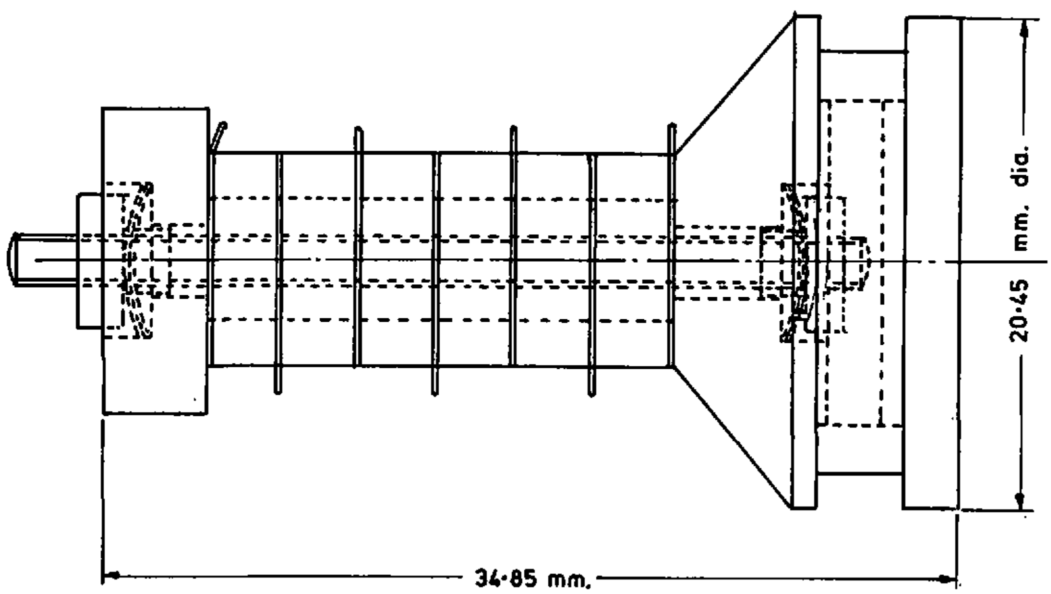

Fig.4 Mechanical assembly

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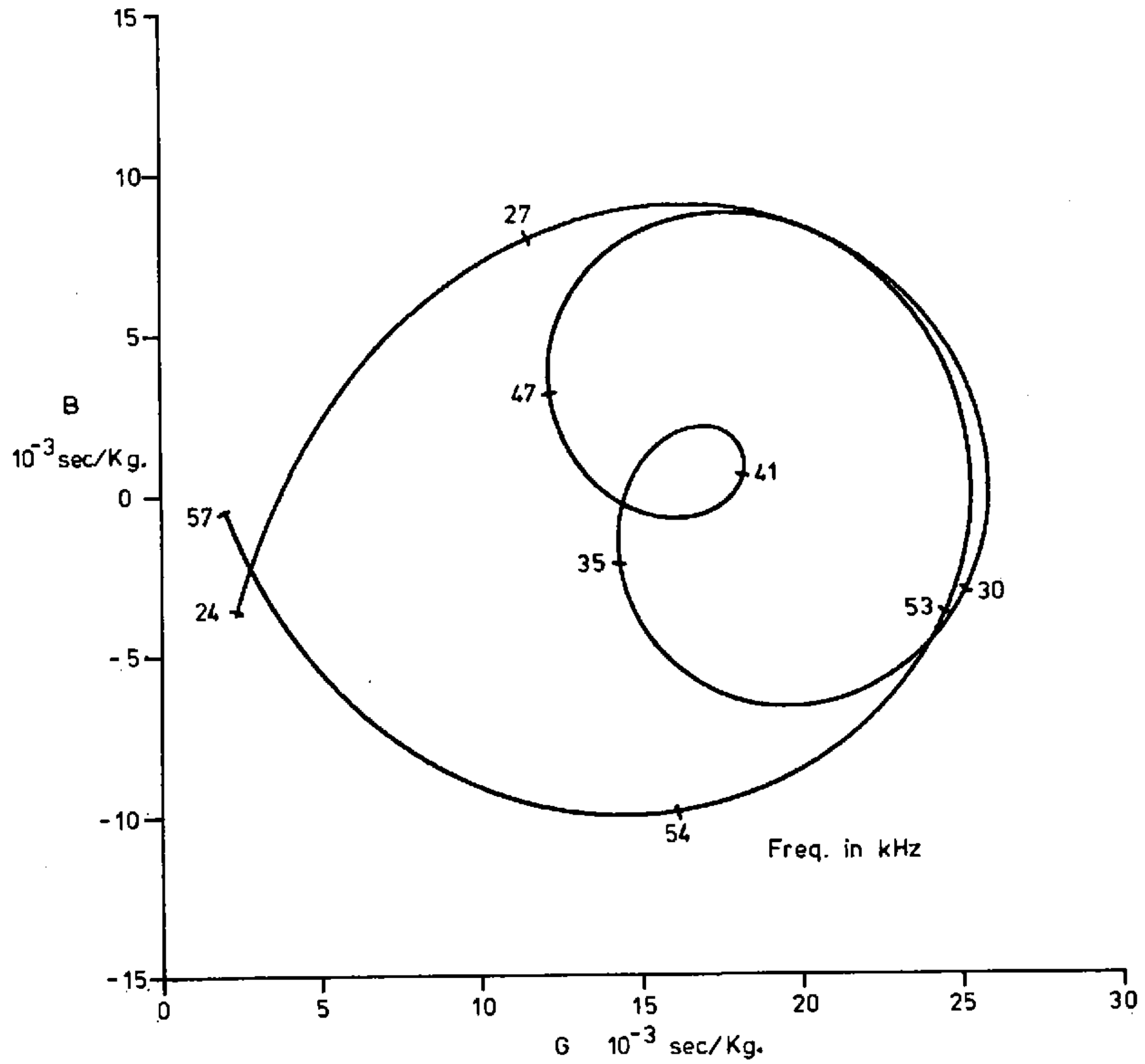

Fig.2 Mechanical admittance with array radiation load

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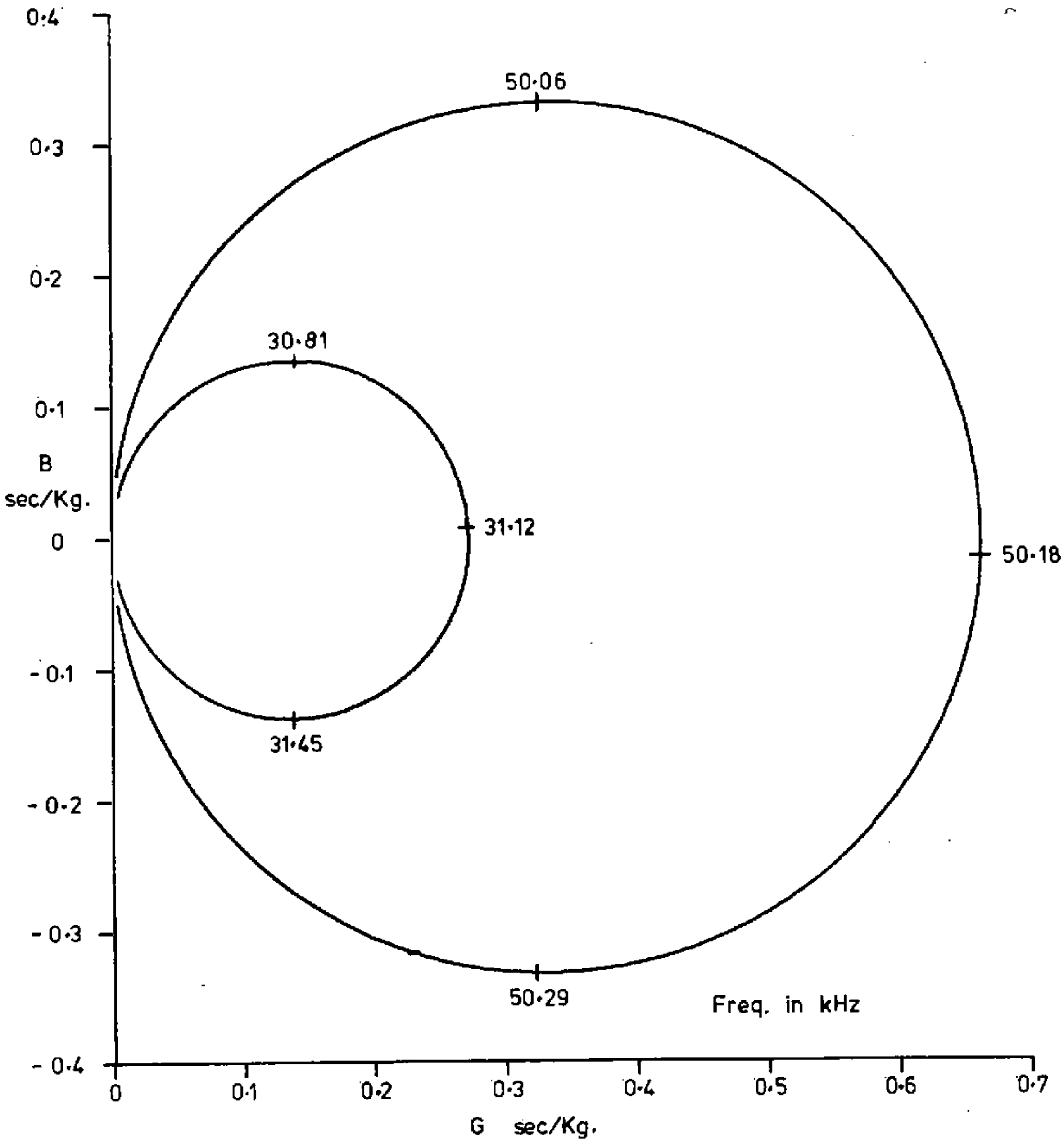

Fig.3 Mechanical admittance with isolated head load

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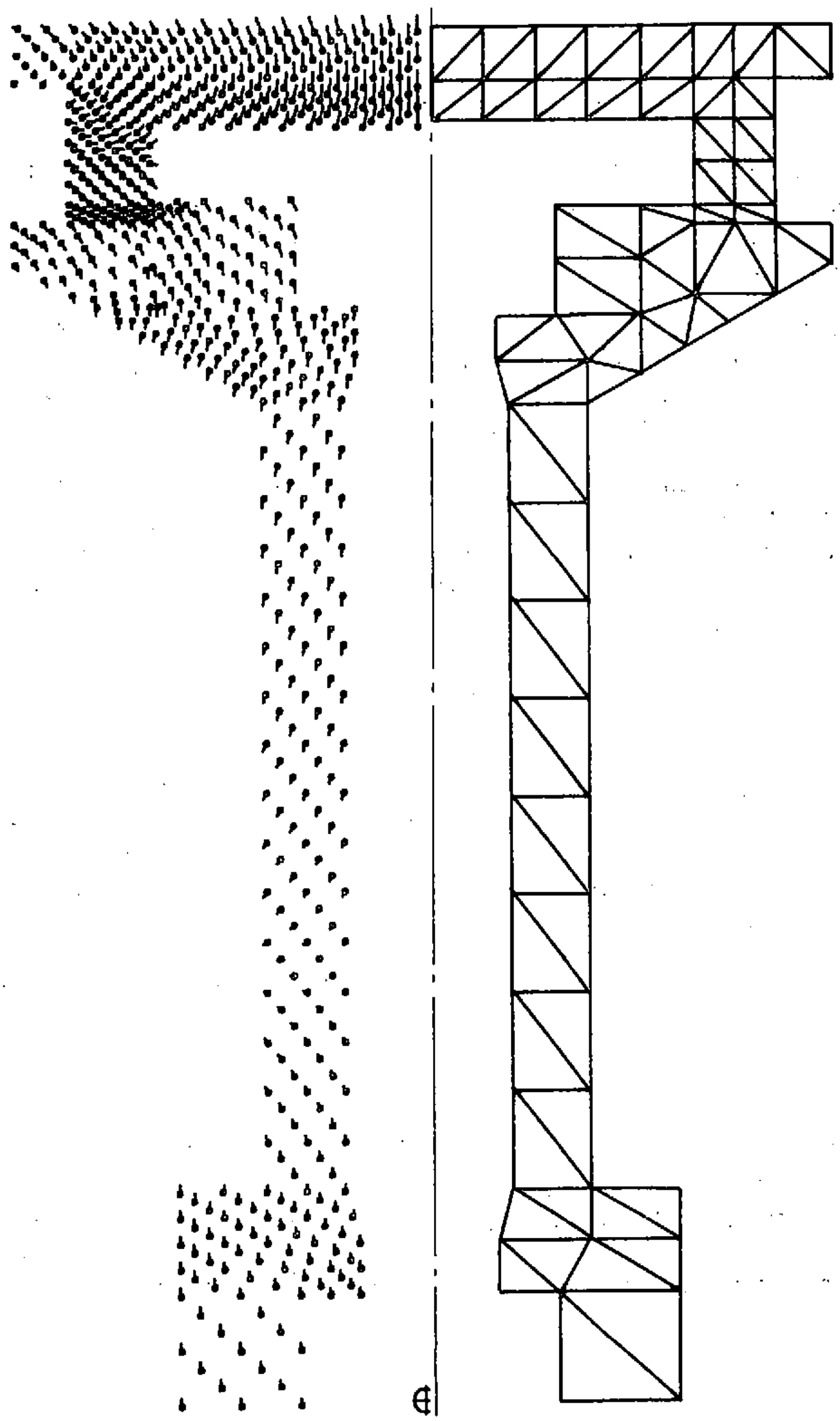

Fig.5 Finite Element Analysis :
Mesh of 6 noded triangular elements and scaled displacements at nodes and within elements close to upper resonant frequency