# FLEXURAL MODE ACOUSTIC TRANSDUCERS

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## Introduction

This investigation was carried out because a need had arisen for a transducer, to work in the frequency range 5 to 50 kHz, which would be as small and light as possible for a given face area.

## General principles

A basic fact about transmitting transducers is that, to obtain any worhhwhile efficiency of operation it is generally necessary to use some kind of mechanical resonance. The purpose of this is to provide magnification of the motion from the driving element (e.g. a lead-zirconate plate) to the active face of the transducer. Resonance is used to match the high acoustic impedance of the PZT to the relatively low impedance presented to the moving face by the fluid medium. The need for this is greatest when transducers are required to have face dimensions that are small compared with the acoustic wavelength in a liquid or when the transducer is required for use in air as the medium.

Resonance involves the alternating interchange of energy between two forms of energy storage. In a mechanical system these are, respectively, kinetic energy stored in moving masses and strain energy stored in the elastic deformation of springs. In an electrical system the respective energy storages take place in magnetic and electric fields. As far as the mathematical analysis of resonant systems is concerned it makes very little difference which kind of energy storages are involved; the differential equations are the same and the solutions are of the same general form, It is for this reason that electrical equivalent circuits can be used to demonstrate the bahaviour of emchanical systems. The reverse is also true of course and the choice of midel is simply one of personal preference. The only justification for introducing electrical equivalent circuits into a discussion of mechanical systems is that these are nowadays sufficient;y familiar to be a useful aid to the visualization of what is really the solution of sets of differential equations. The only point to watch is that, in transducer design, we are dealing with electro-mechanical systems which involve both mechanical and electrical energy storages. So it is necessary to distinguish between true circuits elements carrying real currents and voltages, and notional ones which represent mechanical effects. A logical procedure is to identify kinetic energy with energy stored in a magnetic field and strain energy with that in an electric field. Inductance then represents mass, with velocity as the analogue of electric current, while capacitance represents compliance (defined as linear displacement per unit force) and force is equivalent to electrical voltage. All the usual terminology of electric circuit theory can then be translated directly into mechanical terms, and vice-versa. For example,circuit impedance appears as the ratio of force to velocity and the frequency of resonance of a mass-spring mechanical circuit is inversely proportional to the square root of the product of mass and compliance.

In the 'sandwich' type of transducer (see Fig, 1) the strain energy storage is provided mainly by direct tension and compression stresses. For an exact analysis of such a system the various parts have to be treated as mechanical transmission lines, each having distributed compliance and mass. In this case however, where one of the objectives is to keep the overall dimensions of the device as small as possible, the dimensions of component parts are likely to be small compared with the wavelength of waves of mechanical deformation within them, so that it is sufficiently accurate to trate the energy storages as if they were provided by lumped masses and springs, as indicated diagrammatically in Fig. 3 (a). Fig. 2 shows a simplified electrical analogue circuit, assuming that the mass of the PZT plates is negligibly small.

Losses at interfaces

One of the problems in the design of a 'sandwich' transducer is that of providing the mechanical coupling between the driving element and the rest of the structure. A common method is to use an adhesive such as epoxy resin for this purpose, with a central clamping bolt which applies a static compressive stress exceeding the peak alternating stress which can occur in use. The idea is to ensure that the bond is never subjected to tension. Each adhesive layer has a mechanical compliance and these appear in the equivalent circuit as small shunt capacitances. Moreover, the mechanical deformation in the bond is not purely elastic; it is generally accompanied by energy losses which have to be represented by leakage resistance, as shown in Fig. 2. Unfortunately this additional leakage occurs at a point in the circuit where the impedance is high. In mechanical terms, at resonance the combination of the mass and spring on each side of the driving element presents a high mechanical impedance; i.e. there are large forces and small displacements. So a very small amount of 'lost motion' or 'mechanical leakage' in an adhesive layer can lead to large losses.

An interesting experiment is to use an epoxy resin bond as the mechanical coupling agent and to monitor the overall efficiency of the transducer as the bond gradually sets. The usual result is that there is a gradual reduction in the total loss as the hardening process takes place. A likely explanation is that the main stress transmitted through the interface is accompanied by 'Poisson Ratio' effects which, for dissimilar materials, will tend to cause a certain amount of lateral motion of one surface relative to the other. While the coupling medium is still liquid this sliding motion can occur, causing frictional losses. The hardening resin grips the surfaces and is itself stressed as they try to move. In other words the friction loss is replaced, to some extent, by elastic energy storage in the solid bond. This theory is borne out by the fact that hardening of the bond is also accompanied by a slight fall in the resonant frequency, brought about presumably by the additional energy storage in the resin.

This conclusion carries with it the disturbing implication that the bond is subjected to alternating shear stresses, in addition to the expected main compressional stress. Shear stresses in materials like epoxy resins are a potential source of deterioration and eventual failure of the transducer. So, on grounds of reliability, there is a good case for avoiding solid bonds and for using a substance such as silicone fluid as the mechanical coupling medium, even if this leads to some loss of efficiency.

Flexural-mode transducers

In a 'sandwich' transducer the resonant frequency is given approximately by $\omega_0$ where

$$\omega_0^2 (C_1+C_2)\left(\frac{L_1 L_2}{L_1+L_2}\right) = 1$$

i.e. when the total mechanical compliance resonates with the 'parallel' combination of the two masses. To obtain a low resonant frequency the product of mass and compliance has to be large, so if the total mass is to be kept small, it follows that the compliance must be large. The trouble with the 'sandwich' arrangement is that the compression springs are too stiff when practical materials are used.

Attention was therefore turned to the alternative 'flexural-mode' or 'bender' arrangement in which the mechanical springs are formed by the bending of metal parts instead of direct compression (see Fig. 3 (b)). It was hoped that this would enable 'softer' springs to be produced, thus permitting low resonant frequencies to be attained with relatively small masses.

The simplest 'bender' arrangement is the bimorph PZT plate, which bends bodily so that the elastic energy storage is formed by deformation of the plate itself and the mass is provided by its own distributed mass. A difficulty is that of driving a flat piston face from the flexural motion of the driving plate. Another type of 'bender' is the 'top-hat' form described by Craster (1). The idea is to form a rigid bond between a PZT disc and a metal disc and to pole the PZT in its thickness mode. The 'Poisson Ratio' forces mentioned earlier tend to deform the discs into a 'dome' shape. The composite disc is supported at its edges and it executes 'dome-shaped' vibrations, with the direction of curvature alternating in sign. One objection is that, although a kind of piston motion is achieved, the velocity distribution is far from uniform. It is a maximum at the centre and falls off, actually reversing in sign before the edges are reached. A more serious objection is that the mechanical coupling relies on the existence of shear stresses in a solid bond.

So the author decided to seek a new design which would incorporate the flexural-mode spring idea, while retaining the clamping bolt and enabling a fluid coupling medium to be used. The result is shown in Fig. 4. The mechanical resonant circuit now involves a 'flywheel' shape in which the rim acts as the mass and the spring is formed by the web. The purpose of the central boss is to ensure that the motion in the region of the interface is almost entirely axial so that shear effects are avoided. Experimental models, using different metals and varying proportions of the 'flywheel' shape gave encouraging results. A practical drawback of this arrangement is that the moving part takes the form of a rim, rather than the required flat piston face, so that an additional disc would have to be attached to the rim for this purpose. This meant that an additional interface was going to be needed anyway and it was realized that it might be better to return to the conventional 'sandwich' and to incorporate the 'flywheel' idea in the form of a separate additional spring element. This led finally to the arrangement shown in Fig. 5. It might appear that the effect of the softer spring might be masked by the stiffness of the clamping bolt but calculation showed that this could be avoided by suitable choice of dimensions and materials. The unit shown in Fig. 5 used aluminium for the head and the spring element, with brass for the tail mass. Its resonant frequency was 10 kHz.

## Non-linear distortion

A point that may be worth mentioning is that, at the time this work was done, the author was concerned with the use of acoustic non-linear effects as a method of detecting small gas-bubbles in liquids. For this reason there was great interest in obtaining transducers with exceptionally small amounts of intrinsic non-linearity. It was established experimentally that the main source of such distortion, in a'sandwich' type transducer, is in the interface layers. Many epoxy resin bonds were testing with this in mind and it was found that the measurement of second-harmonic distortion provided a very sensitive indication of the state of the bond. The point of general interest here is that this provides a method of detecting the deterioration, caused presumably by shear-stress fatigue, which can sometimes occur; and this can be recognized long before any other symptoms of incipient failure are noticed. Generally the use of a fluid interface layer, instead of a rigid bond, produces considerably higher harmonic distortion. In most other applications, apart from the rather special bubble-detection one, this is unlikely to matter however.

## Reference

1. W. F. CRASTER (1971) Paper presented at B.A.S. Spring Meeting, Design of bender trnaducers and their relevance to lightweight design.

## Acknowledgement

Information contained in this paper was first published in Proc. Ultrasonics International Conference, London, 1975. The diagrams are reproduced from that publication.

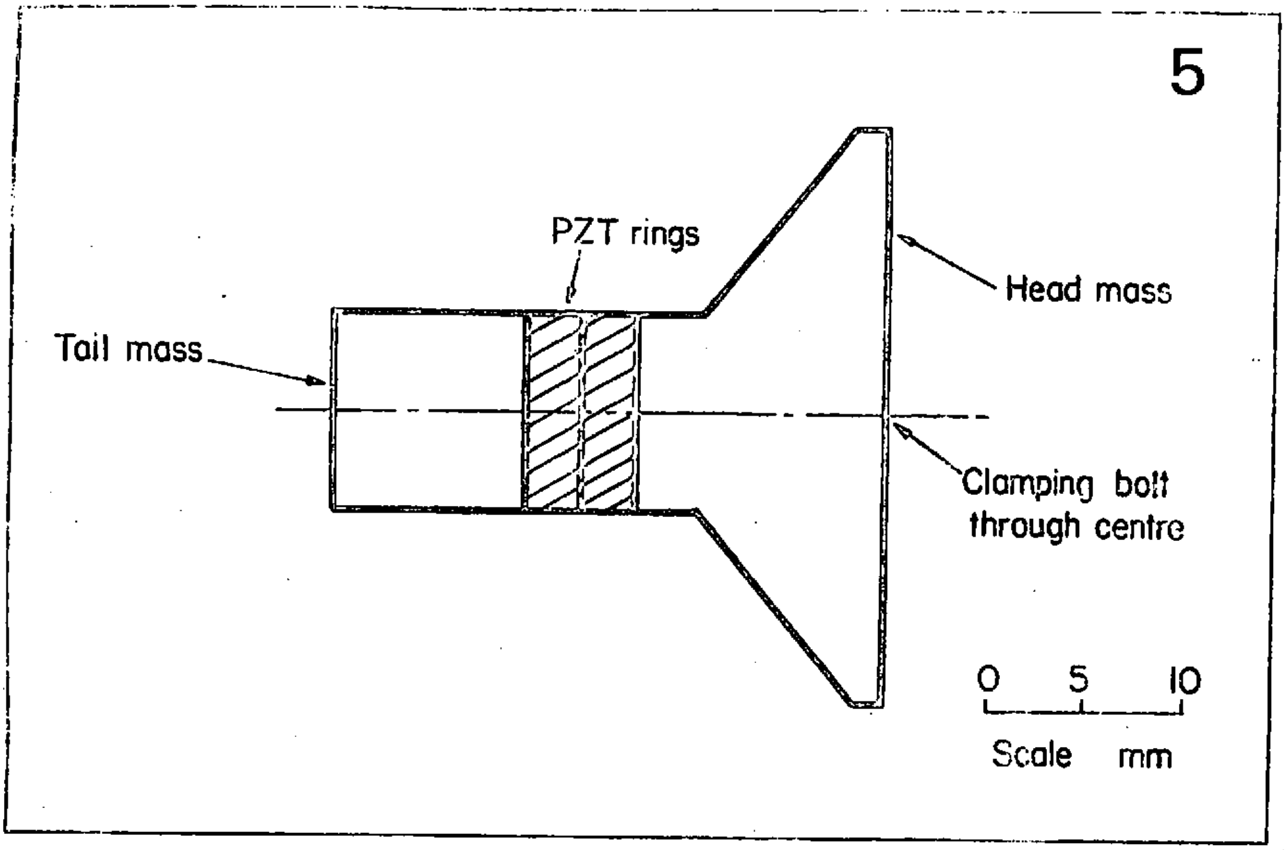


Fig. 1 'Sandwich' type transducer (general configuration, drawn to scale). Aluminium head; brass tail; frequency 25 kHz.

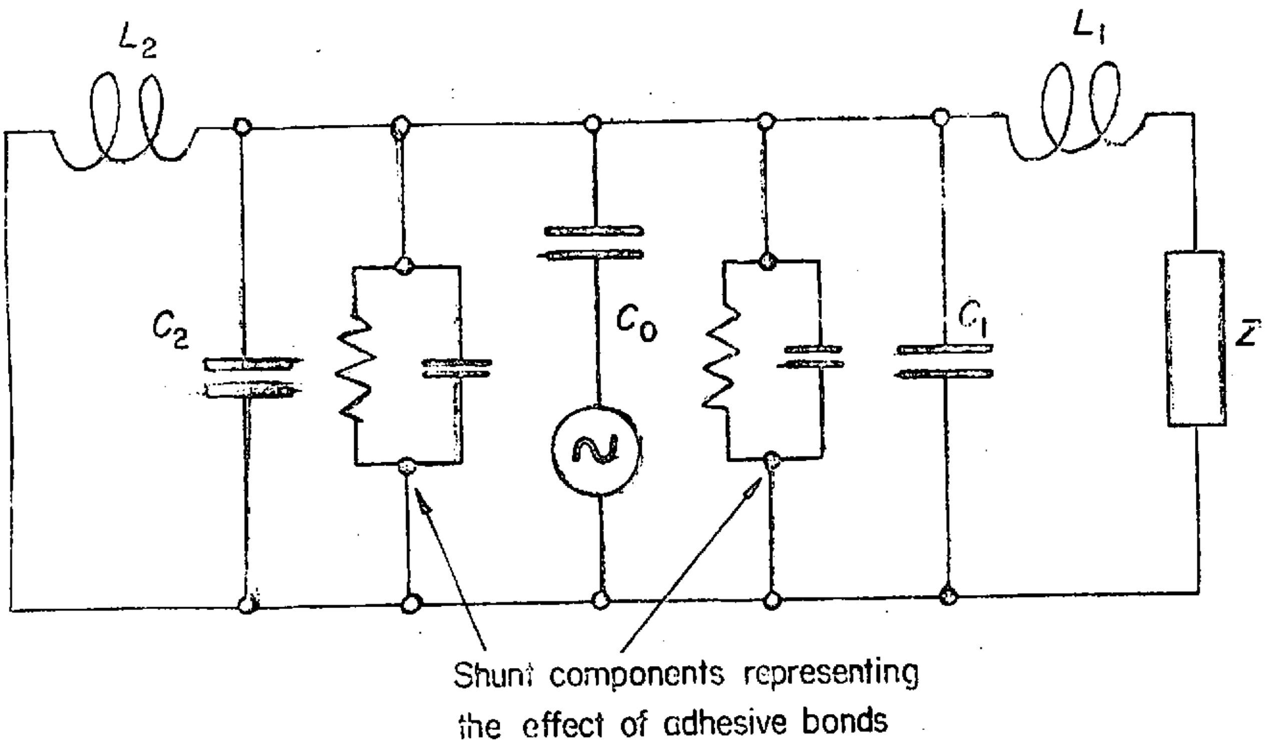


Fig. 2 Electrical analogue circuit of 'sandwich' type transducer. $L_1$ and $L_2$ represent the head and tail masses, respectively. $C_1$ and $C_2$ represent their associated compliances. $C_0$ represents the total effective compliance of the PZT driving element; $Z$ represents the acoustic loading due to the fluid medium

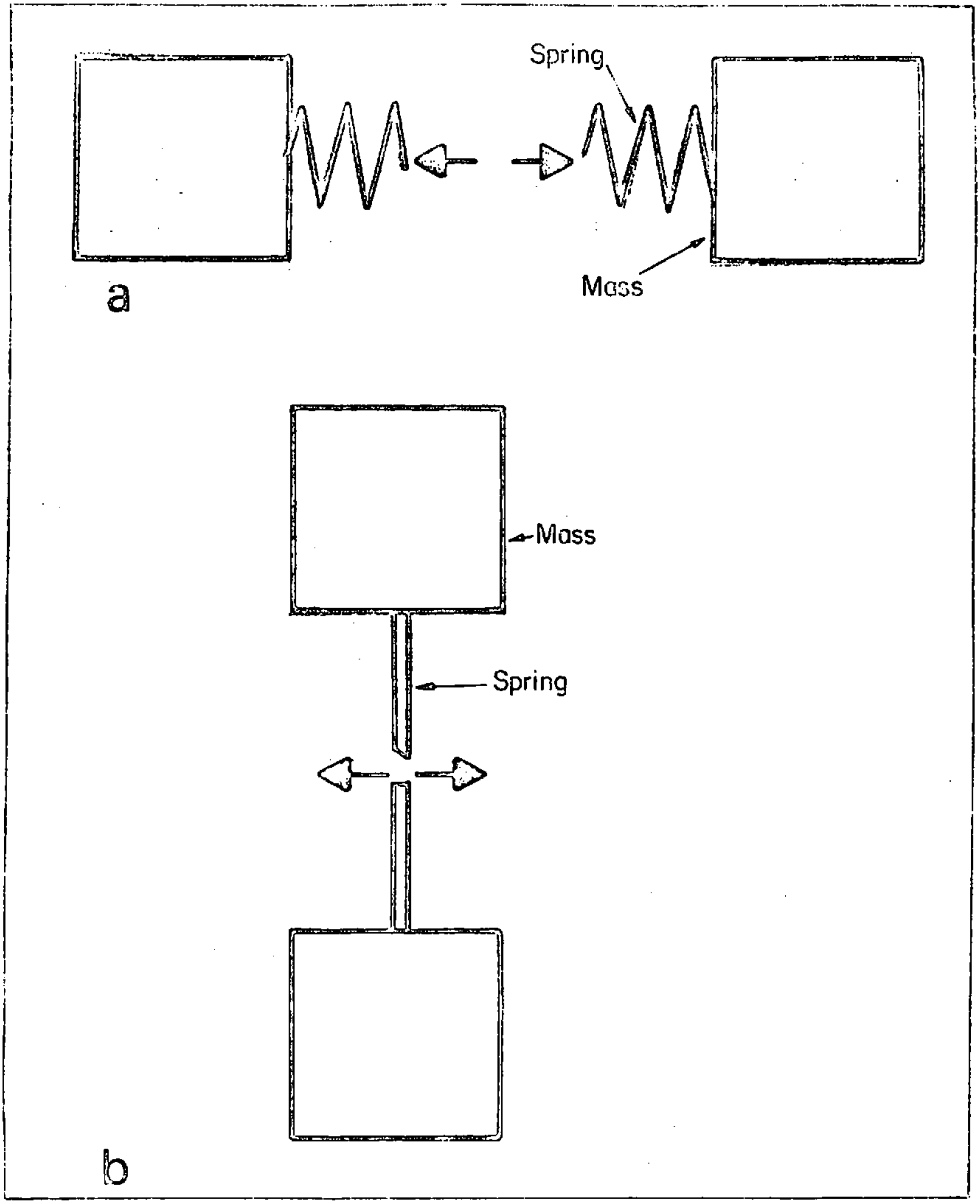


Fig. 3 Diagrams of 'lumped' mechanical resonant systems;-

(a) 'sandwich' type

(b) 'flexural' type

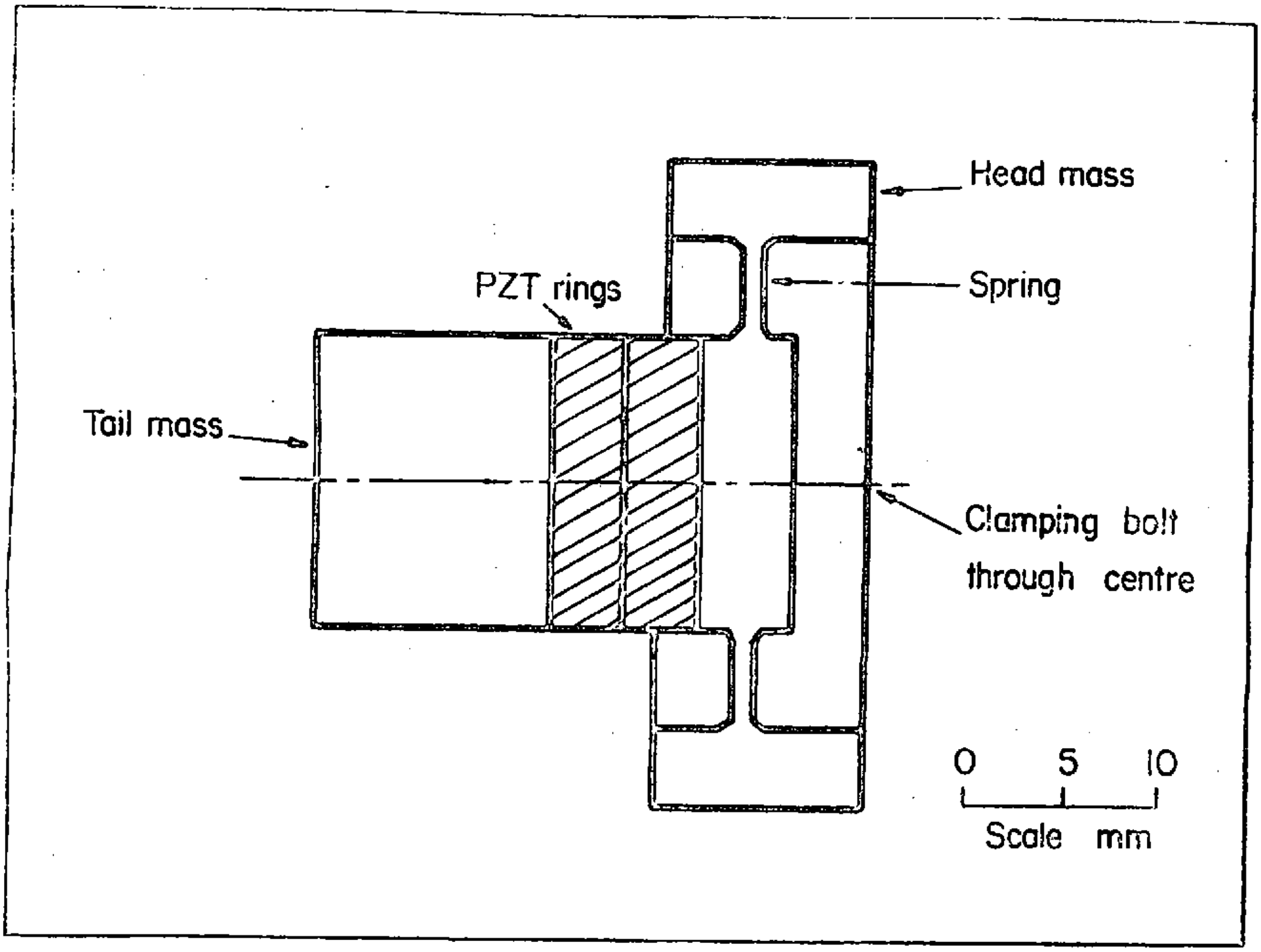


Fig. 4 'Flywheel' transducer configuration, drawn to scale.
Aluminium head; brass tail; frequency 5.5 kHz.

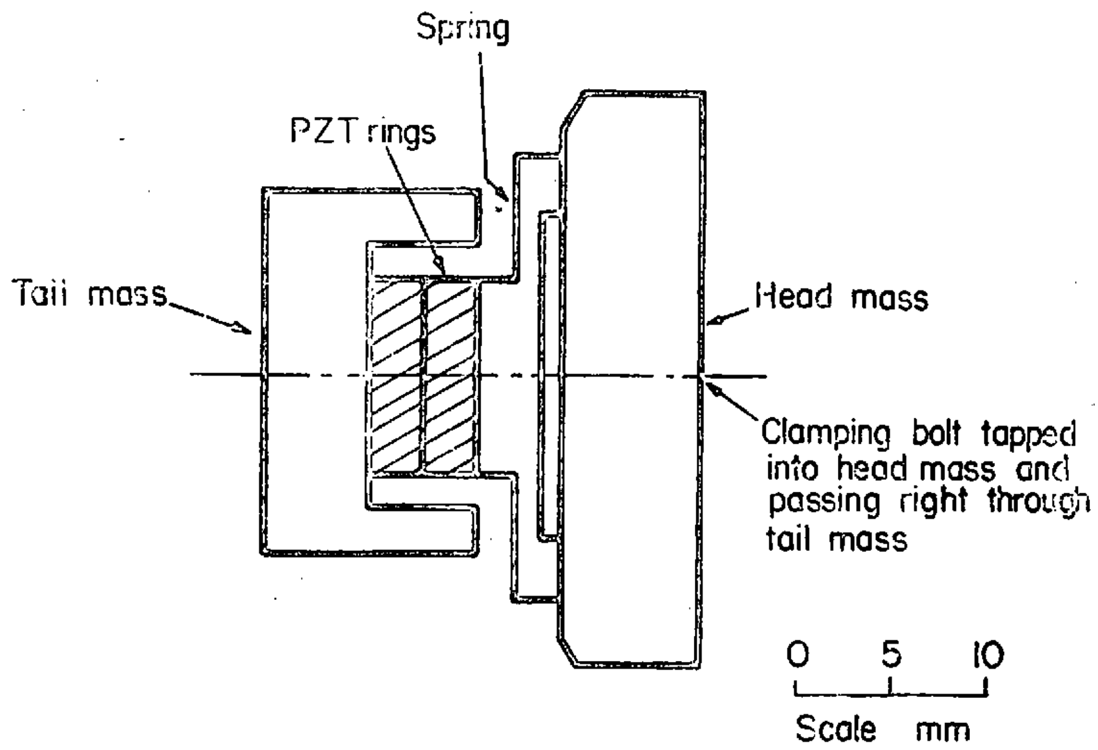


Fig. 5 Modified 'sandwich' type, incorporating additional flexural spring, drawn to scale. Aluminium head; brass tail; frequency 10 kHz.