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(57) Abstract: Embodiments of the present invention relate to an acoustic pump operable using a first axial resonant mode.

## Device

### Field of the invention

Embodiments of the present invention relate to a device and method of operating the same and, more particularly, to displacement pumps that use fluid resonance.

### 5 Background to the invention

Positive displacement pumps work with a constrained volume of fluid, such as, for example, in a cylinder in a piston pump. They are primarily used for applications where high pressure is important. Positive displacement pumps have a distinct cycle during which air is drawn into a chamber, compressed, then expelled. This cycle is usually controlled by mechanical  
10 valves that open and close ports in the chamber.

In contrast, Kinetic pumps work by accelerating unconstrained fluid volumes, such as, for example, air going through a fan. They are usually used for applications where high flow rate is important. Kinetic pumps are usually steady flow devices (non cyclic operation) and do not need to use valves on inlet and outlet ports.

15 For the purposes of this patent, a pump may be considered as any physical device that transfers energy to a fluid by application of mechanical work. The energy transferred will be in the form of potential energy (pressure) primarily due to a change of the fluid volume, and kinetic energy due to an increase in fluid velocity. The pumping process will typically also increase the temperature of the fluid.

20 Known acoustic pumps, that is, positive displacement pumps that use fluid resonance in the pump chamber to enable a cyclic pumping function without using mechanical valves, use the bulk or Helmholtz mode, to enable cyclic operation. In this mode, the wavelength of pressure oscillations in the chamber are significantly longer than the largest length dimension of the chamber. Therefore, the pressure in the chamber varies with time but is  
25 spatially uniform throughout the chamber. While this mode of operation enables devices to be realised, engineering constraints mean that it is difficult to reduce the physical scale of these devices to operate them at high frequency and to decrease the required installed volume needed to deliver a given amount of power (form factor).

Acoustic pumps are referred to by various names including Synthetic Jet Actuator, Zero Net  
30 Mass Flux Actuator, or Massless Jet Actuators. The key underlying physical process is the use of acoustic resonance in the chamber to establish a flow inlet/outlet cycle at a port in the

pump chamber. While the net mass flow in and out of the chamber is zero, differences in flow patterns during the inlet and outlet part of the cycle means that there is a net flow of fluid momentum away from the device which provides the required pumping action.

#### Summary of invention

- 5     Embodiments of the present invention provide a device comprising a chamber bearing an actuable member, having an actuating boundary to excite an acoustic mode of the fluid in the chamber, and a port; characterised in that the device comprises a control system having an actuator to actuate the actuable member, the control system and actuable member being arranged to excite or induce a first or higher order acoustic mode by periodically varying the  
10     geometry of the chamber.

- Embodiments of the present invention provide a practical means to exploit higher order modes of oscillation of the pump chamber to allow at least one or more of higher frequency operation and reduced form factor. High harmonic modes are characterised by oscillation wavelengths that are small compared to the length scale of the chamber. This means that  
15     the pressure oscillations in the chamber vary both in space and time.

Preferred embodiments excite these modes using an oscillating boundary with a mode shape that provides effective coupling with the acoustic wave in the chamber.

#### Brief description of the drawings

- Embodiments of the present invention will now be described by way of example only with  
20     reference to the accompanying drawings in which:

figure 1 shows schematically an acoustic pump according to the prior art;

figure 2 depicts an acoustic pump according to embodiments of the present invention ;

figure 3 illustrates an array of actuators according to embodiments of the present invention;

figure 4 shows an acoustic pump according to an embodiment of the present invention;

- 25     figure 5 shows an acoustic pump according to an embodiment of the present invention;

figure 6 depicts acoustic pumps according to embodiments of the present invention;

figure 7 illustrates acoustic pump actuation according to embodiments of the present invention;

figure 8 shows an embodiment of an acoustic pump;

figure 9 illustrates frequencies responses of a plurality of acoustic pumps according to various embodiments;

figure 10 shows embodiments of a pair of devices;

5 figure 11 depicts a series arrangement of devices according to an embodiment;

figure 12 illustrates a device and control system according to an embodiment;

figure 13 shows experimental results derived from testing a device according to an embodiment; and

10 figures 14a-d illustrate a preferred embodiment of a device in plan, isometric, exploded and sectional views

#### Detailed description of embodiments

Figure 1 shows a generic arrangement of an acoustic pump 100. The pump 100 comprises an acoustic chamber 102 with a port 104 that connects fluid enclosed in the chamber with the fluid outside of the pump 100. The internal boundary surfaces 106 to 110 of the  
15 chamber are arranged to reflect acoustic energy without significant loss such that the chamber exhibits a relatively undamped acoustic resonance, that is, they are sound hard. The precise performance of the chamber can be quantified by specifying the mass per unit area of the material or acoustic impedance. One of the internal boundary surfaces 106 to 110 is defined as an actuating boundary 110. The actuating boundary 110 is sound hard, but  
20 flexible, and can be driven by an external actuator 112 such that mechanical work is done on the fluid in chamber 102. To improve power transfer, the actuating boundary is designed to be a mechanically resonant system with a natural frequency similar to that of the chamber. This is not a straightforward process as it requires someone skilled in the art to match the exit condition driven by the mechanical resonance to an acoustic mode with zero mass flow  
25 at the closed end.

One skilled in the art appreciates that existing forms of acoustic pumps are typically designed to be operated at a frequency close to natural frequency of the bulk mode given by the combination of acoustic chamber and outlet port, that is, they are operable in the Helmholtz mode. The Helmholtz mode of operation may be understood physically to be  
30 associated with oscillation of a mass of fluid within the exit port of the chamber against a

spring provided by the elastic compression of the fluid within the chamber. The acoustic natural frequency of the device when operating at a bulk/Helmholtz/low order is

$$f_H = \frac{v}{2\pi} \sqrt{\frac{A}{V_0 L}}$$

approximated by the following equation:

where  $v$  is the speed of sound in the fluid,  $A$  is the cross sectional area of the outlet port,  $V_0$  is the volume of the acoustic chamber and  $L$  is the effective length of the neck of the outlet port taking into account the motion of the air just outside the neck. The effective length can be specified by the physical length of the neck plus an end correction suitable for that neck shape. End corrections can also be specified as complex numbers in order to take account of the losses in the flow caused by the neck. Since the acoustic natural frequency only depends on the chamber volume rather than its shape, the choice of chamber geometry and the mode shape of the actuating boundary are relatively unimportant. However, the neck length and shape are important parameters independent from the frequency because they determine the acoustic efficiency of the system. Furthermore, since the pressure is spatially uniform in the chamber, the location of the outlet port around the chamber boundary is unimportant.

In contrast to the foregoing, when operating an acoustic pump using higher order modes of the acoustic chamber according to embodiments of the present invention, i.e. modes with a natural frequency greater than the Helmholtz mode, one skilled in the art will appreciate that the spatial arrangement of the boundaries and outlet port will establish a wave mode whose natural frequency is reasonably separated in frequency from the natural frequencies of other modes such that it is possible to supply energy via the actuating boundary to this mode at the exclusion of other modes, and whose mode shape can be reasonably matched with the mode shape of the actuating part of the boundary such that both correspond to a velocity antinode (maximum) at the outlet port. This can be quantified by careful modelling of the acoustic modes, vibrational modes and crucially the exit condition. Since the pressure in the chamber is spatially non-uniform for higher order modes, the output of the device varies depending on the location of the outlet port. Therefore, the outlet pressure-flow characteristics of embodiments of the device can be varied by varying the location of the outlet port, which may be of benefit in some applications.

A further advantage of higher order mode operation is that the natural frequency of the chamber acoustic mode is significantly less sensitive to the effective neck length of the outlet port compared to devices using a Helmholtz like resonance. This provides flexibility when

the effective neck length is predetermined by the application, such as, for example, when mounting pump devices beneath a structural skin that may be of fixed thickness for strength reasons.

Figure 2 shows a number of rectangular acoustic pumps 200a to 200e according to embodiments of the present invention shown in cross section. The depths of the pumps 200a to 200e into the page is arbitrary, but one skilled in the art will appreciate that higher order acoustic modes based on the length into the page will exist and, therefore, that it may be desirable to provide frequency separation between such modes and the primary longitudinal higher order modes of interest by ensuring that the length of the device is large in comparison with its height and its depth. In all of the embodiments illustrated, the actuating boundary 202a to 202e is illustrated as a cantilever beam 204a to 204e, which is driven by an actuator located at a position indicated by the label 'F'. Points 1, 2 and 3 refer to possible locations for outlet ports. The pairs of curves 206a to 206e illustrate respective mode shapes of the zeroth order acoustic mode for the chamber, which for this geometry will be a plane wave travelling parallel to the longer axis of the rectangle. A velocity node (zero amplitude) indicates a point where the amplitude of the acoustic velocity is a minimum and the acoustic pressure a maximum. It can be appreciated that the nodes are located at 208a to 208e.

According to the first embodiment of an acoustic pump 200a, the acoustic chamber 210a has an actuating boundary 204a on the top, fixed boundaries 212a and 214a on the left and bottom, and an open boundary 216a on the right. One skilled in the art will appreciate that the shape of the first higher order acoustic mode will be approximated by a quarter wavelength with a velocity node 208a (minimum) at the left hand boundary and a velocity antinode (maximum) at the right hand (open) boundary 216a. In the embodiment illustrated, the free end 218a of the cantilever beam 204a is chosen to be collocated with the velocity antinode of the acoustic wave to improve the coupling between the first bending mode of the beam and the acoustic wave. This requires careful matching of the mode shape such that the large velocities seen at the exit induced by the vibrational mode correspond to acoustic modes with zero mass flow at the closed end such that the acoustic and vibrational modes complement each other rather than compete with each other. One skilled in the art will appreciate that using port 3 as the outlet port of the pump 202a would give a low pressure, high volume flow rate output, using a port at location 1 would give a high pressure, low volume flow rate output and that using port 2, or any other port located therebetween, that is between ports 1 and 3, would provide a compromise between pressure and volume flow rate.

A second embodiment of an acoustic pump 200b shows a similar arrangement to that in the first embodiment 202a, with the difference that the fixed boundary 214a on the bottom edge of the chamber has been replaced with an additional actuating boundary 214b operating in anti-phase to the upper boundary 204b. This configuration is advantageous in that the pair of cantilevers 204b and 214b can be arranged in a tuning fork configuration with the actuator element (not shown) placed between the tines of the tuning fork. A further illustration of such an embodiment will be described below with reference to figure 8.

A third embodiment of an acoustic pump 200c is illustrated that is the same as the first embodiment 200a, but for the right hand boundary 216c being closed. Therefore, the first higher order acoustic mode is a half wave length with velocity nodes 208c at the left 212c and right 216c hand boundaries and with a natural frequency twice that of the first embodiment 200a. The third embodiment 200c allows low pressure high flow rate fluid to flow from port position 2 and high pressure low flow rate fluid from ports 1 and 3. One skilled in the art will appreciate that the mode shape of the acoustic wave for a half wave in the chamber may be better matched to the mode shape of the second bending mode of the cantilever beam which has a velocity maximum near the centre of the beam.

A fourth embodiment of an acoustic pump 200d is illustrated in which the left 212d and right 216d boundaries are open, which provides velocity antinodes at either end of the acoustic chamber.

A fifth embodiment of an acoustic pump 200e is shown in which two cantilever beams are used to provide actuation as is the case in the third acoustic pump 200c.

Figure 3 shows an array 300 of acoustic pumps, which can be made by stacking a number of acoustic pumps 302 to 310 back to back. One skilled in the art appreciates that embodiments can be realised in which actuating boundaries can be shared by adjacent acoustic pumps. Such embodiments have the advantage of reducing the required installation volume of the array. Operation of a pair of devices with a shared boundary means that the flow outlet from each device is in antiphase. This will have the benefit of reducing the noise produced by the array 300. For example, the flow from the outlets 312 and 314 of acoustic pumps 302 and 304 will be in antiphase. In the illustrated embodiment one skilled in the art can appreciate that the shared boundaries are fixed. However, the shared boundaries could equally well be actuating boundaries. One skilled in the art appreciates that whilst the analysis in the appendix is for a single actuating boundary it can be readily extended to the case of an actuating boundary on both sides.

Embodiments of the invention can be realised in which a cantilever used to provide an actuating boundary has a variable cross-section such as, for example, an arbitrary distribution of cross section along its length. Figure 4 shows an embodiment of an acoustic pump 400 using of a tapered beam 402 against a cavity 404 closed at a first end 406, that is the built end, of the beam and open at the free end 408 of the beam 402. Using a tapered beam 402 allows the overall beam mass to be reduced compared to the mass of a uniform cross section beam of the same natural frequency. The mode shape for the first bending mode of such a tapered beam 402 will be slightly different to that of a uniform beam with greater displacement at the unsupported end of the beam as compared to the rest of the beam since the moment of inertia about the fixed end will be lower, which may be beneficial for some applications because it has the potential to induce even greater velocities at the free end.

Referring to figure 5, there is shown an embodiment of an acoustic pump 500 having an acoustic chamber 502 with a varying cross-sectional area distribution along its length. It can be appreciated that the cavity 502 tapers inwardly towards the built end 504 of the cavity. Varying at least one of the geometry of the chamber 502 and the beam 506 along its length allows one skilled in the art to at least improve, and preferably optimise, the energy transfer between the mechanical and acoustic domains according to the theory given in the appendix and hence improve the performance. One skilled in the art will appreciate that the desired performance will depend upon the application. Typically, the performance of devices according to embodiments of the present invention would be measured by the net momentum imparted to the surrounding fluid or the net energy imparted.

Figure 6 shows various embodiments 600 for providing an actuation force to the actuating boundary of the acoustic pumps described above. Embodiments can be realised in which the actuation force  $F$  can be normal to the actuating boundary as in cases a), that is, within or without the chamber 602 or tangential to the boundary as can be appreciated from case b). If a piezo-ceramic material is used for the actuator then for case a) the material would be used in the d33 mode where the strain is in the same direction as the applied voltage. For case b) the piezo material would be used in the d31 mode in which the strain is normal to the direction of the applied voltage. For a) it is advantageous that the actuator contacts the actuating surface via a flexure or point application (as shown in i) such that undesirable bending moments are not transferred to the piezo-ceramic material. Locating the actuator between two symmetric cantilever elements is advantageous in that external clamping forces are not required to react actuation loads from the actuator. The actuation arrangement shown in b) is commonly found in piezo-ceramic diaphragm and bimorph

actuators. In this arrangement the actuator deforms the structure to which it is attached via a shear force that causes the structure to bend out of the plane of actuation. For transferring a shear force, the actuator must be bonded directly to the actuating member and the strain of the member should equal the strain in the actuator. Since piezo-ceramic materials are very strain limited, e.g. 0.5%, then this may limit the strain and hence the deflection of the actuating boundary. Therefore, for applications where the radius of curvature of the actuating boundary is small (large deflections) such that strains that are compared to the maximum strain allowable for piezo-ceramic materials are introduced at the inner and outer surface of the actuating boundary, direct stress actuation as shown in Figure 6a is preferred.

To operate efficiently acoustic pumps according to embodiments of the present invention, the boundaries of the acoustic chamber need to be acoustically sealed from the surroundings apart from at the port or ports. The simplest solution to implementing a sealed actuating boundary is to use a diaphragm fixed around its perimeter to the non actuating boundary, and this type of design is widely used for traditional Synthetic Jet Actuator type acoustic pumps. By the nature of the circumferential clamping arrangement, the displaced shape of the diaphragm involves two degrees of curvature. This tends to make diaphragms relatively stiff, which can present problems as devices are reduced in size since the thickness of the diaphragm may have to be reduced to impractical levels to keep its natural frequency sufficiently low to be matched with an acoustic chamber natural frequency.

However, embodiments of the present invention use actuated boundaries in the form of cantilevers, i.e. rectangular plates that are fixed relative to the unactuated boundaries along one edge. The natural frequency of the first bending mode of a cantilever of a given length built in at one end is approximately 3 times lower than the natural frequency of an equivalent diaphragm of the same thickness with radius equal to the length of the cantilever.

Therefore, at small scales consistent with devices working at ultrasonic frequencies, e.g. around 20 kHz, where reduced material thicknesses may increase cost and reduce performance, embodiments preferably use partially supported actuating boundaries in the form a cantilevered plate rather than a fully supported actuating boundary in the form of a diaphragm.

One skilled in the art appreciates that using cantilevered actuating boundaries introduces the problem of sealing around the free edges. Embodiments of the present invention address this via two solutions as can be appreciated from figure 7. Figure 7 shows two embodiments of devices. A first device 702 is based on providing a sufficiently narrow gap between the relatively moving parts to achieve a viscous seal. It can be appreciated that the first device 702 comprises a number of fixed boundaries 704 to 708 and an actuating member

710 disposed to define a chamber 712. A second device 714 uses flexure elements 716 and 718 that allow large relative displacements of an actuating member 720 in a reciprocating movement relative to a fixed boundary 724 thereby defining a variable geometry chamber 726 that is sealed from the surroundings. One skilled in the art appreciates that the first 702 and second 714 devices are viewed along their respective longitudinal axes. The above embodiments of the first 702 and second 714 devices could be applied to any of the embodiments of the present invention described herein.

Figure 8 depicts an isometric cross sectional view of a device 800 according to an embodiment. One skilled in the art can appreciate that the device 800 is in the form a tuning fork with an actuator 802 between the tines 804, and side plates 806 (only one shown) to create a sealed acoustic chamber 808 in the volume enclosed between the tines 804.

In the above embodiments, the natural frequencies of the mechanical and acoustic components of an acoustic pump both scale inversely with length, i.e. if a particular device geometry is halved in linear length scale, both the natural frequency of the mechanical elements and acoustic elements will increase by a factor of 2. For a device of a given geometry, the natural frequencies of the mechanical and acoustic elements will vary with the materials used. For the mechanical components, the natural frequency will be proportional

to  $\sqrt{\frac{E}{\rho}}$  where  $E$  is the material's Young's modulus and  $\rho$  is the material's density. For small scale devices operating at ultrasonic frequencies, it is likely that a material such

silicon, which has relatively high values of  $\sqrt{\frac{E}{\rho}}$  compared to other materials, would be a good choice, although any other material such as metals with a high stiffness to density ratio could be used. For the fluid domain, the natural frequency is proportional to the speed of the sound in the fluid. One skilled in the art appreciates that the fluids could be liquid or gas. Liquids typically have a greater speed of sound as compared to gases at the same temperature. For example, water has a speed of sound approximately 4 times higher than air at the same temperature. Consequently, the acoustic natural frequency of the device for water will be approximately 4 times higher than it is for air as the working fluid.

One skilled in the art appreciates that the power transfer between two oscillating systems is maximised when the impedance of each system is matched. Embodiments of the present invention are arranged to optimise the output velocity from an acoustic pump. A demonstration of this is provided by a numerical experiment based on a finite element

simulation of a 2d acoustic pump comprising a thin rectangular acoustic chamber with one side comprised of a actuating boundary provided by a cantilever. According to an embodiment, the material for the mechanical elements was Aluminum with density  $2700\text{kg/m}^3$  and Young's Modulus of 70 GPa. The fluid was air at 1 bar pressure at  $20^\circ\text{C}$ .

5 The dimensions are given on the chart below. The effects of varying the thickness of the cantilever on the velocity output of the device as a function of cantilever excitation frequency are shown in Figure 9. It can be appreciated that for the device simulated the peak output velocity is obtained for a cantilever thickness of 3mm.

Referring to figure 10, there is shown schematically a pair 1000 of embodiments of devices 1002 and 1004 according to embodiments of the present invention. A first device 1002 of the pair comprises a block base 1006 bearing first 1008 and second 1010 actuating members, preferably tines, having an actuator 1012 disposed therebetween. The actuator 1012 is preferably a pzt actuator element. The device 1002 has a chamber 1014 and an outlet port 1016. The second device 1004 comprises first 1018 and second 1020 actuating members having first 1022 and second 1024 flexures defining a chamber 1026 having an outlet port 1028. In both embodiments, the gaps between the actuating members was 2 mm and the length of the chamber 40 mm. The tine width was 5 mm and the flexure material was a thin self adhesive plastic film (parcel tape).

The embodiments were designed to operate at 2khz. For a device designed to work at 20kHz, the dimensions are simply scaled by 1/10, i.e. all linear dimensions are multiplied by 0.1.

Figure 11 shows a pair 1100 of devices serially configured according to an embodiment of the present invention.

A first device 1102 is housed in a fluid reservoir 1104 having an inlet 1106 and an outlet 1108. The first device 1102 could be any device described herein. In the embodiment shown the first device 1102 comprises first 1110 and second 1112 actuating members defining a chamber 1114 and an outlet port 1116. An actuator 1118 is disposed between the actuator members 1110 and 1112.

A second device 1102 is housed in a fluid reservoir 1104' having an inlet 1106' and an outlet 1108'. The first device 1102' could be any device described herein. In the embodiment shown the first device 1102' comprises first 1110' and second 1112' actuating members defining a chamber 1114' and an outlet port 1116'. An actuator 1118' is disposed between the actuator members 1110' and 1112'.

The chambers 1114 and 1114' are in fluid communication via a conduit 1120.

Figure 12 shows an embodiment of an arrangement 1200 according to an embodiment of the present invention. The arrangement 1200 comprises a device 1202 having a pair of actuating members 1204 and 1206 defining a chamber 1208 with an outlet 1209 and having an actuating element 1210 disposed therebetween. The actuating element 1210 is coupled to a control system 1212. The control system 1212 provides an appropriate signal 1214 to the actuating element 1210 to cause the actuating members to oscillate and induce the modes described above according to a driving signal. Depending on the design of the pump, it may be desirable to excite a number of different modes of vibration at varying amplitude. To do this an electrical driving voltage applied to the actuating element would be synthesized from the addition of a number of preferably sinusoidal (i.e. signals containing a single frequency) voltage signals at frequencies corresponding to the frequencies of the required modes of vibration and amplitudes consistent with the desired relative contribution of the particular mode shape to the overall mode of vibration of the device.

Figure 13 shows a frequency response graph 1300 of the variation of actuating member tip displacement and jet velocity, both in m/s, with frequency in Hz derived from repeated experimental measurements made while the actuator was excited from 0 Hz to 16 kHz. The applicable conditions and parameters under which the experiment was conducted are

(1) the actuating element was a PZT, having dimensions of 5 x 5 x 2 mm

(2) the actuating members were tapering tines

(3) the outlet or front aperture had dimensions 5 x 2 mm

(4) rear-side aperture was circa 0.3 of (3) above

(5) chamber sides were closed

(6) frequency sweep resolution was 5 Hz

(7) the operating temperature was 22C.

Referring to figure 14(a), there is shown a view 1400 of a preferred embodiment of a device 1402. The device 1402 comprises a three plate assembly. The outer plates 1404 and 1406 comprise a pair of cantilever actuators 1408 and 1410. The gaps between the cantilever actuators 1408 and 1410 and the rest of the outer plates 1404 and 1406 are sealed using respective flexural seals 1412 and 1414. A chamber plate 1416 is disposed between the

two outer plates 1404 and 1406. The chamber plate 1416 defines a chamber 1418. The chamber plate 1416 comprises at least one and preferably two ports 1420 and 1422. An actuating device 1424 is housed within the chamber 1418 and adapted for influencing movement of the cantilever actuators 1408 and 1410 in response to excitation thereof.

- 5 Preferred embodiments use a piezo actuator that is electrically insulated. The actuating device 1424 is preferably disposed at one end of the chamber 1418, with appropriate electrically coupling(s) 1426 being via one or more than one through hole (not shown) in the chamber plate 1416.

- 10 The mechanical length of the cantilever actuators 1408 and 1410 is adapted or set according to a desired resonant mode for a given plate or actuator thickness. In preferred embodiments the thickness of the cantilever actuators corresponds to the thickness of the actuating device, but embodiments are not limited thereto. Similarly, the width of the chamber corresponds to the width of the actuating device, but embodiments are not limited thereto.

- 15 The outer plates 1404 and 1406 are bonded to the chamber plate 1416 such that the fixed ends are disposed over the actuating device. For embodiments that use a piezo actuating device, the piezo actuator required a preload. Therefore, the piezo actuator is first bonded to one cantilever, then the second cantilever is loaded, preferably at the tip, to provide the appropriate free load at the fixed end and the piezo actuator is bonded to the second  
20 cantilever 1410 whilst loaded thereby filling any gap between the cantilever and the piezo.

- Single port embodiments can be realised that would act as both inlet and outlet. Such an arrangement might find application in micro pump. Double port embodiments can be realised, which would enable one port to act as the inlet and the other port acting as an outlet. Pumped flow through the chamber would then be possible via a couple of check  
25 valves or alternatively, for a no moving parts solution, manufacture one port geometry with area contraction ratio and the other with expansion ratio.

- One skilled in the art appreciates that the chamber is sealed externally with the flexible membranes 1412 and 1414 that span the gap between each cantilever beam and the external structure of the respective cantilever plate. It can be appreciated that the  
30 membranes are preferably kinked or otherwise shaped to accommodate movement of the cantilever actuators 1408 and 1410 without loss of seal. Alternatively, embodiments can be realised that replace one of the cantilever plates with a rigid base plate sealing one side of the chamber, which may introduce a performance penalty but without loss of functionality.

The flexural seals 1412 and 1414 can be flat or shaped. One skilled in the art appreciates that the flexural seals can introduce a degree of damping. Furthermore, any such damping is preferably minimised and adapted to ensure that it does not interfere with the movement of the cantilever actuators 1408 and 1410.

- 5 One skilled in the art appreciates that a device according to the present can find application as a pressure source and/or with a single port at one end, as follows. Exciting the chamber at the resonant frequency or at a desired resonant mode establishes oscillatory pressure antinodes; at least one of which could be used as a fluctuating dynamic pressure source. In single port embodiments, the port is positioned such that or the resonant mode is arranged  
10 such that the port and an antinode coincide.

The oscillatory pressure drives fluid pumped through the port and can act as inlet for half the cycle and outlet during the other half. The fluid escaping the chamber is the useable pumped volume flow rate directed away from the port, the fluid entering the chamber is drawn from the ambient volume in the local vicinity of the port.

- 15 One skilled in the art appreciates that devices according to embodiments of the present invention have a potential advantage that their mechanical design is scalable to small size/high frequency operation. Still further, their rectangular or cuboidal geometry provides a more efficient use of space (higher volume to maximum linear dimension ratio) compared to circular diaphragm based designs.

- 20 Embodiments of the present invention have at least one or more than one of the following advantages taken jointly and severally in any and all combinations

- Low noise due to ultrasonic operation, which can be relevant for air applications. Prior art micropumps (commercial and academic) are typically a factor of 10 short regarding target frequencies of operation, for example, they operate at 2kHz when  
25 20kHz is desirable;
- Cost – embodiments of the present invention are suitable for mass production using MEMS fabrication;
- Form factor - embodiments of the present invention scale well to high frequency and/or with package size;
- 30 • low voltage, since embodiments can be realised that use 40v operation or less. Still further, embodiments can be realised using 12v or less. Using a stack piezo electric element as the actuator device can enable lower voltage operation;
- power consumption such be at least as good or better than prior art diaphragm pumps;

- environmental flexibility;
  - embodiments can have a wide range of configurations and/or can be more easily optimised for a wider range of pressure/flow options than conventional micropumps; and
- 5      • potentially disposable.

Embodiments of the invention can be applied in any of the following areas:

1.. Medical applications:

- Drug administration into body, via a patch, intravenously or as an in-body implant;
- 10      • As an in body pump such as, for example, a blood pump
- artificial urethra and/or sphincters
  - intelligent non-contact disinfectant dispensers
  - Personal (hand held) nebulisers

2. Industrial applications:

- 15      • Feeding lubricants to point of need such as, for example, in linear actuators or micro-machining where minute amounts of cutting lubricants are needed;
  - Food and drink industry applications such as, for example, dosing of flavourings and/or fragrances in food & drink dispensers. For example, within a coffee making context the small size (low price / disposability) allows integration directly into a water
- 20      reservoir and prevents cross contamination between substances and/or gives better hygiene;
- in consumer products for controlling dosing cosmetics and personal care treatments to improve efficiency in industrial applications
  - so called 'air care' devices in which businesses and smaller consumer devices need
- 25      small amounts of scent oil per dosing cycle of a room fragrance. In addition to being able to modulate a scent impression over time, embodiments could reduce the cost of expensive fragrances and out-perform capillary-based passive solutions, which are currently most often used in disposable air freshening systems, with more accurate and consistent delivery.

30

3. Electronic applications:

- pumps in 'micro fuel cells'
- as part of a cooling system

- within smart phones and other microelectronics consumer devices to provide low volume air flow, with a small form factor and low power consumption;
- chip and/or server or other computer/device cooling achieved by spraying an evaporative liquid cooling;
- 5 • cooling for LED lighting

4. Lab on a chip:

- for gas Sensing
- for gas micro-mixing

5. Creating boundary layer effects:

- within an aerospace context
- within a wind turbine

Throughout the description and claims of this specification, the words "comprise" and  
15 "contain" and variations of them mean "including but not limited to", and they are not  
intended to (and do not) exclude other moieties, additives, components, integers or steps.  
Throughout the description and claims of this specification, the singular encompasses the  
plural unless the context otherwise requires. In particular, where the indefinite article is  
used, the specification is to be understood as contemplating plurality as well as singularity,  
20 unless the context requires otherwise.

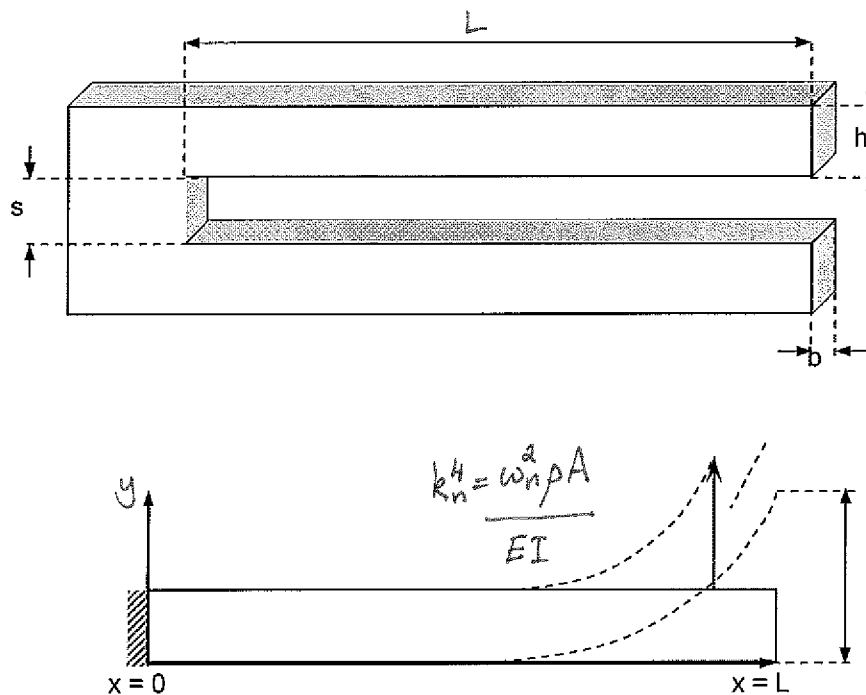
Features, integers, characteristics, compounds, chemical moieties or groups described in  
conjunction with a particular aspect, embodiment or example of the invention are to be  
understood to be applicable to any other aspect, embodiment or example described herein  
unless incompatible therewith. All of the features disclosed in this specification (including  
25 any accompanying claims, abstract and drawings), and/or all of the steps of any method or  
process so disclosed, may be combined in any combination, except combinations where at  
least some of such features and/or steps are mutually exclusive. The invention is not  
restricted to the details of any foregoing embodiments. The invention extends to any novel  
one, or any novel combination, of the features disclosed in this specification (including any  
30 accompanying claims, abstract and drawings), or to any novel one, or any novel  
combination, of the steps of any method or process so disclosed.

The reader's attention is directed to all papers and documents which are filed concurrently

with or previous to this specification in connection with this application and which are open to public inspection with this specification, and the contents of all such papers and documents are incorporated herein by reference.

## Appendix

### Device Theory : Forked Actuator Prototype



## Nomenclature

|       |                                           |           |
|-------|-------------------------------------------|-----------|
| A     | Surface area of applied load on the beam  | $m^2$     |
| $A_n$ | Constant                                  |           |
| b     | Beam width                                | m         |
| $B_n$ | Constant                                  |           |
| c     | Speed of sound of the fluid in the cavity | $ms^{-1}$ |
| $C_n$ | Constant                                  |           |
| d     | Orifice diameter                          | m         |
| $D_n$ | Constant                                  |           |
| E     | Young's modulus of beam material          | Pa        |
| $E_n$ | Constant                                  |           |
| $f_n$ | Beam natural resonant frequency           | Hz        |

|                   |                                                        |            |
|-------------------|--------------------------------------------------------|------------|
| $F_n$             | Constant                                               |            |
| $G_n$             | Constant                                               |            |
| $h$               | Beam depth                                             | m          |
| $H_n$             | Constant                                               |            |
| $I$               | Beam second moment of area; $I = (bh^3)/12$            | $m^4$      |
| $k_n$             | Wave number                                            |            |
| $L$               | Beam length                                            | m          |
| $p$               | Acoustic pressure in the cavity                        | Pa         |
| $q$               | Applied load on the beam                               | N          |
| $s$               | Separation distance between the tines                  | m          |
| $t$               | Time                                                   | s          |
| $T$               | Period of single beam oscillation                      | s          |
| $u$               |                                                        |            |
| $w$               | Beam deflection                                        | m          |
| $x$               | Distance along the length of the beam                  | m          |
| $X$               | Beam deflection                                        | m          |
| $y$               | Distance along the depth of the beam                   | m          |
| $z$               | Distance along the width of the beam                   | m          |
|                   |                                                        |            |
| <b>Subscripts</b> |                                                        |            |
| $n$               | Eigenfrequency mode number                             |            |
|                   |                                                        |            |
| <b>Greeks</b>     |                                                        |            |
| $\zeta_n$         | Constant                                               |            |
| $\rho$            | Density                                                | $kgm^{-3}$ |
| $\omega_n$        | Beam natural resonant frequency; $\omega_n = 2\pi f_n$ | Hz         |

## Vibration of the beam

The beam bending equation is given by:

$$5 \quad \frac{d^2}{dx^2} \left( EI \frac{d^2 w}{dx^2} \right) = q \quad (1)$$

Where:

E = Young's modulus

I = second moment of area

10 w = beam deflection

q = applied load

15 The above equation is the well known and well established Euler-Bernoulli Bending equation and is valid provided the displacement is small which in practice means that the peak deflection is a small fraction (less than, for example, 5%) of the over length which it has to be in this case because of the cavity depth since the depth of the cavity is already small in comparison with the cantilever length.

20 For a vibrating beam, there is no applied load (linear assumption for the acoustics). However, this could be provided as an additional correction for very high amplitudes, which in practice would be amplitudes greater than about 110dB). Mathematically, this would be the particular integral, compared to the complementary function provided below. Nevertheless, there is acceleration of the beam, rather than a static beam, which leads to an acceleration of the beam balanced to the forces. Therefore, the internal beam forces are matched to the beam acceleration:

$$25 \quad -\rho A \frac{\partial^2 w}{\partial t^2} = \frac{d^2}{dx^2} \left[ EI \frac{d^2 w}{dx^2} \right] \quad (2)$$

Equation (2) is solved using the separation of variables:

Let  $w = X(x)T(t)$

30 However, it should be noted that any number of similar pairs of functions could be added to the above, such as, for example,

$$W = X_1(x)T_1(t) + X_2(x)T_2(t) + \dots$$

- 5 It is also possible that the solution cannot be separated in this way. However, in the present case, one pair is sufficient to find a solution for the equation that matches the boundary conditions.

$$\frac{d^2}{dx^2}(EI) \left[ \frac{X''}{X} \right] + EI \left[ \frac{X'''}{X} \right] = k_n^4$$

$$\frac{d^2 T}{dt^2} + \omega_n^2 T = 0$$

10

- 15 In the above equations, it is important to note that the choice of constants was arbitrary to satisfy the differential equations. They could have been positive or negative or, indeed, zero and still satisfy Euler Bernoulli extended for vibration. However, for a harmonic solution, the time function needs to be harmonic, which implies a negative constant,  $-\omega_n^2$

giving the second equation above. The remaining constant,  $k_n^4$ , is chosen in order to ensure that the coupled system satisfies the overall bending equation for the beam. For harmonic excitation, this requires  $k_n^4$  to be positive. This constrains the bending modes to the frequency of oscillation.

- 20 From which:

$$T = e^{i\omega t} \text{ or } e^{-i\omega t}$$

- 25 Choose  $T = e^{i\omega t}$  since positive time and positive frequency can be chosen. It will be noted that there could be a constant multiplier, but this would be superfluous since it would only ever appear as a product with another constant.

- 30 If it is further assumed that EI is constant, which will be true for all beams of uniform cross section manufactured from one material. For the case of a tapered beam, this will also represent a good approximation provided the tapering is small in comparison with the length of the beam. Where this is not the case, the system can still be solved, but this assumption would not be valid and special treatment would be required.:

$$X(x) = A_n \cos(k_n x) + B_n \sin(k_n x) + C_n \cosh(k_n x) + D_n \sinh(k_n x) \quad (3)$$

each term of which represents an independent solution of the fundamental differential equation. As the equation is 4<sup>th</sup> order, this completely describes the behaviour provided the initial separation of variables was possible. To find the relevant constants, boundary conditions should be applied, but it should be noted that the amplitude will not be able to be determined because it depends upon the applied driving load.

The boundary conditions are as follows.:

1.  $X = 0$  at  $x = 0$  (fixed end) which should be fairly obvious.

$$\rightarrow A_n = -C_n$$

2.  $\frac{\partial X}{\partial x} = 0$  on  $x = 0$  (smooth gradient). This is a standard boundary condition that comes about because the end is not able to rotate.

$$\rightarrow -A_n k_n \sin(k_n x) + B_n k_n \cos(k_n x)$$

$$-A_n k_n \sinh(k_n x) + D_n k_n \cosh(k_n x) = 0 \quad \text{on } x = 0$$

$$\rightarrow B_n = -D_n$$

$$\text{i.e. } X(x) = A_n [\cos(k_n x) - \cosh(k_n x)] + B_n [\sin(k_n x) - \sinh(k_n x)]$$

3.  $\frac{\partial^2 X}{\partial x^2} = 0$  on  $x = L$  (no axial acceleration at tip), which is again a standard boundary condition

$$\rightarrow -A_n k_n^2 [\cos(k_n L) + \cosh(k_n L)] - B_n k_n^2 [\sin(k_n L) + \sinh(k_n L)] = 0$$

$$\rightarrow B_n = -A_n \left[ \frac{\cos(k_n L) + \cosh(k_n L)}{\sin(k_n L) + \sinh(k_n L)} \right] \quad (4)$$

$$4. \quad \frac{\partial^3 X}{\partial x^3} = 0 \quad \text{on } x = L, \text{ which is also a standard boundary condition.}$$

$$5 \quad \rightarrow A_n k_n^3 [\sin(k_n L) - \sinh(k_n L)] - B_n k_n^3 [\cos(k_n L) + \cosh(k_n L)] = 0$$

$$\rightarrow B_n = \left[ \frac{\sin(k_n L) - \sinh(k_n L)}{\cos(k_n L) + \cosh(k_n L)} \right] A_n \quad (5)$$

10 Equation (5) is not a contradiction of equation (4) but rather an additional condition. In other words, the constant,  $k_n$ , must be such that both (4) and (5) are satisfied.

Hence, the condition for  $k_n$  is:

$$15 \quad \left[ \frac{\sin(k_n L) - \sinh(k_n L)}{\cos(k_n L) + \cosh(k_n L)} \right] = - \left[ \frac{\cos(k_n L) + \cosh(k_n L)}{\sin(k_n L) + \sinh(k_n L)} \right]$$

$$\cos^2(k_n L) + \sin^2(k_n L) + 1 + 2 \cos(k_n L) \cosh(k_n L) = 0$$

$$\text{i.e. } 2 + \cos(k_n L) [e^{k_n L} + e^{-k_n L}] = 0$$

20

For the geometry given:

$$n \quad k_n L \quad l_n \quad f_n$$

|   |       |         |        |
|---|-------|---------|--------|
| 1 | 1.875 | 12,600  | 2005   |
| 2 | 4.694 | 78,950  | 12,570 |
| 3 | 7.855 | 221,000 | 35,122 |

### Acoustics of the cavity

5

The 3D wave equation is given by:

$$\frac{\partial^2 p}{\partial t^2} - \bar{c}^2 \nabla^2 p = 0$$

10 where:

$\bar{c}^2$  = speed of sound in the air in the cavity

p = acoustic pressure in the cavity

15 This can also be solved by separation of variables, but this time four separated functions are combined, that is,

$$p = p_1 + p_2 + p_3 + p_4$$

20 where:

$$p_1 = X_1(x)Y_1(y)Z_1(z)e^{i\omega t}$$

$$p_2 = X_2(x)Y_2(y)Z_2(z)e^{i\omega t}$$

$$p_3 = X_3(x)Y_3(y)Z_3(z)e^{i\omega t}$$

25  $p_4 = X_4(x)Y_4(y)Z_4(z)e^{i\omega t}$

Four functions are needed because the acoustic waves must satisfy a variety of boundary conditions. Firstly, they must match the velocity of the beam. Since the beam satisfies a 4<sup>th</sup> order equation, two separate paired functions are needed to match them. Additionally, the acoustic waves must match whatever axial boundary condition apply from the cavity, bringing in the possibility of two more waves. These can be separated from the other two solutions by choosing zero velocity on the beam (giving the correct velocity when added to the other two solutions). . At this point it is worth noting that a large number of modes formally satisfy the wave equation before the boundary conditions are applied. Furthermore, in order to satisfy the correct velocity on the surface of the cantilever, an acoustic mode must exist whose axial variation matches the cantilever rather than propagating at the undisturbed speed of sound. Despite this, however, any number of modes exist that do propagate at the undisturbed speed of sound. These modes will be essential to also match the conditions at the left hand end of the cavity, whether or not closed or open. It is this combination of acoustic modes that gives the complicated behaviour seen in the graphs later.

Note each function satisfies the wave equation, i.e.:

$$\frac{\partial^2 p_1}{\partial t^2} - \bar{c}^2 \nabla^2 p_1 = 0, \quad \frac{\partial^2 p_2}{\partial t^2} - \bar{c}^2 \nabla^2 p_2 = 0, \quad \frac{\partial^2 p_3}{\partial t^2} - \bar{c}^2 \nabla^2 p_3 = 0 \quad \text{and} \quad \frac{\partial^2 p_4}{\partial t^2} - \bar{c}^2 \nabla^2 p_4 = 0$$

from which we have:

$$-\frac{T\omega_n^2}{\bar{c}^2} - \left[ \frac{X_1''}{X_1} + \frac{Y_1''}{Y_1} + \frac{Z_1''}{Z_1} \right] = 0 \quad \text{which means} \quad \frac{X_1''}{X_1} \text{ is a constant, as is } \frac{Y_1''}{Y_1}$$

and  $\frac{Z_1''}{Z_1}$ .

A negative constant for  $X_1$  (say  $-k_n^2$ ) gives  $X_1 = C_n \cos(k_n x) + D_n \sin(k_n x)$  from which

$$-\left(\frac{\omega}{c}\right)^2 + k_n^2 = \frac{Y_1''}{Y_1} + \frac{Z_1''}{Z_1} \text{ follows.}$$

For 2D propagation  $\frac{Z_1''}{Z_1} = 0$ . Therefore,  $\frac{Y_1''}{Y_1} - \left(\frac{\omega}{c}\right)^2 + k_n^2$ . Furthermore, the velocity at  $y = d$  is the velocity of the beam. Since this must be true for all values of  $x$ , it must have the same function in  $x$ .

Hence,  $\frac{X_2''}{X_2} = +k_n^2 \rightarrow X_2 = E_n \cosh(k_n x) + F_n \sin(k_n x)$  at  $y = d$  the velocity of the vibrating beam, given by the time derivative of the displacement, should be matched to the vertical acoustic velocity.

5 For  $Y_2$ ,  $k_n < \frac{w}{L}$  so

$$Y_1 = \cos(k_n y) \quad \text{so} \quad u_y = \frac{\partial p}{\partial y} = 0 \quad \text{on } y = 0$$

and  $Y_2 = \cosh(\alpha_n y)$

10

where  $\alpha_n = \sqrt{\left(\frac{w}{L}\right)^2 - k_n^2}$ , which gives

$$p = e^{i\omega t} \left[ (C_n \cos(k_n x) + D_n \sin(k_n x)) \cos(\alpha_n y) + (E_n \cosh(k_n x) + F_n \sinh(k_n x)) \cosh(\alpha_n y) \right]$$

15 At  $y = d$   $u = \frac{\partial w}{\partial t} = \frac{-1}{\rho i \omega} \frac{\partial p}{\partial y} ;$

$$\rho w^2 A_n [\cos(k_n x) - \cosh(k_n x)] + B_n [\sin(k_n x) - \sinh(k_n x)] = -\alpha_n C_n \cosh(k_n x) \sin(\alpha_n d) + \alpha_n E_n \sinh(\alpha_n d) \cosh(\alpha_n x) - \alpha_n D_n \sin(\alpha_n d) \sin(k_n x) - \alpha_n F_n \sinh(\alpha_n d) \sinh(\alpha_n x)$$

20

$$C_n = \left[ \frac{-\rho w^2}{\alpha_n \sin(\alpha_n d)} \right] A_n$$

$$E_n = \left[ \frac{-\rho\omega^2}{\alpha_n \sinh(\alpha_n d)} \right] A_n$$

$$D_n = \left[ \frac{-\rho\omega^2}{\alpha_n \sin(\alpha_n d)} \right] B_n$$

$$5 \quad D_n = \left[ \frac{-\rho\omega^2}{\alpha_n \sinh(\alpha_n d)} \right] B_n$$

i.e.

$$10 \quad p_1 + p_2 = \frac{\rho\omega^2 A_n e^{i\omega t}}{\alpha_n} \left[ \left( \frac{\cos(k_n L) + \cosh(k_n L)}{\sin(k_n L) + \sinh(k_n L)} \right) \left( \frac{\sin(k_n x)}{\sin(\alpha_n d)} + \frac{\sinh(k_n x)}{\sinh(\alpha_n d)} \right) \right. \\ \left. - \left( \frac{\cos(k_n x)}{\sin(\alpha_n d)} + \frac{\cosh(k_n x)}{\sinh(\alpha_n d)} \right) \right]$$

and the acoustic velocity induced by the beam alone (essentially with no cavity to further constrain the acoustic waves) is:

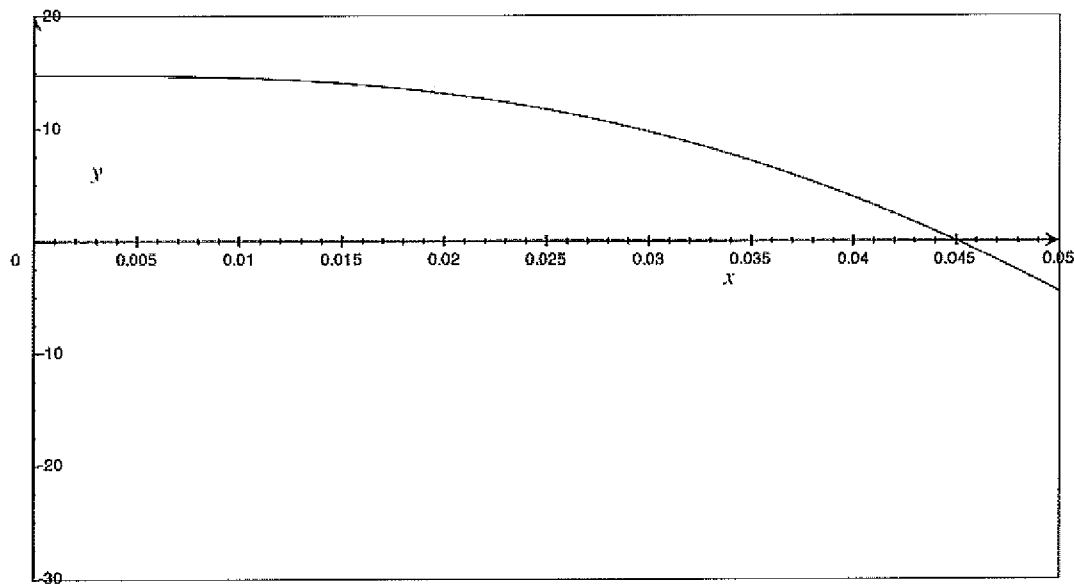
15

$$u_{beam} = u_1 + u_2 = \frac{-i\rho\omega k_n A_n e^{i\omega t}}{\alpha_n} \left[ \left( \frac{\cos(k_n L) + \cosh(k_n L)}{\sin(k_n L) + \sinh(k_n L)} \right) \left( \frac{\cos(k_n x)}{\sin(\alpha_n d)} + \frac{\cosh(k_n x)}{\sinh(\alpha_n d)} \right) \right. \\ \left. + \left( \frac{\sin(k_n x)}{\sin(\alpha_n d)} - \frac{\sinh(k_n x)}{\sinh(\alpha_n d)} \right) \right]$$

20

See figures 1-3 for the acoustic velocity modes induced by the first three bending modes. It should be noted here that the acoustic velocities described by the graphs of figures 1-3 given immediately below are the induced acoustic velocities in the axial direction and, therefore, not the displacement of the beam. The graphs of figures 1-3 given immediately below represent the beam forcing alone.

- 5 Additional constraints to the cavity, such as closing the end, would alter this velocity.. Notice that the important first bending mode naturally induces a velocity at the end of the beam (which would be zero if closed). This induces the velocities seen in figure 4.



10 Figure 1: X-axis axial position, y-axis deflection (for arbitrary driving). First vibrational mode

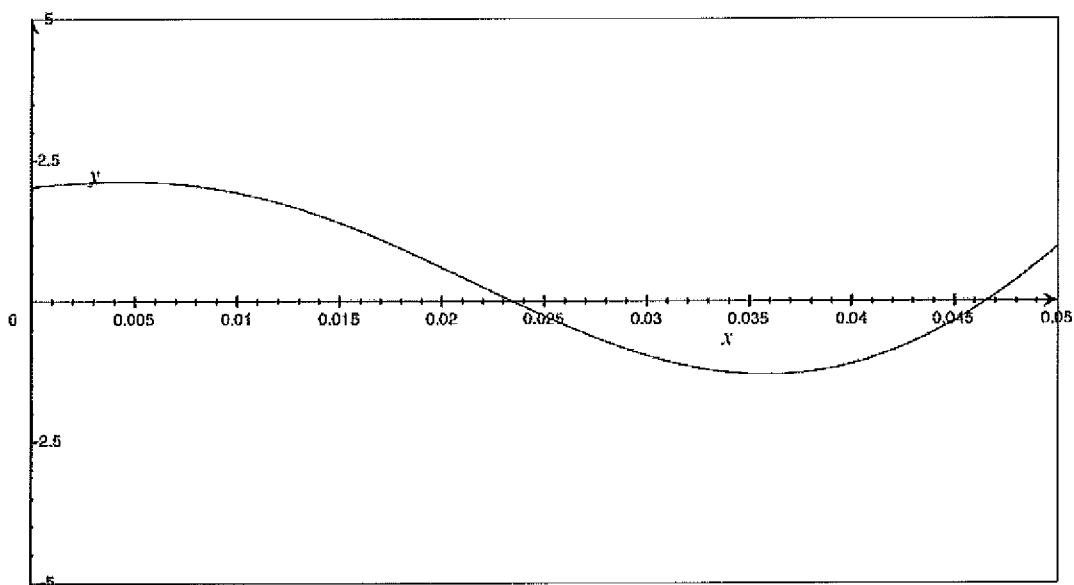


Figure 2: Second vibrational mode

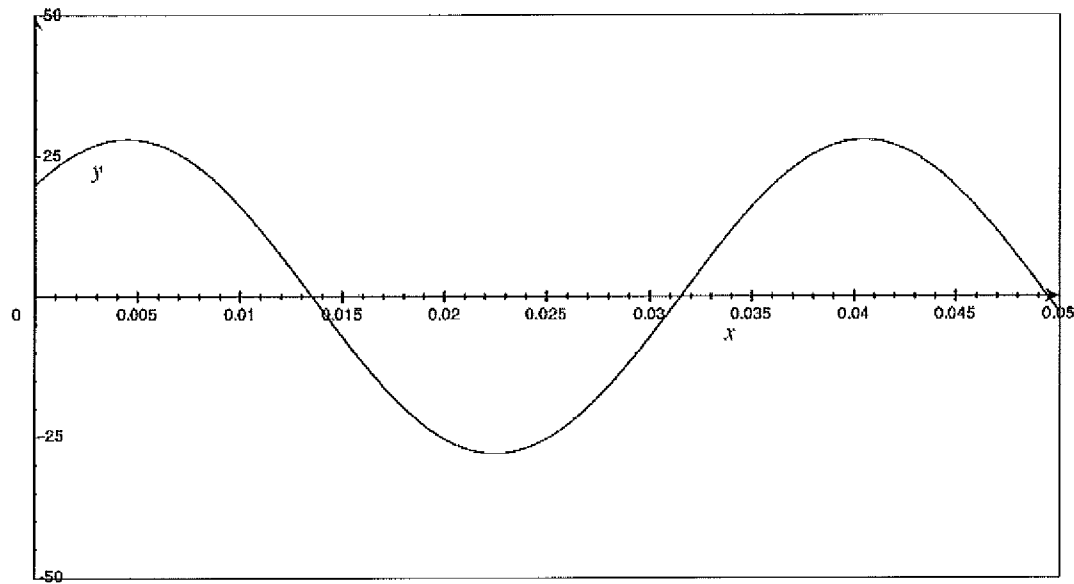


Figure 3: Third vibrational mode.

5

In the event that the ends are constrained acoustically, such as by closing the system, one must add:

$$u_{ac} = \frac{-i\rho\omega k_n A_n e^{i\omega t}}{\alpha_n} \left[ \left( \frac{\cos(k_n L) + \cosh(k_n L)}{\sin(k_n L) + \sinh(k_n L)} \right) \left( \frac{\cos(k_n x)}{\sin(\alpha_n d)} + \frac{\cosh(k_n x)}{\sinh(\alpha_n d)} \right) + \left( \frac{\sin(k_n x)}{\sin(\alpha_n d)} - \frac{\sinh(k_n x)}{\sinh(\alpha_n d)} \right) \right] + A e^{-i\omega x/c} + B e^{i\omega x/c}$$

10 For open at the other end we obtain figure 4 (which is closed at  $x = 0$ , hence the zero velocity).

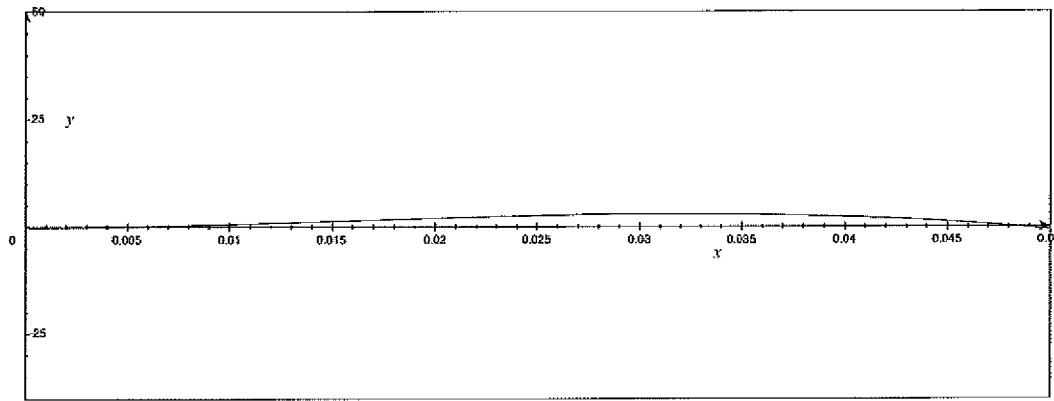


Figure 4: Acoustic velocity in the x-direction. In this case, the vibrational mode of the beam induces axial (x-) velocities in the cavity. However, the cavity is constrained by a closed end and so an additional acoustic mode is set up in the cavity such that the overall flow field satisfies all of the boundary conditions especially no flow at the closed end and matching y-velocity on the vibrating cantilever.

The frequency response for any of these can be obtained by combining the equations above with a particular integral to give the following graph:

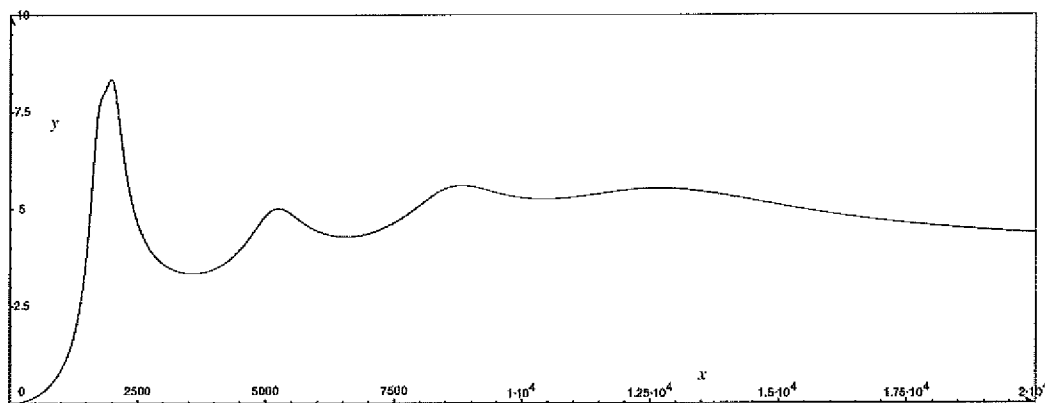
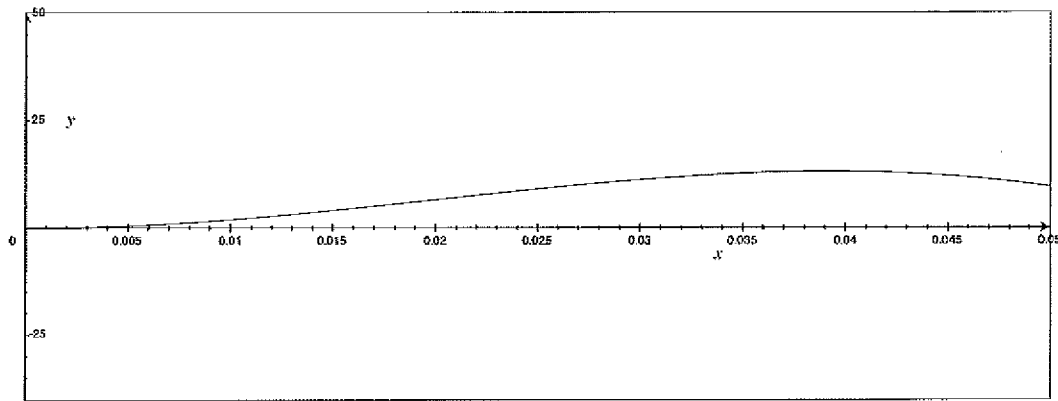


Figure 5: Frequency response. X-axis driving frequency, The y-axis here specifies the acoustic response and is proportional to the maximum x-velocity in the cavity. The precise amplitude will depend upon the magnitude of the driving, but the figure does indicate the response seen as a function of frequency for equal driving at all frequencies. First peak is merge of vibrational mode and acoustic, subsequent peaks are damped.

One skilled in the art appreciates that:

- 1) The interaction is a complex one with the coupling resulting in a separation for the acoustic mode (the quarter wave acoustic resonance cancels with the vibrational one rather than augments.)
- 2) An understanding of the equations is required for an understanding of how to control this. For example, by partially closing the open end, we require higher order modes with acoustic velocities (not shown in the graph) which are normal to the axis along the cavity. Motion in both orthogonal directions are possible. This alters the wavenumber, leading to, for example, flows like figure 6, which is more desirable because it gives large exit

5 velocities (at the right hand end of figure 6) whilst allowing the left hand end to be closed, giving the designer more freedom in the design. Partially closing off the open end is easy to do practically. To obtain desirable velocities such as in the graph of figure 6, we can alter the cavity dimensions. By choosing the cavity dimensions and the material properties of the cantilever independently, one skilled in the art can independently choose the optimal driving frequency. A practical issue is that it is mechanically challenging to get the mechanical frequencies high enough to match the first(higher order) acoustic mode, so if we have to drive it even faster then this will be even more difficult.



10 Figure 6. Acoustic axial velocity with partially closed boundary.

### Claims

- 5 1. A device comprising a chamber bearing an actuable member, having an actuating boundary to excite an acoustic mode of the fluid in the chamber, and a port; characterised in that the device comprises a control system having an actuator to actuate the actuable member, the control system and actuable member being arranged to excite or induce a first or higher order acoustic mode by periodically varying the geometry of the chamber.
- 10 2. A device as claimed in claim 1 wherein the actuable member comprises an elongated member clamped at one end in a cantilever arrangement.
3. A device as claimed in claim 2, wherein the elongate member comprises a rectangular plate.
4. A device as claimed in any preceding claim, wherein the actuable member has at least a portion thereof having a substantially uniform cross-section.
- 15 5. A device as claimed in any preceding claim, wherein the actuable member has at least a portion thereof having a varying cross-section.
6. A device as claimed in claim 5, wherein the varying cross-section forms a progressively tapering actuating member towards a displacement end of the actuable member.
- 20 7. A device as claimed in any preceding claim wherein the chamber has a longitudinal axis and at least one transverse axis; the longitudinal axis and traverse axes being arranged such that a frequency of a first order acoustic mode along the longitudinal axis is distinct and lower than a frequency of a first order along said at least one transverse axis.
- 25 8. A device as claimed in any preceding claim, wherein the actuating member bears a first acoustic impedance and a fluid within the chamber bears a second acoustic impedance; the first and second impedances having a predeterminable relationship.
9. A device as claimed in claim 8, wherein the first impedance substantially matches the second impedance.
10. A device as claimed in any preceding claim, wherein the chamber comprises at least one port.
- 30 11. A device as claimed in claim 10, wherein the chamber comprises at least a pair of ports.
12. A device as claimed in claim 11, wherein a first port of the pair of ports is an inlet port and a second port of the pairs of ports is an outlet port.
- 35 13. A device as claimed in any preceding claim, wherein the chamber comprises a second actuable member bearing a respective, second, actuating boundary.

14. A device as claimed in claim 13, wherein the control system comprises a second actuator arranged to actuate the second actuable member.
15. A device as claimed in either of claims 13 and 14 wherein the control system is operable to induce anti-phase movement of the first and second actuable members.
- 5 16. A device as claimed in any preceding claim, wherein the chamber comprises at least one fixed boundary.
17. A device as claimed in claim 16, wherein the chamber comprises at least two fixed boundaries.
- 10 18. A device as claimed in any preceding claim, wherein the chamber comprises at least one open boundary.
19. A device as claimed in claim 18, wherein the chamber comprises a pair of open boundaries.
20. A device as claimed in any preceding claim, wherein the actuator comprises at least one of an electromagnetic actuator and a piezoelectric actuator.
- 15 21. A device as claimed in any preceding claim, wherein the actuator is disposed distally relative to a displacement end or proximally relative to a fixed end of the actuating member.
22. An array of devices comprising a number of devices as claimed in any preceding claim.
- 20 23. An array as claimed in claim 22, wherein the control system is adapted to actuate actuable members of adjacent devices antisymmetrically.
24. An array as claimed in either of claims 22 and 23, wherein the number of devices comprises a linear array of said devices.
- 25 25. An array as claimed in claim 24, wherein the number of devices comprises an  $n \times m$  array of said number of devices.
26. A control or actuation surface bearing an array of devices as claimed in any preceding claim.
27. A device substantially as described herein with reference to and/or as illustrated in the accompanying drawings.
- 30 28. A method substantially as described herein with reference to and/or as illustrated in the accompanying drawings.
29. A control system substantially as described herein with reference to and/or as illustrated in the accompanying drawings.

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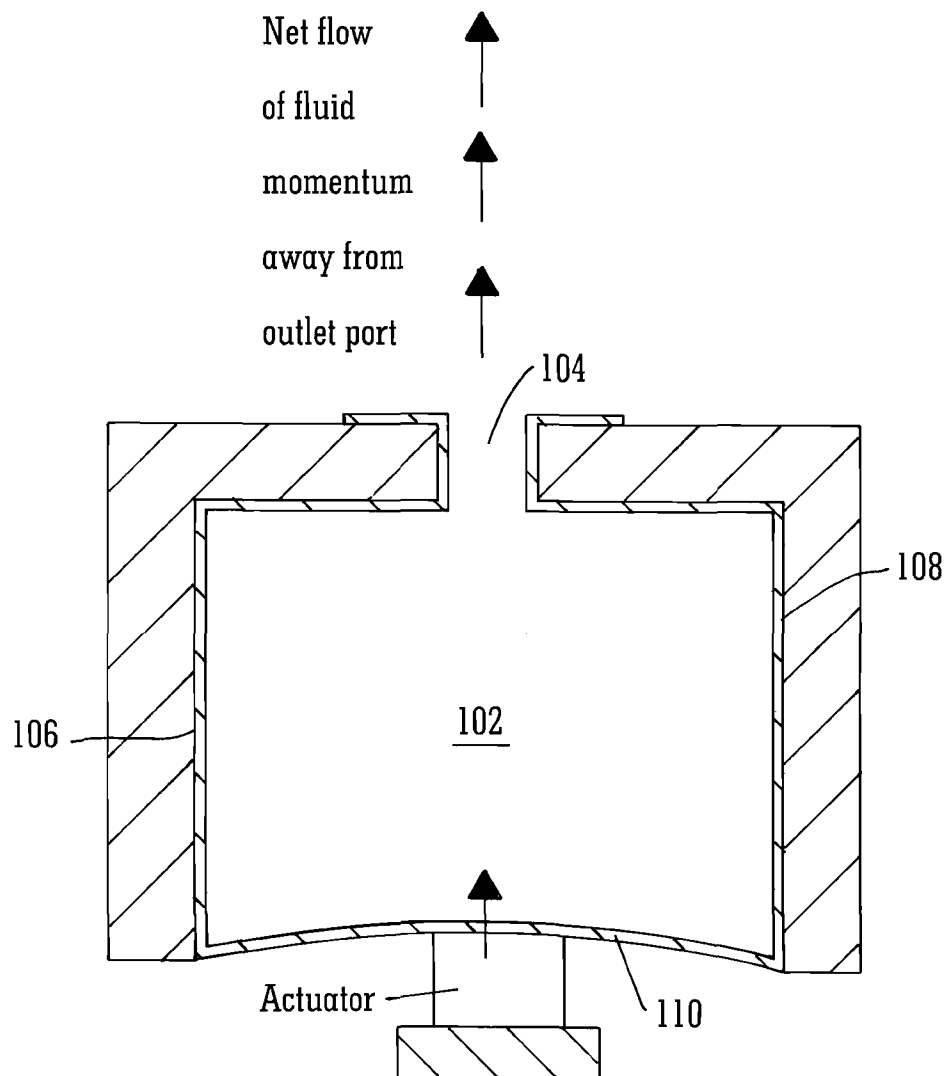


FIG. 1

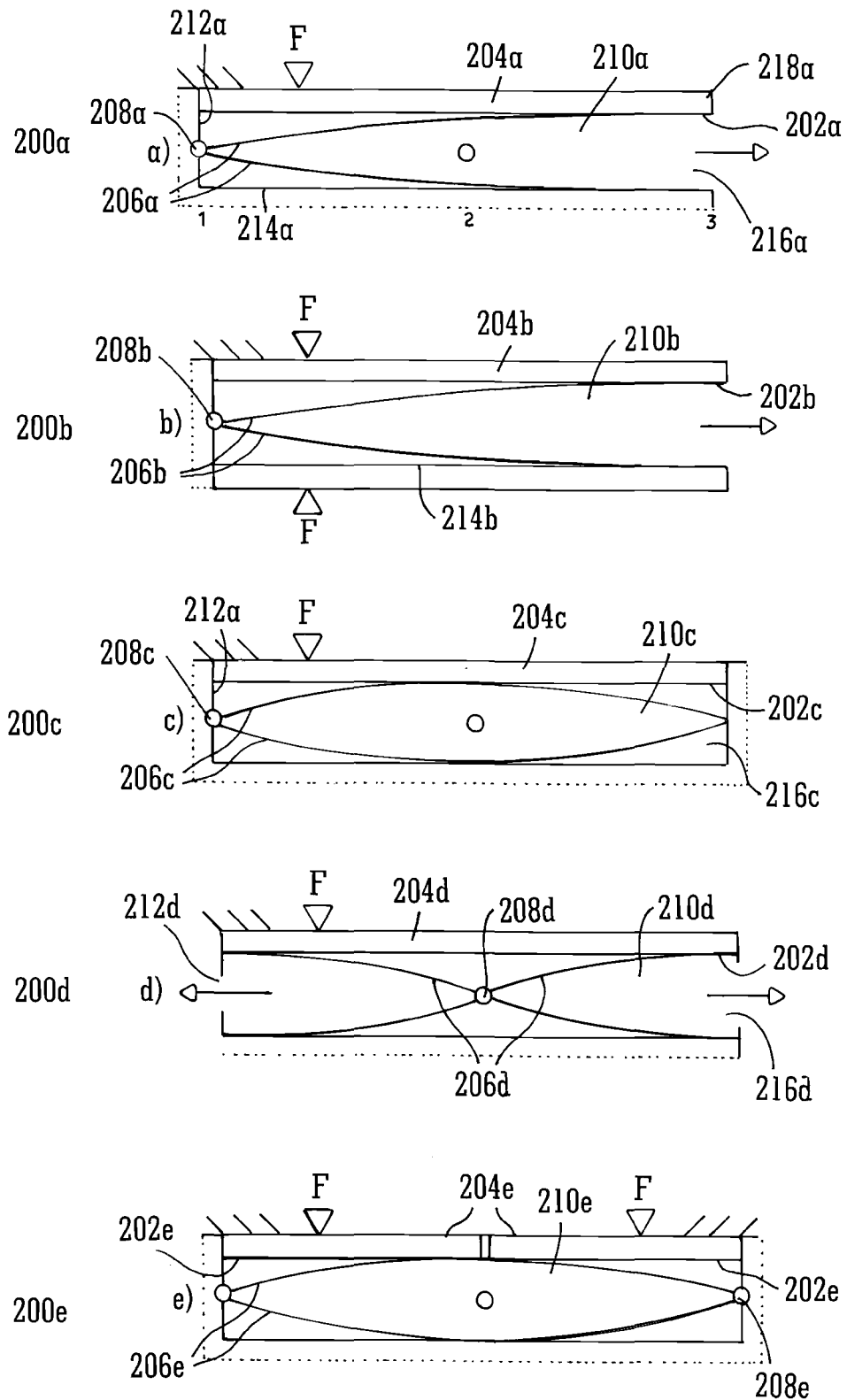


FIG. 2

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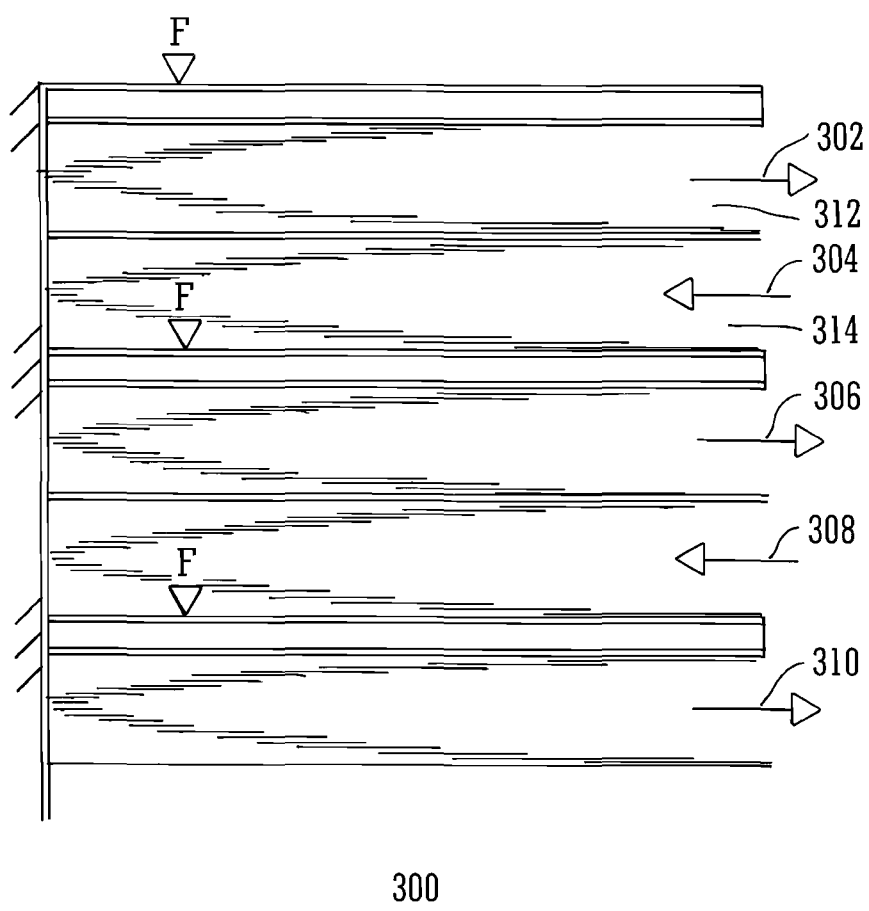


FIG. 3

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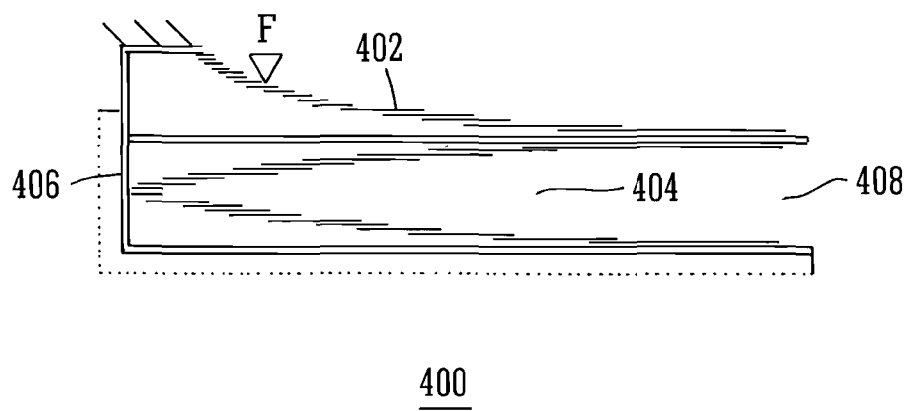


FIG. 4

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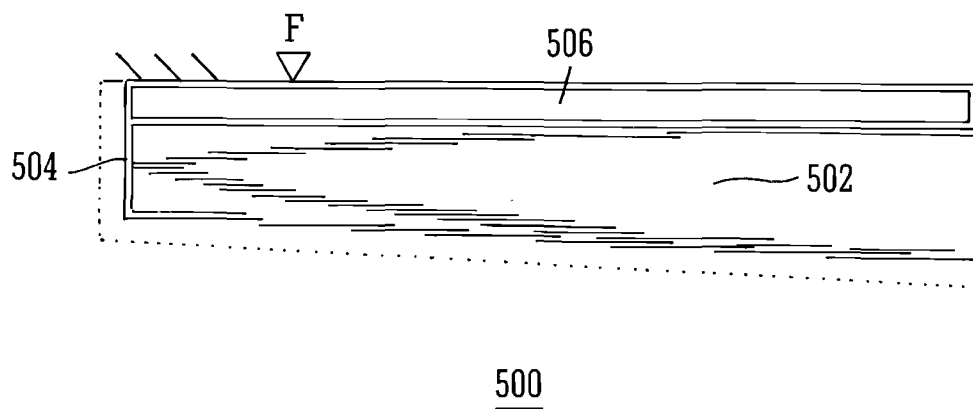


FIG. 5

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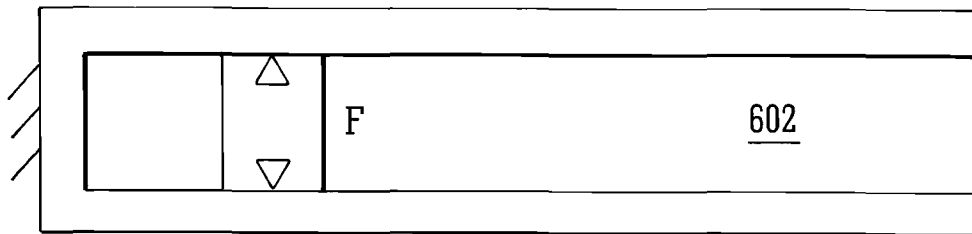
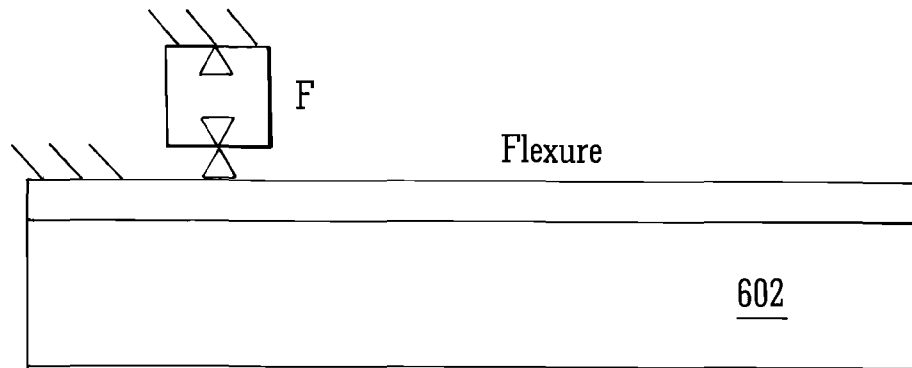


FIG. 6a

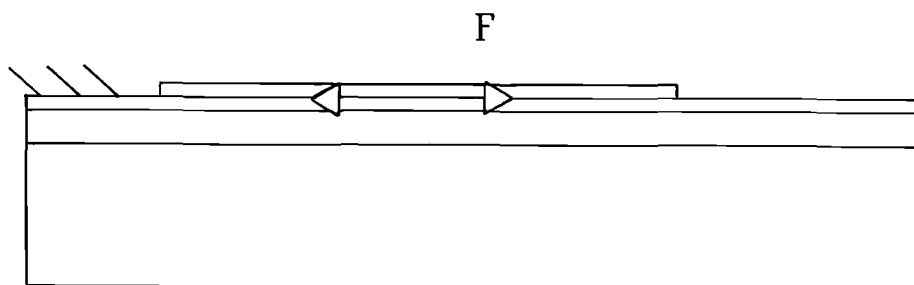


FIG. 6b

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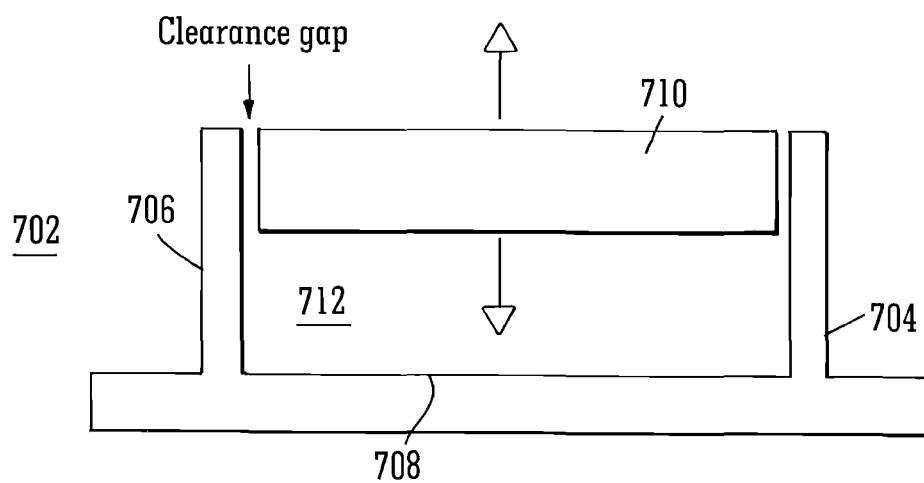


FIG. 7a

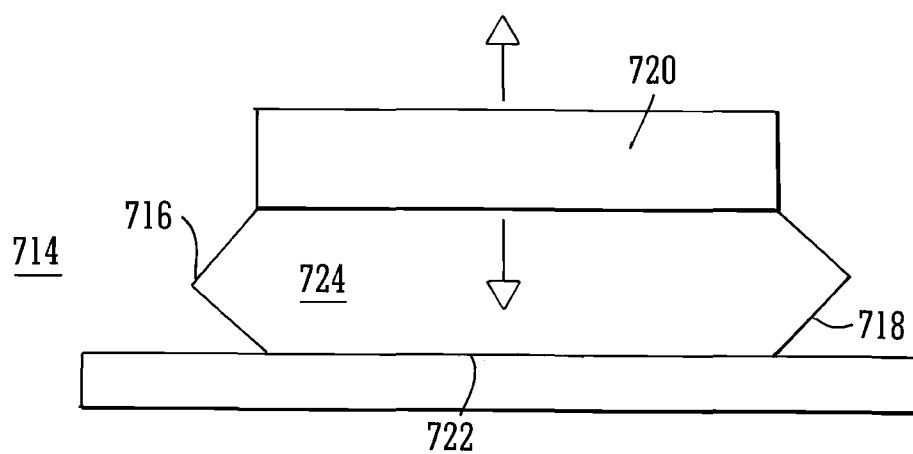


FIG. 7b

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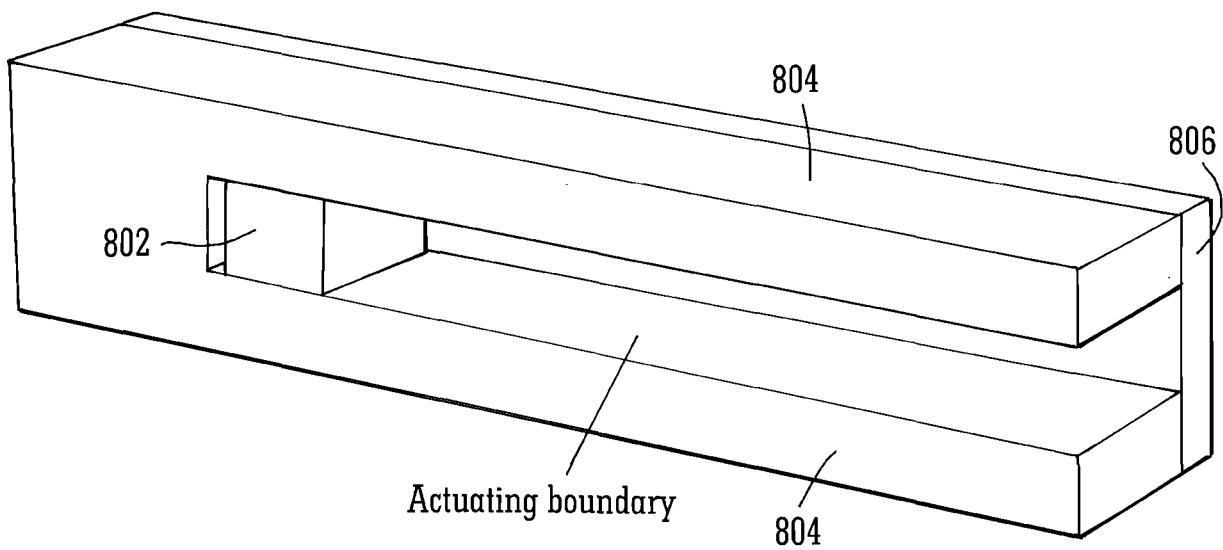
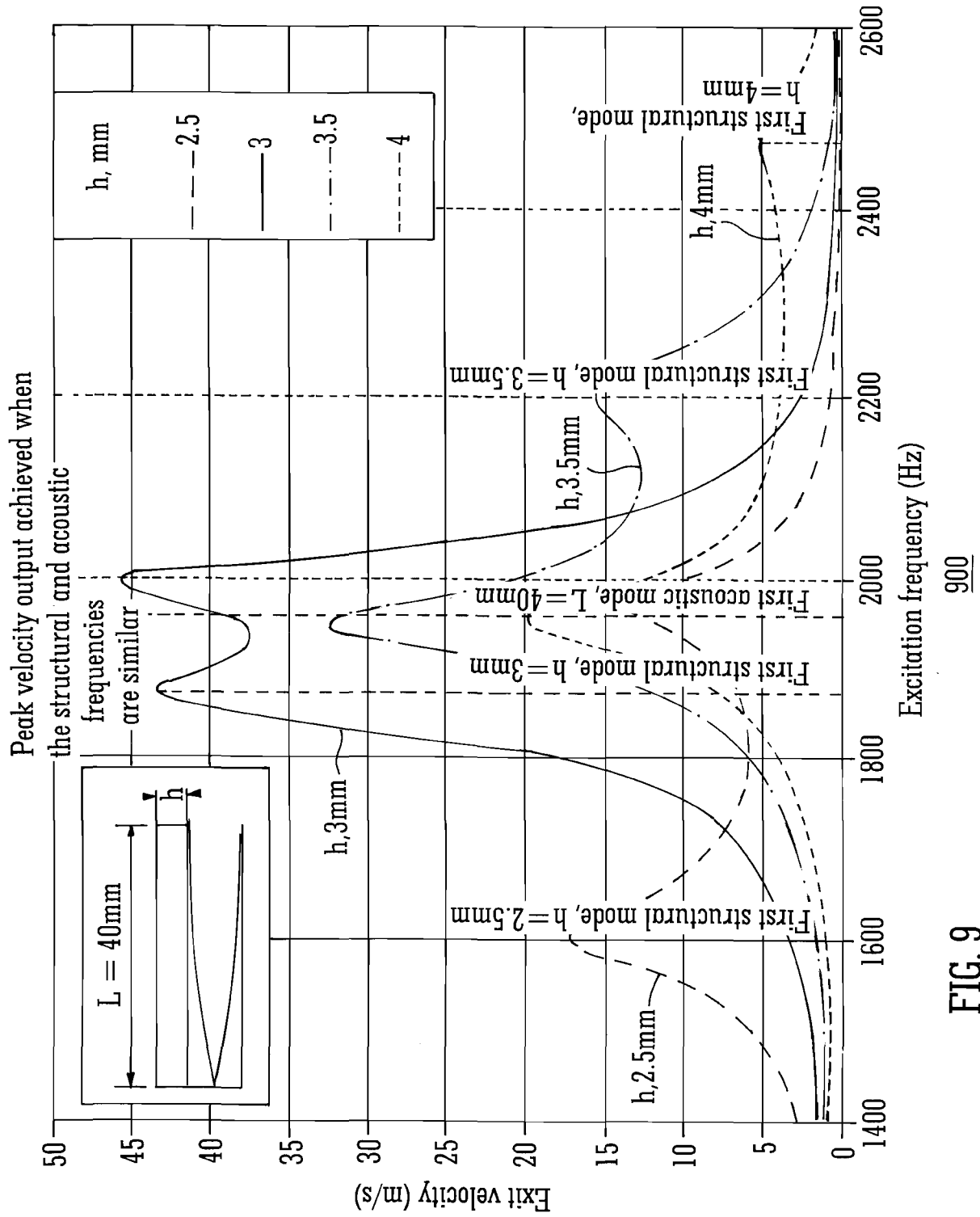


FIG. 8



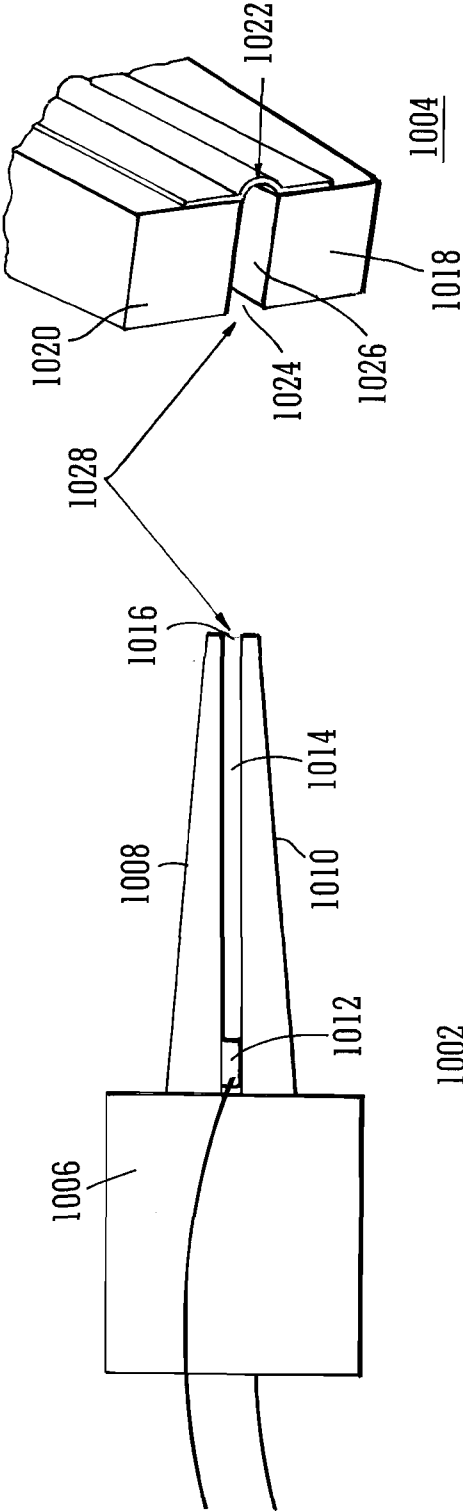


FIG. 10

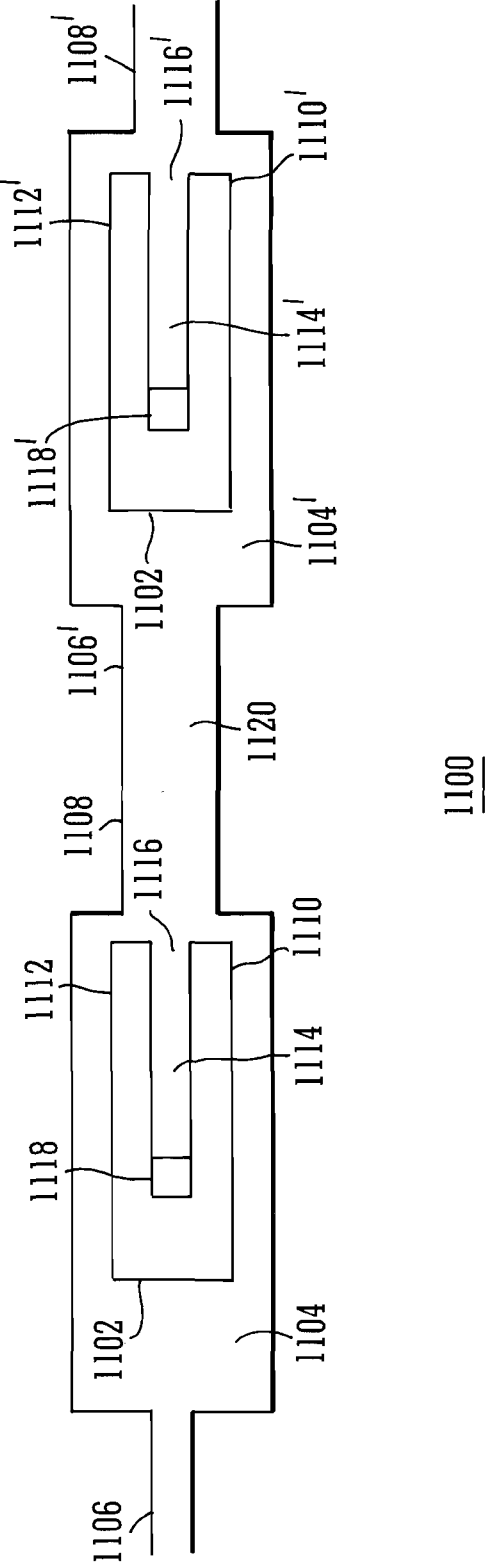


FIG. 11

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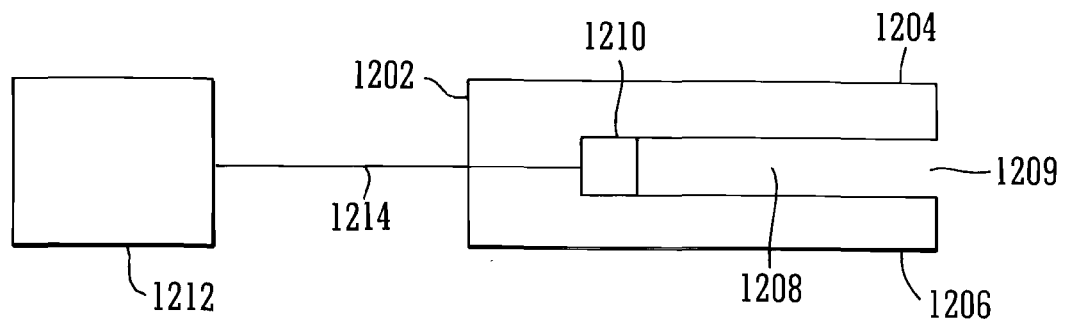


FIG. 12

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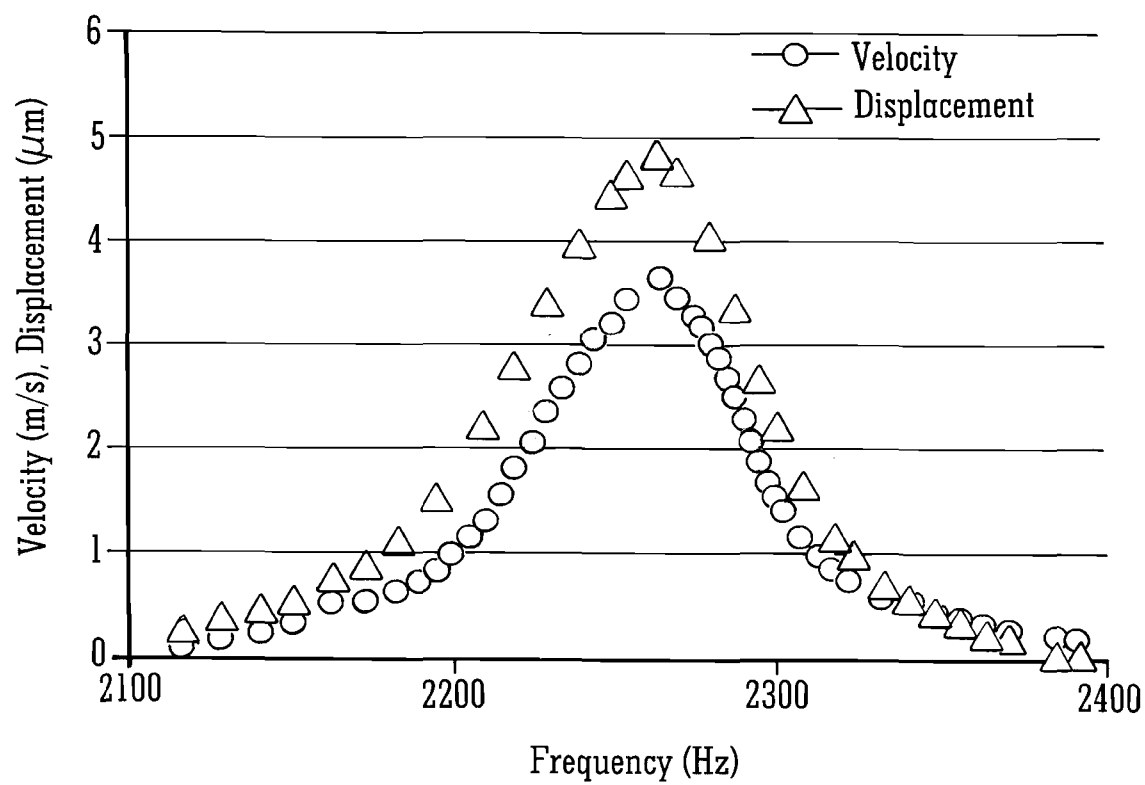


FIG. 13

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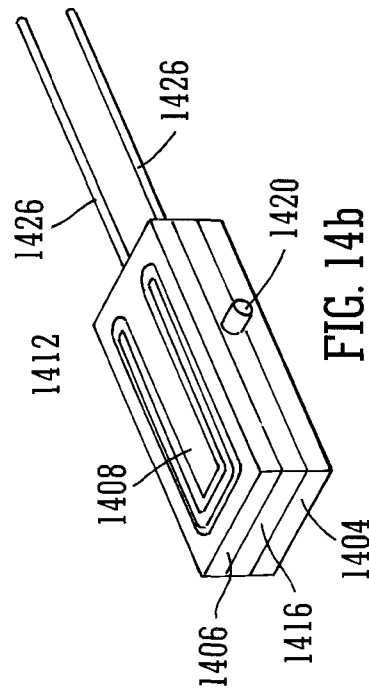


FIG. 14b

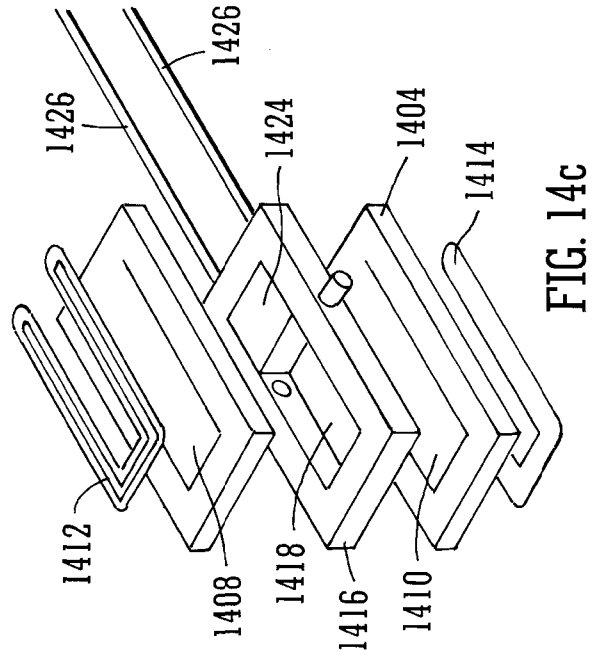


FIG. 14c

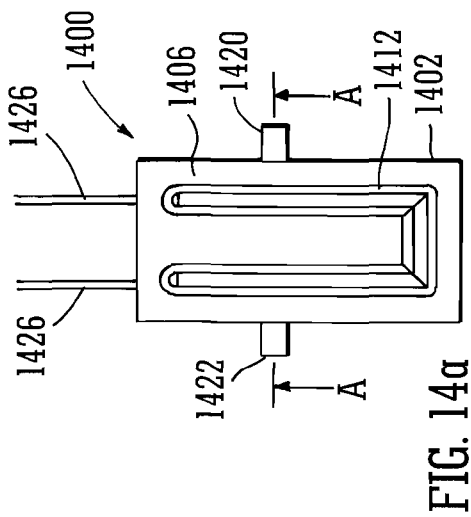


FIG. 14a

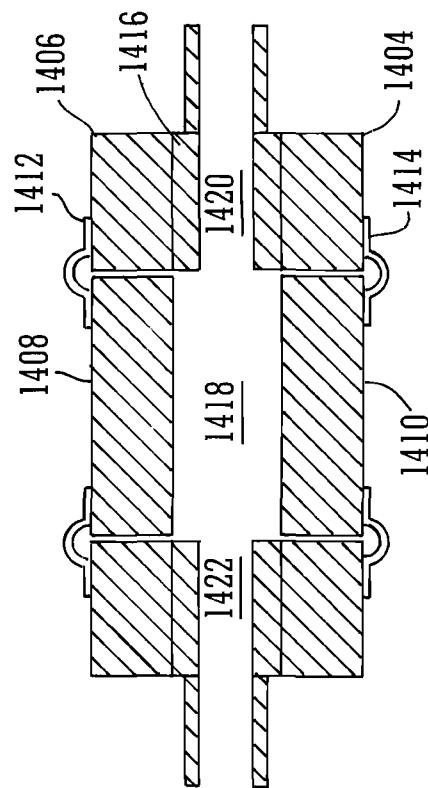


FIG. 14d

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