May 8, 1962 A. G. BODINE, JR 3,033,158

APPARATUS FOR TRANSMITTING SONIC VIBRATIONS INTO LIQUID BODIES

Filed Feb. 17, 1960 3 Sheets-Sheet 1

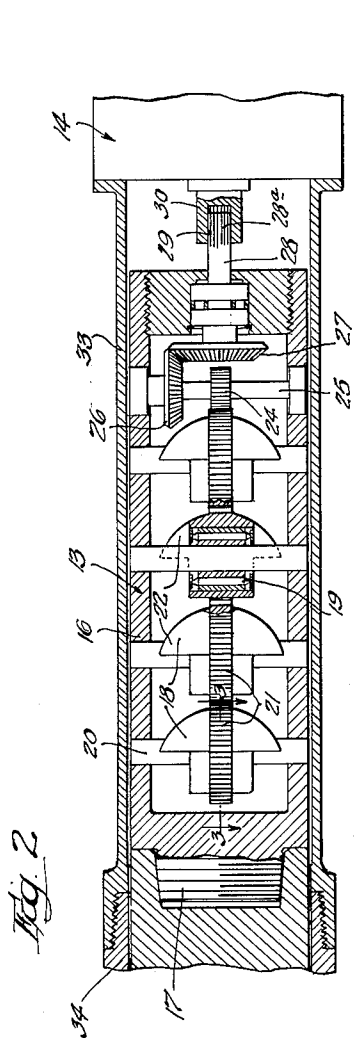


Fig. 2

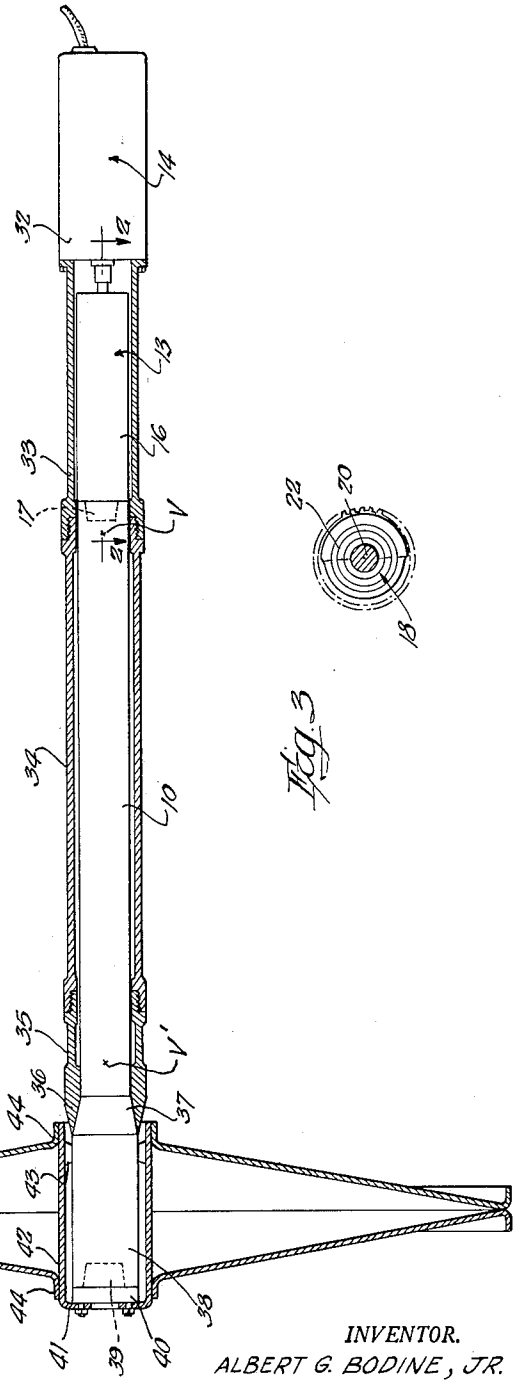


Fig. 1

Fig. 3

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APPARATUS FOR TRANSMITTING SONIC VIBRATIONS INTO LIQUID BODIES

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Fig. 4

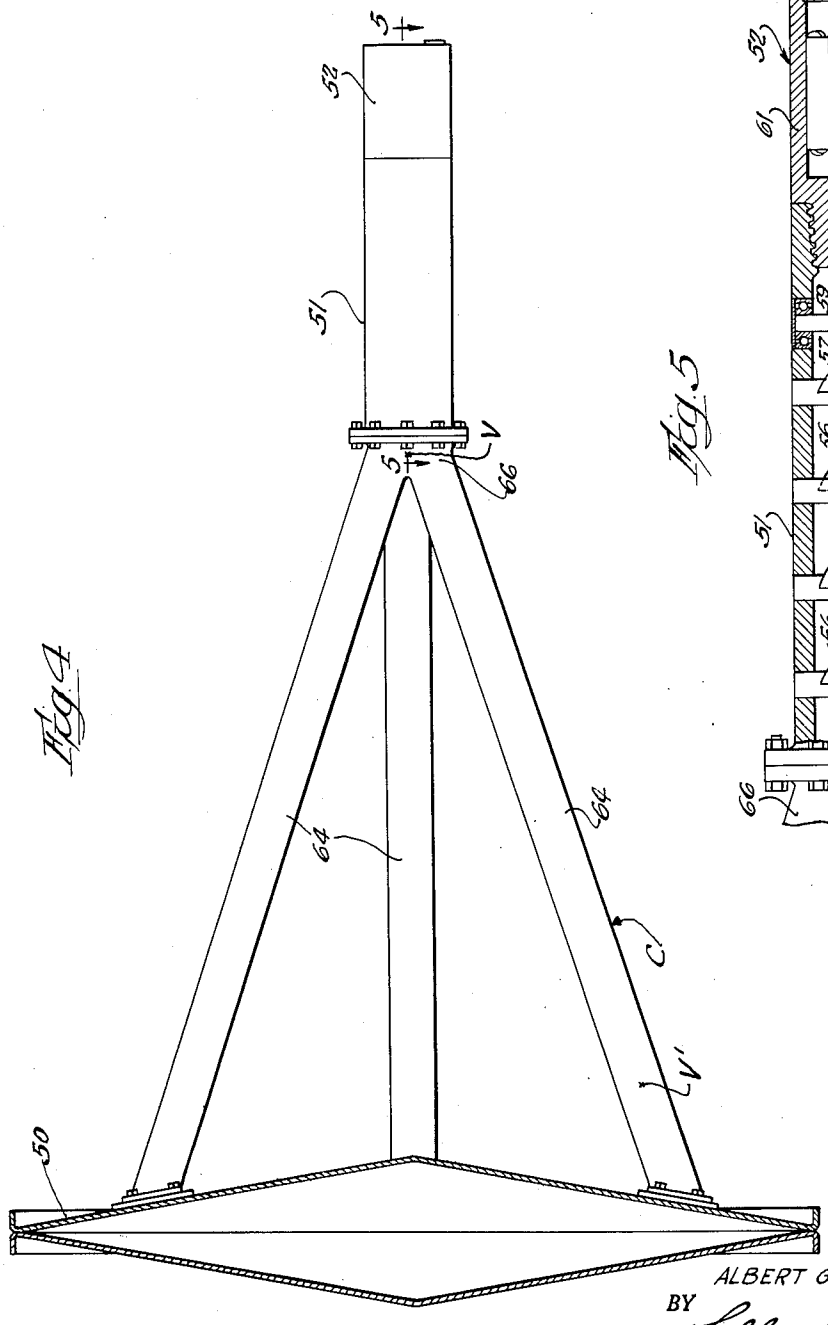
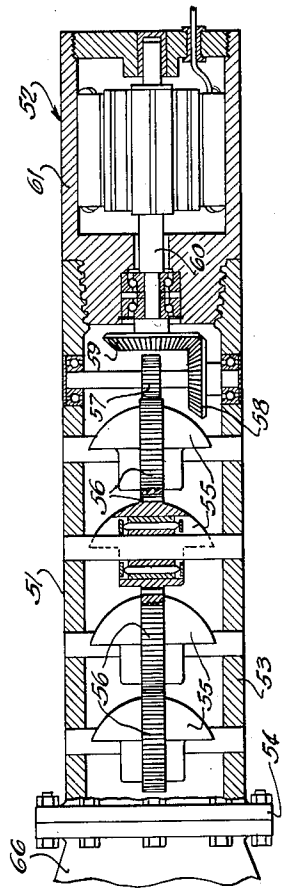


Fig. 5



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APPARATUS FOR TRANSMITTING SONIC VIBRATIONS INTO LIQUID BODIES

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Fig. 6

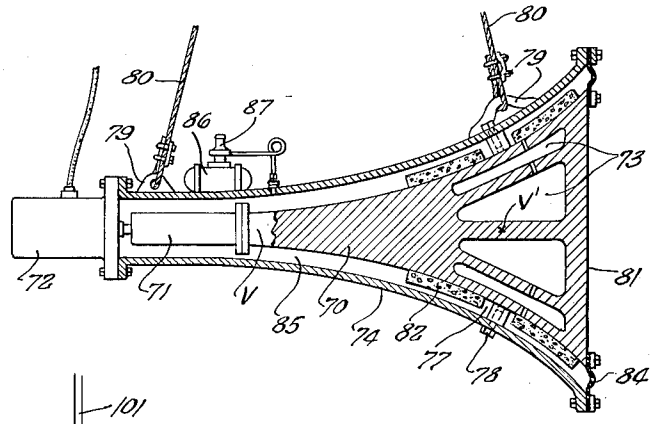


Fig. 7

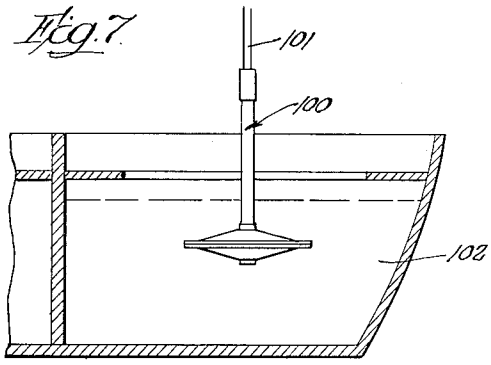


Fig. 8

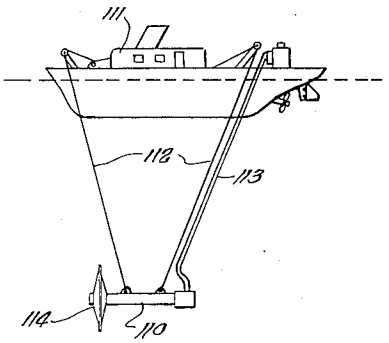
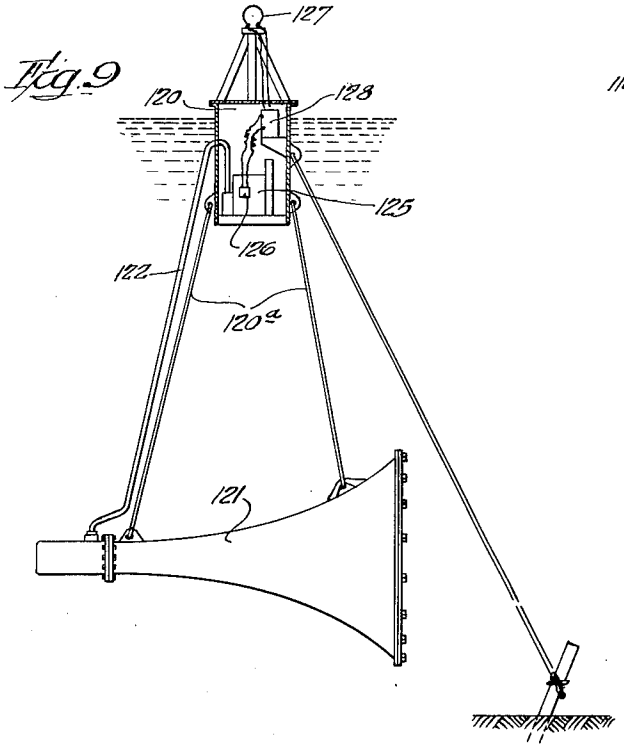


Fig. 9



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3,033,158
APPARATUS FOR TRANSMITTING SONIC VIBRATIONS INTO LIQUID BODIES

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Filed Feb. 17, 1960, Ser. No. 9,349
7 Claims. (Cl. 116—137)

This invention relates generally to high power acoustic apparatus for generating high energy sound waves of relatively low frequency in liquids, for such purposes as sonic liquid treatment in large process tanks, sonically cleaning large liquid tanks, etc. The invention is useful also for transmitting high energy pressure waves for substantial distances through extended bodies of water, such as the ocean, to create an underwater sound field useful in submarine detection.

This application is a continuation-in-part of my copending application Serial No. 825,117, filed July 6, 1959, entitled Method and Apparatus for Generating and Transmitting Sonic Vibrations, now Patent No. 2,960,314, which was a continuation-in-part of my application Serial No. 484,627, filed January 28, 1955, now abandoned.

The invention contemplates very high sonic power, relatively low frequency, which, without intention of limitation, may be typically between 200 and 2000 cycles per second, and rugged mechanical oscillators for generating the sound waves.

Objects of the invention include the provision of improved sonic processing or wave generating apparatus characterized by extraordinary power, relatively low frequency, simple and rugged mechanical oscillators of low cost but high reliability, unusually high efficiency for a sonic processing machine, excellent frequency stability, and ready portability.

In accordance with the invention, there is provided a vibratory sound wave radiator in coupling contact with the liquid, preferably in the general form of a flat cone; and to assure good loading of the radiator by the liquid, the dimension across the surface of the radiator is made at least a major fraction of a wave length of the sound wave to be radiated thereby, measured in the liquid. An optimum dimension is substantially one wave length, and a dimension up to two wave lengths, or even more, might in some cases be used, though at such dimension the radiator may be overly large for practical purposes, for reasons that will appear.

Assuming a radiator of optimum dimensions, therefore, i.e., one wave length across, and assuming a frequency of 400 cycles per second, which is a desirable frequency for many applications, the diameter of the radiator, using a value of 4800 feet per second for the velocity of sound in water, is of the order of 12 feet. A vibratory radiator of such scale, immersed in liquid, is heavily loaded, and has a high loaded mechanical impedance. By definition, the mechanical impedance of the liquid-loaded radiator is the complex quotient of the driving force on the radiator divided by the linear velocity. This factor for such a radiator, as described, is quite high.

One problem presented is to drive this high impedance radiator effectively. Simple mechanical oscillators or force generators of high power types do not characteristically operate at the necessary high output impedance for reasonable match to the impedance of the radiator. One type of oscillator that is eminently suitable, excepting for this problem, particularly for the high power and frequency range in contemplation, is disclosed in several forms in my aforementioned U.S. Patent No. 2,960,314. This oscillator is of a mechanical type, involving a cyclically driven relatively small inertia mass body which is oscillative at relatively high velocity in a predetermined path relative to a body of substantially larger mass which

supports it through suitable constraining means, such as a bearing. The small oscillative mass reacts through such bearing means to exert an reactive oscillating force on the supporting body, causing it to vibrate through a small amplitude. Because of the relatively large mass of the supporting body as compared with the mass of the small oscillative body, there is a large velocity reduction, and correlative force gain, between the small oscillative body and the supporting body. In consequence, the force generator so constituted has a fairly ample output impedance; however, it is still so poorly adjusted to the loaded impedance of the radiator that effective direct drive of the latter by the generator would not be feasible.

In addition, and as a further major consideration, a force generator of the class described can, in accordance with principles disclosed in my aforementioned U.S. Patent No. 2,960,314, be very advantageously controlled to operate at the selected resonant frequency, with good frequency stability, high Q, and an important energy storage property (fly-wheel effect), by coupling it directly to an elastically vibratory half-wavelength bar structure, which vibrates at the desired operating frequency in a resonant standing wave pattern, and which is in turn coupled to the device to be driven, in this case, the sound wave radiator. The coupling between the generator and this half-wave standing wave bar structure is made to a velocity antinode at one end of the latter, where impedance is not too large, and is, in fact, fairly well adjusted to the output impedance of the generator.

Under such conditions the oscillator assembly moves with large vibratory velocity, and thus has high power output. The velocity antinode at the opposite end of the bar structure may then be coupled to the device to be driven. By coupling to a velocity antinode of the bar structure, where impedance is not too high, and is of the order of that of the generator, an important advantage is gained, in that the bar structure, vibrating at substantial amplitude at the antinode coupling point, exerts a back reaction through the supporting body of the generator and its bearing means, or other constraining means for the small oscillative inertia body of the generator, which, further supposing a properly governed drive effort on the oscillative body, constrains that body to oscillate at closely controlled frequency which is just under the frequency for peak resonance. It is critically important that the driving force on the oscillative body be less than that which would take the oscillative mass to or over the basic resonant frequency of the system. Under such conditions, operating frequency is stabilized, and a fairly high frequency operation made possible for a powerful form of mechanical oscillator.

There still remains, however, the problem of impedance adjustment between that of the force generator and the high impedance of the large liquid loaded radiator. The half-wavelength resonant standing wave bar, disclosed in my U.S. Patent No. 2,960,314 and referred to hereinabove, permits effective operation of the generator, but does not, without further improvement, furnish a sufficiently high impedance at its output end for effective drive of this large radiator.

A prerequisite to a good understanding of the present invention, in the aspect of meeting the problem stated in the preceding paragraph, is a full understanding of a half-wavelength standing wave bar, and its impedance characteristics, and this subject will next be given attention.

Assume an elongated steel (elastic) bar, of uniform cross section, suspended in a "free-free" state, i.e., supported at its mid-point, and free for longitudinal vibration at both ends. Assume further an alternating force generator coupled to one end of the bar, so as to apply a sinusoidal alternating force thereto in a direction longitudinally of the bar, and at a frequency f equal to $s/2L$,

where s is the speed of sound in the material of the bar, and L is the length of the bar. Under these conditions, alternating sinusoidal waves of compression and tension are launched by the force generator into the bar. These travel the length of the bar, are reflected from the far end thereof as waves of unlike kind (i.e., a wave of compression is reflected as a wave of tension, and vice versa), and by virtue of interference between the original and reflected waves when $f=s/2L$, a half-wavelength resonant standing wave is established. Under these conditions, longitudinal wave amplitude is cancelled at the mid-point of the bar, and is magnified at the two ends thereof. In effect, the two half portions of the bar alternately elastically elongate and contract, in step with one another, the magnitude of elastic elongation and contraction increasing progressively from zero, or substantially so, at the mid-point to maximum at the ends. The condition at the mid-point of the bar is called a velocity node, and is characterized by minimized cyclic velocity amplitude, and by maximized cyclic stress amplitude. The condition at each end of the bar is called a velocity antinode, and is characterized by maximized cyclic velocity amplitude, and minimized cyclic stress amplitude.

It will be seen that the mechanical impedance of the velocity antinodes of the halfwave bar is relatively low, and at the velocity node thereof is extremely high. There is a gradual transition of mechanical impedance from high at the velocity node to low at the velocity antinodes. At the velocity antinodes, the impedance is well adjusted to the output impedance of a mechanical cyclic force generator of the type herein above mentioned, but is far too low for effective drive of the liquid loaded sound radiator.

The present invention meets this last problem by reducing the quarter-wavelength portion of the bar between the velocity node and the coupling to the radiator to 5–10% of its original quarter-wavelength, or in other words to the approximate range of typically $\frac{1}{20}$ to $\frac{1}{40}$ wavelength. The extremity of the shortened one-fifth to one-tenth wavelength portion of the bar is then directly coupled to the sound wave radiator. The bar in this form, so coupled to the radiator, still operates with a standing wave pattern, at the original resonant frequency, but the pattern from the node out to the coupling point to the radiator is a small fraction of a quarter-wavelength. And at the extremity, which is close spaced to the node, the mechanical impedance is very high, and of the order of that of the liquid loaded radiator.

Thus I have accomplished the difficult problem of driving a high impedance liquid loaded sound radiator of large effective area from a simple mechanical oscillator of the type mentioned, with good impedance adjustment between the generator and the radiator, and with preservation of the desirable back reaction effect which stabilizes the force generator and holds it closely at the predetermined operating frequency, typically just slightly under the peak resonance frequency for the system as a whole.

With reference now to the drawings:

FIG. 1 is a view of an illustrative apparatus in accordance with the invention, partly in side elevation, and partly in longitudinal medial section;

FIG. 2 is an enlarged section on line 2—2 of FIG. 1;

FIG. 3 is a section on line 3—3 of FIG. 2;

FIG. 4 is a side elevational view of a modification of the apparatus of FIG. 1, the radiator being shown in section on a diameter thereof;

FIG. 5 is an enlarged longitudinal section on line 5—5 of FIG. 4;

FIG. 6 is a side elevation of another modification, parts being shown in longitudinal medial section;

FIG. 7 shows an apparatus in accordance with the invention in combination with a liquid body in a tank;

FIG. 8 shows an apparatus in accordance with the invention supported in the ocean from a ship; and

FIG. 9 shows an apparatus in accordance with the invention supported in the ocean by a buoy.

In FIGS. 1, 2 and 3, an elastic bar **10**, preferably steel and in the form of an elongated cylinder, mounts at one end a sound wave radiator **11**, in this case in the form of two flat sheet metal cones, **12**, back to back. Secured to the opposite end of a bar **10** is an alternating force generator **13**. The force generator **13** is of a type containing a small oscillative inertia body constrained to move in a predetermined cyclic path, a number of suitable examples of which are disclosed in my aforementioned application Serial No. 825,117. However, I here show a type having eccentrically weighted rotors driven through appropriate gearing from an electric motor **14**, preferably an induction motor.

Force generator **13** comprises a cylindric body or case **16** having at one end a threaded coupling pin **17** screwed into a threaded socket at the adjacent end of bar **10**, so as to afford a secure, rigid coupling between the generator case and bar **10**. Inside case **16** is a series of eccentrically weighted rotors **18** rotatably mounted through suitable bearings **19** on shafts **20** set into case **16**. Spur gears **21** on the peripheries of the rotors mesh with one another, and by inspection, it will be seen that the rotors are arranged so that their eccentric weights **22** all move longitudinally of bar **10** in synchronism, so that unbalanced longitudinal components of force are additive, but with alternate rotors turning in opposite directions, so as to cancel transverse force components.

The spur gear **21** on the upper rotor is driven by spur gear **24** on the cross shaft **25** journalled at its ends in case **16**, and driven in turn through bevel gear **26** meshing with bevel gear **27** on an axial shaft **28** journalled in the top end of the case. Shaft **28** has a splined end portion **28*a***, engaged by a splined socket **29** in the end of shaft **30** of the aforementioned drive motor **14**.

The case **32** of motor **14** is fastened to one end of a sleeve **33** surrounding the force generator, and coupled at its opposite end to one end of a tubular jacket **34** that surrounds bar **10** and extends toward radiator **11**. The opposite end of jacket **34** is connected to one end of coupling sleeve **35**, the other end of which has an internally tapered end portion **36** joined by a firm taper fit to a complementary tapered surface **37** on bar **10**, the bar being of somewhat enlarged diameter beyond this taper, as at **38**, within the confines of cones **12**, as shown.

Screwed into a socket in the end of bar portion **38** is a plug **39** supporting an end cap **40**, which furnishes a stud mounting for the inwardly turned end portion **41** of a cone mounting sleeve **42** annularly spaced around bar portion **38** and supported therefrom by ribs **43**. The cones **12** are furnished with inner flanges **44** which are welded to sleeve **42**. At their peripheries, the cones **12** contact one another, and are suitably connected together, as by welding. The outer rims of these cones **12** are preferably furnished with turned stiffening flanges **46**.

As described in the introductory paragraphs hereto, the sound radiator constituted by cones **12** has an effective diameter preferably of the order of a wavelength of sound in the liquid medium in which the apparatus is to operate. For water, this diameter dimension is accordingly, for a frequency of 400 cycles per second, about 12 feet. The bar **10**, calculated on the basis of the speed of sound in steel, for a frequency of 400 cycles per second, has a length of about 10 feet.

The apparatus is immersed in a body of liquid, in any one of a number of situations, some of which will be particularized hereinafter, so that radiator **11** is liquid loaded, and when caused to vibrate effectively in contact with the liquid, radiates a sound wave therefrom.

The electric drive motor drives the shaft of the force generator, causing rotation of the unbalanced rotors; and the longitudinal components of the cyclic reactive forces of these rotors are transmitted through the rotor shafts to the generator case, where they are additive to generate a longitudinal cyclic force on the generator case, which is in turn applied to the end of bar **10**. The case of the

force generator, and the end portion of bar **10**, being at a velocity antinode V of the bar, vibrate longitudinally of the axis of the bar, and the impedance at the coupling point between generator case and bar is elevated substantially over that of the combination of unbalanced rotors, but materially less than that of the liquid loaded radiator **11**. The velocity node of the bar locates itself at V^1, close to the radiator **11**, and the distance between V and V^1 is a quarter-wavelength. At the sound radiator extremity of the bar is a fairly high impedance point, where the velocity cycle is of relatively small amplitude, and the stress cycle is of relatively high amplitude. The radiator **11** is connected to the bar **10** at this high impedance point and is driven with a corresponding impedance characteristic. This high impedance is well adjusted to the loaded impedance of the radiator **11**, and the latter is therefore effectively driven from the force generator, so as to radiate sound waves at high power.

An advantage of the apparatus of FIGS. 1 and 2 is particularly to be noted, in that whereas the force generator must undergo vibration with the bar **10**, the splined connection between the motor shaft and the force generator shaft permits the motor to remain stationary in space, supported by the jacket **34**, which is also stationary by virtue of being connected to the bar **10** at a node of the latter.

The drive motor for the force generator has been described as preferably an induction motor. By using an induction motor with substantial armature "slip" in its rotating field, the driving force exerted thereby on the force generator is readily held at a magnitude less than the threshold value where the frequency generated by the force generator breaks over the peak of resonance, as more fully explained in my application Serial No. 825,117. This is a feature of marked advantage, giving the system very good frequency stability. Alternative expedients within the scope of the invention are available to assure this frequency stability. For example, there can be a torque-responsive engine-generator combination for supplying electric power to the driving motor for the force generator, which driving motor need not be in such instance an induction motor. Under such conditions, the torque responsiveness of the engine substitutes for the torque responsiveness of the induction motor, so that the motor generator can be closely coupled, or non-slip, in nature. An important feature of my force generator-resonator combination is that such frequency stability greatly adds to the controllability of a remotely controlled servo-governor.

It might here be mentioned that it is advantageous in all casese to fabricate the radiator structure of stiff plate or cone members forming an assembly having its first resonant frequency above the resonant frequency of the resonant bar structure. This keeps the radiator structure from "flapping" owing to resonant frequencies equal to or below the operating frequency of the system.

FIGS. 4 and 5 show a modified resonant bar structure, and modified mounting of the periodic force generator and drive motor. A cone type sound radiator of the type of that of FIG. 1 is shown at **50**. A force generator is shown at **51**, and an electric drive motor at **52**, the modified resonant far structure being designated generally at C.

The force generator is of the type of that shown in FIGS. 1–3, having case **53** bolted at **54** to the bar structure, unbalanced rotors **55** geared together by gears **56** and driven through gears **57**, **58** and **59** from motor shaft **60**, the motor case **61** being in this instance screwed into the end of the generator case, as shown.

The bar structure C in this instance comprises a plurality of elastic cylindrical rods or tubes **64**, integrally joined into a single head **66** which is bolted at **54** to the force generator case.

The rods **64** diverge from head **66** at a typical angle as shown, and their extremities are furnished with mounting plates **69** bolted to the rearward side of the radiator assembly in the region of the line of inertia thereof. These may be two diametrically opposite rods **64**, or four, as here shown, or a greater number. In fact, there may be sufficient of such rods to form a full cone, or an integral cone may be used. Of course, the more rods, the less will be their individual cross section areas.

The apparatus of FIGS. 4 and 5 operates with a velocity antinode at V, a velocity node in each rod **64** at V^1, relatively close spaced to the radiator, and a region of high impedance at the junctions of each rod with the radiator. The standing wave patterns in the rods **64** are alike, and of course similar to those in the bar **10** of the embodiment of FIGS. 1–3. The advantage of the bar structure of FIGS. 4 and 5 is the location of the point of radiator drive well out on the cone assembly where the drive effort results in minimum bending of the latter.

In FIG. 6 is shown a modified form of the invention in which the resonant bar structure **70** is flared out at one end to constitute the sound radiator. To the small end of the bar structure **70** is bolted the case of the periodic force generator **71**, whose drive shaft is driven from the drive shaft of electric motor **72**, in an arrangement generally like that of FIGS. 1–3. The bar structure **70** is flared from its small end in the general form of an exponential horn. It may be a casting of a material such as anodized aluminum, and may be cast with cavities **73** for lightness. A horn-like shell **74**, configured to the general outline of the bar structure **70**, surrounds the sides of the bar structure with clearance, and encloses force generator **71**, being bolted to the case of motor **72**, and to the bar structure in the plane of the velocity node V^1 of the latter, as by means of spacing webs **77** and screws **78**. The shell **74** is here shown as formed with eyes **79** to which supporting cables **80** may be conveniently connected. In resonant operation, a velocity antinode V appears at the small end of the bar structure, a region of high impedance appears at the front radiating face **81** of the bar structure. The mechanical impedance characteristics are similar to those of the previously described embodiments.

Operation is in general the same as that of the embodiments of FIGS. 1–5, with the exception that the exponential shape tends to make the acoustic structure more unidirectional as regards the sound wave radiation pattern.

To improve the unidirectional wave radiation quality, provisions may be made for decoupling the back side of the horn-shaped bar structure, so that substantially only the front face **81** is acoustically coupled to the liquid, and thus so that radiation is primarily out from, and along the axis of, the main radiating surface **81**. To this end, I coat the back surface of the flared portion of the bar structure with a layer **82** of cellular elastomer such as cellular rubber. This material tends to compress and expand in step with vibration of the bar structure, and effectively prevents back sound wave radiation. Various decoupling provisions for a similar purpose were disclosed in my Patent No. 2,717,763.

At greater depths, the cellular material tends to become collapsed, and a more positive decoupling means is required if a monopole radiation pattern is to be achieved. In such case, an annular flexible diaphragm **84** is fastened to the periphery of the front face **81** of the flared bar structure and to the periphery of the horn-shaped shell **74**, so as to form a closed gas space **85** between the back of the flared bar structure and the shell. The jacket or shell **74** is substantially stationary, being connected to the resonant bar structure **70** at a node of the latter. This expedient effectively decouples the back side of the bar structure **70** from the liquid outside. To assure that the diaphragm **84** will not be subject to an appreciable pressure differential, a conventional diver's gas storage bottle **86**, with a conventional aqua-lung pressure regulator **87**, feeds gas to the space inside the shell or jacket, at a pressure equal to that of the submergence pressure.

In FIG. 7 I have shown a sound wave radiation apparatus **100** in accordance with the invention submerged by a cable **101** in a liquid body **102** contained in a tank **103**, which may, for example, be a tank section, in a tanker ship. This operates to radiate powerful sound waves in the liquid body. This apparatus is capable of removing solid foreign material which has accumulated on and adhered to the inside surface of the tank. Moreover, the apparatus is especially effective for mixing the contents, and preventing settling out of solids. The apparatus can be permanently installed in an ocean liner or other tank, as well as being inserted occasionally for short duration processing. In the case of ships tanks, the invention used as illustrated is especially effective because of its high power. And because of its moderate frequency, sonic energy loss through the hull is minimized.

FIG. 8 shows my sound wave radiator apparatus **110** in combination with an ocean vessel **111**, for locating the position and orientation of the sound radiator in the water. Here the apparatus is suspended by cables **112** from conventional boom facilities on the vessel. The electric power source may also be located on the ship, and power fed to the apparatus by cable **113**. The dashed lines in the figure represent the sound radiation pattern if a simple form of my invention be used, giving radiation from the back as well as the front of the radiator **114**. This is a dipole acoustic pattern, which places the vessel advantageously in a null region, and which is of advantage for establishing a standing wave pattern between a plurality of such ships.

FIG. 9 shows a monopole version of resonator-radiator in accordance with the invention, given a radiation pattern primarily in one direction only. It illustrates an anchored buoy **120** for supporting the resonator-radiator **121** of the invention through cables **120***a*. Here, the power source may be on land, with an electric cable run out to the buoy, and thence, as at **122**, down to the motor of the apparatus **121**. Using a single power source on land to feed a number of such units, the operation thereof can be conveniently correlated as regards frequency and/or phase. As an alternative, the buoy may accommodate a conventional engine-generator **125** within it, supplying electric power for the electric motor of the apparatus. Here, additional governing means **126**, of conventional type, can be radio controlled through antenna **127** and conventional sensor **128** to govern the speed of the engine, and therefore the frequency of the radiated wave.

It should be understood that a plurality of my resonator-radiator units can be closely spaced, so as to produce a beam, as taught in my Patent No. 2,745,507.

In the foregoing specification and in the claims, I have used the expression cyclic mechanical force generator. It is to be understood that such a device is of the broad class comprising an inertia mass which moves through predetermined stroke limits with a component of movement relative to the case or body of the generator, and with constraining means between the inertia mass and the core or body of the generator either as disclosed herein, or in the form of any of various bearing arrangements, some variations of which are disclosed in my U.S. Patent No. 2,960,314.

The drawings and description will be understood to be merely illustrative of and not restrictive on the invention considered in its broader aspects, and many modifications are therefore possible within the scope of the broader of the appended claims.

I claim:

1. A device for radiating sound waves into a body of liquid, comprising; an elongated member having a driving end and a radiating end, said driving end having a source of cyclic sound waves coupled thereto and said radiating end having means defining an enlarged liquid-contacting surface, the sound waves produced by said source having a velocity node spaced from said radiating surface a distance less than a quarter wave length of said waves.

2. A device as defined in claim 1 wherein said member is of generally horn shape and comprises a single piece of elastic metal composition, the large end of said horn being said liquid-contacting surface.

3. A device as defined in claim 1 wherein said liquid-contacting surface is of a transverse dimension equal to a major fraction of a wave length in said liquid at the frequency of said source.

4. A device as defined in claim 1 wherein said source of cyclic sound waves comprises a mass element movable cyclically and means reactively coupling said mass element to said driving end of said member.

5. A device as defined in claim 1 wherein said source comprises a body fixed to said driving end of said member and a mass element mounted on said body for oscillation relative to said body with a component of movement in the direction of length of said member whereby said body receives a reactive oscillating force for transmission to said driving end of said member.

6. A device as defined in claim 1 wherein said driving end is farther from a velocity node in said member than is said radiating end.

7. A device as defined in claim 1 wherein the length of said member is greater than a quarter wavelength of said waves and wherein said velocity node is between said driving end and said radiating end.

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