June 29, 1965 A. G. BODINE 3,191,537

SONIC PUMP IMPELLER

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Fig. 1.

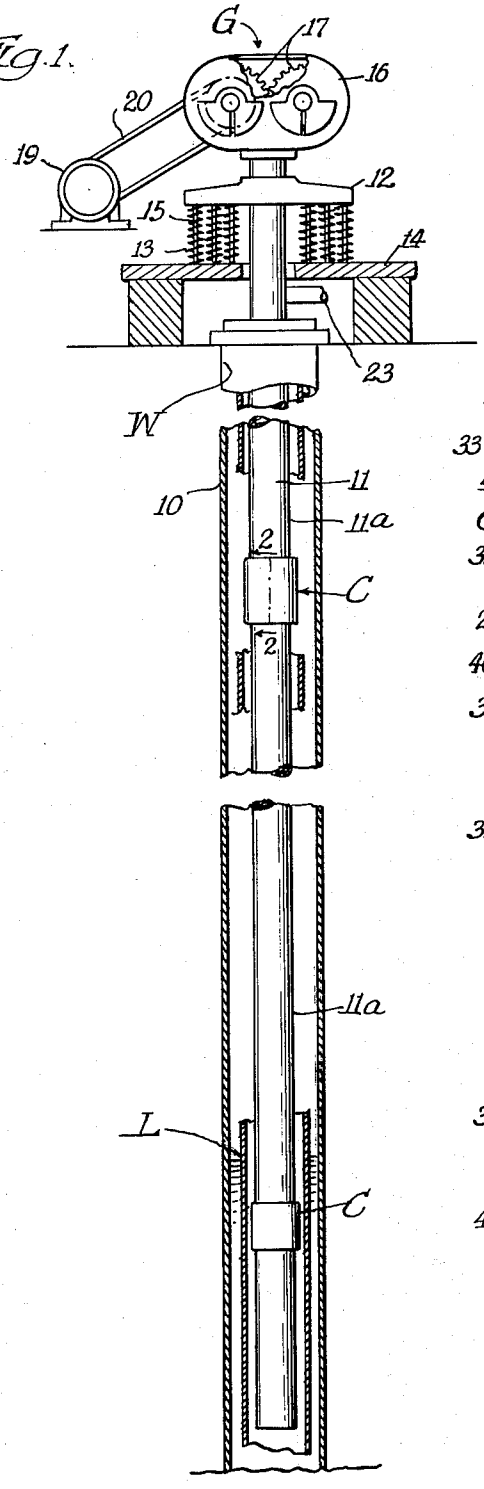



Fig. 2.

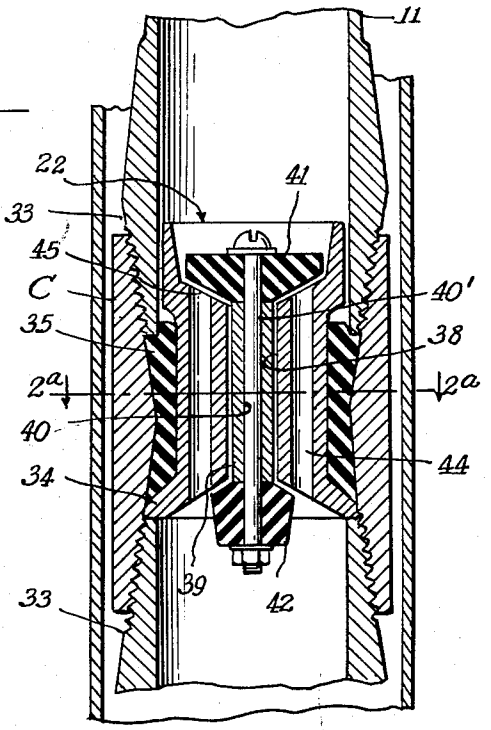
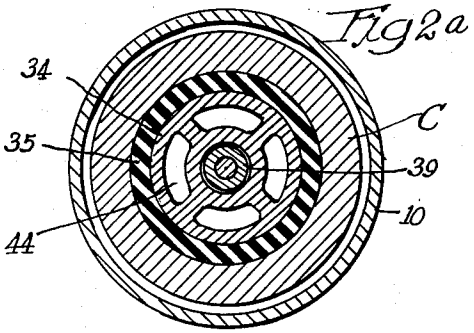



Fig 2a



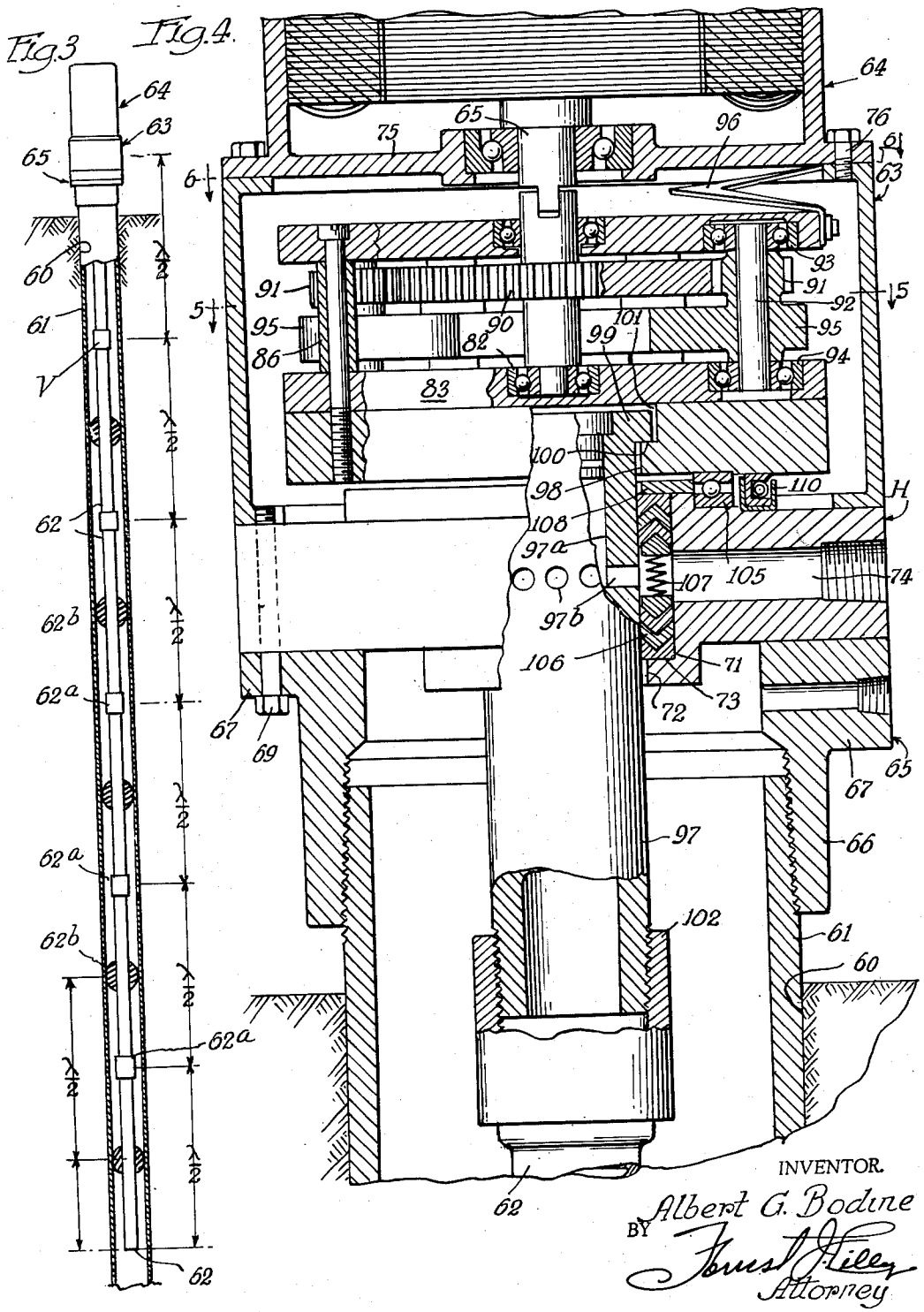

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Fig.3 Fig.4.




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Fig. 5.

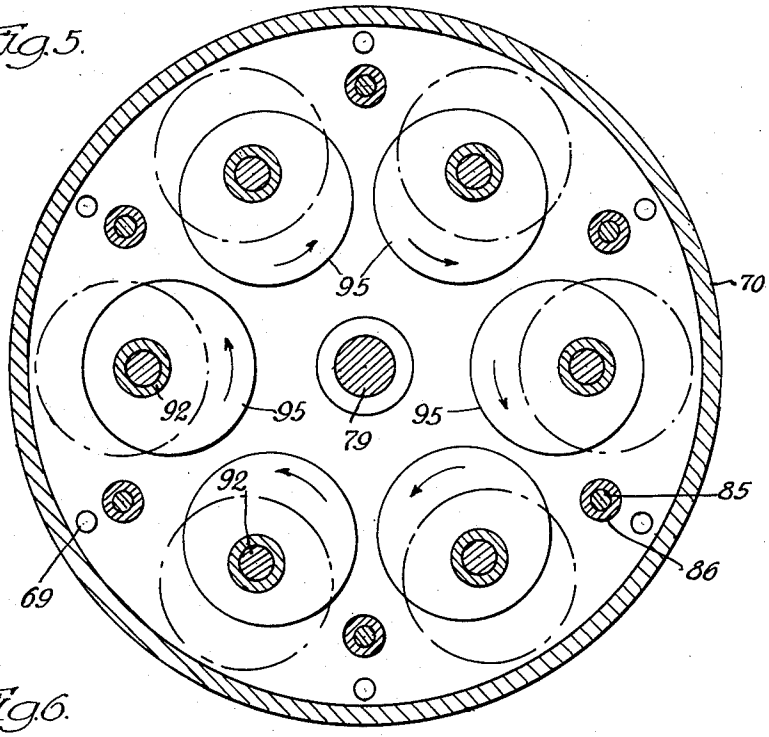
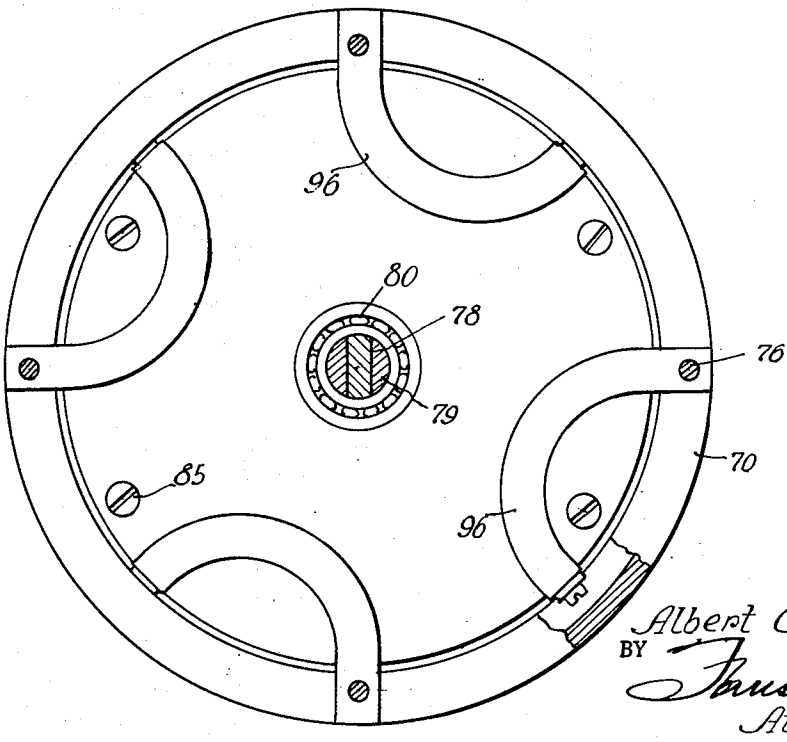



Fig. 6.




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Fig. 7.

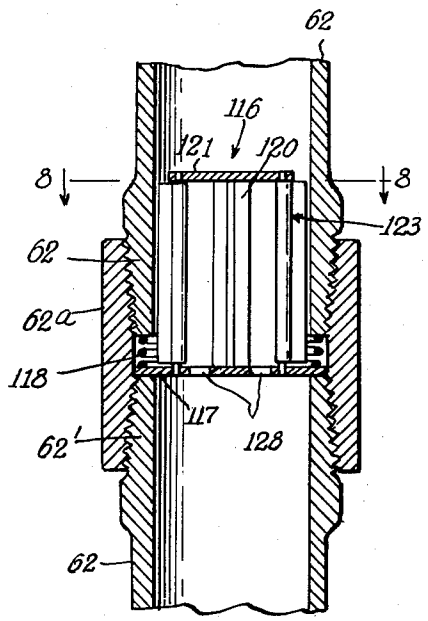



Fig 8.

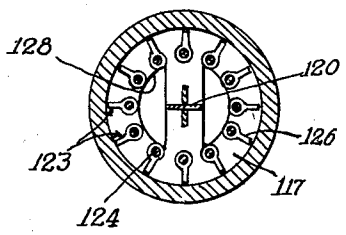



Fig. 8a

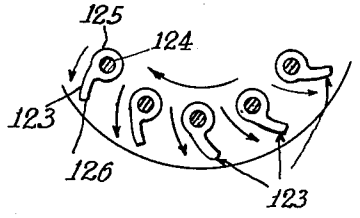



Fig. 8b

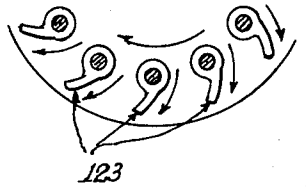
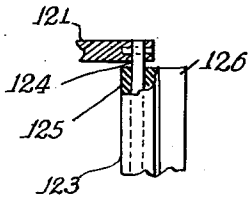



Fig. 9.




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United States Patent Office 3,191,537 Patented June 29, 1965



# 3,191,537
# SONIC PUMP IMPELLER

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Filed May 26, 1964, Ser. No. 370,377
5 Claims. (Cl. 103—43)

This application is a continuation-in-part of my co-pending application, Serial Number 815,510, filed May 25, 1959, now U.S. Patent No. 3,143,970, for Method and Apparatus for Pumping Fluids by Oscillatory Impeller Action, and is a continuation-in-part also of my co-pending application, Serial No. 353,205, filed March 19, 1964, for System for Pumping from Sandy Wells with Sonic Pump.

This invention is concerned with sonic deep well pumps, particularly with the fluid impelling member therein. In sonic pumps, for example, as disclosed in my prior Patent No. 2,444,916, the impelling member may typically be a check valve, or it may be a vane. Such impelling member is actuated by elastic vibration of a column. I have found that a desirable stability of fluid pumping performance is accomplished if the fluid impelling member or element, actuated by the elastic column, is made of a material having an acoustic impedance substantially less than the acoustic impedance of the material of which the column is made. In this consideration impedance may be represented by the equation $Z=\rho c$, wherein Z is impedance, $\rho$ is the density of the material, and $c$ is the speed of sound therein.

In some forms of the sonic pump the fluid impelling element is a member somewhat akin to a valve seat, or valve body. In cooperation therewith we find a periodic closure element, which moves relative to the impelling element so as to perform a check valve type of function. In many such cases the closure element and the valve seat element together form a fluid impelling element when the closure element is closed or in engagement with the seat element. This normally occurs once during each cycle of the sonic wave action. During the remaining part of the cycle the closure element disengages from the seat element so as to allow flow of the previously impelled liquid body.

I have found a distinct advantage in having either the seat or closure element, or both, be of a material having a lower acoustic impedance than the actuating column. A very desirable effect takes place with such a combination. Apparently elastic wave energy which is delivered from the elastic column, of typically steel material, is applied much more uniformly to the total fluid impelling structure, and the fluid, if the intervening seat or closure element is of substantially lower acoustic impedance.

Apparently there is "feedback" acoustic vibration between the closure element and the elastic column if the closure element and the intervening seat element is of high acoustic impedance material. On the other hand, a beneficial effect takes place when the lower impedance material is used in the closure element or the seat element. Apparently there is no longer a feedback excitation whereby the closure element can partake of high frequency vibrations. The closure element moves with much more uniform action against the seat element. One very beneficial effect, in addition to more steady pumping rate, is a very much reduced wear of the fluid-impelling-valve combination.

In addition, I find that the lower impedance material in the impeller structure generates less wasteful shock energy, and useless high frequency components, in the liquid column. In this connection I find it an additional advantage if this impedance is as low as that of the liquid. For instance, some rubber compounds have $\rho c$ about the same as many liquids, such as water or oil.



The above described advantages of material impedance are also substantially applicable if the impeller element is a cyclic vane actuated by an elastic column, which is a variation of the above described sonic pump action.

Presently preferred illustrative embodiments of the invention are shown in the accompanying drawings, in which:

FIG. 1 is a diagrammatic longitudinal sectional view, with parts shown in elevation, of a sonic pumping system making use of the invention.

FIG. 2 is a view of a coupling incorporating a check valve, the view being taken on line 3—3 of FIG. 1.

FIG. 2*a* is an end view of FIG. 2.

FIGS. 3–9 show vane impeller embodiments which are described in detail.

In FIG. 1 an oil well bore is indicated by the letter W, and a well casing within said bore by numeral **10**, the lower end portion of the casing adjacent the productive formation being understood to be perforated in the usual manner. The pump tubing string **11**, understood to be composed of elastic material, as steel, is suspended in well bore W from platform **12**, its lower end reaching downwardly to the region of the liquid L to be pumped from the well bore. The tubing string is made up in lengths or stands **11***a*, coupled end to end by couplings such as C. Platform **12** is resiliently mounted on vertical coil springs **13** standing on ground-supported platform **14**, springs **13** being guided by vertical rods **15** set into platform **14** and slidable in platform **12**.

Mounted on platform **12** or on the upper end of tubing **11** extending thereabove, is a vibration generator or vibrator G comprising a housing **16** containing a device for vibrating the platform **12** and the upper end of the tubing **11**, thereby exerting a vertical oscillating force upon the upper end of the tubing **11**. The means for generating vibrations contained within housing **16** may be of any type, but that here shown is a simple type having meshing oppositely rotating spur gears **17** carrying eccentric weights **18**, which balance out horizontal vibrations but are additive to produce a substantial resultant oscillatory force in a vertical direction. The driving pulley of the vibrator, mounted on the shaft for one of the spur gears, is driven by electric motor **19** through belt **20**. Since this vibrator is employed to generate elastic waves in the pump tubing which are in the same nature as sound waves, and travel with the speed of sound waves in the pipe, I may refer to this vibrator as a sonic wave generator.

The oscillating force applied to the upper end of the elastic tubing **11** by the sonic wave generator launches alternating deformation waves of tension and compression down the tubing, traveling in the tubing with the speed of sound. It must be understood that the tubing is not vertically reciprocated in a bodily manner. On the contrary, the vertically oscillating force applied to the upper end of the tubing by the sonic wave generator is of sufficiently high frequency (for instance, although without implying any limitation on the invention, 20 cycles per second for a 4,000 foot tubing) as to make that type of operation impossible. Instead, longitudinal elastic deformation waves of compression and tension, of wavelength actually shorter than the length of the tubing string, travel down the tubing string, causing each transverse section thereof to oscillate vertically with each passage of a wave.

At certain of the tubing couplings there are located fluid impelling and check valve elements **22** (see FIG. 2), as will be described in more particular hereinafter. The general operation of the pump is described in my aforementioned Patent No. 2,444,912, and reference is also made to my Patent No. 2,702,559. It will suffice here to say simply that the pump operates by vertically vibratory influences applied to the fluid impelling and check valve element **22**, the vertically vibratory impulses being ob-

tained from the elastic vibration waves transmitted down the pump tubing string. The fluid impelling and check valve means are located in vertically vibratory regions of the tubing string, and in this manner, as more fully described in my aforesaid patents, well fluids are pumped up the tubing string, to be delivered via outlet pipe 23.

As disclosed in my aforementioned Patent No. 2,444,912 on sonic pumps, the pump tubing string 11 contains one of more fluid impelling units, illustratively in the form of check valves such as indicated at 22, and the lowermost of which is on the lower extremity of the pump tubing. A typical check valve 22 is shown in FIGS. 2 and 2*a*, and it will be understood that a suitable number of such check valves may be employed. Such a valve is located at and positioned by a conventional coupling collar C such as commonly used to join adjacent lengths of pump tubing to form the complete tubing string. Thus, a coupling collar C screwthreadedly joins the upset end portions 33 on the adjacent ends of two lengths of pipe. A generally tubular valve body 34 is positioned within tubing collar C and within the upset end portions of the adjacent lengths of tubing 11. Valve body 34 is recessed on the outside to accommodate rubber ring 35 which is compressed when the coupling is made, in an obvious fashion, to position the valve body. The valve body has a central bore 38, which slidably receives a tubular stem 39 formed with a central longitudinal bore 40 to receive a long bolt 40' which clamps upper and lower rubber valveheads or disks 41 and 42, respectively against the corresponding ends of the stem 39. Longitudinal passageways 44 are formed through valve body 34, opening into the lower end of the valve body outside rubber head 42, and opening at the top inside the rubber head 41. The conical upper surface 45 of the valve body through which opens the passageway 44 is, as here shown, formed on an angle which conforms to the angular lower surface of the rubber disk 41, and the latter, in moving onto and off of surface 45, acts as a valve. Surface 45 acts as the coacting valve seat. It will be seen that the length of stem 39 is such as to permit the rubber valve head 41 to seat against valve seat 45, so as to close off passageway 44, or to be elevated sufficiently to open good flow passages from passageway 44 around the outer periphery of the valve element 41 to the tubing above.

The use of rubber having a number between the approximate range of 40 to 70 on the Shore scale for the valve element 41, and especially with a plastic such as nylon for the member affording valve seat 45, gives an impelling element which is especially effective for applying kinetic energy to the fluid stream in accordance with this invention, and also to resist abrasion from pumped sand.

Another principal present form and application of the invention is also in the field of deep oil well pumping, and an illustrative embodiment of the invention for that field is shown in FIGS. 3–9, inclusive. In FIG. 3, which is somewhat diagrammatic, numeral 60 designates a well bore, lined with casing 61, understood to be perforated in the usual manner at its lower end, and numeral 62 designates a string of elastic pump tubing which is torsionally oscillated by means of torsional oscillation generator 63 located above the ground surface. Tubing string 62 is made up of lengths of steel tubing connected by conventional couplings 62*a*, and is provided with a plurality of bearing elements 62*b* designed to centralize the tubing in the casing, and to afford low-friction and non-wearing contact therewith. They may be composed of a plastic, such as a fabric filled phenolic. Since these bearing elements form no part of the present invention, no detail disclosure is deemed necessary herein. Generator 63 is driven from an electric motor 64 (see also FIG. 4) arranged with its shaft 65 vertical, as indicated. The generator 63 is mounted on casing head H, which is in turn mounted on the upper end of casing 61.

Considering casing head H in more particular, it will be seen from FIG. 4 to be made up of a lower member 65 comprising a ring 66 which is internally threaded and screwed onto the threaded upper end of casing 61, and is provided at the top with a flange 67, together with an upper relatively thick plate 68. Screws 69 read upwardly through flange 67 and through plate 68 to thread into an internal flange at the bottom of generator housing 70 to secure these members tightly in assembly. A vertical axial bore 71 extends downwardly in plate 68, meeting a reduced counterbore 72 to form a shoulder 73 adapted to seat later described packing. A radial bore 74 extends outwardly in plate 68 from bore 71, being screwthreaded at its outer end for reception of a coupling on the end of the flow line, not shown.

The housing for electric drive motor 64 has a lower end plate 75 which is connected at 76 to the upper inwardly turned flange of generator housing 70. The lower end plate of the drive motor thus serves as the upper end plate for generator 63. Motor shaft 65 extends downwardly into generator housing 63, and is coupled at 78 to axial generator shaft 79. Shaft 79 is journalled near its upper end in a bearing 80 seated in an upper oscillator end plate 81, and has a lower reduced extremity journalled in a bearing 82 seated in a frame plate 83 spaced below plate 81. The plate 83 rests on top of a heavy tubing suspension plate 84, and plates 81, 83, and 84 are secured in assembly by long screws 85, a spacer sleeve 86 being used between plates 81 and 83.

Fixed on shaft 79 just below frame plate 81 is a spur gear 90, which meshes with a plurality of smaller spur gears 91, here six in number, and understood to be equally spaced around gear 90. The gears 91 are tight on shafts 92 journalled in bearings 93 and 94 in plates 81 and 83, respectively. Integral with gears 91 are eccentrically-weighted rotors 95, and these are phased so that their unbalanced masses approach and recede from the central axis of the generator in unison wtih one another, FIGS. 4 and 5 showing them at their innermost positions in full lines, and with their outermost positions indicated in phantom lines in FIG. 5. It will be clear, accordingly, that the forces exerted by the unbalanced rotors along lines radial to the central axis of the generator will counterbalance one another. On the other hand, each pair of diametrically opposed rotors exerts an alternating force couple. The only difference in the case of the present embodiment is that there are three pairs of diametrically opposed unbalanced rotors, and it will be evident that the alternating force couples of the several pairs of rotors will be in phase, and therefore additive. The generator thus produces a strong torsionally oscillating force couple or torque which is evidently applied through the gear shafts 92 to the assembly of frame plates 81, 83, and 84, causing an oscillation of the latter as will appear. Preferably, though not necessarily, a spring element 96 may connect housing 63 with the oscillatory generator assembly, thus establishing a normal centralized position for the generator, and preventing progressive rotation thereof within the housing. In some installations it is desirable to eliminate this constraint, and to allow the tubing to rotate slowly, so as to distribute wear.

Tubing string 62 is supported from lower generator plate 84 through a tubular stem 97, which extends through a central bore 98 in plate 84, the upper extremity of said stem 97 having an external flange or head 99 resting on a shoulder 100 in plate 84 formed by an enlarged counterbore 101 extending downwardly into plate 84. Stem 97, which has a longitudinal bore 97*a*, and ports 97*b* communicating said bore to outflow passage 74, is coupled to the upper end of tubing string 62 as by coupling 102. The entire weight of the tubing string is thus borne by shoulder 100 on generator plate 84. Plate 84 is in turn supported for anti-friction oscillation by a bearing 105 seated on casing head H. Stem 97 is packed within casing head member 68 by means of chevron packings 106 placed in bore 71 above and below radial outflow passage 74, coil springs such as indicated

at 107 holding the packings properly compressed against shoulder 73 at the bottom, and a retainer plate 108 at the top. A packing ring 110 is preferably also used between bottom generator plate 84 and the casing head outside bearing 105, as indicated at FIG. 4.

The oscillatory torque generated within the frame assembly of generator 63 is exerted on the upper end portion of stem 97, and therefore on the tubing string 62 below, the weight of the tubing string assuring a sufficiently tight frictional engagement between generator plate 84 and tubing flange 90 to assure that the oscillatory torque so developed is applied to stem 97 and the tubing string 62 below. Thus, the generator is torsionally loaded by the tubing string, and oscillates torsionally through a limited angular displacement as determined by the torsional elastic stiffness and moment of inertia of the tubing string.

Directing attention now to the tubing string itself, it will be seen that there is exerted on the upper end thereof an alternating force couple. This results in a wave of torsional elastic displacement being transmitted down the tubing string with the velocity of torsional sound waves in the material of the tubing string. The tubing string is fabricated of elastic material, as steel, with good elastic fatigue properties. If the generator is driven by its drive motor 64 at a frequency corresponding to a frequency for torsional wave resonance in the tubing string, the torsional wave so sent down the tubing string is reflected from the lower end thereof and arrives back at the top end of the tubing in proper phase to be reinforced by a succeeding torsional force couple then being delivered from the generator, and a condition of torsional standing wave resonance is thereby attained. Under such circumstances, velocity antinodes V (regions of maximum torsional displacement), will appear at the extremities of the tubing and at half-wavelength spaced points therealong, as indicated in FIG. 3. In the case of such standing wave operation, couplings 62*a* are preferably located at the antinodes, for a reason to appear presently. Intervening between these velocity antinodes are regions of minimized or zero torsional displacement, called nodes. It is, incidentally, preferable to locate the bearing elements 62*b* at the nodes, as indicated diagrammatically in FIG. 3. On opposite sides of each node, the tubing is twisted in opposite directions, the angular displacement increasing progressively from each node to each antinode. Thus different portions of the tubing are torsionally displaced in opposite directions; and it will be seen that successive half-wavelength lengths of tubing are twisted in reverse directions, the points of zero twist occurring at the nodes. With such a resonant, standing wave system, it becomes advantageous, as will appear hereinafter, to locate the subsequently described fluid impellers within the tubing couplings 62*a* at the antinodes, where the torsional motion is maximized. Other effective modes of resonance include patterns which do not necessarily coincide with tubing section lengths.

It is, however, found quite practicable to operate the pumping system of FIGS. 3 to 6 without necessarily establishing resonance. In this case, torsional traveling waves are propagated the length of the tubing, and the points of maximum torsional displacement travel longitudinally of the tubing at the speed of sound. These longitudinally travelling torsional waves torsionally oscillate the fluid impellers, and effect pumping as will subsequently appear.

One illustrative form of tubing impeller adapted for installation in the tubing string 62 is shown in FIGS. 7 to 9. The tubing string is indicated at 62, and one of the tubing couplings 62*a*, for example, the lowermost coupling of FIG. 3, is shown. A vane assembly 116 is mounted in the tubing in the region of the coupling collar 62*a*. In this instance, it includes a lower base plate or disc 117 seated on the upper end of the lower, threaded tubing coupling pin 62', and retained thereagainst by means of a coil compression spring 118 engaging the lower end of the tubing coupling pin 62'' immediately above. A standard 120 is mounted in the center of disc 117, and supports at its top an upper disc 121, of somewhat less diameter than the inside diameter of the tubing string, whereby to afford an annular flow passage therebetween. A plurality of fluid impeller vanes 123 are mounted on vertical pins 124 set tightly at the top in disc 121 and at the bottom in disc 117. The impeller vanes 123 have hubs 125 fitted tightly on pins 124, as by being molded thereon, and have vane or blade portions 126 projecting from the hubs in a direction normally radial of the tubing. These vanes are flexible, and may be composed of rubber, or suitable plastic such as nylon, Teflon, etc., thus having an acoustic impedance which is lower than that of the tubing 62.

Disc 117 is apertured at 128 for upward fluid flow from below into the space between the vanes and the central standard 120. The fluid flow path is upwardly through apertures 128, outwardly between the vanes, and thence upwardly via annular space 122.

As the portion of the tubing in the region of the vane assembly 116 is torsionally oscillated, the vanes, acting against the fluid therebetween, propel the fluid radially outward. The relief of the resulting pressure is upward via the annular channel 122, and the fluid thus flows upwardly therethrough, and so upwards in the pump tubing, to be delivered from the casing head via passage 74. The flexing action of the vanes, in one direction and then the other as the impeller oscillates torsionally, is well illustrated in FIGS. 8*a* and 8*b*. This flexing action contributes a propulsive effect similar to that of a swim fin, or a fishes tail. In some situations, a single impeller unit may be employed, located near the lower end of the pump tubing, and at a velocity antinode of the torsional standing wave is such a wave is to be established. Usually, however, a number of the impeller units are used along the tubing; and if a standing wave is established, these are most advantageously at the velocity antinodes. They may be at half-wavelength spacing, or integral multiple thereof. Furthermore, in deep wells there are no true nodes, and the impellers are effective at any location along the elastic column. The series of impellers gives a series booster performance that is important to effective pumping in most wells.

The invention has now been described in certain present illustrative forms, but it is obviously capable of modification and of being carried into practice in various other forms, without departing from the spirit and scope of the appended claims.

I claim:

1. In a deep well sonic pump, the combination of:
an elastic column extending down into said well, which column is capable of transmitting elastic wave energy therealong,
an elastic wave generator coupled to said column for applying elastic vibrations thereto at an elastic wave frequency of said column,
a fluid impelling element driving connected to said column so as to be actuated by said elastic wave energy, said fluid impelling element being oriented within a conduit passage having an inlet and an outlet so as to have fluid impelling contact with fluid in said conduit,
said fluid impelling element thereby providing an acoustic coupling between said elastic column and said fluid,
and said fluid impelling element comprising a material having an acoustic impedance value which is less than that of the material of said elastic column.

2. The subject matter of claim 1, in which said fluid impelling element is a check valve closure element.

3. The subject matter of claim 1, in which said fluid impelling element is a check valve seat element.

4. The subject matter of claim 1, in which said fluid impelling element is a flexible vane.

5. The subject matter of claim 1, in which said fluid impelling element has an acoustic impedance equivalent to that for the pumped liquid.



**References Cited by the Examiner**

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ROBERT M. WALKER, *Examiner.*