

Jan. 18, 1966

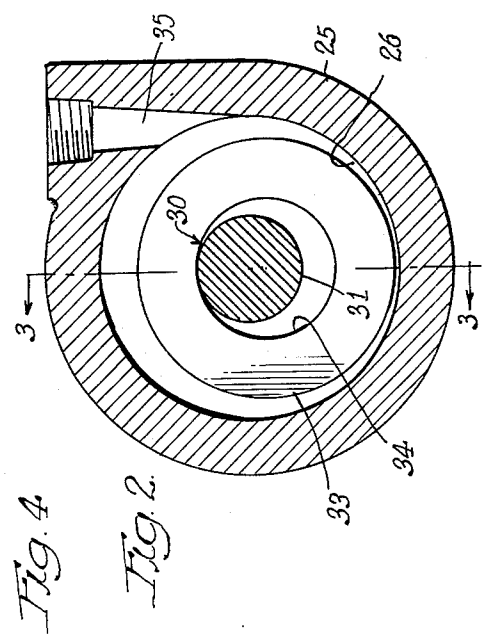
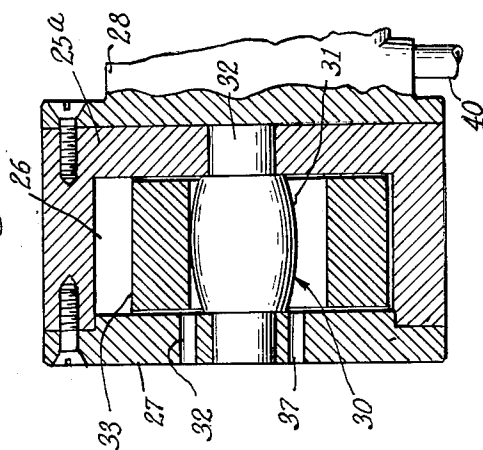
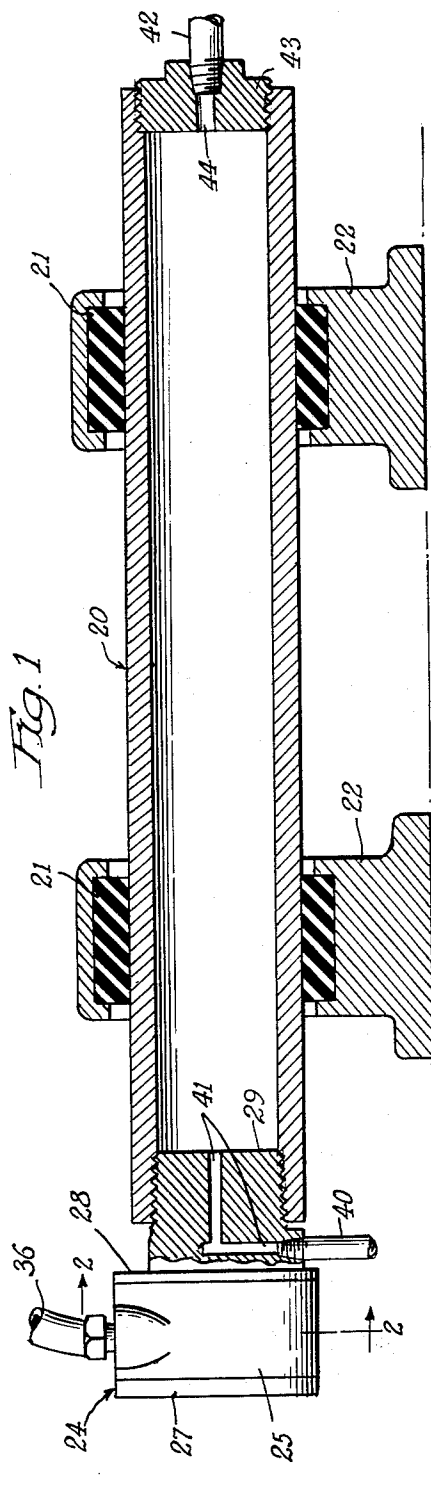
A. G. BODINE, JR

3,229,961

WHIRLING RING SONIC GENERATOR

Filed Oct. 8, 1964

2 Sheets-Sheet 1



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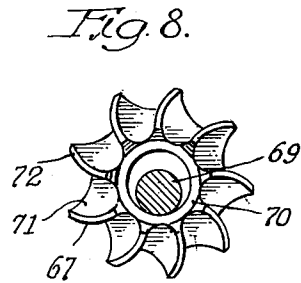
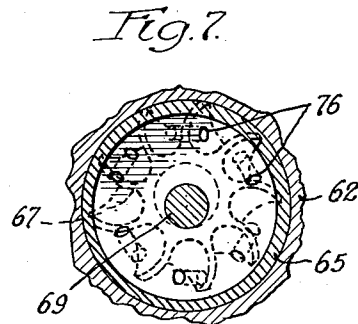
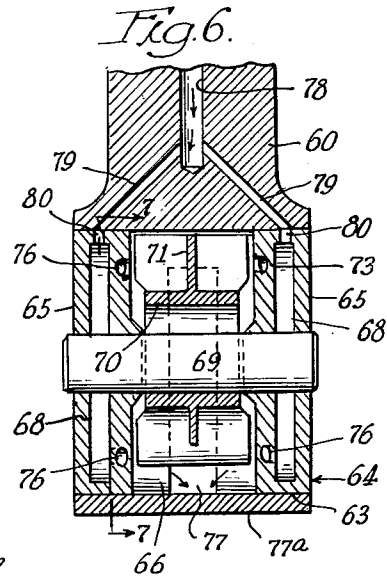
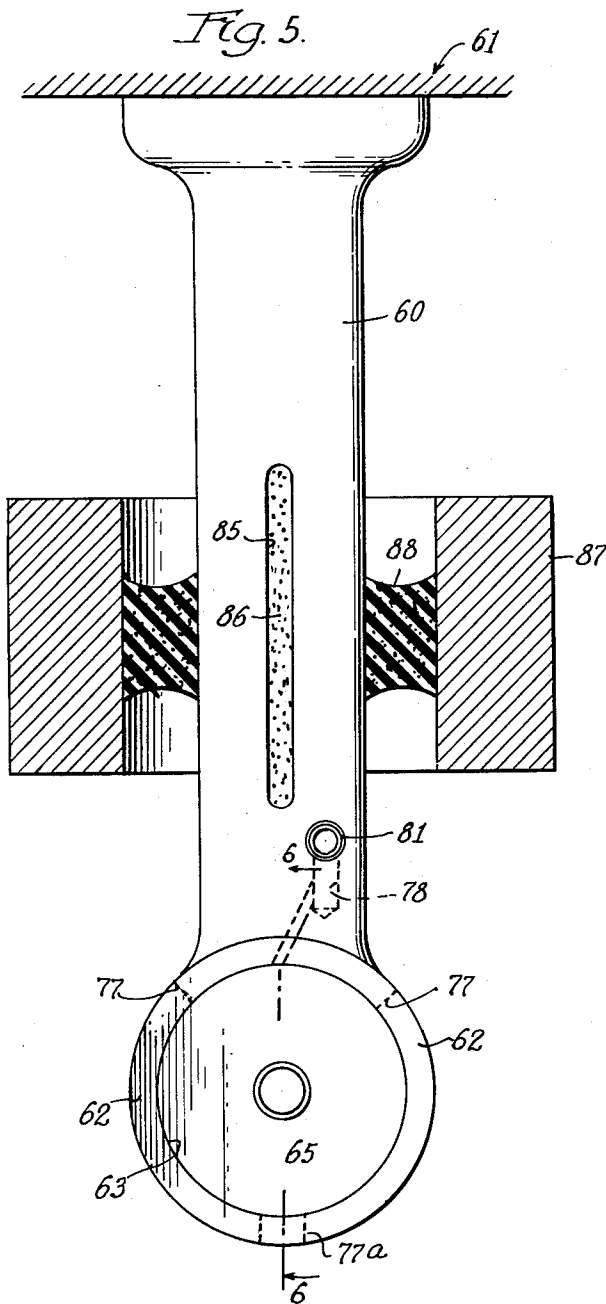
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WHIRLING RING SONIC GENERATOR

Filed Oct. 8, 1964

2 Sheets-Sheet 2



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3,229,961

WHIRLING RING SONIC GENERATOR

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3 Claims. (Cl. 259-1)

This application is a continuation-in-part of my application entitled Apparatus for Generating and Transmitting Sonic Vibrations, Serial No. 55,537, filed September 12, 1960, now Patent No. 3,153,530, which was a division of my application entitled Method and Apparatus for Generating and Transmitting Sonic Vibrations, Serial No. 825,117, filed July 6, 1959, now Patent No. 2,960,314. Said application Serial No. 825,117 was a continuation-in-part of my prior and parent application Serial No. 484,627, filed January 28, 1955, entitled Apparatus for Generating and Transmitting Sonic Vibrations, now abandoned.

This invention relates generally to methods and apparatus for the generation and transmission of relatively high power by means of intense sonic vibrations, particularly for generation or transmission of sonic vibrations in resonant vibratory mechanical devices, either elastically deformable vibratory bodies of the distributed constant class, or elastically supported bodily vibratory devices of lumped constant characteristics.

A large number of industrial uses for high power sonic vibrations have been discovered. It is known, for example, that intense sonic energy may be applied to gases, liquids or solids to produce certain desired chemical or physical effects. Many types of power tools or other equipment are operated by sonic energy of high intensity. One illustrative example involves a longitudinally extended elastic bar, in which a longitudinal resonant standing wave is set up and maintained, so that an end portion of the bar becomes the location of a velocity antinode of such standing wave, and is utilized to vibrate a bit or other tool against the work. Other modes of vibration, such as lateral or gyratory, are within the scope of the invention. Other illustrative examples will appear in the ensuing description.

The bodies or devices to be sonically vibrated at resonance are often characterized by high acoustic impedance. They vibrate with great force, and with small velocity amplitude. The problem of driving these devices, i.e., the provision of an effective vibration generator suited thereto, is often very difficult, particularly in view of the fact that ordinary practically available sources of motivating power operate at low impedance, characterized by driver elements moving with relatively low force but substantial velocity. Ordinary low impedance drivers are incapable of driving high impedance devices because of the mismatch of impedance. The efficiency of transduction has been characteristically low.

The common sonic generators, such as magnetostriction bars, crystals, etc., are capable of a motion of only a few feet per second by reason of limitations set by elastic strain limits, which motion I have found to be entirely inadequate for high power applications.

Mechanical generators are known which have the requisite motional characteristics, but suffer from complexity and a host of ensuing problems. Any degree of complexity of moving parts results in various vibratory interactions taking place at high frequency between these parts, with consequent high energy loss and frequent destruction of parts in high stress applications. At very high frequencies gears will chatter, bearing separators seize and fracture, and the individual balls or rollers of antifriction bearings are forced to rotate so fast they become unstable in their motion. Plain journal bearings seize and overheat. The power of previously known generators has

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been relatively low, particularly at the higher frequencies, and the ruggedness required of an industrial machine has been lacking. Many proposed industrial applications of sonic power have been correspondingly handicapped.

It is accordingly the primary object of the present invention to provide novel and improved sonic vibration generating apparatus particularly suited to various industrial applications and characterized by relatively high power output, efficiency of transduction, simplicity and ruggedness.

The invention is practiced in systems involving the driving of an inertia mass rotor in an orbital path under guiding constraint of a bearing means, whereby a periodic force impulse is exerted on the latter, and the coupling of this bearing means to a vibratory device or system having a resonant frequency range whereby said periodic force impulse, or a component thereof, is effective to vibrate said device in said range. Two types of such devices are possible, a ball or rotor spinning in a circular raceway, and a ring spinning on a pin. I have discovered certain inventive features and important advantages in the second type, and these will be pointed out hereinafter. In these generators, the rotor is driven at an orbital frequency which generates a vibration frequency in the range of resonance for the driven vibratory device, I have discovered that the driven vibratory device, when so vibrating in its resonance range, with its vibration amplitude amplified by resonance, back-reacts with the orbital rotor, strongly constraining the rotor to an orbital periodicity corresponding to its own resonant frequency. I have further discovered that the apparatus tends inherently to operate on the low side of the frequency for peak resonant amplitude, and further, that the whole apparatus, driven vibratory device and orbiting rotor, tends to lock in synchronously slightly below the frequency for peak resonant amplitude. The orbiting rotor is strongly constrained to produce this frequency, and although it could of course be strongly enough driven to reach a threshold condition where it would reach and break over peak amplitude resonance frequency, considerable increase in driving effort is required before this unwanted condition occurs. In this connection, it is to be understood that the driving effort on the rotor is limited to a value below such threshold condition. The rotor is hence guarded from overspeeding and destroying itself or its housing when operated at high frequency.

In addition to these effects, the constraint which keeps the frequency of the orbiting rotor to the low side of the resonance curve (amplitude vs. frequency) of the vibratory driven device is effective to establish a phase angle between the rotor motion and the motion of the vibrating device wherein maximum power is delivered from the rotor to the vibrating device for a given power input to the rotor.

It will be evident that such an orbiting rotor generator has high output impedance, while being operable by motive power at low impedance, impedance being understood to be proportional to the ratio of force to velocity. Considering the output side of the generator, where the race or bearing for the orbiting rotor is coupled to the vibratory driven device, it will be seen that force will be high owing to the high magnitude of centrifugal force, while the stroke amplitude, and therefore the velocity amplitude, will obviously be low. The desirable high output impedance for the resonant system is therefore attained. Impedance is generally thought of in connection with alternating phenomena such as alternating forces, in comparison with resulting velocity amplitude. The motive power source used in the present instance is typically a continuous air jet, rather than an alternating entity. Nevertheless, the continuous air jet has the characteristic of relatively lower force and relatively high velocity, and is,

broadly speaking, a form of power having a low impedance quality. The generator of the invention thus fulfills the requirement of operating off a low impedance form of power, and delivering power at high impedance.

The illustrative embodiments chosen for disclosure herein are of the type wherein the resonantly driven device is of the "distributed constant" system type wherein the parameters of mass and elasticity governing the resonant vibration frequency are distributed throughout all or a significant part of the vibratory system, as in the ideal example of a tuning fork. By contrast, a "lumped constant" system is one wherein the parameters of mass and elasticity governing the resonant frequency are largely concentrated or localized in discrete elements such as intercoupled masses and springs, respectively. Of course, these are idealized classifications. Practical systems usually are mixtures of the two. Thus, practical systems wherein the parameters of mass and elasticity are preponderantly "distributed" will also very commonly have local concentrations of mass, with small capability for elastic vibration therein; while practical systems wherein mass and elasticity are preponderantly localized, as in intercoupled spring and mass elements, will invariably have certain "distributed constant" qualities in view of mass inherently present in spring elements, and elasticity inherently present in mass elements. Thus, the resonantly driven devices of the invention may embody such distributed constant elements as an elastic bar, in which either transverse gyratory, or longitudinal standing wave action may be set up by the vibration generator. Such bar may be a solid rod, or it may be tubular, as a steel pipe. The term "bar" is often used in the field of acoustics in connection with discussions of elastic wave propagation, without reference to the cross-sectional form of the bar, and the term will be so used herein, both in the specification and claims, thus generically comprehending hollow rods, or pipes, as well as solid rods, I-beams, and other structural shapes.

This invention is based upon the discovery that an orbiting rotor in the form of a ring whirling around a bearing means in the form of a pin is an especially effective and well operating sonic generator when combined with a resonant elastic member. In such a device it is possible to provide a small difference in diameter between the pin and the hole in the ring, so that the contact point makes a number of revolutions around the pin while the ring turns once about its own axis, thus accomplishing a frequency step-up, and good high-frequency sonic vibration. In addition, with such combination, having small differences in the two diameters, and thus small eccentricity of the orbiting mass, the impulse force for a given mass of ring can be fairly low, thus permitting a very massive ring, for a given total impulse force. This possible massiveness of the orbiting rotor, when using the ring form of orbiting oscillator, gives an orbiting rotor of very high inertia (high acoustic Q) and consequent speed constancy.

It can thus be seen that the ring lets one use a heavy rotor, with small orbit so that unbalanced bearing loads are minimized. This heavy rotor, especially with its large outside diameter, is a very good flywheel, thus tending further to run at constant r.p.s. in spite of irregularities in the power supply or the load. This augmented frequency stabilizing function is unique and of especial value when used in combination with a resonant driven member. A much more powerful sonic generator can thus be used in combination with a given sized resonant member, and still stay locked onto the resonant frequency of the resonant member. This latter cooperation is a function having no parallel to draw upon, from non-resonant systems.

The ring and pin generator has a further important advantage over ball and raceway generators. For example, assuming a ball in a raceway, and a large ratio of ball diameter to raceway diameter, so as to give desirably high impulse, and assuming a fluid jet directed tangentially into

the raceway, it can readily be seen that the jet injected immediately behind the point of contact between the ball and the raceway will impinge on the outermost portion of the ball (the portion nearest the raceway), and rearwardly of its point of contact with the raceway. The ball is thereby driven forward along the raceway. It will be noted, however, that the surface of the ball receiving the jet stream will, because of rolling action on the raceway, be moving against the direction of the jet stream. It is even more important to note that at other points around the raceway, and about the perimeter of the ball, the fluid stream is also moving contrary to the direction of motion of the ball surface. And, in the general region opposite the point of contact between the ball and race, the fluid stream is moving opposite to the direction of movement of the ball as a whole. The result is general turbulence, resistance to fluid flow, conflicting drive efforts on different portions of the ball, instability, including lateral wobble, and a positive refusal to deliver efficient or high performance output.

With this specific problem in mind, it is a particular and further object of the invention to provide a fluid driven mechanical vibration generator of the general class discussed in the foregoing, which, however, is so physically reorganized that all portions of its fluid driven element move in the general direction of the fluid stream or streams impinging thereon.

According to the invention, the ring is fluid driven by one or more fluid jets oriented to project fluid streams generally tangentially toward the ring, and these jets are discharged from nozzles typically in the wall of a surrounding cylindrical casing whose outer wall is preferably of just sufficient inside diameter to accommodate the gyrating ring. The fluid used for the driving of the ring may be a gas, liquid or steam. In practice, air is very effective.

For the purpose of some applications, the ring may advantageously have vanes or buckets to receive the fluid jets. When the ring engages the pin, the center of gravity of the ring is offset a short distance from the center of gravity of the pin. When impinged upon by the air jets, the ring does not spin on the pin like a wheel with a plain bearing on a shaft, but rolls on the pin without substantial slippage, and gyrates. The center of gravity of the ring moves in a circular path around the center of the pin and the ring exerts a centrifugal force on the pin in a direction line extending through the center of the pin and the center of gravity of the ring. A force vector acts on the pin along this direction line, and of course rotates about the center of the pin with the center of gravity of the gyrating ring.

In this ring and pin generator, the fluid jets all strike the ring generally tangentially and in the general direction of the desired motion of the ring. No portion of the ring moves in opposition to the contacting fluid stream spinning around the casing, as is the case in ball and raceway type generators. Accordingly, resistance to fluid flow is very materially reduced, and the fluid can be forced through the device at greatly increased velocity, permitting, in turn, greatly increased gyration frequency.

Another unique advantage peculiar to the fluid-driven ring and pin form of generator, particularly where the gyrating ring is arranged to have only small clearance with the wall of the confining casing, is that, in contradistinction to the ball and raceway generator, the rotating element (ring) is driven by a positive displacement effect. Thus, consider the ring to be in contact with the pin, so that its periphery, at a point 180° removed from the pin, is then near or adjacent the side wall of the casing. A jet stream, directed into the crescent-shaped space between the casing wall and this near point of the ring, then acts tangentially against the periphery of the ring to move the ring on and out of the way of the jet stream, which is otherwise blocked off. The ring thus functions as a positive displacement device.

The ring and pin form of generator has some additional advantages over the ball and raceway form, one of which is that the gyratory ring is of larger diameter than an orbital ball for comparable impulse. The ring can therefore be laterally guided at an increased radius, as later more particularly described, and its lateral stability can therefore be more effectively controlled. This is a problem peculiar to fluid drive in this type of generator. The ball and raceway type of generator utilizing fluid stream drive is particularly subject to lateral instability and wobble. Better lateral guidance in the ring and pin type, when properly proportioned, as set forth presently, aids lateral stability. The ring is thus adequately constrained laterally notwithstanding variations or flutter in the fluid drive. Another advantage is that the ring and pin generator can be practically designed with a closer ratio of diameters of circular bearing surfaces than can the ball and raceway generators, meaning that greater bearing area can be afforded in the case of the ring and pin. Another advantage is the inherent compactness and concentration of mass in the pin, considered as the vibration takeoff means, as compared with a ball race.

Another unique feature is that a ring of fairly large mass, and corresponding impulse force, can be used with a desirably short path of travel around the bearing means or pin, leading to the possibility of higher frequencies of operation for a given upper limit of rolling speed of the contact point between the ring and the pin. It is thus possible to drive the resonant member at a desirably high frequency of resonant operation while still having a sufficiently low rolling friction so that the rotor remains fully responsive to the back reaction of the bar at this frequency.

Still another benefit inherent in such a system is that, with an orbiting rotor of the ring type, the effective orbiting mass can be fairly large in relation to the orbital path of the center of gravity of the ring. The heavy mass then tends to hold intimately in contact with the bearing surface on the pin, in spite of any non-linear shock impulses which might be transmitted back along the elastic vibration transmitting member from the vibratory work load. The system thus tends to continue to generate a good sinusoidal vibration pattern in spite of various non-linear environmental factors.

An important discovery of the present invention is that the unique advantages outlined in the foregoing, particularly the sonic frequency stability, maximum force impulse with minimum rolling friction, and high available sonic frequency, are all most particularly effective and available to the maximum degree when the ring is made relatively narrow in axial dimension in comparison to its diameter. A practical or working criterion is that, for the benefit of the present discoveries to be substantially availed of, the axial length of the ring should be less than its external diameter. Such a relatively narrow orbiting ring, I have found, can attain frequencies of operation in relatively high ranges, such as fifteen or more kilocycles per second, because it can be guided into a true and steady path, without having to combat large axial components of forces due to minor irregularities in the shaft, such as irregularities in its diameter. Such a ring also, for similar reasons, has high frequency stability, which is also particularly important with resonant systems, when instability of the oscillator can cause generation of unwanted harmonics.

The invention will be further described in connection with the following detailed description of certain illustrative embodiments thereof, wherein further objects and features of the invention will appear, reference being had to the accompanying drawings, in which:

FIG. 1 is a longitudinal sectional view of one embodiment of the invention;

FIG. 2 is a transverse section taken on line 2—2 of FIG. 1;

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

FIG. 4 is a diagrammatic view illustrative of a standing wave action characteristic of the apparatus of FIG. 1;

FIG. 5 is a side elevation, partly in medial section, of a modified form of the invention;

FIG. 6 is a section on line 6—6 of FIG. 5;

FIG. 7 is a section on line 7—7 of FIG. 6; and

FIG. 8 is a detail side elevation of the rotor of FIGS. 5 and 6.

FIGS. 1 to 4 show one illustrative form of the invention applied to the problem of supplying intense sonic energy to liquids or gases to produce desired physical or chemical effects. Many industrial uses for such a process are known and described in the literature and need not be further discussed herein.

The elastic bar is here in the form of a tube 20, composed of an elastic material, such as steel, which is carried by spaced rubber blocks or sleeves 21 supported by mountings 22, and these blocks 21 are such as will permit a substantial degree of elastic vibration in all directions in planes transverse of the tube. The tube does not rotate bodily, but portions thereof spaced from the nodal point or points of a standing wave set up in the tube gyrate in a circular path by elastic bending of portions of the tube from its neutral position (see FIG. 4). Such circular motion or gyration is a form of harmonic vibration, being the resultant of two components of linear transverse harmonic vibration occurring at right angles to one another with 90° phase difference. The rubber blocks 21 will be seen to comprise compliant mountings permitting such gyratory action.

The vibration generator, generally designated by numeral 24, comprises a cylindric housing 25 formed with a cylindric chamber 26, preferably though not necessarily, coaxial with the tube 20. This housing 25 is formed with one integral side closure wall 25a, and its opposite side is fitted with a removable closure wall 27. A flanged fitting 28 is secured to the wall 25a, and has a threaded projection 29 screwed into the corresponding end of tube 20. A center bearing pin or axle 30 of circular cross-section, preferably formed with a central crowned or barrel-shaped portion 31, has reduced end portions 32 set tightly into the walls 25a and 27. The periphery of this axle 30 provides a circular rolling bearing surface, which is surrounded by an orbital inertia rotor or roller in the form of a relatively narrow and relatively massive ring 33, having a circular and smooth-surfaced central opening 34 of somewhat larger diameter than that of pin 30. This ring preferably has a relatively small diameter opening relative to the outside diameter of the pin 30, so that the external periphery of the pin is not greatly exceeded by the internal diameter of the ring. As shown, the diameters are in the ratios of approximately 3 to 4, and the lengths of the paths around the pin and around the inside of the ring are therefore also typically in the ratio of approximately 3 to 4. Essentially, the diameter of the ring is greater than its axial length, which defines the condition under which the substantial benefits of the invention become importantly available. Still further, the ring 30, as mentioned above, is relatively massive, having a typical radial thickness (in the case of FIGS. 1-3) which is approximately half its axial length. In some applications the outer periphery of the ring has a small clearance with the periphery of the cavity 26 when hanging on the axle 30, or spinning thereabout.

It is important that the side closure walls 25a and 27 be in parallel planes at right angles to the axis of the pins 30, and also that they have close tolerance spacing from opposite sides of the ring 33. Thereby, axial play, wobble, and instability of the ring whirling on the pin is avoided by the thrust bearing and lateral guiding functions fulfilled by the inside surfaces of the walls 25a and 27. In FIG. 3, the clearances are shown disproportionately large for clarity of illustration. In practice,

the clearance on each side is advantageously reduced to something of the order of 0.002" for a ring of, say, 2 inches in diameter.

The inertia ring 33 is caused to roll on its axle 30 by a fluid jet, either air under pressure, steam, or a liquid, introduced through an injection nozzle 35 formed in the housing 25 tangential to the periphery of the circular cavity 26, such fluid being introduced to the nozzle 35 via a hose 36 coupled thereto. The spent driving fluid may be discharged from the chamber 26 in any desired manner; as here shown, it is vented to atmosphere via orifices 37 formed in closure plate 27 as close to the center of the chamber 26 as possible.

The tangentially introduced fluid causes the inertia ring 33 to roll on the axle 30, and the centrifugal force exerted by the rolling ring on the axle 30, and thence transmitted to the housing 25, elastically bends the proximate end portion of the tube 20 and moves it around in a circular path. As earlier pointed out, this motion of the end portion of the tube is a form of harmonic vibration, being the resultant of two perpendicular transverse linear harmonic vibrations in quadrature.

FIG. 4 shows with some exaggeration, the tube 20 undergoing gyratory elastic motion characteristic of the standing wave for the fundamental resonant frequency of the tube for longitudinally propagated transverse elastic waves. It will be understood from known principles that the standing wave diagrammatically indicated in FIG. 4 results from the transmission down the length of the tube, from the generator 24, of transversely oriented elastic deformation waves, which are reflected from the far end of the tube, and through interference with a succeeding forwardly propagated wave, the standing wave is established somewhat as indicated. It will be seen that nodal points occur at sections of the tube approximately one-fourth the length of the tube from each of its ends, while the two ends of the tube are at antinodes of the standing wave.

The speed of rotation of the inertia ring 33 about the axle 30 is in the first instance determined by the fluid jet which drives it. I have discovered, however, that presupposing a drive of the inertia ring at a number of revolutions per second about the axle 30 which approaches or approximates the resonant frequency of the tube 20 for the described transverse mode of vibration so that the tube 20 and the gyrator housing 24 connected to one end thereof will describe circles of augmented amplitude at the antinodal points, the inertia ring 33 then unexpectedly tends strongly to "lock in" at that frequency, i.e., to spin at a number of cycles per second equal to the resonant frequency for the tube 20 and housing 24. I have further found that the spin speed of the ring tends to lag slightly behind the precise resonant frequency for peak resonant amplitude, or in other words, stays on the low side of the resonance curve. Under these conditions, any tendency for overspeeding of the inertia ring with increased pressure on its driving jet is strongly resisted. The resonantly gyrating tube thus exerts a back reaction on the inertia ring, holding the ring at resonant periodicity, but on the low side of the frequency for peak amplitude, thereby preventing it from overspeeding. In other words, under the constraint imposed by the described back reaction from the resonantly vibrating elastic member, the spin frequency of the ring is held to the resonant frequency of the elastic member, slipping to a degree within the driving fluid stream, the fluid jet thus acting as a slip-type drive.

As stated earlier, the ring is closely confined, with small working clearance, between the walls 25a and 27. Also, the length of pin within and contacted by the ring is relatively short, so that axial disturbances or influences on the whirling ring are small. Still further, the ring has a short axial length relative to its diameter, affording a previously unforeseen advantage as regards stability, and

resulting in increase in availability of all the advantages of the ring and pin form of generator.

The generator of FIGS. 1 to 4 will be seen to have the advantageous characteristics and features clearly described in the introductory part of this specification, and a complete repetition thereof at this point will serve no additional purpose. Suffice it to say that the ring and pin generator as described in the foregoing exhibits excellent behavior in the high frequency ranges, being, first of all, capable of being driven at high frequency, further, having both high flywheel effect, and good frequency stability, and finally, yielding a good vibratory force impulse with minimum rolling friction.

The apparatus of FIGS. 1 to 4 is here shown as equipped with means for introducing a fluid to one end of the tube and discharging it from the other. Thus, an inlet tube 40 coupled to a passageway 41 in member 29 introduces the fluid to be treated to one end of tube 20 and an outlet tube 42 mounted in a plug 43 screwed into the opposite end of the tube 20 communicates via a passageway 44 with tube 20 to withdraw treated fluid. It will be understood that fluid within the tube 20 is subjected to sonic agitation. Various known industrial processes capable of making use of such sonic frequency agitation of liquids or gases form no part of the present invention and need not be described herein.

It will further be understood that whereas the generator is shown in FIGS. 1 to 4 as applied in a manner to generate gyratory vibrations, it may be used to generate vibrations of other kinds, typically longitudinal vibrations, as for example in FIGS. 5 to 8.

FIGS. 5 to 8 show an advanced form of sonic vibration generator utilizing an orbital rotor of the fluid-driven, whirling-ring type, the generator being illustratively shown as coupled to an end portion of an elastic bar whose opposite end is coupled to a work load, in an arrangement such that a longitudinal sonic standing wave is set up in the bar. This form of generator is particularly adapted for relatively high frequency applications, and as illustrative of its capabilities, I have operated such a vibration generator at a frequency in excess of 15,000 cycles per second. A frequency of this order is possible because the rotor is made of thin cross-sections and is of very light weight.

In FIGS. 5 and 6, the elastic bar is designated generally at 60. This bar is understood to be composed of some good elastic material such as a good grade of steel or alloy steel. The upper end of this bar 60 is flange-connected, as by bolts, or silver soldering, to the diagrammatically illustrated work load 61, which may be the lower, vibratory wall of a liquid tank whose contents are to be sonically treated, or may be any other device to which vibratory action is to be applied. The lower end of bar 60 is formed with a cylindrical hub 62, into whose bore 63 is press-fitted the sonic vibration generator generally designated at 64. Two circular or disk-like side plates 65 are press-fitted into bore 63 at a spacing to provide a cylindrical chamber 66 for the presently described rotor 67. The side plates 65 are hollowed out to provide manifold cavities 68, and tightly mount an axial bearing pin 69 which extends across rotor chamber 66. Rotor 67, which can be fabricated of ball bearing steel, comprises a cylindrical hub 70 receiving pin 69 with a clearance of the typical proportions shown. The outside diameter of this hub 70 exceeds the axial length thereof. Integral with hub 70 is a medial, radially extending web 71, on opposite sides of which are impeller vanes 72, formed integrally with hub 70 and web 71. These vanes 72 are curved, as shown best in FIG. 8, and are of substantial area, preferably projecting laterally beyond the ends of hub 70. The vanes are formed with radial, outside edges 73 parallel and very closely spaced to the inside surfaces of plates 65, the clearance being preferably of the order of .001 inch on each side, so as to prevent the rotor wobbling or otherwise becoming unstable in its high speed gyration. I have also found it desirable to provide

a very high polish on the inside surfaces of the side plates so as to reduce friction between the rotor and these surfaces to a minimum.

A plurality of nozzle bores 76 are drilled through the inside walls of plates 65 between manifold chambers 68 and rotor chamber 66. These are preferably placed in a circular pattern around pin 69 in a pattern such as indicated in FIG. 7, and are oriented in a tangential direction with reference to rotor chamber 66. They are also preferably so positioned that the fluid jets delivered therefrom impinge on the vanes 72, as near as possible to their innermost junction with the rotor hub, so as to cause the air streams to flow radially outward along the vanes as they deliver energy thereto. It will be evident that this desirable condition can be more fully realized than as shown in FIG. 7 if the rotor hub has less clearance with the pin 69; an excellent performance has been attained in practice with very considerably less clearance proportions than as shown in the illustrative embodiment. The spent air flows off the peripheries of the rotor vanes, and is exhausted via discharge ports 77 formed in the periphery of hub 62, and spaced by a rib 77a. In this case, the ports 77 are formed medially in hub 62, and extend throughout an angle greater than 180°, here substantially three-quarters of a full circle.

The air manifold chambers 68 are fed via a bore 78 in bar 60 leading from an air inlet to a pair of branch passages 79 communicating with ports 80 in side plates 65 opening into cavities 68. A suitable air conduit connection is made to bore 78 through the side of bar 60 as indicated at 81.

The device thus described operates in accordance with principles and with the unique advantages discussed hereabove. This particular whirling ring rotor is adapted for very high frequency operation. A further advantage is that the impeller vanes formed on the rotor add greatly to the efficiency of the generator, since they extract considerably more work or energy from the nozzle jet streams than can a simple ring or ball.

It will be observed that in the device of FIGS. 5 to 8 a longitudinal resonant standing wave is set up in the bar 60 with use of but a single orbiting rotor generator having both longitudinal and lateral components of vibration. It is found in practice, however, that substantial vibration amplitude in the bar 60 can be made to occur only at the longitudinal resonant mode of vibration of the bar by selection of rotor size to give a powerful impulse only up in this range, and in the absence of capability for vibration in strong lateral modes at the frequency of the longitudinal mode, the component of force delivered laterally by the rotor causes very little lateral vibration.

The invention further provides a means for reducing or virtually eliminating any tendency for premature resonant lock in at unwanted modes of vibration below the desired operating frequency, such as lateral modes, or resonant bouncing modes against the load wherein the bar vibrates bodily. It will be evident to those skilled in the art that the bar 60 of FIG. 5 would have a lateral mode of vibration with a resonant frequency lower than its first longitudinal mode. It would be undesirable to permit such a lateral mode to take over control of the rotor, holding back its ability to climb in frequency up to the desired longitudinal mode. The rotor could be driven hard enough to pass the frequency of the unwanted lateral mode, but it is preferable to suppress vibration at any such mode.

To prevent or suppress a lateral mode of such strength, I incorporate vibration damping material at a strategic location relative to the bar. For example, and as shown in FIG. 5, I provide the bar 60 in its central region with a longitudinal slot 85, and place therein a body 86 of viscous, pliable damping material, such as tar, pitch, a thermoplastic, or partially vulcanized rubber. The material introduces shear viscosity damping under the conditions of bending with any tendency for lateral vibration bending modes, so as to prevent any substantial tendency for resonant lock in at such a mode.

To prevent unwanted low frequency longitudinal modes of vibration, such as a resonant bouncing mode wherein the bar vibrates as a whole against the spring action of the load, I preferably surround the bar, in the region of the desired wave pattern (which is the longitudinal center point of the bar in the case of FIG. 5) with an inertia mass ring 87, and I interpose between this ring 87 and the bar a ring-like body 88 of damping material of the same viscous, pliable nature as the already described damping body 86. This damping material, in combination with the inertia ring 87, damps out all longitudinal resonant modes excepting the mode which locates a node of the wave pattern at the damping body.

Both embodiments of the invention now disclosed will be seen to embody the improvements of the invention particularly stressed hereinabove. It will be understood, of course, that the embodiments illustrated herein are for illustrative purposes only and that various changes in design, structure, and arrangement may be made within the spirit and scope of the appended claims.

I claim:

1. In a sonic vibration system, the combination of: an elastically vibratory system having a resonant frequency and a part free to vibrate at said frequency, and sonic vibration generating means for generating vibration in said system at said resonant frequency thereof comprised of a bearing pin fixed to said free part of said system to vibrate therewith, an inertia ring surrounding and guided by said bearing pin for turning in an orbital path, and driving means for driving said inertia ring around said path to vibrate said pin at a frequency in the region of said resonant frequency, said inertia ring having an inside diameter greater than the outside diameter of said bearing pin so that said ring swings about said pin with rolling bearing contact action, and said inertia ring having an outside diameter, which is not substantially less than the axial length thereof.

2. The apparatus of claim 1, including also parallel ring-guide surfaces at right angles to the axis of said pin and on opposite sides of said ring, with close tolerance spacing to said ring, so as to act as lateral bearing guides constraining said ring against material wobble and axial play.

3. The subject matter of claim 1, wherein said means for driving said ring comprises means for directing a jet of air toward the outside periphery of said ring with a component of velocity tangential thereto so as to impinge on the ring and cause it to gyrate about said pin.

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