Dec. 3, 1968 A. G. BODINE 3,414,203

APPARATUS FOR CRUSHING ROCK MATERIAL AND THE LIKE UTILIZING COMPLEX SONIC WAVE ACTION

Filed Nov. 7, 1966 4 Sheets-Sheet 1

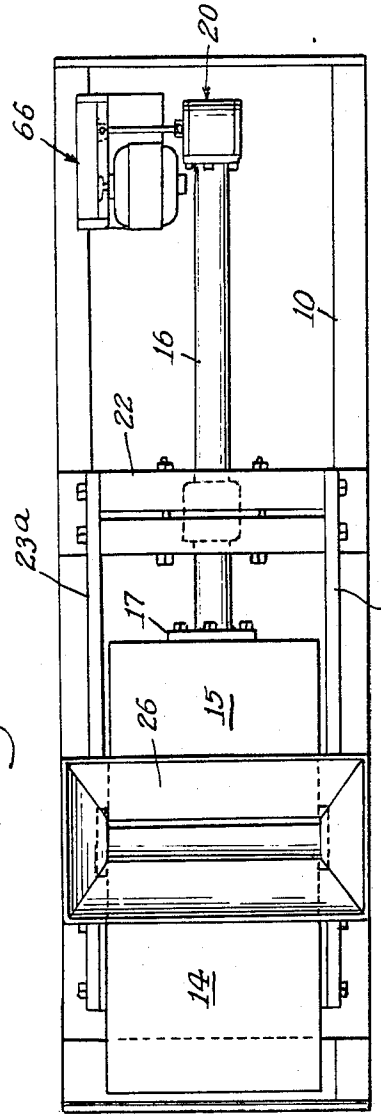



Fig. 1

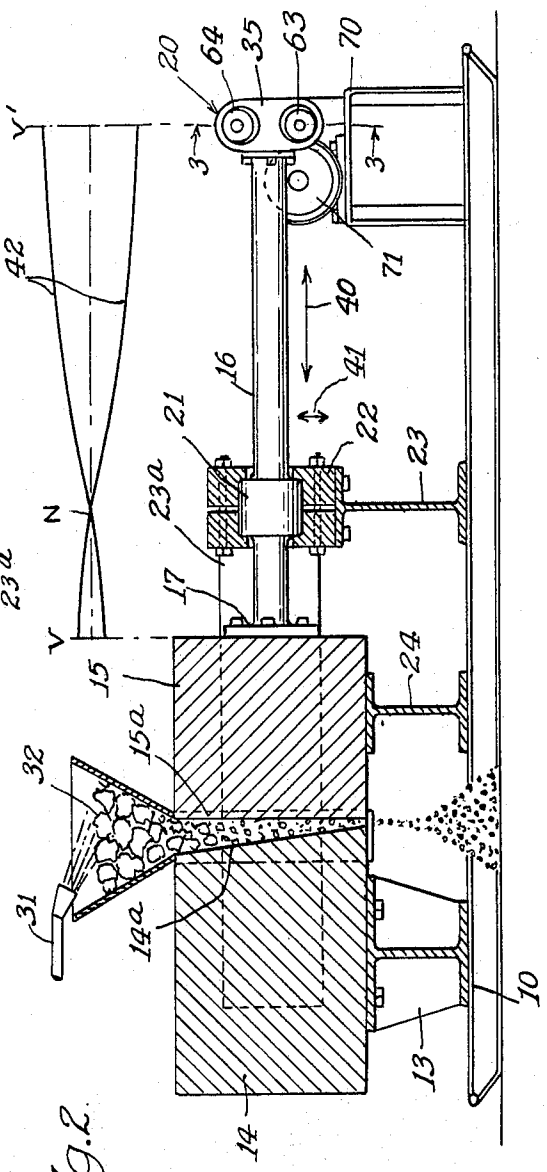



Fig. 2

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Fig. 3

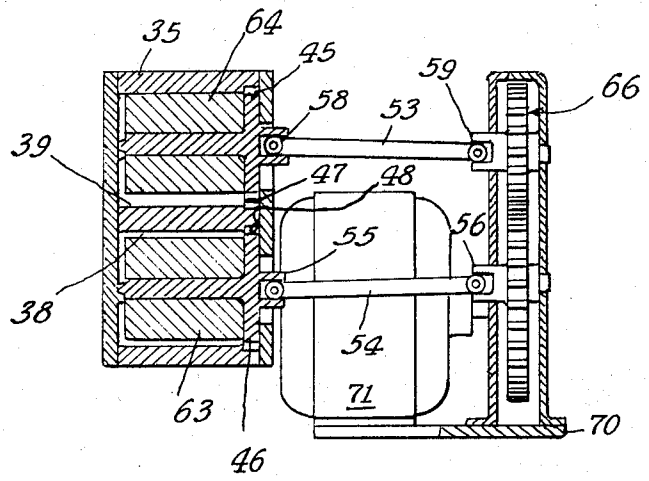
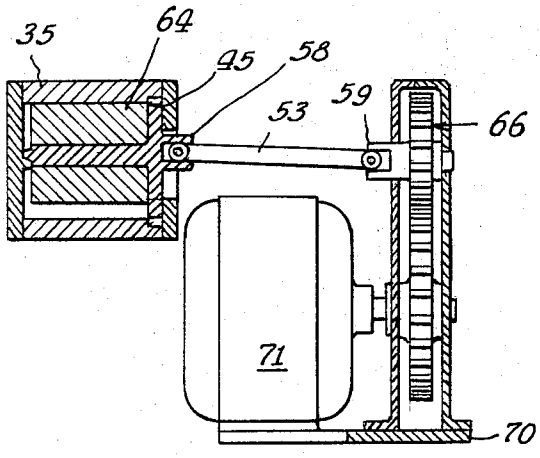



Fig. 4




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APPARATUS FOR CRUSHING ROCK MATERIAL AND THE LIKE UTILIZING COMPLEX SONIC WAVE ACTION

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Fig. 5

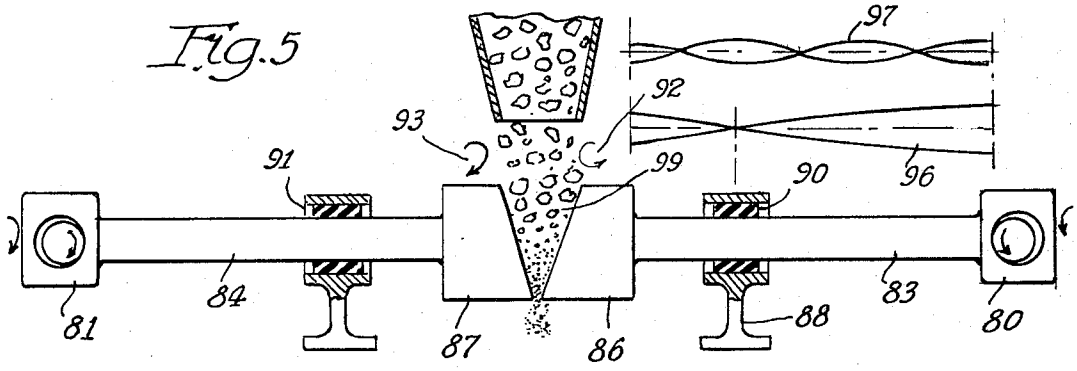
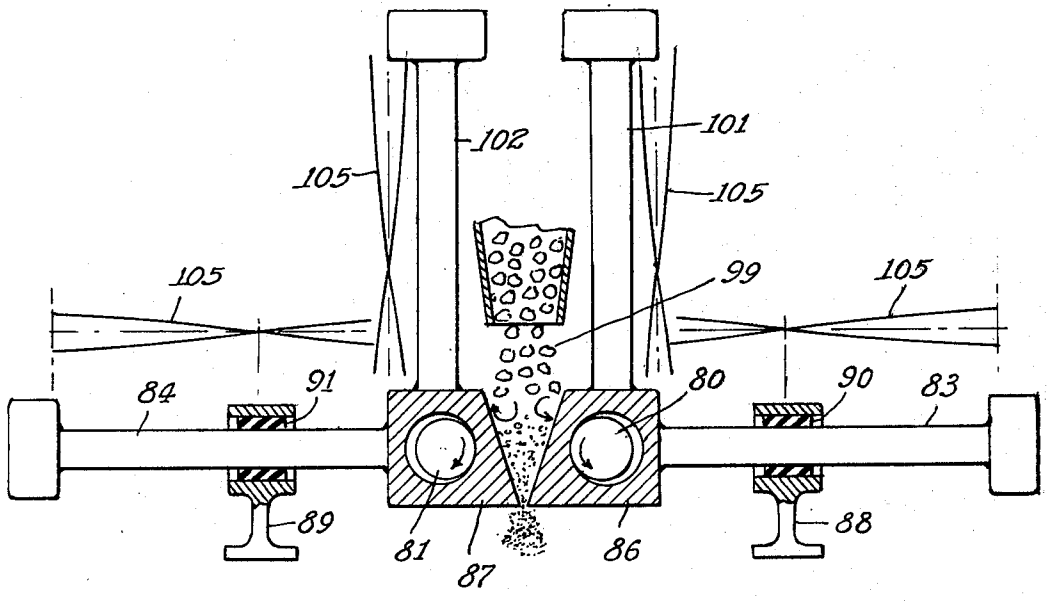



Fig. 6




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APPARATUS FOR CRUSHING ROCK MATERIAL AND THE LIKE UTILIZING COMPLEX SONIC WAVE ACTION

Filed Nov. 7, 1966 4 Sheets-Sheet 4

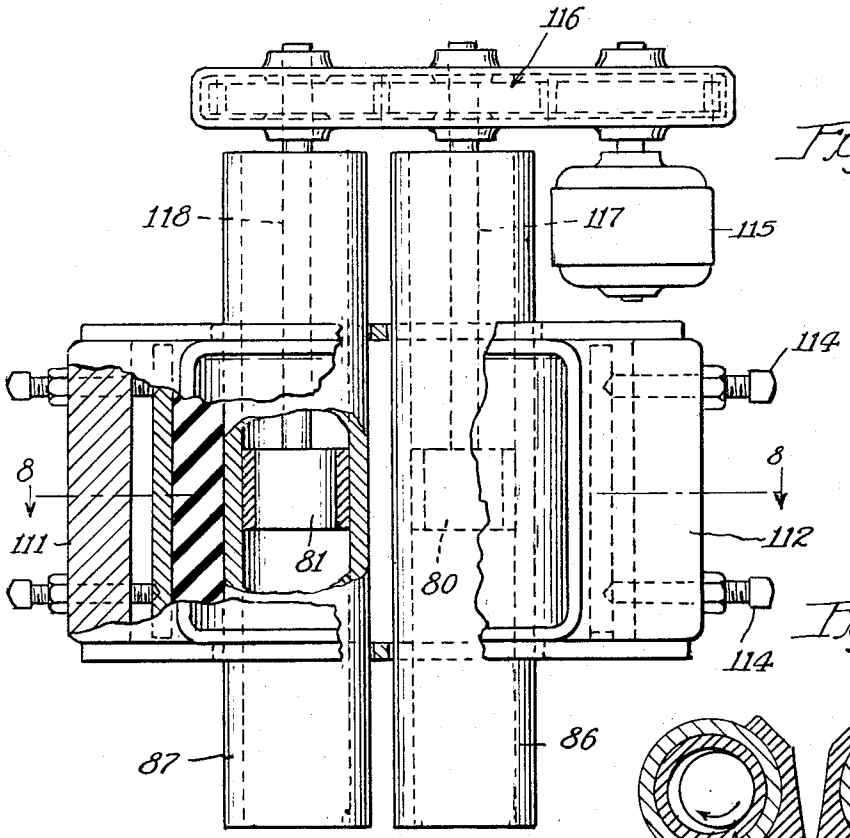



Fig. 7.

Fig. 9.

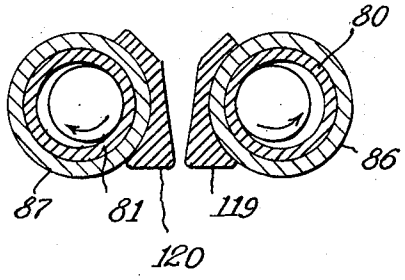
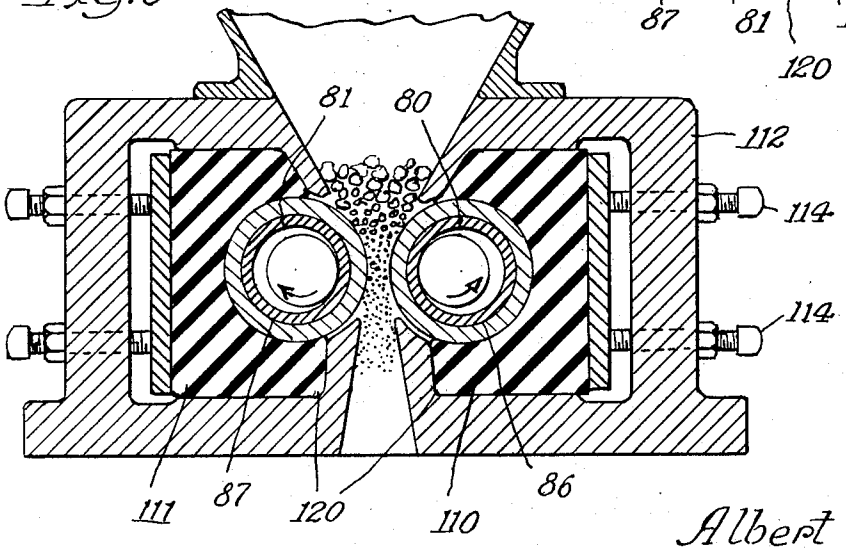



Fig. 8




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United States Patent Office 3,414,203 Patented Dec. 3, 1968



3,414,203

# APPARATUS FOR CRUSHING ROCK MATERIAL AND THE LIKE UTILIZING COMPLEX SONIC WAVE ACTION

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Continuation-in-part of application Ser. No. 341,608, Jan. 31, 1964. This application Nov. 7, 1966, Ser. No. 592,517

10 Claims. (Cl. 241—202)

This application is a continuation-in-part of my copending application Ser. No. 341,608, filed Jan. 31, 1964, now Patent No. 3,284,010, entitled Crushing Process and Apparatus With Sonic Wave Action.

This invention relates to apparatus for crushing rock material and the like utilizing complex sonic wave action and more particularly to such apparatus in which sonic energy is used both to crush the material and to efficiently feed such material through the crushing jaws.

In the aforementioned copending application, apparatus is described in which sonic energy is applied to one or both of a pair of crushing jaws which form part of a resonant vibration circuit. Rock material to be crushed is passed between the jaws through a wedge-shaped passage formed therebetween and acted upon by such energy so that such rock material is efficiently crushed to a particle size determined by the dimensions of the exit end of the passage. In the apparatus described in the aforementioned patent application, the sonic energy is generated along a single axis so as to cause solely longitudinal jaw vibration, this in interests of obtaining maximum utilization of the sonic energy for the crushing operation. It has been found that while the use of such a single-axis vibratory mode provides highly efficient crushing action, certain difficulties are encountered in some situations in properly passing the rock material through the wedge-shaped jaws, in some cases the rock material popping up out of the jaws. Further, in depending solely upon gravity feed, the taper angle of the feed passage must be limited in the interests of efficient feed-through, thus placing a distinct limitation on the particle size reduction capabilities of the crushing apparatus.

The apparatus of this invention, by utilizing a complex force wave pattern having a downward thrusting force component, enables a substantial increase in the throughput of the material being crushed and makes possible the utilization of a subestantially wider angle between the crushing jaws thereby increasing the rock particle size reduction capabilities of the apparatus. In addition, the problems encountered with rock material being forced upwardly out from between the crushing jaws in the crushing process are substantially eliminated by virtue of the downward force components.

The improvement is achieved in the apparatus of this invention by positioning, operating and designing the force generation system so that the sonic energy has a complex gyratory wave pattern including both longitudinal and transverse force components. In certain embodiments of the device of the invention complex wave motion having relatively large-amplitude longitudinal components as compared with the transverse components are utilized while in other embodiments, apparatus for generating forces having transverse components of the same order of the longitudinal components thereof are described, the particular type of complex wave action utilized depending upon the particular application requirements involved. In certain embodiments of the device of the invention, the orbiting mass oscillator units which are utilized to generate the sonic energy are mounted in the jaws themselves to enable the efficient driving of such oscillator units with a short shaft from the single drive motor.



It is therefore an object of this invention to provide an improved sonic rock crushing apparatus.

It is another object of this invention to provide sonic rock crushing apparatus in which complex sonic wave action is utilized to improve the feeding of material to be crushed through the crushing jaws.

It is still another object of this invention to increase the particle size reduction capability of sonic rock crushing apparatus.

It is still a further object of this invention to increase the throughput rate of sonic rock crushing apparatus.

It is still a further object of this invention to decrease the likelihood of material being crushed popping out from between the jaws of a sonic rock crusher.

Other objects of this invention will become apparent from the following description taken in connection with the accompanying drawings, of which:

FIG. 1 is a top plan view of a first embodiment of the device of the invention;

FIG. 2 is a side elevational view in cross-section of the embodiment shown in FIG. 1;

FIG. 3 is a side elevational view in cross-section of an oscillator unit that may be utilized in the embodiment of FIGS. 1 and 2;

FIG. 4 is a side elevational view in cross-section of an oscillator unit that may be utilized in lieu of the dual oscillator shown in connection with first embodiment of the device of the invention;

FIG. 5 is a drawing illustrating the basic structural components of a second embodiment of the device of the invention;

FIG. 6 is a drawing illustrating the basic structural components of a third embodiment of the device of the invention;

FIG. 7 is a top plan view with partial cut-away section of a fourth embodiment of the device of the invention;

FIG. 8 is a cross-sectional view taken along a plane indicated by 8—8 of FIG. 7; and

FIG. 9 is a cross-sectional view in elevation illustrating a replaceable jaw device that may be utilized in conjunction with the embodiment of FIGS. 7 and 8.

It has been found most helpful in analyzing the operation of the device of this invention to analogize the acoustically vibrating circuit involved to an equivalent electrical circuit. This sort of approach to analysis is well known to those skilled in the art and is described, for example, in Chapter 2, of Sonics, by Hueter and Bolt, published in 1955 by John Wiley and Sons. In making such an analogy, force, F, is equated with electrical voltage, E; velocity of vibration, $u$, is equated with electrical current, $i$; mechanical compliance, $C_m$, is equated with electrical capacitance, $C_e$; mass, M, is equated with electrical inductance, L; mechanical resistance (friction), $R_m$, is equated with electrical resistance, R; and mechanical impedance, $Z_m$, is equated with electrical impedance, $Z_e$.

Thus, it can be shown that if a member is elastically vibrated by means of an acuostical sinusoidal force, $F_0 \sin \omega t$, ($\omega$ being equal to $2\pi$ times the frequency of vibration), that

$$Z_m = R_m + j\left(\omega M - \frac{1}{\omega C_m}\right) = \frac{F_0 \sin \omega t}{u} \quad (1)$$

Where $\omega M$ is equal to $1/\omega C_m$, a resonant condition exists, and the effective mechanical impedance, $Z_m$, is equal to the mechanical resistance, $R_m$, the reactive impedance components $\omega M$ and $1/\omega C_m$ cancelling each other out. Under such a resonant condition, velocity of vibration, $u$, is at a maximum, power factor is unity, and energy is most efficiently delivered to a load to which the resonant system may be coupled.

It is important to note the significance of the attain-

ment of high acoustical "Q" in the resonant system being driven, to increase the efficiency of the vibration thereof and to provide a maximum amount of energy for the grinding operation. As for an equivalent electrical circuit, the "Q" of an acoustical vibration circuit is defined as the sharpness of resonance thereof and is indicative of the ratio of the energy stored in each vibration cycle to the energy used in each such cycle. "Q" is mathematically equated to the ratio between $\omega M$ and $\omega R_m$. Thus, the effective "Q" of the vibrating circuit can be maximized to make for highly efficient high amplitude vibration by minimizing the effect of friction in the circuit and/or maximizing the effect of mass in such circuit.

Of significance in the implementation of the method and devices of this invention, is the high acceleration of the components of the elastic resonant system that can be achieved at sonic frequencies. The acceleration of a vibrating mass is a function of the square of the frequency of the drive signal times the amplitude of vibration. This can be shown as follows:

The instantaneous displacement, $y$, of a sinusoidally vibrating mass can be represented by the following equation:

$$y=Y \cos \omega t \qquad (2)$$

where Y is the maximum displacement in the vibration cycle and $\omega$ is equal to $2\pi f$, $f$ being the frequency of vibration.

The acceleration, $a$, of the mass can be obtained by differentiating Equation 2 twice, as follows:

$$a=\frac{d^2y}{dt^2}=-Y\omega^2 \cos (\omega t) \qquad (3)$$

The acceleration, $a$, thus is a function of $Y\times(2\pi f)^2$. At resonance, Y is at a maximum and thus even at moderately high sonic frequencies, very high accelerations are achieved making for correspondingly high vibrational forces at the grinding interfaces.

In considering the significance of the parameters described in connection with Equation 1, it should be kept in mind that the total effective resistance, mass, and compliance in the acoustical vibration circuit are represented in the equation and that these parameters may be distributed throughout the system rather than being lumped in any one component or portion thereof.

It is also to be noted that an orbiting mass oscillator may be utilized in the device of the invention that automatically adjusts its output frequency to maintain resonance with changes in the characteristics of the load. Thus, in the face of changes in the effective mass and compliance presented by the load, the system automatically is maintained in optimum resonant operation by virtue of the "lock in" characteristics of applicant's unique orbiting mass oscillator. The orbiting mass oscillator automatically changes not only its frequency but its phase angle and therefore its power factor with changes in the resistive impedance load to assure optimum efficiency of operation at all times.

Referring now to FIGS. 1 and 2, a first embdoiment of the device of the invention is illustrated. This first embodiment is structurally the same as one of the embodiments described in the aforementioned application of which this application is a continuation-in-part, except for the phasing of the oscillator rotors. Stationary jaw member **14** which provides a high inertia mass is supported on I-beam support **13** which in turn is mounted on base member **10**. Jaw member **14** has the inertia necessary to withstand or absorb a large periodic force impulse in the operation of the crusher without substantial yield or vibration. Mounted opposite stationary jaw member **14** is vibratory jaw member **15** which is in the general form of a rectangular block. Jaw member **15** is supported on I-beam **24** which in turn is supported on base member **10**. Coupled to vibratory jaw member **15** by means of coupler flange **17** is elastic shaft member **16** which is preferably made of an elastic steel. Shaft member **16** is supported on I-beam support **23** which is mounted on base **10** by means of split mounting block **22**. Side straps **23***a* are fastened at opposite ends thereof to block **22** and stationary jaw **14** to steady these members.



Attached to the end of elastic vibratory shaft **16** is orbiting mass oscillator **20**. Orbiting mass oscillator **20** comprises a casing **35** having a pair of rotors **63** and **64** mounted for rotation in races formed therein. Rotors **63** and **64** are driven in opposite directions at the same rotation speed by means of motor **71** through gear train **66**. Motor **71** is supported on stand **70** which is amounted on base **10**.

Referring now additionally to FIG. 3, rotors **63** and **64** are phased with respect to each other so that one of the rotors is slightly ahead of the other in its rotation cycle. Thus, when, for example, rotor **64** is in the "12 o'clock" position, rotor **63** may be in the "7 o'clock" position (instead of the "6 o'clock" position as in the aforementioned application of which this is a continuation-in-part). By offsetting the phasal relationship of the rotors in this manner, a gyratory force is imparted to elastic vibratory shaft **16** causing it to vibrate in an elliptical path. The gyratory force pattern can be vectorially resolved into a larger force vector along the longitudinal axis of shaft **16** and a smaller force vector transverse to this longitudinal axis, such force vectors being indicated by arrows **40** and **41**, respectively. The rotation speed of motor **71** is adjusted to cause resonant vibration of the vibration system including shaft **16** and vibratory jaw **15** to set up a standing wave pattern in shaft **16** as indicated by graph lines **42** with a velocity antinode indicated by V at the oscillator end of the shaft being transformed into a relatively low velocity indicated by V at the end of the shaft coupled to the high mass crushing jaw **15**.

Sonic generator **20** may be of a type described in my Patent No. 3,217,551, issued Nov. 16, 1965, which is illustrated schematically in FIG. 2. Briefly, this oscillator comprises a pair of rotors **63** and **64** which are mounted in housing **35** for rotation about separate races **38** and **39** formed by the inner walls of the housing. The rotors have gear rings **45** and **46** attached thereto which matingly engage gear rings **47** and **48**, respectively, formed in the housing. Rotors **63** and **64** are driven in opopsite directions by means of motor **71** which drives gear train **66** which in turn is coupled to shafts **53** and **54** through universal joints **59** and **56** respectively. Shafts **53** and **54** are coupled to rotors **64** and **63**, respectively, through universal joints **58** and **55**. As already noted, rotors **63** and **64** are phasally arranged with respect to each other so that while a large part of the transverse vibrational components effectively cancel each other out, leaving a predominantly longitudinal vibration mode in the output, there still remains a significant amount of transverse vibrational component as indicated by arrows **41**, the importance of which will now be explained in connection with FIG. 2.

Rock material **32** to be crushed is fed into hopper **26** from feeder **31**. This material passes from the hopper into a wedge-shaped passage formed between the sloped wall **14***a* of stationary jaw member **14** and the straight wall **15***a* of vibratory jaw member **15**. Jaw member **15** resonantly vibrates towards and away from jaw member **14***a* in the directions indicated by arrows **40** with relatively high force. The vibratory energy is transferred to the rock material and the rock material is rapidly fatigued and shattered. At the same time as the force component **40** of the resonant vibrational energy is achieving the crushing action, the force component **41** acts to move the rock material downwardly through the jaws and is phased with a sort of "pawing" action. Thus the rock material is kept moving through the jaws in an efficient manner.

If a greater downward force component is desired as,

for example, where greater throughput is required or greater reduction in the particle size of the rock material is to be achieved with jaws having a wider angle, a single oscillator unit may be utilized as shown in FIG. 4 in place of the dual oscillator unit of FIG. 3. The single rotor **64** is driven by motor **71** through gear train **66** in same manner as for the oscillator unit illustrated in FIG. 3. In this instance, however, the vertical force component is practically equal to the horizontal force component. It is to be noted that while the output of the single oscillator unit is in the form of a circular gyratory force pattern, that in view of the greater stiffness or mass handling ability of shaft **16** in its longitudinal dimension than its lateral dimension, greater amplitude longitudinal vibration occurs than transverse vibration resulting in an elliptical gyration of the shaft with greater amplitude in the longitudinal than the transverse mode.

Referring now to FIG. 5, a second embodiment of the device of the invention is illustrated. In this embodiment, two separate orbiting mass oscillators **80** and **81** are mounted at the ends of resonant bars **83** and **84** respectively. The opposite ends of resonant bars **83** and **84** are attached to jaw members **86** and **87** respectively. Resonant bars **83** and **84** are supported on associated support members **88** and **89** and are vibrationally isolated therefrom by means of isolator members **90** and **91**. The rotors of oscillators **80** and **81** are rotated at the same speed in opposite directions by motor means (not shown) to produce resonant gyratory vibration of their associated bar members. This results in a cooperative substantially circular motion of each of jaw members **86** and **87** as indicated by arrows **92** and **93**, respectively. Longitudinal resonant vibration is set up in each of the bar members as indicated by resonant pattern **96** while transverse standing wave vibration is set up therein as indicated by wave pattern **97**. This embodiment has an advantage in that in view of the cooperative substantially circular motion of the jaws relative sliding action therebetween is avoided, thereby greatly reducing friction effects and wearing of the jaw faces. Also the cooperative downward motion of the jaws tends to give the rock material **99** a maximum downward impelling force thereby greatly increasing the pumping action. This increases the throughput and enables the use of a wider jaw angle thereby making possible a greater reduction in the size of the rock particles. Further, the tendency of the rock material **99** to pop out from between the jaws is lessened. In the embodiment of FIG. 5, the mass of the resonant bars **83** and **84** is relatively large as compared with that of jaw members **86** and **87**. This tends to greatly increase the relative magnitude of the downward or lateral wave portion of the composite wave which, as already noted, provides distinct advantages where greater downward pumping force is required.

Referring now to FIG. 6, a third embodiment of the device of the invention is illustrated. This embodiment differs from the previous embodiment in that the oscillators **80** and **81** are mounted right in the jaw members **86** and **87** themselves and in that additional resonant bar members **101** and **102** are attached to the jaw members to enhance the downward resonant vibration component. Resonant bar members **83** and **84** as for the previous embodiment are supported on supports **88** and **89** and vibrationally isolated therefrom by vibration isolators **90** and **91**. The additional resonant bar members **101** and **102** extend vertically from the tops of jaws members **86** and **87**, respectively, and are welcomed thereto.

This embodiment has the advantage that in view of the fact that the rotors are relatively close to each other, it is relatively easy to drive them from a single motor with an arrangement of the type illustrated in FIG. 3. The use of separate resonant bars for each of the vertical and horizontal force components assures optimum force output and highly efficient vibration thereby providing maximum force output at the crushing jaws. The positioning of the oscillators right in the jaws themselves with the resonant bars extending outwardly from the jaws enables a mechanization with four separate resonant vibration systems utilizing only two oscillators which are driven by a single drive motor. It is to be noted that all four resonant bars are preferably designed to resonate longitudinally at the same frequency so that with the oscillators driven by a single motor, optimum resonant operation is achieved with a resultant in-phase vibration of the jaws. The simple resonant vibration of the bar members is indicated by graph lines **105**.

Referring now to FIGS. 7 and 8, a fourth embodiment of the device of the invention is shown. In this embodiment, the jaw members **86** and **87** take the form of elastic tubular members with the oscillator units **80** and **81** being mounted directly therein. Here again the oscillator units may be of the type shown in FIG. 3. The oscillator rotors are driven by means of motor **115** through gear train **116** which is coupled to oscillator drive shafts **117** and **118**. The jaw members **86** and **87** are supported in housing **112** and are vibrationally isolated from the housing by means of isolator pads **110** and **111** which locate the jaw members by means of bolts **114**. In this embodiment, the jaws are resonantly vibrated in a wave pattern comprising two quadrature related waves transverse to the longitudinal axis of such jaws. This results in a circular in-phase motion of the jaw members. This provides a highly effective crushing action and also highly effective downward pumping action. Oscillator units **80** and **81** may be press-fitted into their associated jaw members.

Referring now to FIG. 9, a modification of the embodiment of FIGS. 7 and 8 wherein removable jaws are attached to tubular members **86** and **87** is illustrated. Jaw members **119** and **120** are welded to the sides of tubular members **86** and **87** and are utilized to perform the crushing action rather than the sides of the tubular members. Otherwise the structure and operation are identical as described for the previous embodiment. This embodiment has the advantage in that the jaws can be removed and repaired and replaced as necessary, thus avoiding the necessity for replacing the elongated tube members themselves.

The apparatus of this invention, thus by providing a downward pumping action in addition to the sideways crushing action, affords the advantages of greater throughput and a capability for greater reduction in rock size in a single operation than heretofore possible and avoids the popping of rock material out from between the jaws with the vibratory crushing action.

While the device of the invention has been described and illustrated in detail, it is to be clearly understood that this is intended by way of illustration and example only, and is not to be taken by way of limitation, the spirit and scope of the invention being limited only by the terms of the following claims.

I claim:

**1.** In a crushing apparatus:

a pair of opposed crusher members, at least one of which is vibratory toward and away from the other;

means for feeding material to be crushed between said jaws;

a sonic wave generator adapted to deliver a complex alternating output force; and

an elastically vibratory wave transmission system comprising a member of solid elastic material having a range of elastic vibration intercoupled between said generator and said vibratory jaw so as to receive said alternating force, undergo corresponding elastic vibration and impart vibration to said vibratory jaw;

said sonic wave generator having a complex gyratory force output which vibrates said jaw both sidewise and in up and down directions.

**2.** The apparatus as recited in claim **1** wherein said sonic wave generator comprises a pair of orbiting mass oscillators and means for rotating said oscillators in op-

posite directions at the same speed, said oscillators being phasally offset with respect to each other to generate both longitudinal and transverse phased force components in said wave transmission system.

3. The apparatus as recited in claim 1 wherein both of said jaws are vibratory toward and away from each other and wherein said sonic wave generator comprises separate oscillator means for each of said jaws, said elastically vibratory wave system comprising a similar mem- said jaws and each of said oscillator means respectively, ber of solid elastic material intercoupled between each of and means for driving said oscillator means to cause substantially circular vibration of said jaws.

4. In a crusher apparatus:

a pair of opposed crusher members, both of said crusher members being vibratory toward and away from each other,

means for feeding material to be crushed between said crusher members,

a sonic wave generator mounted on each of said crusher members,

a separate elastic member of solid elastic material having a range of elastic vibration coupled to each of said crusher members, each of said crusher members and an associated one of said elastic members forming a separate vibration system, and

means for driving said wave generators to cause complex sonic vibration of said crusher members to produce cooperative vibration thereof both towards and away from each other and cooperaitvely in an upward direction.

5. The crusher apparatus of claim 4 and including a pair of mutually perpendicular solid elastic members coupled to each of said crusher members, each of said elastic members forming a separate resonant vibration system with an associated crusher member to cause circular vibration of said crusher members.



6. The crusher apparatus of claim 5 wherein one of the elastic members of each of said pairs thereof is oriented vertically and the others of said elastic members are oriented horizontally.

7. The crusher apparatus of claim 4 wherein said crusher members and said elastic members are comprised of tubular members positioned alongside each other.

8. The crusher apparatus of claim 7 wherein said sonic wave generators comprise orbiting mass oscillators mounted within each of said tubular members.

9. The crusher apparatus of claim 7 and additionally including removable jaw members attached to said tubular members.

10. The crusher apparatus of claim 4 wherein said elastic members have a large mass as compared with that of said crusher members.


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F. T. YOST, *Assistant Examiner.*

# UNITED STATES PATENT OFFICE

# CERTIFICATE OF CORRECTION

Patent No. 3,414,203 December 3, 1968

Albert G. Bodine

It is certified that error appears in the above identified patent and that said Letters Patent are hereby corrected as shown below:

Column 6, line 66, "vibration" should read -- vibrations --. Column 7, line 10, cancel "said jaws and each of said oscillator means respectively," and insert the same after "each of" in line 11; same column 7, line 30, after "upward" insert -- downward --.

Signed and sealed this 17th day of March 1970.

(SEAL)
Attest:

Edward M. Fletcher, Jr.
Attesting Officer

WILLIAM E. SCHUYLER, JR.
Commissioner of Patents