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METHOD AND APPARATUS FOR ROCK CRUSHING
UTILIZING SONIC WAVE ACTION
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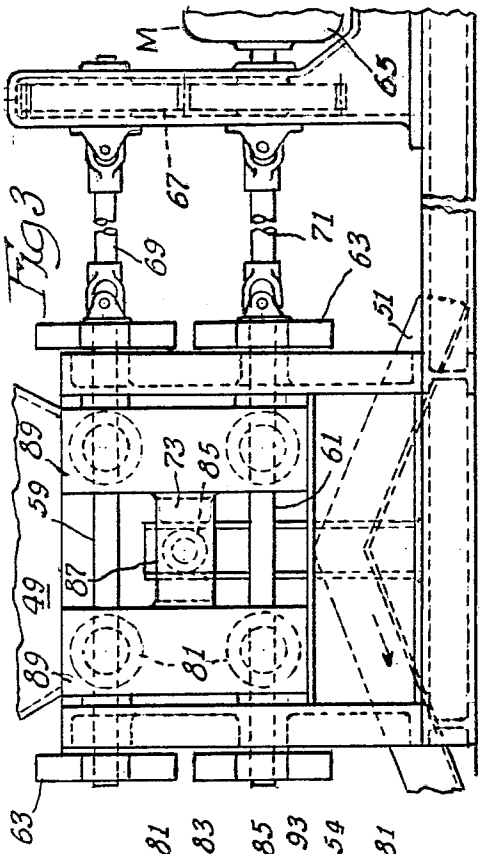


Fig 3

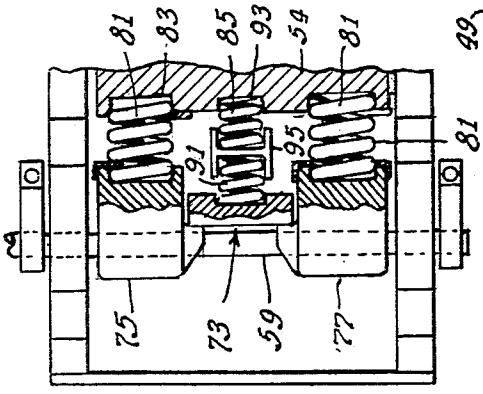


Fig. 2

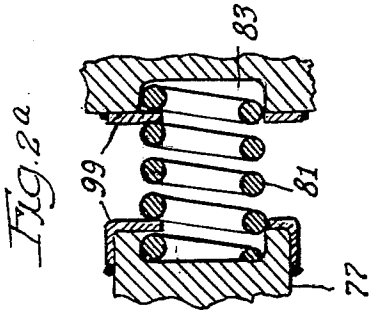


Fig. 2a

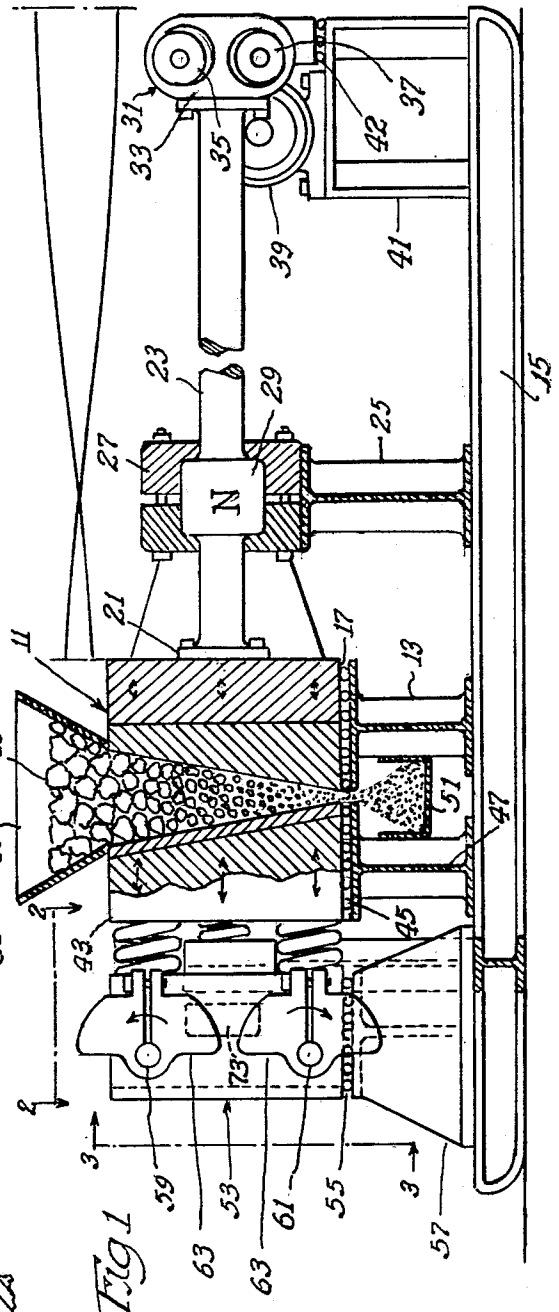


Fig 1

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METHOD AND APPARATUS FOR ROCK CRUSHING UTILIZING SONIC WAVE ACTION

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ABSTRACT OF THE DISCLOSURE

A method and apparatus for rock crushing utilizing sonic wave action, comprising applying to a rock crusher having two jaws a resonant sonic energy force of relatively high frequency to the jaw combination while simultaneously applying a resonant sonic energy of a much lower frequency to the jaw combination.

The herein invention is an improvement upon the method of rock crushing disclosed in U.S. Patents Nos. 3,131,878 and 3,284,010 of May 5, 1964 and Nov. 8, 1966, respectively, of the same inventor, and that disclosed in copending application Ser. No. 592,517, filed Nov. 7, 1966, now Patent No. 3,414,203, by the same inventor. In the issued patents there is disclosed a method of crushing rocks utilizing sonic wave action wherein 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 materials is efficiently crushed to a particle size determined by the dimensions of the exit end of the passage. In the apparatus described in the patent, the sonic energy is generated along a single axis so as to cause solely longitudinal jaw vibration. It was found, however, that while the use of such a single axis vibratory mode provided highly efficient crushing action, difficulties were encountered in some situations improperly passing the rock material through the wedge-shaped jaws. In some cases, the rock material popped out of the jaws. Further, in depending solely upon gravity feed the taper angle of the feed passage had to be limited, placing a resultant limitation on the partical size reduction capabilities of the apparatus.

The copending application referred to overcame these deficiencies by utilizing a complex force wave pattern having a downwardly thrusting force component. The improvement in the copending application consisted in positioning, operating and designing the force generation system so that the sonic energy had a complex gyratory wave pattern including both longitudinal and transverse force components. In certain embodiments of the copending application, a complex wave motion having relatively large-amplitude longitudinal components as compared with the transverse components was utilized. In other embodiments, apparatus were described for generating forces having transverse components of the same order as the longitudinal components thereof. In these embodiments, as indicated, the desired end result was to achieve an additional downward thrust component that aided the rock in passing through the crusher jaws.

While the aforegoing patent and copending application are very effective for a wide range of materials, there are certain materials which become agitated by the crushing process in such a way that the material develops a dynamic degree of suspension between the jaws. This results in the material dynamically spinning away from the crushing jaws. This effect results from the fact that the sonic energy activates the material being crushed in such

a manner that the material achieves a dynamic fluidized state tending to counteract the action of gravity so as to spin away from the jaws. Under these conditions, the jaw strikes the material only at the extreme stroke of the jaw. During the interval when the jaw is retracting from the material, the particles do not have time to fall so as to be in a position for engagement by the jaw on its next stroke toward the material. Additionally, as can be seen, the sonic activation of the jaw can be so fast that the material cannot fall very far between the impacts of the jaw. Since it does not fall very quickly, then the material remains in a dynamically suspended state a short distance away from the main position of the jaw. This result can even occur in the improvement disclosed in the copending application where a downward force vector is achieved. However, as can be appreciated, with the downward vector the material in a dynamic fluidized state and away from the jaws cannot fully benefit from this vector force so as to adequately move down and pass through the jaws in the desired manner.

The present invention overcomes the foregoing problems resulting from the utilization of the devices described. This is accomplished in the herein invention by superimposing a lower frequency upon the jaws of the device, which in turn modulates the higher frequency imposed upon the jaws. Thus, in the system of this invention, the jaw combination is experiencing two simultaneous vibrations. The low-frequency vibration causes a low-frequency excursion of the two jaws relative to each other and then back again. When the low-frequency jaw retracts, the material between the jaws is allowed to fall a substantial distance between the jaws before the next movement of the jaw toward the high-frequency one. To accomplish this effect, in one embodiment a jaw is constructed in identically the same manner as described in either the aforementioned patent or copending application, and is driven by orbiting-mass resonators as described therein. The opposing jaw of the rock crusher in this embodiment then is driven through orbiting masses acting upon heavy springs, which in turn are connected to the jaws. The springs present necessary lumped compliance to the system so as to achieve within a compact system the necessary low-frequency vibrations desired. The invention will be explained in more detail with reference to the following drawings and description in which:

FIG. 1 is a partially sectioned view of the rock crushing apparatus of this invention;

FIG. 2 is taken along lines 2—2 of FIG. 1 showing the connection of the orbiting masses to the low-frequency vibratory jaw of the device;

FIG. 2*a* is an enlarged sectional view of the spring connecting the low-frequency orbiting masses to the jaws of the device; and

FIG. 3 is a rear side view taken along lines 3—3 of FIG. 1 particularly depicting the drive mechanism for the low-frequency orbiting masses.

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 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 acoustical sinusoidal force, $F_0 \sin \omega t$ (ω being equal to 2π 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 ωM 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 ωM 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 couple.

It is important to note the significance of the attainment 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 ωM and ωR_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 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:

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

where Y is the maximum displacement in the vibration cycle and ω 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:

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

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 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.

Turning now to FIG. 1, there is illustrated one form of the device of this invention. A vibratory jaw **11** rests on an I-beam support **13** which in turn is affixed to a base **15** for the device. Unlike the prior apparatus the high-frequency jaw **11** is mounted on a plurality of roller bearings **17** which permit relative movement between it and the I-beam **13** during the vibration. It has been found that the utilization of such bearings **17** prevents undue dissipation of the vibratory energy into the structure through the I-beam support **13**, maximizing its use for crushing the rock **19**. Coupled to the vibratory jaw **11** by means of a coupler flange **21** is elastic shaft member **23** which is preferably made of an elastic steel. Shaft member **23** is supported on I-beam support **25** which is mounted on base **15** by means of a split mounting block **27**. Surrounding the shaft **23** within the mounting block **27** is a mounting collar **29**.

Attached to the end of the elastic vibratory shaft **23** is an orbiting-mass oscillator **31**. The orbiting-mass oscillator **31** comprises a casing **33** having a pair of rotors **35** and **37** mounted for rotation in races formed therein. Rotors **35** and **37** are driven in opposite directions at the same rotation speed by means of motor **39** through a gear train (not shown). Motor **39** is supported on a stand **41** which is mounted on base **15**. The rotor housing **33** is additionally supported on stand **41** through roller bearings **42**. The rollers **42** prevent dissipation of the vibratory force generated in the housing **33** into the support **41**.

A description of the operation of the oscillator **31** and the specifics of its construction will not be given since it is fully described in the aforementioned issued patents. As indicated, the herein invention can operate in the manner disclosed in those patents, or alternatively can utilize the improvement described in the copending application. As previously mentioned, the improvement of the copending application, which is incorporated herein by reference, involves the rotors **35** and **37** being 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 vibrational mode in the output, there still remains a significant amount of transverse vibrational component which serves to aid the rock passing through the jaws of the device.

The low-frequency vibratory jaw **54** is disposed oppositely of the high-frequency one **11**, and rests on rollers **45** which in turn are supported by an I-beam structure **47**, in the exact same manner as described with regard to the high-frequency jaw. A hopper **49** connected between the two jaws at the top thereof directs the material to be crushed **19** between the jaws, while a ramp **51** shown in dotted outline particularly in FIG. 3, is connected between the I-beam support structures **13** and **47**, respectively, to carry the crushed rock away from the jaws. Disposed adjacent the low-frequency jaw **43** is an oscillator structure **53**, resting on rollers **55** which in turn are supported by a large I-beam structure **57**. Passing through the structure **53** are two axles **59** and **61**. Eccentric weights **63** are mounted on each end of each axle, thus there are a total of four such weights as particularly seen in FIG. 3. Disposed adjacent to the structure **53** as shown in FIG. 3, is a motor **65** which drives through a gear train **67** connecting rods **69** and **71**. Connecting rods **69** and **71** in turn drive axles **59** and **61**, respectively, turning the eccentic weights in the direction shown by the arrows of FIG. 1. As seen in FIG. 1, the two weights are driven in opposite directions. This will accomplish a force in the transverse axis while cancelling out any force sin the vertical axis. Thus, there is no vertical vibratory action. Mounted on each shaft and in engagement therewith is a large housing **73** which can, for example, be a casting. The housing **73** has two receptacles **75** and **77** for containing large springs **81**. The springs **81** in turn are seated and fastened within recesses **83** provided in the low-frequency jaw **43**. Thus, there are four springs **81** held in four receptacles in relation to the axles **69** and **71**. The four springs **81** are shown in dotted outline particularly in FIG. 3. Additionally, as can be seen from

FIGS. 2 and 3 particularly, in the center of the four large springs there is disposed a smaller center spring **85**. One end of smaller spring **85** is seated in a block **87** supported from back plates **89**. The small spring **85** is split in two sections, **91** and **93**, with an I-beam support **95** therebetween. The I-beam support **95** in turn is affixed to the base portion **15** of the apparatus, as particularly seen in FIG. 3.

In order to obtain the desired low frequency in the jaws **43** without having a compliant system an extremely large mass would be necessitated. In order to overcome the necessity for such large mass, the four large springs have been incorporated to add lump compliance to the system whereby the springs can furnish enough compliance to effect the desired low-frequency resonant vibration in that portion of the system. In other words, the jaws **43** together with the eccentric weights and large springs and their support comprise a lumped mass system with the springs adding significant compliance so as to be able to generate a resonant low-frequency vibration in the system without necessitating an extremely large mass or a very long bar functioning in the manner of bar **23**. The small spring **85** is merely used to maintain a centering position for the four springs relative to the jaws **43**. Thus, in effect, it is a balancing spring serving to prevent uneven action on one side or one of the four centering springs; and serving to maintain the jaw spacing. The springs are affixed in a manner shown in FIG. 2*a*, wherein welded plates **99** intersect the coils of the spring locking them in the recesses within the receptacle members **75** and **77** and jaw **43**.

Thus, as can be appreciated, the jaw combination of this invention is experiencing two simultaneous vibrations, the low-frequency vibrations cause a low-frequency excursion of jaw **43** toward jaw **11**, and back again. The low frequency of the jaw **43** is of a sufficiently low speed so as to allow the crushing material **19** to fall a substantial distance during the time the jaw **43** is retracted relative to the opposite jaw **11** on each cycle. Then, as jaw **43** returns toward jaw **11**, the rock or other material **19** which is being treated falls far enough so that it is squeezed very vigorously by the low-frequency action.

Concurrently, while the above described squeezing action of the low-frequency jaw **43** is occurring, material **19** is simultaneously subjected to the high-frequency vibration of the main crushing jaw **11**. Thus, in effect, the high-frequency vibrations are applied in periodic pulses or duty intervals during which time the rock is held with substantial force by the low-frequency vibration. In other words, what is occurring is that when the low-frequency jaw is at the point of its excursion nearest the high-frequency jaw **11**, there is coupling of energy therebetween and through the material **19** so that the high-frequency energy from jaw **11** is transmitted through the material thereby crushing it during the impact of that jaw. When the low-frequency jaw alternatively is retracted during its excursion from the high-frequency jaw, the energy between the jaws is uncoupled allowing the material **19** to fall. During the uncoupled period, the load presented by the material **19** is decreased so significantly that the characteristic of the orbiting-mass oscillator system of the high-frequency jaw will build up and increase an amplitude. In other words, during the excursion of the low-frequency jaw **43** away from the high-frequency jaw, the system can store up resonant sonic energy in that the oscillator **31** is no longer presented with a high load. Now when the low-frequency jaw **43** moves toward jaw **11**, a high load is then presented. However, the amplitude of the oscillator has increased just prior to this point so that a larger force is effectively transmitted to the material **19**.

The vibrations in the high-frequency system are not ultrasonic as has been indicated in the previously filed application and issued patent, referred to above. The high-frequency vibrations contemplated can, for example, be on the order of 100 to 200 c.p.s. The low-frequency vibrations experienced in the jaw **43** can typically be on the order of 5 to 20 c.p.s., depending upon the design of the machine and the material being treated. In some instances, the low-frequency vibrations can be on the order of only two to three cycles per second. This is particularly so in larger machinery.

One further advantage of this invention is that the high-frequency vibrations experienced in jaw **11** greatly aid the falling of the material **19** through the jaws during the period when the low-frequency jaw **43** is retracted. Thus, the material is not only aided by the natural effect of gravity but also by the sustained high-frequnecy vibration which, though not acting to greatly crush the material during the retraction of the jaw **43** due to the uncoupling effect, however does transmit sufficient energy to overcome static friction between the material and the faces of the jaw thereby aiding the gravity effect.

It will be appreciated that, if desired, this invention can be practiced by having one jaw stationary, and the other jaw partaking of both the low-frequency, large-excursion, "admittance" vibration, in combination with the higher frequency crushing vibration.

While 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 it 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, each being vibratory toward and away from the other;
- means for resonantly vibrating one crusher member at a high sonic frequency; and
- means for simultaneously resonantly vibrating one crusher member at a significantly lower sonic frequency.

2. The crushing apparatus of claim **1** wherein said means for vibrating at a high frequency is connected to one crusher member while said means for vibrating at a low frequency is connected to the other crusher member.

3. The apparatus as recited in claim **1** wherein said means for resonantly vibrating one member at a high sonic frequency comprises:
- orbiting-mass oscillator means, and
- a member of solid elastic material connecting said oscillator means to said crusher member for transmitting elastic vibration thereto.

4. The apparatus as recited in claim **2** wherein said means for resonantly vibrating said second crusher member at a low frequency comprises:
- orbiting-mass oscillator means, and
- springs coupling said oscillator to said second crushing member.

5. The apparatus of claim **4** wherein said springs comprise:
- four equidistantly spaced heavy springs and a fifth lighter weight spring centrally disposed relative to the said four springs.

6. The apparatus of claim **4** wherein said oscillator means comprises:
- eccentric weights and
- means for rotating said weights.

7. The apparatus of claim **6** further comprising:
- a support structure;
- two vertically displaced axles mounted in said structure with an eccentric weight mounted on the ends of each axle; and
- means rotatably mounted on said axles engaging said springs on one end thereof.

8. A method of crushing material between opposed jaws, comprising:

directing material to be crushed between said jaws, and subjecting at least one of said jaws to resonant vibration of a predetermined frequency, while simultaneously applying a resonant vibration of a substantially lower frequency to at least one of said jaws.

9. The method of claim **8** wherein the higher and lower frequency vibrations are applied to the same jaw while maintaining the opposing jaw is a fixed position.

10. The method of claim **8** wherein the high and low frequency resonant vibrations are separately applied to opposite jaws.

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U.S. Cl. X.R.

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