Jan. 29, 1963 H. P. KAUTZ 3,075,713

CRUSHER JAWS

Filed May 15, 1961 2 Sheets-Sheet 1

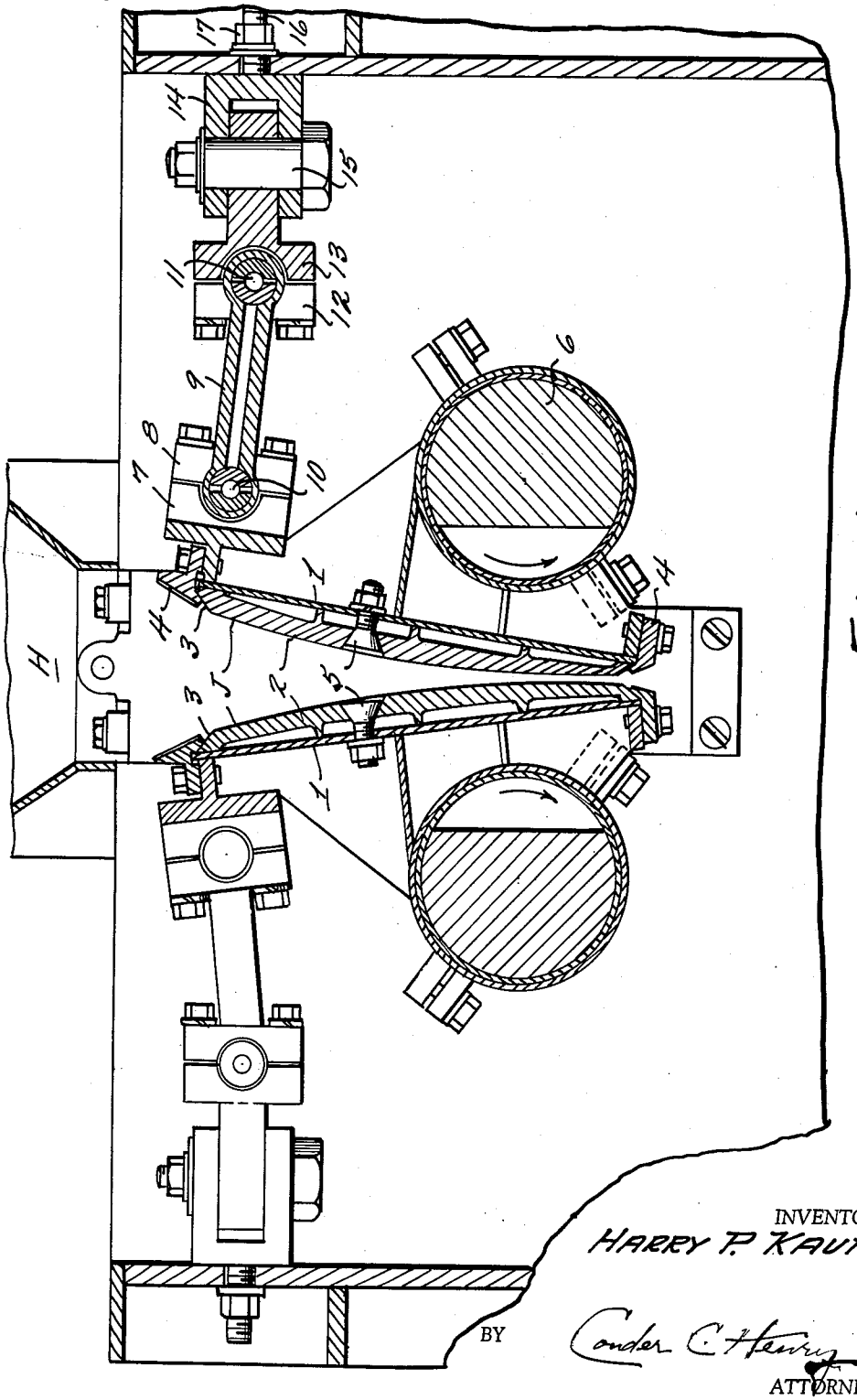


FIG. 1

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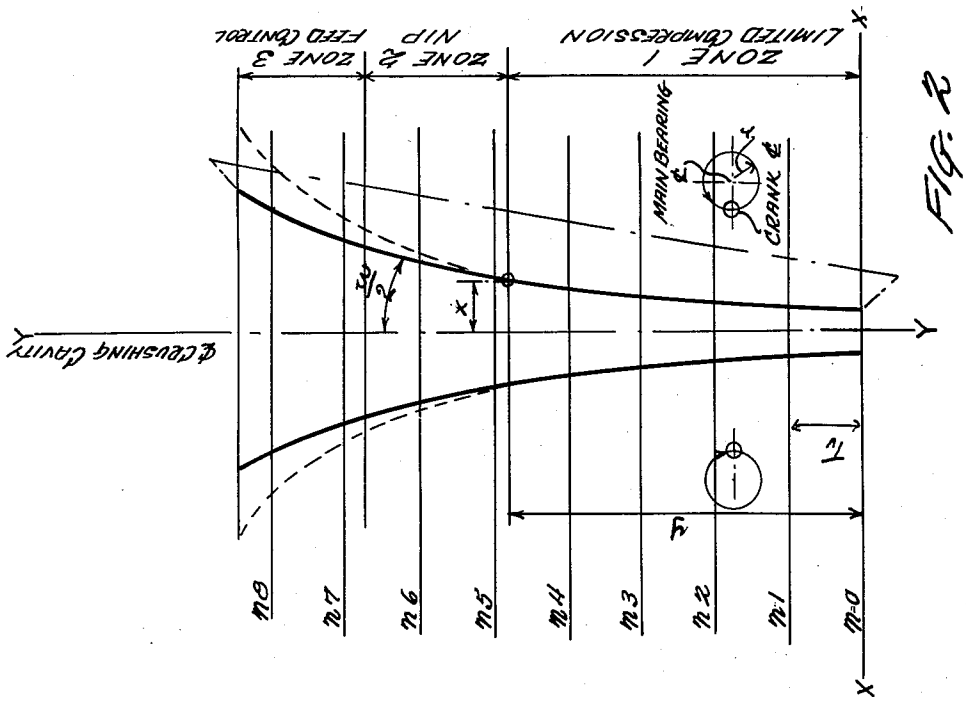


FIG. 2

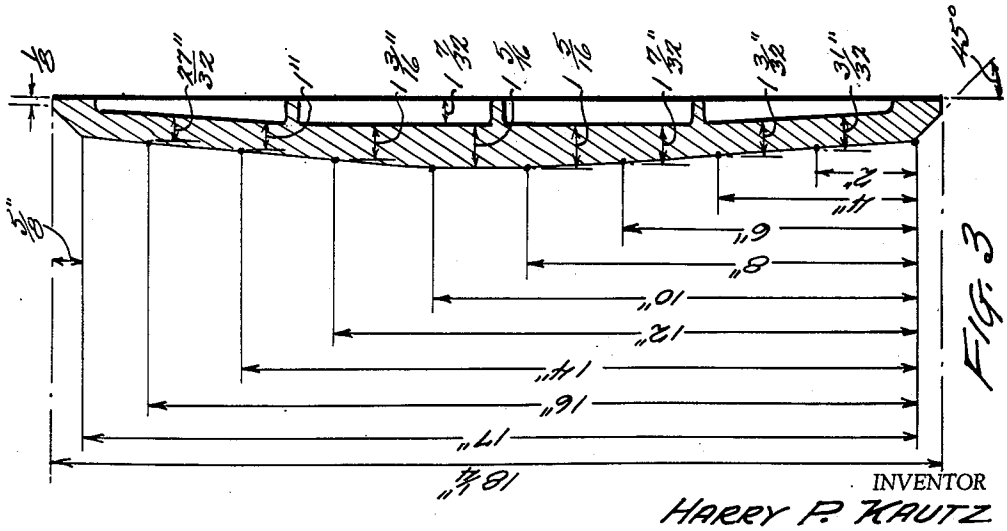


FIG. 3

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CRUSHER JAWS

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My invention relates to improvements in crusher jaws for use in ore crushers generally and particularly for use in connection with the type of crusher disclosed in the patent to Nutting, No. 2,865,570, granted December 23, 1958.

The Nutting patent reveals a method and means for cyclic propulsion and selective acceleration of crushable material through the eccentrically driven jaws of a jaw crusher independently of the forces of gravity and also reveals size-selectivity control of such material while passing through the jaws. After constructing a working model of the Nutting machine and testing the same, I have found it surprisingly efficient but capable of improvement in the important respect hereinafter discussed.

By examining the wear patterns on the test machine I have derived a practical and logical mathematical formulae to specify the optimum jaw plate surface profiles by providing a way to determine rectilinear coordinate values for points through which the profile lines of the jaw faces pass.

The principal object of my invention, therefore, is to provide jaw wear plates for crushers having faces shaped to secure maximum efficiency and to decrease plate wear.

Another object of my invention is to provide interchangeable wear plates shaped as indicated for the individual models and sizes of crushers as well as to meet the specific requirements for feed and product.

For the most efficient conveying, crushing, and sizing of rock-like material in any ore crusher on optimum shaped crushing zone is required. Such optimum shape depends, among other things, upon such factors as feed and product sizes, machine size, frequency of compression, radius of eccentricity, and crushing zone depth.

The efficiency of crushers depends on these many factors, including not only the considerations of the quantity and the size reduction of particles, but also including the considerations of the power required and the need for durable, serviceable, and inexpensive machine parts. In general, the most efficient crushers are those which accomplish the duty to which they are assigned, at the least total cost. The total cost being the sum of costs for materials and labor which are required to manufacture, install, operate, and maintain, and including the cost of power to drive the equipment. To provide the least total cost, it may be necessary to provide machines which are relatively expensive to manufacture, but which provide means for more than compensatory savings in power and in the materials required to operate and maintain them.

Crusher design should be directed from the manifold considerations to provide maximum efficiency, and, in general, the basic rule can be stated as that design which accomplishes the duty in the least space without overstressing any of the parts of the machine. The least space required depends fundamentally upon the maximum geometric rate at which the crushing surfaces may converse from inlet to discharge. To avoid the overstressing of machine parts, whether the machine is equipped with an overload relief system, or not, it is important to also design a crushing chamber which will not allow an effective compression stroke which at any level of reference will be greater than that stroke which accomplishes compression to the point if incorpressibility. In general, then, the geometry of a crushing cavity and of crushing surfaces

which converge too rapidly cannot nip, and can also cause compaction of the charge to the degree of incompressibility which overstresses the machine, even though it does afford the shallowest and the lightest jaw structure. Conversely, crushing surfaces which geometrically converge too slowly, are able to nip, and will also prevent overstressing, but will be relatively deep, heavy, and otherwise inefficient.

Also important to the design of jaw surfaces are the factors of speed or the frequency of the crushing cycle, as well as the length of the available compression stroke or the mechanical horizontal throw of the machine. The significance of these factors can be explained by considering the rates of vertical travel of particles through the crusher.

At relatively slow speeds the vertical rate of travel depends almost entirely on the force of gravity; and the distance which particles can fall each cycle depends almost entirely on the geometric rate of convergence of the crushing surfaces; and the effective compression stroke is nearly equal to the mechanically available horizontal throw which is usually provided at specific levels by the combined motions of eccentrics and of toggles. This nonpropulsive action requires that the rate of geometric convergence of the crushing cavity be as high as the nip angle will allow, in order to be able to limit the drop per cycle and thus permit repeated cycles of crushing before discharge. This condition also requires that the close set should not be less than about 70% of the open set, or in other terms, requires that the close set must be about 2½ times greater than the mechanically available horizontal throw at the discharge, if an incompressible charge at that level is to be avoided.

At relatively high speeds the vertical rate of travel of the particles near the level of discharge depends almost entirely on the propulsion action of the jaw plate surfaces; and the effective compression stroke may be only a fraction of the mechanically available horizontal throw. The propulsive action does not require that the rate of geometric convergence of the crushing surfaces be a maximum to be able to limit the vertical progression per cycle of particles, but rather, requires a diminishing rate of convergence to avoid over-compaction, especially at relatively fine close sets.

Because of these relations, high-speed propulsitive machines provided with carefully curved jaw profiles, propel particles through the crushing chamber at rates which are almost directly proportional to speed and to the mechanically available vertical throw without causing over-compaction or over-stress. This, then, has become the basis for efficiency, not only in crusher performance, but also in crusher design.

In other words, by the application of my formulae, I can produce as articles of manufacture improved crusher jaw wear plates having face profiles described in precise mathematical terms which are adapted for many different machines and duties

These and other objects of my invention will become apparent from the specification and drawings forming a part of this application in which:

FIG. 1 is a vertical sectional view through my invention as applied to a Nutting type machine, showing part of a toggle assembly in section;

FIG. 2 shows logarithmic curves depicting the geometry of the material contacting faces or opposed face profiles of the crushing jaw plates illustrated by FIG. 1, and

FIG. 3 shows a jaw plate having a face profile as depicted by FIG. 2.

Referring in detail to such drawings, the mechanism comprises a hopper H positioned above jaws J. Each

jaw comprises a backing plate **1** on which is removably mounted an upwardly diverging convexly curved wear plate **2** having beveled ends **3**. The wear plate, which will hereinafter be more particularly described, is removably secured to the backing plate by any suitable means. As shown, it is clamped at both ends to the backing plate by clamps **4**, and intermediate its ends by countersunk bolts **5**. Each jaw is removably mounted adjacent its lower end for oscillatory movement on the throw or eccentric portion of a crankshaft **6** which has a cut-out portion to provide a counterbalance. Each jaw is also guided and supported at its upper end by a linkage or toggle assembly which will now be described.

At the upper end and fixed to the back of each jaw and extending laterally therefrom its a plate carrying a two-part separable journal comprising half journals **7** and caps **8**, in which one end of a toggle link **9** is pivoted on hollow pin **10**. The other end of said link is likewise pivoted on hollow pin **11** in a two-part separable journal comprising half-journal **13** and cap **12**, which journal is provided with a lateral extension pivoted near the end thereof in a clevis **14** on pivot **15**. The pivot **15** is positioned at right angle to pivot **11**. Thereby both vertical and lateral movement of the link is permitted. The said clevis is provided with a screw threaded extension **16** which is passed through the support, and the stirrup or clevis is held snugly to said support by a nut **17** screwed on the end of extension **16**. Suitable means has been provided for lubricating the bearing surfaces of the toggle assembly.

In operation, the lowermost portions of the opposed jaws move in substantially oval paths because the jaws are mounted on the throw or eccentric portion of the propelling crankshafts while the uppermost portions of the jaws are supported and guided by the toggle assemblies at the center-line of pivot pin **10** in oscillatory-rotary (reciprocating) motion about the center-line of pivot pins **11** in substantially linear upward-and-downward paths approximately in the planes of the jaw faces at about inlet level. Self-alignment of the toggles in the jaws is provided for because they are connected through links **9** to pivots **11**, **15** and **16** disposed in right angular relationship.

Regarding the face profiles of the jaw plates and referring to FIG. 2, insofar as my invention is concerned, such profiles can be expressed best, or at least more accurately, in mathematical terms, i.e., by mathematical formulae. The derivation of such formulae involves consideration of several factors:

First, consideration must be given to the fact that ore crushers, in order to break a substance, do not and need not exert the average pressure required to burst or break a single smooth faced particle of the substance when the compressive forces are carefully and uniformly distributed over the opposing material contacting surfaces of the jaws. This fact, that the required average jaw surface pressures are only a small fraction, say $\frac{1}{100}$, of the pressures which measure the ultimate compressive stress of the substance, is due to the fact that the material contacting jaw surfaces actually apply the force to the irregular surfaces of many particles, which in turn applies the required higher pressure on relatively small areas of many particles.

Also, consideration must be given to bulk density. The density of a solid substance is greater than the density of broken particles of the same substance. For example, the density of silica is about 165 pounds per cubic foot, while the bulk density of broken dry particles of silica varies between 100 and 120 pounds per cubic foot. The voids between particles of dry silica are mainly or usually filled with air. These voids represent the maximum practical compression stroke of the crusher jaws contemplated by me, and constitute between 27% and 39% of the gross volume. It is assumed that these voids are a measure of

the maximum effective stroke (effective horizontal throw) of such machines, based on the knowledge that a greater stroke would impose infinitely greater forces on the machines to effect further size reduction per cycle.

In the Nutting crusher the available horizontal throw at the discharge is maximum and relatively large in comparison to the close set and to particle size. The proper shaping of the jaw plate profiles provides an effective horizontal throw (effective stroke) which is only a small fraction of the actual available horizontal throw which, near the level of the crankshafts, is equal to four times the radius of eccentricity, r.

Further, derivation of my formulae necessitates an examination of the vertical or throughput velocities of particles which must pass through the machine. The vertical throw per cycle, T_v, at the level of discharge is determinable from the equation:

$$T_v=\pi Nr+\frac{g}{2(f+k_1)^2}$$

where $\pi=3.1416$; $N=2$ (the number of propelling crankshafts per machine, in this case 2; r equals the radius of eccentricity; g equals the acceleration of gravity; f equals the frequency; k_1 equals an empirical constant. From existing test performance data, k_1 has been evaluated approximately equal to unity when T_v and r are expressed in feet, f is expressed in cycles per second, and g is expressed as feet per second per second (32.16).

The vertical or throughput rate of the machine may also be expressed as a velocity in units of feet per second, viz., $v=fT_v$.

The value of T_v is then an interval for plotting the progress of particles through the machine. Because the volume rate or throughput is restricted and is determined by the smallest aperture in the crushing cavity, which is the opening at the bottom or discharge level, and for the convenience of notation, the progressive cyclic levels are numbered in FIG. 2 in reverse order, i.e., the last cycle of crushing is the first cycle of notation.

Again referring to FIG. 2, it will be seen that beginning at the lowest level, with the throw or eccentric portions of the crankshafts phased in the position of their nearest approach to each other, the separation of the jaw plates is minimum, or close, and is termed the "close set," and is noted as S_c. This level of measurement is represented by a horizontal line or plane of reference which is the datum level, and which is identified in FIG. 2 as $n=0$, n being defined as the number of cycles prior to the discharge of the particles. Above this datum, consecutive cyclic levels are represented by horizontal lines separated by a distance T_v, which is the vertical throw per cycle. The vertical distance above the datum plane is then noted as nT_v, which is also expressed as y, which is a usual character for graphic representation, where the datum level ($n=0$) becomes the X-axis, and the centerline of crushing cavity becomes the Y-axis. Values of the effective horizontal throw, T_{he}, are then also equal to values of $2x$, or $x=\frac{1}{2}T_{he}$.

In order to limit the compression in progressive cycles of crushing to practical values, I have established a rate of divergence of the jaw profiles toward the feed opening. This rate, based on the ratio of air voids to solids, as well as on observations of wear patterns, is a geometric progression, where the reduction in volume per cycle is directly proportional to the reduction in T_{he}, and to the reduction in values of x per cycle. If the air voids are 29.3% of the gross volume at the level where $n=1$, and these voids are eliminated during the last cycle of crushing, or at the discharge level where $n=0$, we can write:

$$(V_1-V_0)/V_1=0.293$$

and $V_1=1.414V_0$. Because 1.414 is an empirical constant, we can write: $V_1=(k_2)V_0$. By examining the earlier consecutive cycle of crushing, we can write: $V_2=k_2V_1$, and $V_2=(k_2)^2V_0$, and $V_n=(k_2)^nV_0$. Because

volumes are directly proportional to corresponding values of T_{he} and x, we can write: $x_n=(k_2)^n x_o$. Substituting $\frac{1}{2}S_c$ for x_o, $x_n=\frac{1}{2}S_c(k_2)^n$. Because $y=nT_v$ and

$$n=\frac{y}{T_v}$$

we can write $y=T_v(\text{Log } 2x-\text{Log } S_c)/\text{Log } k_2$. Because T_v has already been derived:

$$T_v=\pi Nr+\frac{g}{2(f+k_1)^2}$$

we can by substitution determine coordinate values of x and y. The jaw plate profile defined by this equation diverges from a small to a large angle. The angle of divergence cannot exceed the angle of nip, w, which is the greatest angle which will still nip or grip particles. The tangent of this angle is the coefficient of friction between jaw plate and the particle. The design value of the tangent has been assigned: $\text{Tan}(w/2)=0.2071$, where $w=23°35'$. It may be necessary to provide this straight-line profile at the constant rate of divergence which is equal to the angle of nip to the feed zone of the jaw plate for some anticipated machine models and duties. Also, I propose to provide at the top of the feed zone, an angle of divergence which will exceed the angle of nip, in order to create a zone which repels feed and allows flood or choke feeding of the machine.

As shown on FIG. 2, the foregoing curves apply to the profiles for all values of y from zero to that value of y where the slopes of the curves are numerically equal to the tangent $(w/2)$ or 0.2071, see zone **2** of FIG. 2. At higher levels near the feed point (zone **3**) the angle of divergence is purposely increased to exceed the angle of nip in order to creat a profile which resists over-feeding and allows flood or choke feeding arrangements.

My invention is exemplified by FIG. 3, illustrating a crusher jaw wear plate designed for maximum efficiency at a 4.375 inch feed gap and a 0.273 inch close set and having a face profile line which will pass through points the rectilinear coordinate values of which are determined in accordance with the present disclosure.

While I have described and illustrated the jaws of my invention as including replaceable wear plates, it will be understood that the jaws could be formed as integral units. Therefore, whenever the terms "jaw" or "jaws" are used throughout this specification and claims, it is intended that those terms shall include both integral jaws and jaws provided with wear plates. It should also be understood that jaw crushers other than the Nutting type (which type has two eccentrically driven jaws) are included, recognizing that slower speed, single-acting jaw, and overhead eccentric crushers could have their jaw profiles determined by the formulae, requiring only that values of N would be either zero or one, and that the empirical constants, k_1 and k_2, might need to be re-evaluated.

Having thus described my invention, I claim:

1. As an article of manufacture, a wear plate for the jaw of a jaw crusher driven by propelling crankshafts, the profile of the material contacting face of said wear plate being such that a line depicting the same will pass through

all points determined by the equations: $x=\frac{1}{2}S_c(k_2)^{y/T_v}$; and $y=T_v(\text{Log } 2x-\text{Log } S_c)/\text{Log } k_2$, where

$$T_v=\pi Nr+\frac{g}{2(f+k_1)^2}$$

and S_c equals the close set of jaw faces at the discharge expressed in feet; k_2 is an empirical constant approximately equal to 1.414; π equals 3.1416; k_1 is an empirical constant approximately equal to 1.00 at crushing frequencies above 10 cycles per second; N equals the number of propelling crankshafts; r equals the radius of eccentricity expressed in feet; g equals the acceleration of gravity (32.16 ft./sec.^{2}); f equals the frequency of crushing cycles expressed in cycles per second; x equals values of the absicssa which are measured from the Y-axis and are expressed in feed; y equals values of the ordinate which are measured from the X-axis and are expressed in feet, the X-axis being a theoretical horizontal line at the lowermost ends of opposed jaws when they are in close set position and the Y-axis being a theoretical line extending upwardly at right angles to the X-axis intermediate said jaws when they are in close set position.

2. In an ore crusher comprising laterally spaced facing jaws driven by two propelling crankshafts near their lower ends and providing, when the crusher is in operation, propulsive forces including the force of gravity, the said jaws having in the crushing zone immediately above their discharge end curved material contacting profiles increasingly divergent upwardly from said end to a higher level where the angle included between lines tangent to said profiles at such level is less than or equal to the angle of nip, the curvature of the face of each of said jaws being such that line dipicting the same will pass through all points determined by the equations: $x=\frac{1}{2}S_c(k_2)^{y/T_v}$; and $y=T_v(\text{Log } 2x-\text{Log } S_c)/\text{Log } k_2$, where

$$T_v=\pi Nr+\frac{g}{2(f+k_1)^2}$$

and S_c equals the close set of jaw faces at the discharge expressed in feet; k_2 is an empirical constant approximately equal to 1.414; π equals 3.1416; k_1 is an empirical constant approximately equal to 1.00 at crushing frequencies above 10 cycles per second; N equals the number of propelling crankshafts, in this case 2; r equals the radius of eccentricity expressed in feet; g equals the acceleration of gravity (32.16 ft./sec.^{2}); f equals the frequency of crushing cycles expressed in cycles per second; x equals values of the abscissa which are measured from the Y-axis and are expressed in feet; y equals values of the ordinate which are measured from the X-axis and are expressed in feet, the X-axis being a theoretical horizontal line at the lowermost ends of opposed jaws when they are in close set position and the Y-axis being a theoretical line extending upwardly at right angles to the X-axis intermediate said jaws when they are in close set position.

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