Dec. 10, 1957 A. G. BODINE, JR 2,815,928

DEEP WELL DRILL WITH ELASTIC BIT COUPLER

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Fig. 1

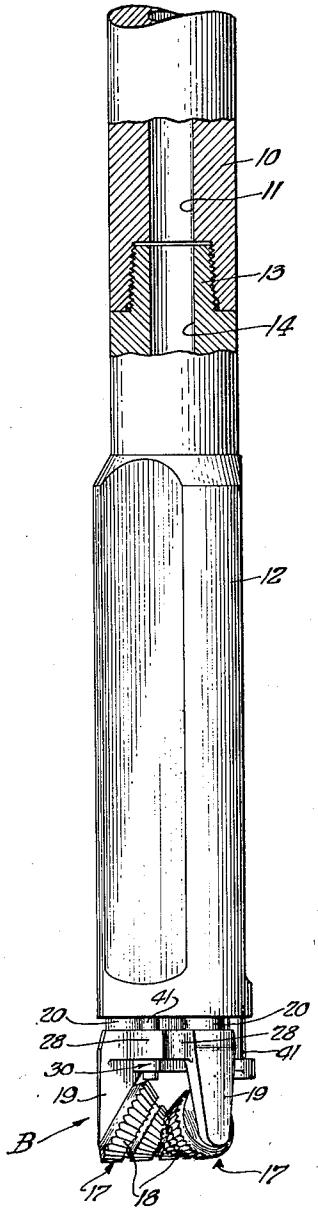


Fig. 5A

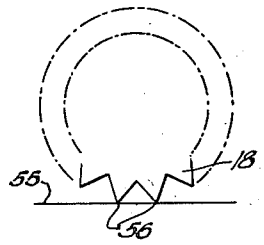


Fig. 5B

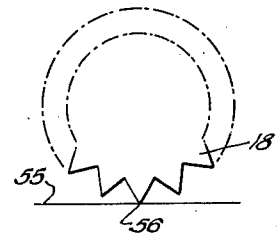
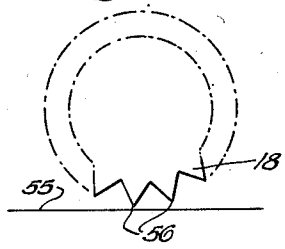


Fig. 5C



INVENTOR.
ALBERT G. BODINE, JR.
BY Forrest Filley
ATTORNEY.

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DEEP WELL DRILL WITH ELASTIC BIT COUPLER

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Fig. 2

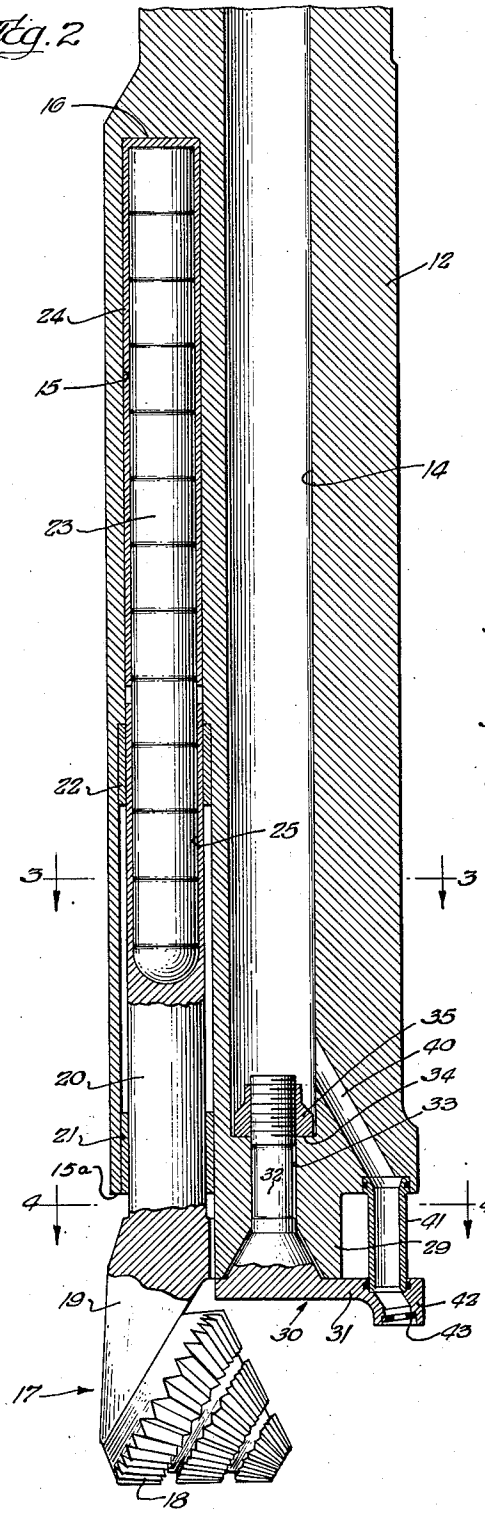


Fig. 6

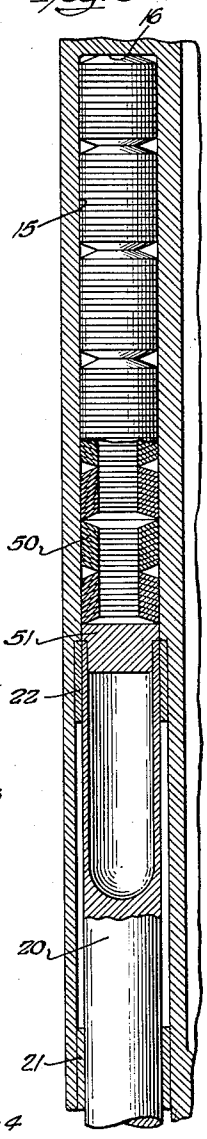
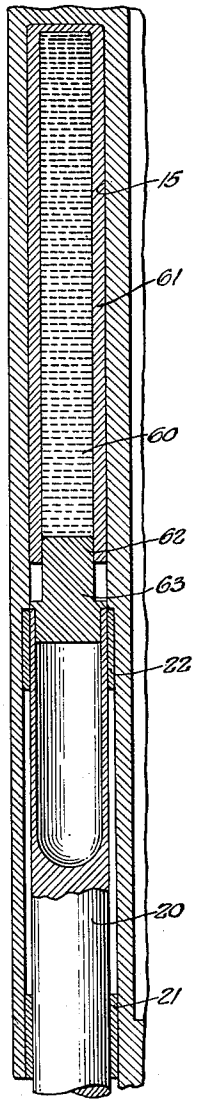


Fig. 7



INVENTOR.
ALBERT G. BODINE, JR.
BY Forrest J. Filley
ATTORNEY.

Dec. 10, 1957 A. G. BODINE, JR 2,815,928

DEEP WELL DRILL WITH ELASTIC BIT COUPLER

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Fig. 3

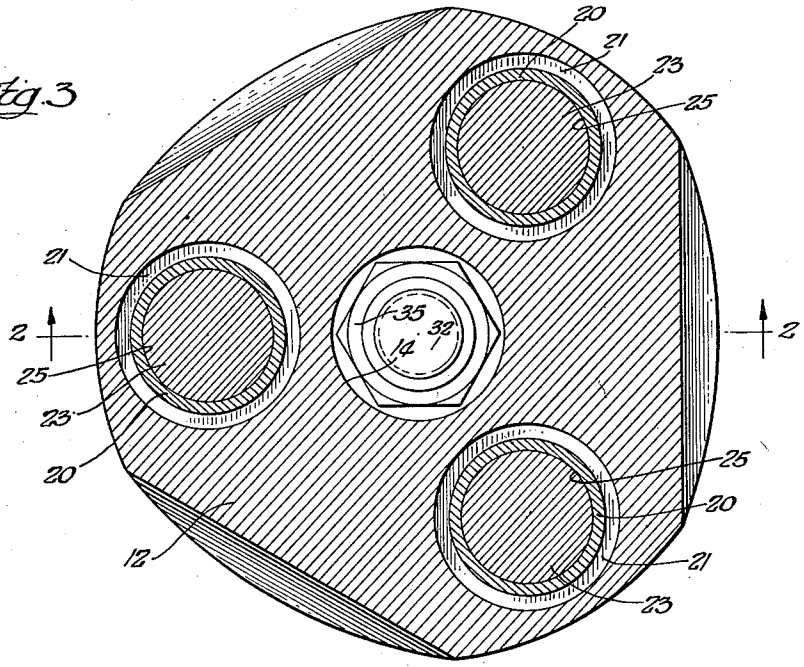
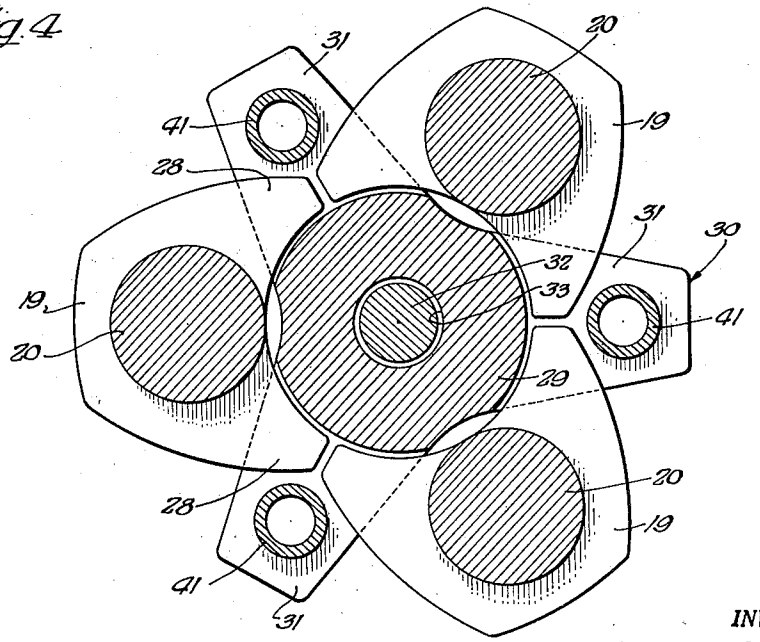


Fig. 4



INVENTOR.
ALBERT G. BODINE, JR.
BY
ATTORNEY.

Dec. 10, 1957 A. G. BODINE, JR 2,815,928

DEEP WELL DRILL WITH ELASTIC BIT COUPLER

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Fig. 9

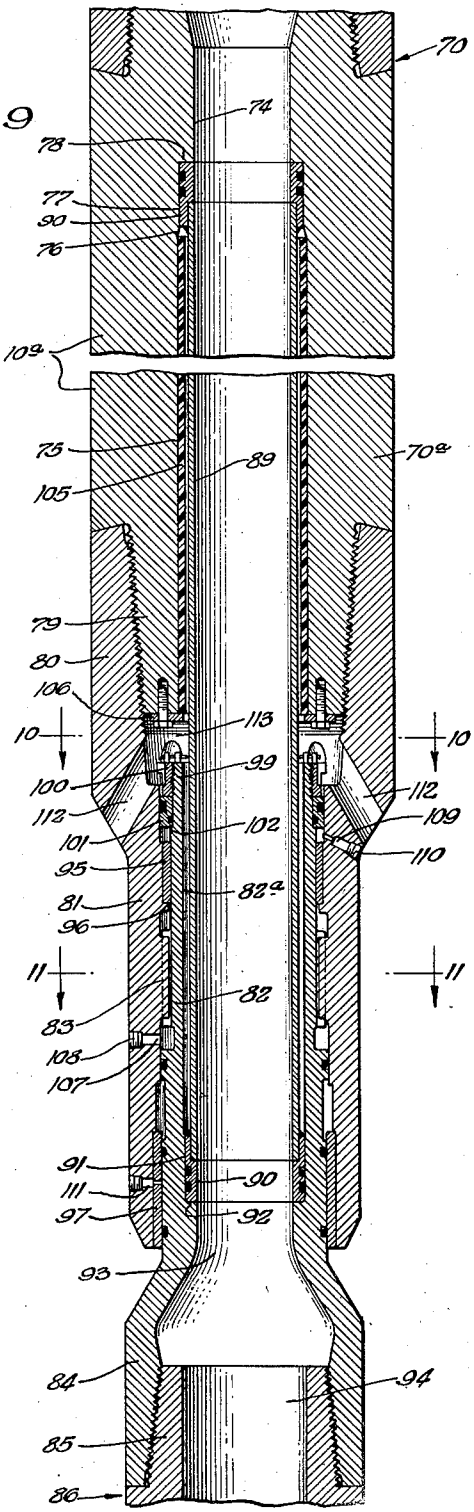
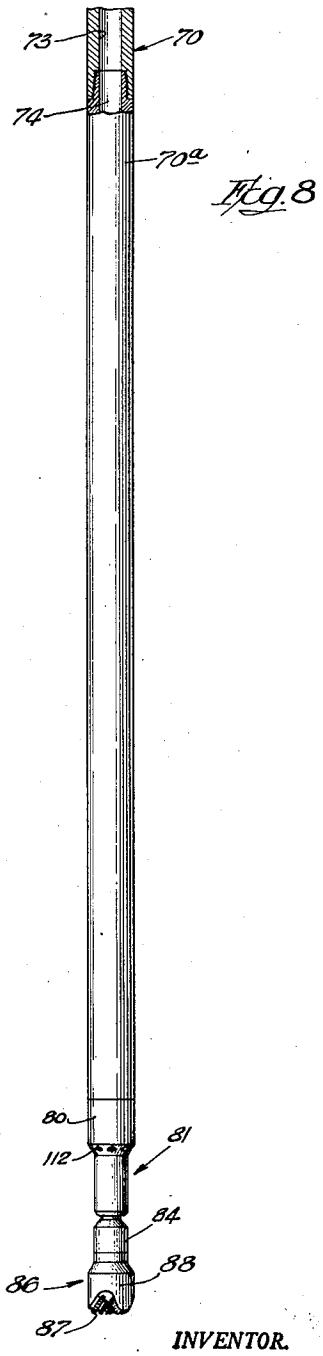


Fig. 8



INVENTOR.
ALBERT G. BODINE, JR.
BY
ATTORNEY.

Dec. 10, 1957 A. G. BODINE, JR 2,815,928

DEEP WELL DRILL WITH ELASTIC BIT COUPLER

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Fig. 11

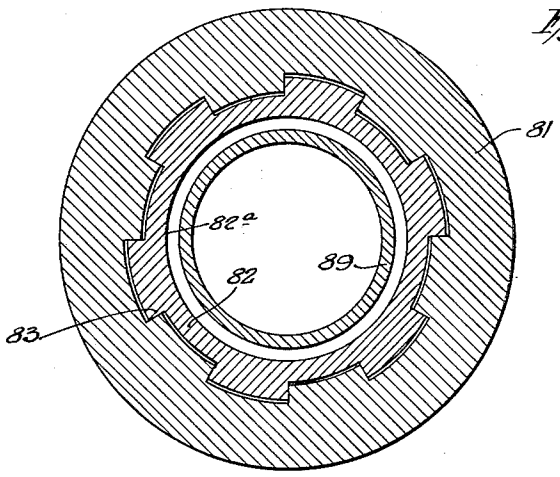
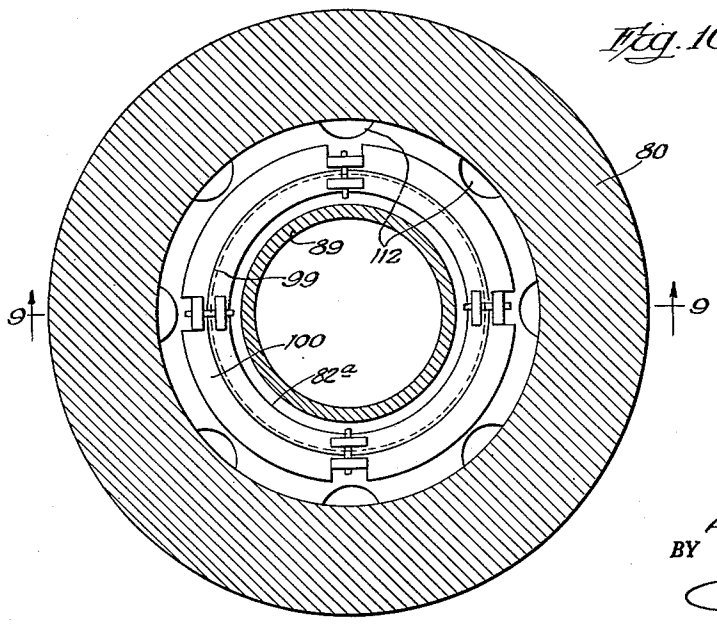


Fig. 10



INVENTOR.
ALBERT G. BODINE, JR.
BY
ATTORNEY

Dec. 10, 1957 A. G. BODINE, JR 2,815,928

DEEP WELL DRILL WITH ELASTIC BIT COUPLER

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Fig. 12

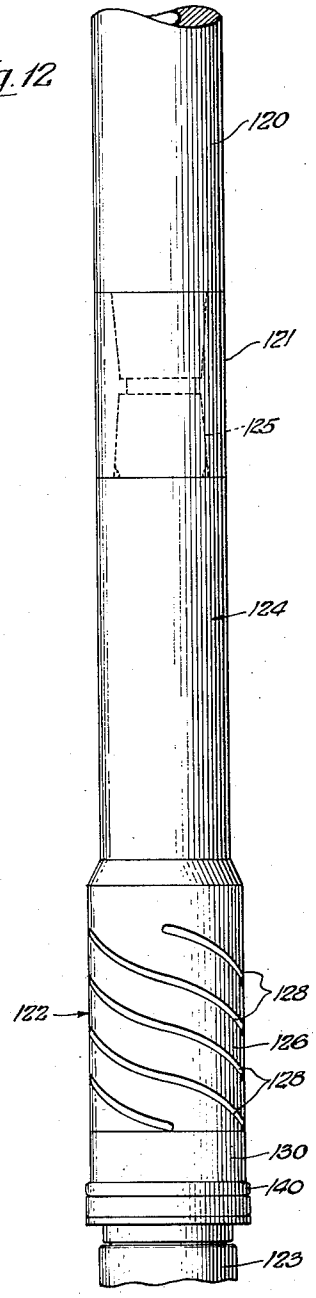
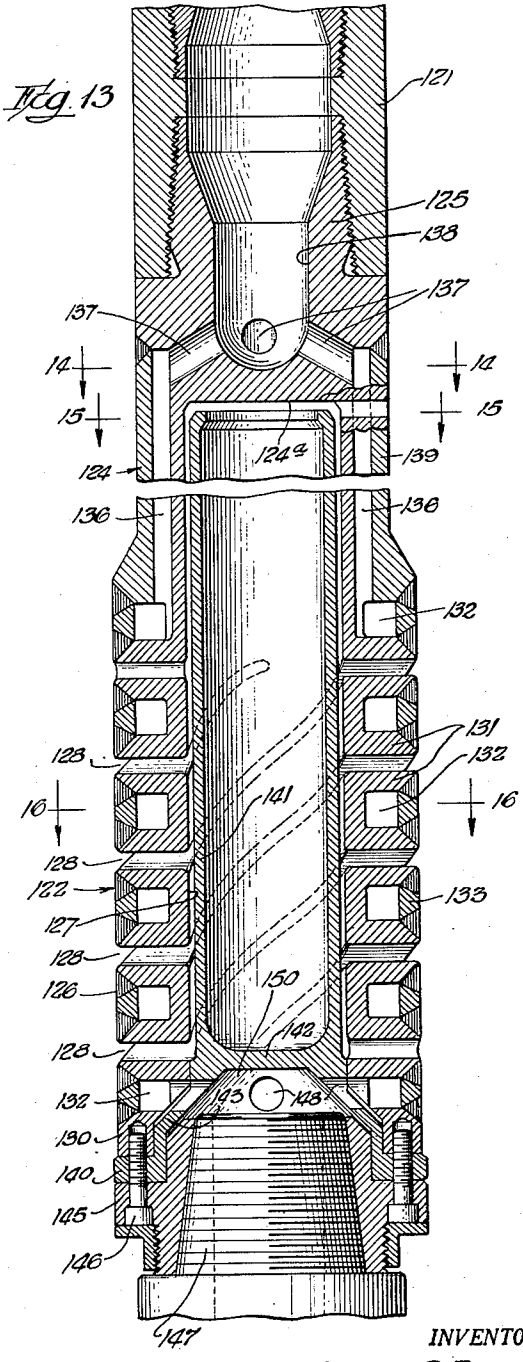


Fig. 13



INVENTOR.
ALBERT G. BODINE, JR.
BY Jarrod J. Filey
ATTORNEY.

Dec. 10, 1957 A. G. BODINE, JR 2,815,928

DEEP WELL DRILL WITH ELASTIC BIT COUPLER

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Fig. 14

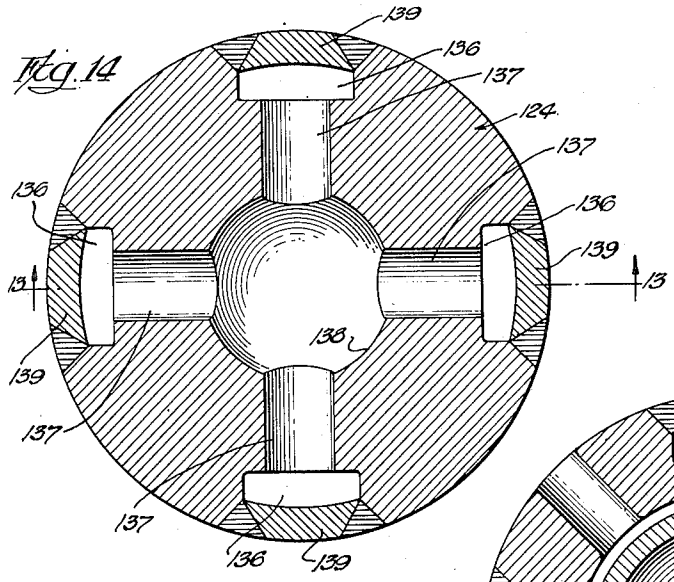


Fig. 15

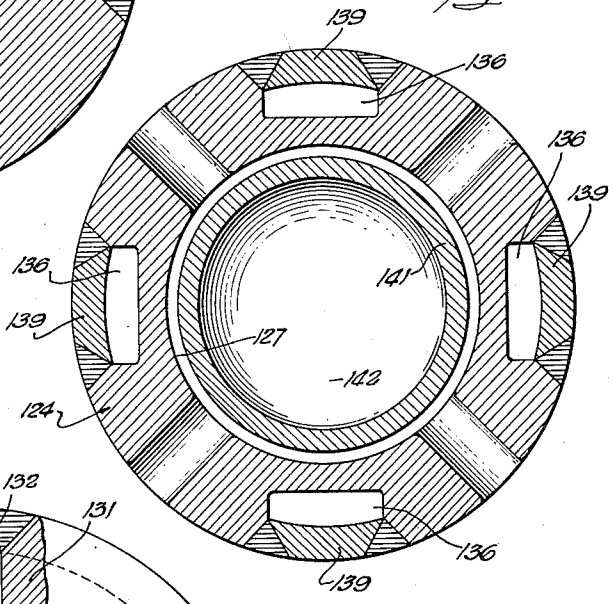
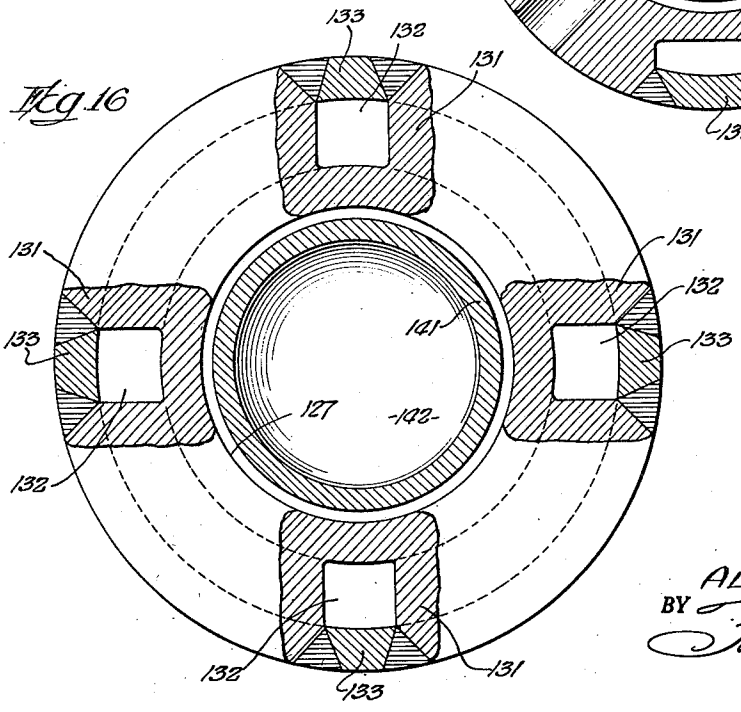


Fig. 16



INVENTOR.
ALBERT G. BODINE, JR.
BY Ernest J. Kelly
ATTORNEY.

Dec. 10, 1957 A. G. BODINE, JR 2,815,928

DEEP WELL DRILL WITH ELASTIC BIT COUPLER

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Fig. 17

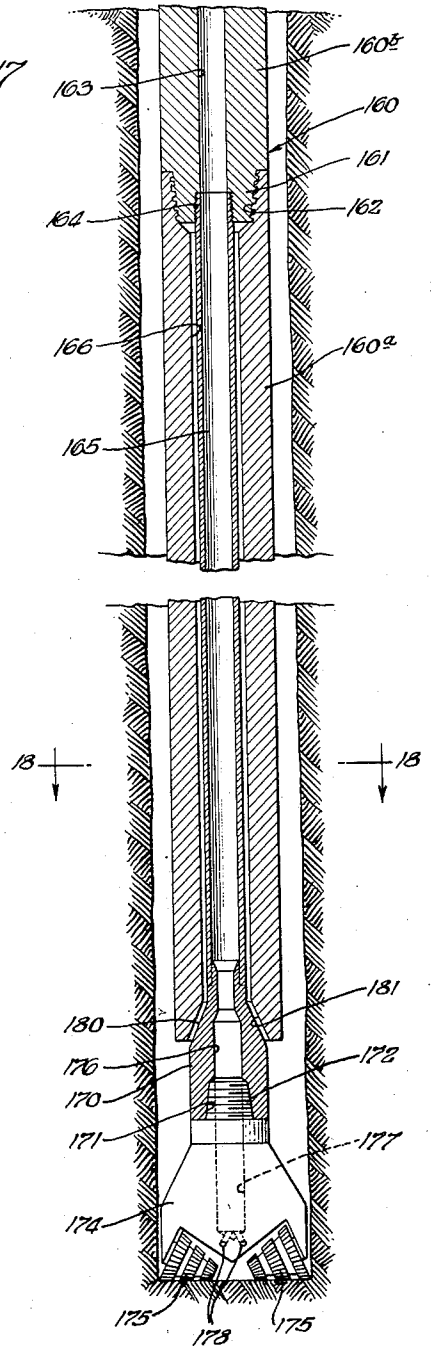
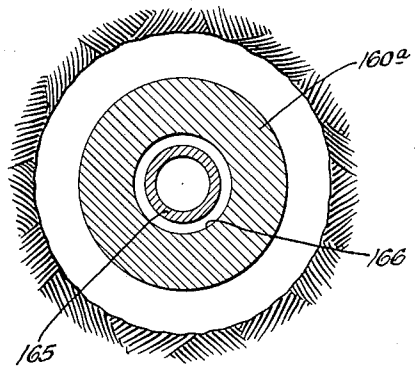


Fig. 18



INVENTOR.
ALBERT G. BODINE, JR.
BY Forrest Kelly
ATTORNEY.

United States Patent Office 2,815,928 Patented Dec. 10, 1957

2,815,928

DEEP WELL DRILL WITH ELASTIC BIT COUPLER

Albert G. Bodine, Jr., Van Nuys, Calif.

Application April 23, 1956, Serial No. 579,868

4 Claims. (Cl. 255—4.4)

This invention relates generally to deep well rock drilling, and more particularly to means for improving the drilling action of conventional rotary rock bits, especially on hard rock formation.

In the operation of the conventional rotary bit of the toothed roller type in deep well hard rock drilling, the toothed bit roller cutters frequently lose proper cutting engagement with the rock at the hole bottom. This condition generally occurs or increases as the hole is drilled to considerable depths, and may not manifest itself for the first few hundred feet. My investigations have shown that this condition can occur from any or a combination of certain causes. First of all, as can readily be seen, the bit and drill collar are elevated a short distance as each roller tooth rolls over the hole bottom, and each such elevation must obviously be followed by a corresponding lowering if the next tooth is to obtain a good "bite" under full loading by the drill collar. However, in practice, each such elevation is not naturally followed instantly by a corresponding drop; and it will be clear that the cutting action ceases or is impaired until effective drop does occur. The bit and collars may also be thrown somewhat sharply upward for a substantial time interval, entirely out of engagement with the rock, owing to and following a particularly pronounced engagement of a tooth with the rock, or with a high spot thereon. A number of cutter tooth engagements with the rock may be missed in either case before the bit is again down in firm cutting engagement with the rock, with the loading of the drill collar fully imposed on the bit and in turn on the rock. Various influences may enter to retard the fall of the system following each such elevation as described. The bit and collars are massive and have high inertia, and are not easily set into vibration at the frequency of cutter tooth engagement with the hole bottom. They cannot drop with greater than one "G", and do not even approach one "G" because of the dampening effect of drilling mud fluid around the collar, and particularly between the bit and the bottom of the hole. At shallow depths, each upward movement of the bit and collars, as a cutter tooth rolls over the hole bottom, results in an elastic wave of compression launched up the drill pipe, and this wave is reflected back down the pipe, and may assist in moving the collar and bit back down. However, at substantial depths, the drill pipe acts (by analogy with electric transmission lines) as an "infinite line," the upwardly traveling wave of compression being attenuated and/or radiated from the top, and failing to return to assist in lowering the collar and bit. They then have no return force other than gravity, and, as pointed out below, drop very sluggishly.

Another influence, somewhat obscure and difficult to understand, may also enter to influence the bit and collars to rise sufficiently to take the normal drill collar loading wholly or partially off the formation. Assume that the bit is so designed, and rotated at such speed, as to result in a fairly high theoretical frequency of cutter

tooth engagement with the hole bottom, say 2000 per minute. This means that the bit and attached drill collars should tend to vibrate longitudinally through a small amplitude at a frequency of 2000 cycles per minute. Under such circumstances, the downwardly facing area of the bit may "couple" acoustically to the surrounding mud fluid, so as to generate therein sound waves of short wavelength traveling up the mud fluid surounding the drill string to the ground surface. The source of this sound wave generation is the vertically projected net downwardly facing area of the bit, which is quite ample. If the bit becomes thus acoustically coupled to the mud, and is driven at a frequency at which sound waves may be generated in the mud, then the bit must generate such sound waves if it is to vibrate in contact with the mud. The cyclic mud pressure against the bit as a result of such tendency to generate sound waves is therefore an influence which limits bit vibration amplitude, and, in practice, there is generally insufficient energy available to overcome this influence, even if there were otherwise enough available for forcing the collars and bit back down in step with cutter tooth engagements with the rock, so that very little high frequency vibration of the bit can occur. The cyclic mud pressure so exerted in an upward direction on the bit may be sufficient to partially or wholly support the drilling assembly, so that the drilling assembly simply "rides" on the mud, with the weight of the bit and collars largely off the hole bottom.

The drilling assembly, i. e., bit and collars, can thus be elevated by engagement of a bit roller tooth with the hole bottom, and some time can elapse before the bit and collars are again down in cutting position; or the bit and collars can be lifted or partially supported by cyclic pressure of the mud fluid owing to acoustic coupling and consequently attempted sound wave generation. The several described effects can occur together. The result is that the bit and collars partake of a long slow, vibratory stroke, with relatively infrequent blows of the cutters against the formation. These blows, while not occurring at a frequency to give the drilling action that should be expected, are nevertheless powerful enough, because of their long vertical drop, to overstress cutters and bearings.

The general object of the invention is accordingly the provision of means for assuring maintenance of engagement of the toothed bit roller cutters with the rock at the hole bottom and maintenance of continuously sustained average loading by the drill collars.

A further object is to provide means for permitting and causing the body of a toothed roller cutter type of bit to rise and fall, without markedly elevating the collars in step with engagements of successive bit cutter teeth with the hole bottom.

A still further object is the provision of a bit as stated in the preceding object, where the average loading of the bit and formation by the drill collars is continuously maintained.

A still further object of the invention is the provision of an improved deep well rotary rock drill of the toothed roller cutter bit type wherein the bit is provided with means giving it an inherent proneness to vibration at a frequency of the order of, or higher than, the frequency of bit cutter tooth engagement with the rock at the normal rate of drill stem rotation, whereby, following each elevation of the bit owing to a cutter tooth engagement with the rock, the bit will be instantly forced back down, so that the next tooth will make cutting engagement with the rock.

According to the present invention, there is interposed between the usual drill collars and the bit tooth element a vibratory frequency responsive elastic coupling mem-

ber, which, in a simple form, can be an elastic column or some other spring member of proper stiffness value, sufficiently compliant to have a range of elastic deformation and restoration at least as great as the amplitude of bit vibration when loaded with the drill collars. Stated differently, this elastic coupling member is of sufficient load carrying capacity for the modulus of the material of which it is composed to carry the impressed compressive load of the collars without undue elastic deformation, and, assuming a columnar type of coupling (such as is used in some species), must be of sufficient length for its modulus and cross sectional area that, under the compressive loading of the collars, it has a range of elastic contraction and restoration at least as great as bit vibration amplitude owing to cutter tooth engagement with the formation. Finally, and most important, the elastic coupling member and the bit, as an elastically vibratory assembly, must have a natural resonant frequency of vibration at least as high as the frequency of roller tooth engagement with the hole bottom at normal speeds of drill collar rotation.

The statement in the paragraph immediately preceding as to the necessary natural resonant frequency of the elastic coupling member and bit is of first importance. If the coupling member and bit assembly is designed with such combination of mass and elasticity that the natural resonant frequency thereof is lower than roller tooth contact frequency, it is then "mass-dominated," and elastic restoration following deformation will not occur in time for vibration at roller tooth contact frequency with the hole bottom to be attained. Instead, a sluggish performance obtains: the bit is elevated by a bit tooth, but then falls slowly, over a protracted time period during which many fully loaded cutter tooth engagements with the rock may be skipped. A mass-dominated coupling acts essentially to impede vibrations at higher frequency than its own resonant frequency. By designing the elastic coupling member and bit for a natural resonant frequency as high or higher than roller tooth contact frequency, however, it is then elasticity-dominated, or "stiffness"-dominated, at elastic deformations and restorations of the coupling occur in step with normal cutter roller tooth engagements with the hole bottom, as is desired. In authoritative acoustic terminology, it has a capacitative response for the operating frequency, because it is "stiffness controlled" which keeps its response in phase, as desired. Obviously, the elastic member, or portion of the apparatus, can be of many different geometric shapes, so long as it provides this all-important frequency response.

The foregoing can be formalized in a simple equation. In terms of frequency,

$$f=\frac{1}{2\pi}\sqrt{\frac{k}{m}} \qquad (1)$$

where m is the equivalent mass of the elastically vibratory system comprising the bit and elastic coupling, k is the equivalent elastic stiffness constant of the vibratory system, being the elastic restoring force when the system has been displaced from its neutral position, and f is the natural resonant frequency of vibration of the elastic coupling and bit. Performance in accordance with the invention is attained when the structure of this elastically vibratory system is designed with such mass m and stiffness k as to have a natural resonant frequency f which is as high as cutter tooth contact frequency with the hole bottom at normal speed of drill collar rotation. In terms of period t, which is the time duration of a single cycle of vibration of the elastically vibratory system (elastic coupling and bit) at its resonant frequency f,

$$t=2\pi\sqrt{\frac{m}{k}} \qquad (2)$$

from which it can be seen that the greater the elastic constant k, and the less the mass m, the shorter will be the "period" of vibration of the system. By designing the structure of the bit and elastic coupling making up this vibratory system for such mass m and elastic stiffness k that its period t is no greater than the time interval required for a bit tooth to roll over the rock at the hole bottom, the elastic coupling will act to force the elevated bit roller back down as fast as permitted by the tooth, so that at the end of the time interval for travel of a given tooth over the rock, the bit is fully down, and the weight of the collars fully imposed on bottom.

The invention will be best understood by now referring to the detailed description of several illustrative embodiments thereof, reference being had to the accompanying drawings, in which:

Fig. 1 is an elevational view, partly in section, showing one embodiment of the invention;

Fig. 2 is a longitudinal sectional view of the embodiment of Fig. 1, taken in accordance with section line **2—2** of Fig. 3;

Fig. 3 is a transverse section taken on line **3—3** of Fig. 2;

Fig. 4 is a section taken on line **4—4** of Fig. 2;

Figs. 5A, 5B, and 5C are diagrams of a toothed roller cutter illustrating several conditions which may be encountered in drilling with toothed roller cutter bits;

Fig. 6 shows a modification of a portion of Fig. 2;

Fig. 7 shows another modification of a portion of Fig. 2;

Fig. 8 is an elevational view, partly sectioned, showing another embodiment of the invention;

Fig. 9 is a longitudinal sectional view of the embodiment of Fig. 8, taken as indicated by section line **9—9** of Fig. 10;

Figs. 10 and 11 are enlarged transverse sections taken on lines **10—10** and **11—11**, respectively, of Fig. 9;

Fig. 12 is a side elevational view of still another embodiment of the invention;

Fig. 13 is a longitudinal sectional view of the embodiment of Fig. 12, being taken in accordance with section line **13—13** of Fig. 14;

Figs. 14, 15 and 16 are transverse sections taken on lines **14—14**, **15—15**, and **16—16**, respectively, of Fig. 13;

Fig. 17 is a longitudinal section of another embodiment; and

Fig. 18 is a section on line **18—18** of Fig. 17.

Referring first to the illustrative embodiment shown in Figs. 1–4, inclusive, numeral **10** designates generally the lower end portion of a usual string of conventional drill collars such as are used in rotary drilling. It is to be understood that these collars are employed in usual numbers and in usual outside and inside diameter and length. Thus they may be of 40-foot length, of a typical diameter of seven inches, and there may be three or four of such collars, coupled together by usual drill collar couplings, the uppermost collar being understood as connected to the usual drill pipe. The several collars are formed with usual bores forming circulation passage **11**.

A bit mounting sub **12** has at its upper end a threaded pin **13** screwed into the box on the lower end of the lowermost collar **10**. This sub **12** has circulation passage **14**, and contains, at 120° spacing around passage **14**, three bores **15** extending upwardly from a downwardly facing surface **15***a* adjacent its lower end, to blind upper ends **16** near the top of the sub. The bit, generally designated at B, comprises three separate and complete independently vibratory bit components **17**, spaced 120° apart, each embodying a toothed roller cone cutter **18** rotatably mounted on an individual carrier shank **19** formed on the lower end of a vertical shaft **20** which is received in one of the three bores **15**, being supported for vertical oscillation in the latter as by means of bushings **21** and **22**. An individual elastic coupling means acts between each bit shaft **20** and the sub **12**, and in the illustrative embodiment shown in Fig. 2, this elastic means comprises a columnar series of low modulus elastic cylinders **23**, composed typically of a plastic such as fabric-phenolic. The cylinders

are thus stacked into a column and this column is confined partially within a guide sleeve 24 pressed into the upper portion of bore 15, and partially within a bore 25 extending downwardly into shaft 20.

Each bit shank 19 as a pair of arcuate wings 28 which partially surround a reduced, cylindric, downwardly extending lower end portion 29 of sub 12, and which are engageable with the latter to prevent undue swinging or rotation of the bit members on the axes of shafts 20 during operation. Some limited rotational movement is, however, sometimes preferred, and the wings 28 may accordingly be designed with limited clearance with respect to member 29 when centered as in Fig. 4.

A spider 30 has three arms 31 reaching radially outward between the bit shanks 19 from a mounting shaft 32 fitted into a bore 33 extending through the lower end portion of sub 12, the bore 33 being of smaller diameter than circulation passage 14, and the juncture of passage 14 with bore 33 providing an upwardly facing annular shoulder 34 which serves as an abutment for a securing nut 35 threaded onto the upper end of shaft 32, all as clearly appears in Fig. 2. The arms 31 of spider 30 engage under the wings 28 of bit shanks 19 to support the bit members and shafts 20 from dropping out of sub 12 while being lowered into or elevated from the drill hole. When the bit members are on bottom, the column of cylinders 23 compresses, so as to develop a clearance space between wings 28 and the spider 30, whereby to accommodate vertical reciprocation of the bit members.

The lower end portion of circulation passage 14 is connected by means of three passageways 40 and sleeves 41, mounted between the lower end of sub 12 and the outer ends of spider arms 31, with circulation discharge nozzles 42 provided at the tips of the arms 31, the nozzles being shown as equipped with properly sized orifice plates 43 to control proper jetting of the ejected fluid.

In the operation of the drill of Figs. 1–4, the drill stem is lowered until the three independently sprung bits are in engagement with the rock at the bottom of the hole, and until a proper loading of the bits by the drill collars is obtained. At proper loading, the three elastic columns of cylindrical blocks will be elastically compressed, and the bit wings 28 accordingly are elevated above the arms of spider 30, the clearance so provided being adequate to accommodate such bit vibration amplitude as is caused by rolling of the toothed cutter cones 18 over the hole bottom. The drill stem is then rotated in the usual way, causing the three cutters to roll over the hole bottom, and it will be seen that the three bits are enabled to vibrate independently of one another as their teeth engage the rock. The full action involved for each bit component and associated elastic coupling component will be explained in the ensuing analysis.

In Figs. 5*a*, 5*b*, and 5*c*, a single toothed bit roller cutter 18 is diagrammatically represented in three different positions which it may assume during operation, the line 55 in each case representing the bottom of the drill hole. In the first position (Fig. 5*a*), two of the cutter teeth 56 are in engagement with the hole bottom. In the second position, the cutter has turned a short distance on its axis, bringing a single tooth 56 into engagement with the hole bottom, and it should be evident that in this position the cutter, together with the bit body or shank and all parts of the drill assembly rigidly connected thereto, are elevated by a short distance as compared with the first position, this following from the fact that the distance from the crest of a cutter tooth to the axis of the cutter is greater than the distance from a line drawn between the crests of two cutter teeth, to the axis of the cutter. The position (Fig. 5*c*) represents a condition often encountered with prior art toothed roller cutter bits, the cutter having become actually separated from the hole bottom, and in such condition, of course, the drill is for the time being inoperative. Several ways have been explained hereinabove in which the condition illustrated in Fig. 5*c* may occur in prior art practices, and it has also been explained how and why the drilling apparatus fails to instantly return downward after reaching the second or third positions, but instead remains elevated for a time, during which many potential tooth engagements with rock may be missed. It will presently be described how the drill of the present invention forces the toothed cutters to alternate properly between the first and second positions shown in Figs. 5*a* and 5*b*, the roller teeth remaining always in pressural engagement with the bottom of the hole.

As explained, a toothed bit roller cutter, in rolling over the hole bottom when the drill collars are rotated, causes the bit and rigidly connected parts above to vibrate in a vertical direction as a result of successive bit tooth engagements with the hole bottom. The elastic coupling means between each bit and the drill collars above is made to be sufficiently compliant and have a range of longitudinally elastic deformation sufficient that it will elastically shorten and elongate during this vibratory action of the bit, permitting the high inertia drill collars to remain substantially stationary as regards longitudinal vibratory motion. The weight of the collars is imposed on and divided between the bits 17 through the several individual elastic coupling means, and the latter must have sufficient load bearing capacity for the modulus of the material of which it is composed to withstand this compressive loading without undue elastic deformation. At the same time, the elastic coupling means is designed to be sufficiently compliant or "soft" to permit elastic deformation in a longitudinal direction, over and above that caused by the compressive loading of the collars, by an amount no less than the bit vibration amplitude. Thus, for the case of Figs. 1–4, each column of elastic cylinders 23 must be of sufficient length, for its value of Young's modulus, and its cross sectional area, to give the required working range of elastic deformation beyond that due to its share of the weight of the drill collars without exceeding the elastic limit of the elastic couplings. This being met, the couplings of course perform without collapse. The phenolic blocks have the advantage of a relatively low modulus, which decreases the necessary length of the column.

Still further, and of primary importance, each elastic coupling means and corresponding bit connected thereto must be designed with such mass and elasticity constants that the coupling means and bit will have a natural resonant frequency of vibration which is at least as high as the frequency of vibration of the corresponding bit component 17 as caused by cutter tooth engagement with the hole bottom at normal speed of drill collar rotation, this being necessary to give the elastically contracted coupling means the ability to expand and force the bits 17 down following elevations thereof by tooth engagements with the rock in a sufficiently short time interval that the bits are enabled to vibrate at tooth contact frequency notwithstanding the existence of the above explained influences tending to retard downward movements of the bits. As an example, for normal drill collar rotation speed in normal rotary drilling, a bit roller cutter would, under ideal conditions, turn on its axis at

$$\frac{ps}{2\pi r}$$

revolutions per second, where s is the R. P. S. of drill collar rotation, p is the perimeter of the bit circle (or, what amounts to the same thing, the perimeter of the hole bored by the bit), and r is the outside radius of a toothed bit roller at its "outside" end, i. e., the portion thereof which travels around the perimeter of the bore hole. The theoretical or ideal bit vibration frequency, i. e., cutter tooth engagement frequency with the hole bottom, f', is then represented by the equation

$$f' = \frac{nps}{2\pi r} \qquad (3)$$

where n is the number of teeth around the cutter at the radius r.

It may also be seen that the theoretical vibration frequency of a toothed roller cutter is a number found by dividing the tooth spacing at the outer, or large diameter end, of the cutter, into the circumference of the hole, and multiplying by the revolutions per second at which the rotary table is driven.

The period t' of bit vibration frequency (time duration of a single idealized cycle of up and down bit travel) is the reciprocal of equation (3):

$$t'=\frac{2\pi r}{nps} \tag{4}$$

Now if each tooth engagement with the hole bottom elevates the bit in a half-period time interval

$$\frac{\pi r}{nps}$$

then the elastic coupling means must be designed for elastic expansion to force the bit back down in the remaining half-period time interval

$$\frac{\pi r}{nps}$$

This it can do only if the natural resonant frequency f of the elastic coupling means and bit is as high as bit vibration frequency f' (bit tooth contact frequency); or, in terms of period, if the natural period t is as small as the period t' for bit tooth contact frequency with the hole bottom. The importance of this consideration can be seen when it is realized that if the natural period of each elastic coupling means and bit **17** is greater than the period t' (for bit tooth contact frequency at normal drill collar rotation), the elastic coupling means then would not be capable of forcing the bit back down in sufficiently short time intervals to bring about bit vibration at the bit tooth contact frequency f', and potential bit tooth engagements with the rock would then be skipped.

In order to design the elastic coupling means and bit for a natural resonant frequency f as high as bit tooth contact frequency f', each coupling means and bit **17** must be physically designed with such lumped mass and mass distribution and the coupling means with such elastic constant k as will satisfy the hereinabove stated Equation 1:

$$f=\frac{1}{2\pi}\sqrt{\frac{k}{m}}$$

with the further requirement that f be at least as great as bit tooth contact frequency with the hole bottom.

Or, making use of both Equations 1 and 3, it is theoretically necessary to satisfy the following relationship:

$$\frac{1}{2\pi}\sqrt{\frac{k}{m}}\geq\frac{nps}{2\pi r}$$

or

$$\sqrt{\frac{k}{m}}\geq\frac{nps}{r} \tag{5}$$

The foregoing discussion is theoretical, and necessarily somewhat idealized, in that the actual bit tooth contact frequency for a given cutter will inevitably depart somewhat from the assumed theoretical value owing to the effect of irregularities of the rock at the hole bottom on the rate of rotation of the cutter. That is to say, certain lags or slippages may occur, as well as certain accelerations, as soft or hard spots, or depressions or high spots, are encountered by the cutters. A practical design must take such deviations into account, and it will therefore be understood that in practice, the value for f as determined by Equation 1 must be made at least equal to actual bit tooth contact frequency, which may range a little above or below the theoretical value as predicted by Equation 3. For safety, a response frequency on the high side is preferred.

The elastic stiffness constant k of each elastic coupling means structure varies inversely with the length of blocks **23**, and the column must therefore be sufficiently short to assure a natural resonant frequency in accordance with Equation 1 which is as high or higher than the order of magnitude of bit tooth contact frequency for the corresponding cutter. At the same time, the column must be long enough in relation to its modulus to assure a range of elastic deformation and restoration as great as bit vibration amplitude.

The lower limit for the resonant frequency of the elastic bit coupler and bit, as defined by Equation 1, is the key to my invention; and many variations in the physical configuration of my elastic coupling member may be made so long as the structural design invariably incorporates such elasticity and mass constants, as explained hereinabove, that the described frequency response characteristic is maintained.

In operation, the bits **17**, bit shafts **20** and elastic coupling means made up of the columns of elastic cylinders **23** are free to vibrate independently of the high inertia sub **12** and collars above. Moreover, each bit **17**, and corresponding bit shaft **20** and column of elastic cylinders **23**, is free to vibrate independently of each other such group. This vibration, as described before, is caused by engagement of successive bit roller cutter teeth with the hole bottom, and occurs at bit tooth contact frequency for the condition of the rock being drilled, and for the speed of rotation of the drill collars, each bit elevation, and corresponding elastic coupling means contraction, as each bit tooth of each cutter engages the hole bottom and lifts the bit, being followed by an elastic expansion of the coupling means and forced lowering of the bit as fast as permitted by the passage of the tooth in question beyond its centered position over the rock. The toothed bit cutters accordingly rise and fall as their teeth engage the rock, but remain in effective contact with the rock, and the sub **12** and high inertia drill collars ride along steadly, with their loading transferred continuously through the several vibrating elastic coupling means to the vibrating bit cutters and thence to the rock.

It is seen that the three sets of vibratory systems comprised of a bit **17**, with its cutter **18**, and corresponding elastic coupling means, operate entirely independently of one another. They may advantageously be designed to operate at different frequencies. By using cutters **18** of different tooth spacings, correspondingly different cutter tooth engagement frequencies are established, and the three vibratory systems may be tuned to respond properly to these three different frequencies. That is to say, each independent vibratory system comprising a bit **17**, with its cutter **18**, and corresponding elastic coupling means, must have a natural resonant frequency as high as, or higher than, the frequency of cutter tooth engagement frequency with the rock for that particular cutter. The resonant frequencies of the three systems may be the same, or different, just so long as each is as high as cutter tooth engagement frequency with the rock for its corresponding cutter. By thus using different numbers of teeth on the different cutters (while retaining the proper lower limits of resonant response frequency for the vibratory systems) tracking of the toothed cutters in indentations in the rock made by other cutters is thereby avoided.

Fig. 6 shows a modified type of elastic coupling means for the drill of Figs. 1–4, made up in this instance of a stack of dished steel spring washers **60** placed between the upper end **16** of bore **15** and a plug **51** fitted into the upper end of bit shaft **20**. In the example shown, these washers **60** are used in successive groups of eight, the disks of alternate groups having their concave sides facing upwardly, and then downwardly, as shown. In this case, it is the shear modulus of the elastic coupling means instead of Young's modulus that is involved in the design of the coupling means. It is here necessary, in a manner analogous to that stated for the embodiment of Figs. 1–4, that the stack of spring washers have such compliance and such

load bearing capacity, in relation to the shear modulus of the material of which the springs are composed, as will withstand the loading imposed by the collars without undue elastic deformation, and with a range of elastic deformation and restoration, while loaded with the collars, which is at least equal to the amplitude of bit vibration. As in the first described embodiment, the elastic coupling means and bit are designed with the required stiffness for capacitative frequency response at bit tooth engagement frequency with the hole bottom.

Fig. 7 shows still another modified form of elastic coupling means for the drill of Figs. 1–4, and consisting in this case of a confined elastic body under compression, subject to elastic volume change in response to bit vibration. Thus the volume or bulk modulus of elasticity is involved in this case, and the elastic body may be composed of any suitable compressible material whose bulk modulus is suitable. Certain viscous liquids such as a silicone grease, or a polymer hydrocarbon, may be used. Thus a body **60** of such material is contained in a cylinder **61** fitted into bore **15**, and a piston **62** on a plug **63** fitted to the upper end of bit shaft **20** is slidably received into cylinder **61** from the bottom so as to engage the body **60**, and so subject it to a compressive pressure. The volume of body **60** and the geometry of the cylinder **61** are made such, with relation to the bulk modulus of the body **60**, that, under the compressive loading imposed by the drill collars, there is afforded the necessary capacitative frequency response and a range of vertical elastic deformation of the body **60** at least equal to the amplitude of bit vibration. Otherwise, operation is as described in connection with the earlier described embodiments.

Reference is next directed to the embodiment shown in Figs. 8–11, showing a form of the invention using an elastic coupling means in the form of a tubular steel column and using a conventional tri-cone bit. Numeral **70** designates generally the lower end portion of the usual string of drill collars, of which the lowermost collar **70***a*, understood to have a typical length of 40 feet, has certain modifications to be explained presently. It is to be understood that these collars are again employed in usual numbers, and in usual outside diameter and length. Thus, the collars may be of 40-foot lengths, of a typical diameter of 7″, and there may be three or four such collars, the uppermost collar being understood as connected to the usual drill pipe. It will further be understood that all of the collars excepting the special lowermost collar **70***a* may be entirely conventional. The several collars above lowermost collar **70***a* are formed with the usual bores forming circulation passage **73**, and this circulation passage is continued downwardly into lowermost collar **70***a* by means of bore **74** extending a short distance down into the collar. A counterbore **75** extends upwardly from the lower end of the collar **70***a* nearly to the lower end of bore **74**, and is slightly reduced at **76**, so as to meet a reduced counterbore **77** which extends upwardly to a juncture with bore **74**, forming an annular downwardly facing shoulder at **78**. The tapered screw-threaded coupling pin **79** on the lower end of collar **70***a* is joined with the screw-threaded box **80** on the upper end of a tubular fitting or sub **81**. A sleeve **82** is telescopically received inside sub **81** and is splined thereto as indicated at **83** (Figs. 9 and 10). Formed on the lower end of sleeve **82** is a coupling box **84** which receives the threaded coupling pin **85** on the upper end of a conventional toothed roller cutter rock bit **86**, the latter being equipped with a plurality of toothed bit roller cutters **87** of any typical conventional design. The illustrative bit **86** indicated in Fig. 8 has three conventional toothed roller cone cutters mounted on a usual bit body **88**.

An elastic coupling means in the form of a long steel tube **89** is received inside collar **70***a*, and extends downwardly into the axial bore **82***a* of sleeve **82**, the bore of tube **89** being seen to form a continuation of the circulation passage **74**. Welded to the upper end of this elastic tube **89** is an upper end ring **90** which fits slidably into collar bore **77** and is adapted to abut shoulder **78**, being provided with grooves and sealing rings, as shown, to seal off against circulation fluid. A similar lower end ring **90** welded onto the lower end of tube **89** is slidably received into a slightly reduced bore **91** at the lower end of bore **82***a* in sleeve **22**, and abuts a shoulder **92** at the lower end of bore **91**. The circulation passage is extended downwardly from the lower end of tube **89** through passageway **93** in sleeve **82** and coupling box **84**, and is continued thence through the usual circulation passage **94** in bit **86**, the fluid being finally discharged through usual ports (not shown) in the bottom of the bit, in any conventional fashion.

A bronze bearing bushing **95** is press-fitted into sub **81** above splines **83** and its lower end engages an upwardly facing shoulder **96** formed in sub **81**. A bearing sleeve **97** is also press-fitted into the lower end of sub **81**, and the splined sleeve member **82** is longitudinally movable through a limited distance on these bearing sleeves **95** and **97**. The surface of sleeve **82** opposite bushing **97** is furnished with suitable seals against external well fluids, as indicated.

The upper extremity of sleeve **82** is externally screw-threaded, as at **99**, and receives an internally screw-threaded ring **100**, which is screwed down into engagement with the upper end of a ring **101** received with a close fit in the annular space between sub **81** and sleeve **82**, the lower end of the ring **101** seating against an upwardly facing shoulder **102** on sleeve **22**. Ring **101** is formed with suitable seals facing the internal surface of sub **81** and the external surface of sleeve **82**, as shown. In Fig. 9, the parts are shown in the position which they assume when the bit is in engagement with the bottom of the drill hole. When the assembly is lifted, so that weight is off bottom, sleeve **82** and the bit carried thereby drop a short distance until ring **101** is seated on the upper end of bushing **95**, and it will be seen that in lowering or elevating of the drill string, the sleeve **82** and bit thus hang from the upper end of bushing **95**. When the bit strikes bottom, the drill collar **70***a* and sub **81** move downward a short distance relative to the bit and sleeve **82** to the position shown in Fig. 9, and it will further be seen that the latter position is determined by the elastic coupling tube **89** and its end rings **87** and **90** then engaging and occupying the full distance between the shoulders **78** and **92**. Also, in the latter position, the weight of the collars and string above is imposed on the upper end of the elastic tube **89**, and transferred from the lower end of tube **89** to the sleeve **82** and thence to the bit.

Preferably, for lateral support of the long and relatively thin elastic tubing **89**, a guide or support tubing **105**, of some suitable material, such as phenolic is placed in the collar bore **75** around the tube **89**, at close annular spacing from the latter, the lower end of the tubing **105** being supported as by retaining ring **106**.

For lubrication of the splines **83** and the upper bearing bushing **95**, a lubricant filler opening **107**, closed by a plug **108**, is located in sub **81** just below the splines, and the sub **81** is also formed, above bearing **95**, with an opening **109** closed by plug **110**, which plug is removed during introduction of grease via opening **107**. The lower bearing bushing **37** may be lubricated via a normally plugged opening **111**.

Ports **112** in the wall of sub **81** establish communication from the space **113** inside said sub, between bit connected sleeve **82** and the lower end of collar **70***a*, to the outside. In the operation of the drill, the volume of this space **113** fluctuates as the sleeve **82** reciprocates

owing to bit vibration, and serious damping would occur without the pressure relief ports **112**.

In operation, in a manner analogous to that given in connection with the earlier described embodiments, the bit **86** and elastic coupling tube **89** are free to vibrate independently of the high inertia collars above, and the sub **81** rigidly connected to the latter, the bit-connected sleeve **82** sliding on the splining **83** between it and the relatively stationary sub **81** as the tube **89** and bit undergo their vibration. This vibration as described before, is caused by engagement of successive bit roller cutter teeth with the hole bottom, and occurs at the bit tooth contact frequency. During rotation of the drill collars, as each bit tooth engages the hole bottom, the bit is lifted causing the elastic coupling means to compress. As the bit tooth rolls off its crown, there follows an elastic expansion of the coupling means and a consequent lowering of the bit which occurs as rapidly as is permitted by the rolling of the tooth in question beyond its centered position over the rock. The toothed bit cutters accordingly rise and fall as their teeth engage the rock, but remain in contact with the rock, and the sub **12** and high inertia drill collars ride along steadily, with their loading transferred continuously through the vibrating elastic coupling means to the vibrating bit cutters and thence to the rock.

The embodiment of Figs. 8–11 again embodies a combination in which the elastic column has sufficient load bearing capacity for its elastic modulus to withstand the compressive loading by the collars without undue elastic deformation, is sufficiently compliant to permit a range of elastic deformation, over and above that caused by its compressive loading, by an amount no less than bit vibration amplitude, and, taken together with the bit, has such mass and elasticity constants that the column and bit have a natural resonant frequency of vibration at least as high as the frequency of cutter tooth engagement with the hole bottom at normal speed of drill collar rotation. It will be seen that with synchronized bit rollers, whose teeth simultaneously engage the bottom of the hole, cutter tooth engagement frequency is the same as though there were only a single toothed cutter. On the other hand, assuming three cutters synchronized with equal phase difference between them, cutter tooth engagement frequency would be multiplied by a factor of three. There is a tendency for the cutters to self synchronize with one another according to the conditions of the first mentioned case. They can of course be mechanically synchronized as desired. Under many conditions of operation, the cutters will operate with random phase relations. Another variable results from the fact that most roller cutters have different numbers of teeth. Moreover, in many cases, owing to high spots on the hole bottom, slanting formation, or other conditions, the primary vibratory drive comes successively from the several cutters as they pass successively over the assumed high spot or the like, in which case the vibration frequency is that owing at any given time to a single cutter. In any case, for operation in accordance with the invention, the resonant response frequency of the elastic column must be as high as that actual cutter tooth engagement frequency with the hole bottom that corresponds with the actual vibration frequency of the bit and elastic coupling means.

Figs. 12–16 show another illustrative embodiment, using a helical form of elastic coupling means, such that the elastic shear modulus of the coupling means is involved. The lower end portion of a usual string of drill collars is indicated at **120**, and coupled thereto, as by coupling **121**, is the upper end of helical elastic member **122** whose lower end carries the bit, fragmentarily indicated at **123**. The bit may be of any conventional or suitable toothed roller cutter type, as before.

The helical elastic coupling means **122** is milled from a solid cylindrical bar of steel. It comprises an upper cylindrical stem **124**, of the same diameter as the drill collars above, formed at the top with threaded coupling pin **125** for connection with the box of coupling **121**. Below stem **124** is a coil portion **126** of enlarged outside diameter, and a bore **127** extends up through this coil portion and up into stem **124** to a blind end **124***a*, as shown. The coil portion **126** is formed, in the present case, by means of four spiral slots **128**, forming a quadruple pitch spring integrally connecting the stem **124** with a base annulus **130**.

The four spiral legs or bars **131** of the quadruple pitch spring are hollow, forming mud fluid passageways **132**. This formation may be provided by first grooving the bars **132**, and then welding spiral filler strips **133** into place, as shown.

The upper ends of the passages **132** communicate with the lower ends of passages **136** (Figs. 13, 14 and 15), and the upper ends of the latter communicate via passages **137** with a bore **138** extending downwardly through coupling pin **125**. It will be understood that the bore **138** is thus in communication with the mud fluid passage extending downwardly through the drill stem and collars above. The passage **136** is shown in this instance as having been formed by vertically grooving the sides of stem **124**, and then welding filler strips **139** in place, as illustrated in Figs. 13, 14 and 15.

Engaging the lower end of annulus **130** at the base of the elastic spring member is a mounting flange **140** of a long mandrel member **141** that extends upwardly, with clearance, through the bore of the coil spring and stem **124**. The lower end of this mandrel **141** is closed with a partition **142**, and is joined to flange **140** by a generally conical wall member **143**. This wall member **143** will be seen to be received snugly within the base annulus **130**. A bit coupling member **145** has a shoulder engaging flange **140**, and extends upwardly with a snug fit inside the wall portion **143**, screws **146** securing the coupling member **145**, flange **140**, and annulus **130** in proper assembly. The coupling member **145** has internal threads to receive the coupling pin **147** at the top of the bit.

Bores **148** through wall **143** and extending into annulus **130** will be seen to communicate with the fluid passages **132** through the several legs of the spring, so that the circulation passage downwardly through the spring is thus continued to the space **150** above the upper end of the coupling pin **147** at the upper end of the bit, to be continued downwardly through the usual circulation passage in the bit, as indicated.

The mandrel member **141** serves as a guide or lateral supporting member for the coil spring to prevent undue lateral deflection or buckling. It also serves as a mass-loading member, which can be interchanged readily with heavier or lighter mandrels to adjust the tuning of the combination.

The operation of this form of invention is in all material respects entirely analogous to that of the earlier described embodiments. As before, the elastic member is designed for sufficient load bearing capacity for its elastic modulus (shear modulus, in this case), to withstand the compressive loading by the collars without undue elastic deformation, and is made to be sufficiently compliant to permit a range of elastic deformation, over and beyond that caused by its compressive loading, by an amount no less than bit vibration amplitude. In addition, the elastic spring member taken together with the bit and associated structure has such mass and elasticity constants that the column and bit have a natural resonant frequency of vibration at least as high as the frequency of cutter tooth engagement with the hole bottom at normal speed of drill collar rotation.

Figs. 17 and 18 show another embodiment of the invention, initially disclosed and claimed in my copending parent application Ser. No. 257,033 (hereinafter more completely identified). This embodiment of the invention is of the same general type as that of Figs. 8–11

embodying an elastic coupling member in the form of a tubular steel column. Referring to Figs. 17 and 18, numeral **160** designates the usual string of drill collars, of which lower collar **160***a* has certain modifications to be explained presently. These collars are again employed in usual numbers, and in usual outside diameter and length. Thus, the collar **160***b* above lowermost collar **160***a* will be understood to be a more or less standard drill collar, while there may be one or more additional collars similar to **160***b* connected above, the uppermost collar being understood as connected to the usual drill pipe. The collars are coupled to one another by conventional coupling pins **161** and boxes **162**, in the usual fashion. Collar **160***b* and the remaining collars above are formed with the usual bores forming circulation passage **163**, and a threaded socket **164** is formed in the lower end of coupling pin **161** of collar **160***b* to receive the threaded upper end of a long, relatively thin walled elastic coupling tube **165**, the bore in the latter forming a continuation of circulation passage **163**. Lowermost collar **160***a* is formed with an enlarged bore **166** to receive this tube **165**, with suitable clearance for operation.

Elastic tube **165** reaches substantially to the lower end of drill collar **160**, where it is welded to a tool joint **170** having a threaded box **171** adapted to receive the pin **172** on the upper end of conventional toothed roller cone rock bit **174**, the latter having conventional toothed conical cutters **175**. Tool joint member **170** and bit **174** have circulation passages **176** and **177**, respectively, the latter leading to mud discharge ports **178** in the bottom of the bit.

Tool joint **170** has a tapered shoulder **180** connecting its cylindrical side wall with the elastic tube **165**, and the collar **160***a* has at its lower end a conical opening **181** spaced from this shoulder **180**, the clearance being sufficient to avoid the possibility of engagement between tool joint and collar during vibratory action of the bit.

The vibratory assembly consisting of the elastic coupling tube tool joint and bit are designed with reference to the frequency of cutter tooth engagement with the hole bottom at normal speed of drill collar rotation in accordance with the theoretical discussions set forth hereinabove in connection with the earlier described embodiments. In addition to the considerations with reference to load bearing capacity of the elastic coupling member, length of elastic tube for its modulus and cross section to give the required working range of elastic deformation beyond that due to the weight of the drill collars, etc., the elastic tube, tool joint **170** and bit are again of such mass and elasticity characteristics as to have an equivalent resonant frequency of vibration which is at least as high as the frequency of cutter tooth engagement with the hole bottom at normal speeds of drill collar rotation, so as to assure that the contracted coupling tube will be able to expand to force the bit down between successive elevations caused by engagement of successive cutter teeth with the hole bottom. Briefly stated, the tube must be sufficiently short to assure a natural resonant frequency as high or higher than bit tooth contact frequency with the hole bottom.

A number of illustrative embodiments of the invention have now been shown and described. It will be understood, however, that these are but illustrative of the invention, and that various other changes and modifications in design, structure and arrangement may be made without departing from the spirit and scope of the invention as defined by the appended claims.

This application is a continuation-in-part of my application entitled Deep Well Drill With Elastic Coupler for Bit, Serial No. 257,033, filed November 19, 1951, allowed November 4, 1955, now abandoned.

I claim:

1. In a rotary well drilling system, the combination of: a drill collar, a toothed bit roller cutter adapted to make successive engagements with the hole bottom during rotary drilling and thereby tending to produce vertical vibration of the cutter when the cutter is forced against the hole bottom, and elastic means including coupling and supporting means between said collar and cutter and arranged to support said collar on said cutter, said elastic means and associated structure being so constructed and of such elastic modulus that it will carry the load of the collar without collapsing and without undue elastic deformation, and having a vertical range of elastic deformation and restoration at least as great as the amplitude of said cutter vibration while loaded by said collar, the equivalent stiffness of said elastic means, which is represented by the factor k, and the equivalent vibratory mass of said combined cutter and elastic means, which is represented by the factor m, being related by the formula

$$f=\frac{1}{2\pi}\sqrt{\frac{k}{m}}$$

wherein f is the equivalent frequency characteristic of said elastic means and cutter, and k and m have such values that f is at least as high as the frequency caused by bit roller-tooth contact at normal speeds of drill collar rotation.

2. In a rotary well drilling system, the combination of: a drill collar, a toothed bit roller cutter adapted to make successive engagements with the hole bottom during rotary drilling and thereby vibrate vertically when forced against the hole bottom, and an elastic vertically vibratory means intercoupled between said collar and said cutter, in such manner as to support said collar on said cutter, and to vibrate in response to cutter vibration, said elastic means being so constructed and arranged and of such elastic modulus as will carry the load imposed by said collar without undue elastic deformation, and having a vertical range of elastic deformation and restoration at least as great as the amplitude of said cutter vibration while loaded by said collar, the equivalent elastic stiffness k and mass m of the vibratory system including said elastic means and cutter being related by the equation

$$f=\frac{1}{2\pi}\sqrt{\frac{k}{m}}$$

where f is the effective resonant vibration frequency characteristic of said vibratory system, and k and m have such values that f is at least as high as the actual effective vibration frequency caused by bit roller tooth contact with the hole bottom at normal speeds of drill collar rotation.

3. In a rotary well drilling system, the combination of: a drill collar, a set of toothed bit roller cutters, each of whose teeth are adapted to make successive engagements with the hole bottom during rotary drilling, whereby the cutters vibrate vertically when forced against the hole bottom, and an elastic vertically vibratory means intercoupled between said collar and said cutters, in such manner as to support said collar on said cutters, and to vibrate in response to cutter vibration, said elastic means being so constructed and arranged and of such elastic modulus as will carry the load imposed by said collar without undue elastic deformation, and having a vertical range of elastic deformation and restoration at least as great as the amplitude of said cutter vibration while loaded by said collar, the equivalent elastic stiffness k and mass m of the vibratory system including said elastic means and cutters being related by the equation

$$f=\frac{1}{2\pi}\sqrt{\frac{k}{m}}$$

where f is the effective resonant vibration frequency characteristic of said vibratory system, and k and m have such values that f is at least as high as the actual

effective vibration frequency of the elastic means caused by bit roller tooth contact with the hole bottom at normal speeds of drill collar rotation.

4. In a rotary well drilling system, the combination of: a drill collar, a plurality of bit roller cutters, each of which is adapted to make successive engagements with the hole bottom during rotary drilling and thereby vibrate vertically when forced against the hole bottom, and a corresponding plurality of elastic independently vertically vibratory means, one for each of said cutters, intercoupled between said collar and the respective cutters, in such manner as to support said collar on said cutters, and to vibrate independently of one another in response to vibration of the respective cutters, said plurality of elastic means being so constructed and of such elastic modulus as will carry between them the total load imposed by said collar without undue elastic deformation, and each having a vertical range of elastic deformation and restoration at least as great as the amplitude of said cutter vibration while loaded by said collar.

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