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(54) **Synchronous vibratory impact hammer.**

(57) A vibratory impact hammer including a hammer body assemblage suspended by rubber mounts for reciprocal axial movement in a support frame, the rubber mounts providing guiding and damping action of the assemblage in either direction of axial movement without extraneous friction forces acting thereupon, and a pair of synchronously driven eccentric weights which are arranged to provide vibratory movement of the assemblage.

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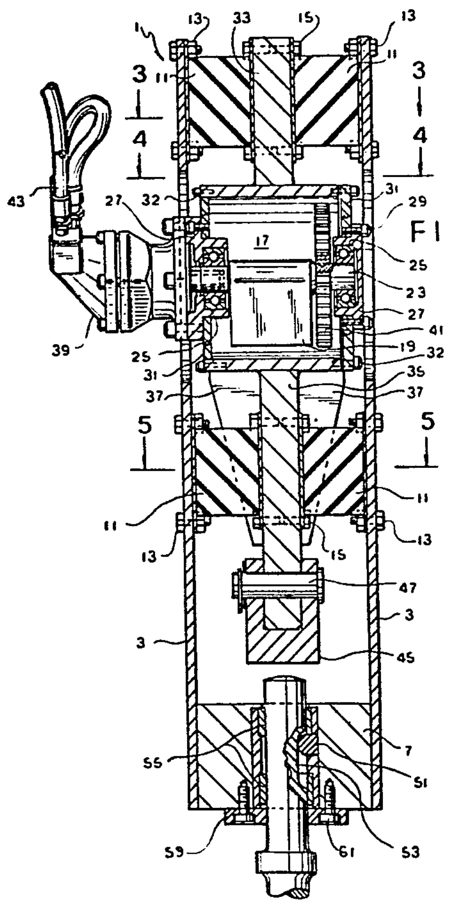
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FIG.2

SYNCHRONOUS VIBRATORY IMPACT HAMMER

This invention relates to a synchronous vibratory hammer employing a driving and a driven eccentric weight arranged to produce vibratory action which may be used for impacting a tool upon a work surface.

The art of vibratory hammers, of the type with which this invention is concerned, is well developed and many different designs have been proposed and employed with varying degrees of success. U.S. Patent No. 3,866,693, dated February 18, 1975 to Bernard A. Century, is representative of one such vibratory hammer.

The subject invention has certain elements in common with the device of the Century patent, however, it differs in at least one important respect, namely, it has no mechanical restraints which can absorb energy, such as would be caused by the guides 186 and 188 of Century's patent. The mechanical restraints in the Century patent are used to control non-linear motion of the hammer element being driven by a single eccentric. The device of the subject invention eliminates the need for such mechanical restraints because of the fact that two eccentrics are used.

The device of the subject invention requires less maintenance than vibratory hammers having non-linear impacting vibrations, which not only shake the hammer supporting mechanism, but are subject to greater wear and breakage.

A primary purpose of the invention is to provide a vibratory hammer with improved operating efficiency, and which minimizes maintenance costs.

These and further purposes and features of the invention will become more apparent from an understanding of the following disclosure with reference to the drawings as set out in the respective figures thereof.

From the drawings, it will be seen that

Fig. 1 is a side elevation view of a vibratory hammer embodying the principles of the invention;

Fig. 2 is an enlarged section view as seen from line 2-2 in Fig. 1;

Figs. 3, 4 and 5 are cross section views as seen from lines 3-3, 4-4 and 5-5 respectively in Fig. 2;

Fig. 6 is an exterior view of a hammer body component used in the device of Fig. 1.

Referring now to Figs. 1 and 2, numeral 1 identifies a vibratory hammer having a support frame consisting of a pair of side plates 3, which are maintained in parallel position by means of tubular space bars 5, welded to the plates, as well as a tool holder element 7, similarly welded thereto. A hammer body assemblage 9 (Fig. 6) is suspended between the side plates 3 by resilient means consisting of four rubber mounts 11 affixed to the side plates and the hammer body assemblage by means of bolts 13 and 15. As will be apparent, the mounts serve as the sole guiding and damping means for the assemblage when the latter is vibrated during tool operation.

The hammer body assemblage 9 includes an eccentric weight chamber 17 enclosing a pair of eccentric weights 19 and 21, mounted upon shafts 23 supported in roller bearings 25 positioned in end caps 27, the latter being secured by bolt means 29 to side members 31 of the eccentric weight chamber 15 by bolts 32. Projecting from the top surface of the eccentric weight chamber 17 and affixed thereto, is an upper arm member 33 adapted to be affixed to the rubber mounts 11 by the bolts 15.

Projecting from the bottom surface of the eccentric weight chamber 7 and affixed thereto, is a lower arm member 35 adapted to the rubber mounts 11 by the bolts 15. Brace members 37 are secured to the sides of the lower arm member 35 and the hammer body assemblage 9, to stabilize the arm member. An hydraulic motor 39, affixed to the end of shaft 23, is provided to rotate the eccentric weight 19. A pair of gears 41, mounted upon the shaft 23, is arranged to transmit rotary motion from the shaft which supports eccentric weight 19, to the shaft which supports eccentric weight 21, so that both weights are rotated at the same speed, but in opposite directions. Hose means 43 supply pressurized hydraulic fluid to the motor 39 when desired, from a power source, not shown.

At the lower extremity of the arm member 35 is a striker plate 45 affixed thereto by means of pin 47. The striker plate is arranged to impact upon a conical tool 49 mounted in the tool holder element 7 as best seen in Fig. 2. Retaining means, including a key 51 projecting into a slot 53 formed in the tool 49, allow reciprocal movement of the tool. The tool slides in bushings 55 supported in a bushing housing 57, the latter positionally maintained against axial movement by a tool stop plate 59, affixed to the end of the tool holder by means of cap screws 61.

Support means for the vibratory hammer 1, are provided by a pivotally attached linkage assemblage 63, which may be operatively positioned by power machinery, e.g., tractor, not shown.

The design parameters of a vibratory hammer built in accordance with the invention disclosed herein, obviously will vary in accordance with the work impact output desired.

It is to be recognized that when a forcing frequency vibrates a mass at its natural frequency, the mass of the forcing frequency generator leads the vibrated mass by 90°. When the forcing frequency is much higher than the natural frequency, the forcing frequency mass could lead the vibrated mass by 180°.

Accordingly, if the leading phase is 135°, the vertical component of centrifugal force of the vibrated mass, coupled with the stored energy of the rubber mounts, will produce maximum impacting on the tool 49.

The optimum phase angle of 135° (θ) is determined by the following equation:

$$\tan\theta = \frac{2\,\zeta\,\frac{f}{fn}}{1 - \frac{f^2}{fn^2}}$$

where

θ = phase angle
ζ = damping factor
f = forcing frequency CPM, RAD/SEC
fn = natural frequency

It can be demonstrated by plotting frequency ratio vs. θ with varying damping factors, that anything less than a critically damped system gives phase angles of approximately 180° at any frequency ratio greater than 1, hence, critical damping of the system is essential for optimum operative results. Critical damping by definition means no over oscillation when a mass is deflected from its static position and returned to the same static position. Critical damping is achieved by a preload, in the present vibratory hammer, by use of the rubber mounts 11.

It is essential, for optimum operation, that the stroke of the hammer be equal to the "in the air" displacement (S"), which is provided by the following equation:

$$S'' = \frac{2\,wr}{W}$$

Where

w = unbalanced weight
r = radius where unbalanced weight is located from the center of rotation
W = total weight vibrated

By application of these formula, a vibratory hammer in accordance with the invention will have the following numeral values, if a work impact output of 200 ft. - lbs. at 1200 rpm is to be achieved:

Stroke	0.4144	inches	(1.052 cm)
Weight of hammer body assemblage (9)	485.52	lbs	(220.23 kg)
w =	60.69	lbs	(27.71 kg)
r =	1.6576	inches	(4.21 cm)
wr =	106.59	in lbs	(192 kg-cm)
wr^2=	166.76	in^2 lbs	(760.8 kg cm^2
K =	3400 #/in.		(239 Kgms/cm^2)

WHAT IS CLAIMED IS:

1. A vibratory impact hammer including a support frame characterized by a hammer body assemblage suspended within the support frame by resilient means arranged to provide guiding and damping action in either direction of axial movement of the hammer body assemblage, said resilient means being the sole means engaging the hammer body assemblage so that extraneous frictional forces are avoided, vibration drive means arranged to vibrate the hammer body assemblage in an axial direction, and a tool reciprocably mounted in the support frame and positioned to receive impact blows of the hammer body assemblage when reciprocated by the vibratory drive means.

2. A vibratory impact hammer as in claim 1, characterized in that said resilient means are rubber mounts.

3. A vibratory impact hammer as in claims1 or 2, characterized in that said rubber mounts are arranged in pairs, one above the hammer body assemblage, the other below the hammer body assemblage.

4. A vibratory impact hammer as in claim 1, characterized in that said vibratory drive means includes a driving eccentric weight and a driven eccentric weight, and a motor means arranged to rotate the weights in synchronism.

5. A vibratory impact hammer as in any one of claims 1-4, characterized in that said eccentric weights are inter-connected by gear means.

6. A vibratory impact hammer as in claim 1, characterized in that a tool holder is provided for the tool, which tool holder includes means for removal of the tool from the hammer.

7. A vibratory impact hammer as in claim 1, characterized in that the phase angle of the forcing frequency leads the vibrated mass frequency by 135^{o}.

8. A vibratory impact hammer as in any one of claims 1-7 characterized in that the stroke of the hammer body assemblage is equal to $2\frac{wr}{W}$, wherein w = unbalanced weight, r = radius where unbalance is located from the center of rotation, and W = total weight vibrated.

9. A vibratory impact hammer as in any one of claims 1-8, characterized in that a hammer with a work impact output of 200 ft. - lbs. at 200 rpm, would have a stroke of 0.4144 inches (1.052 cm) and design parameters as follows:

W = 485.52 lbs (220.23 Kgms)

w = 60.69 lbs (27.71 Kgms)

r = 1.6576 inches (4.21 cm)

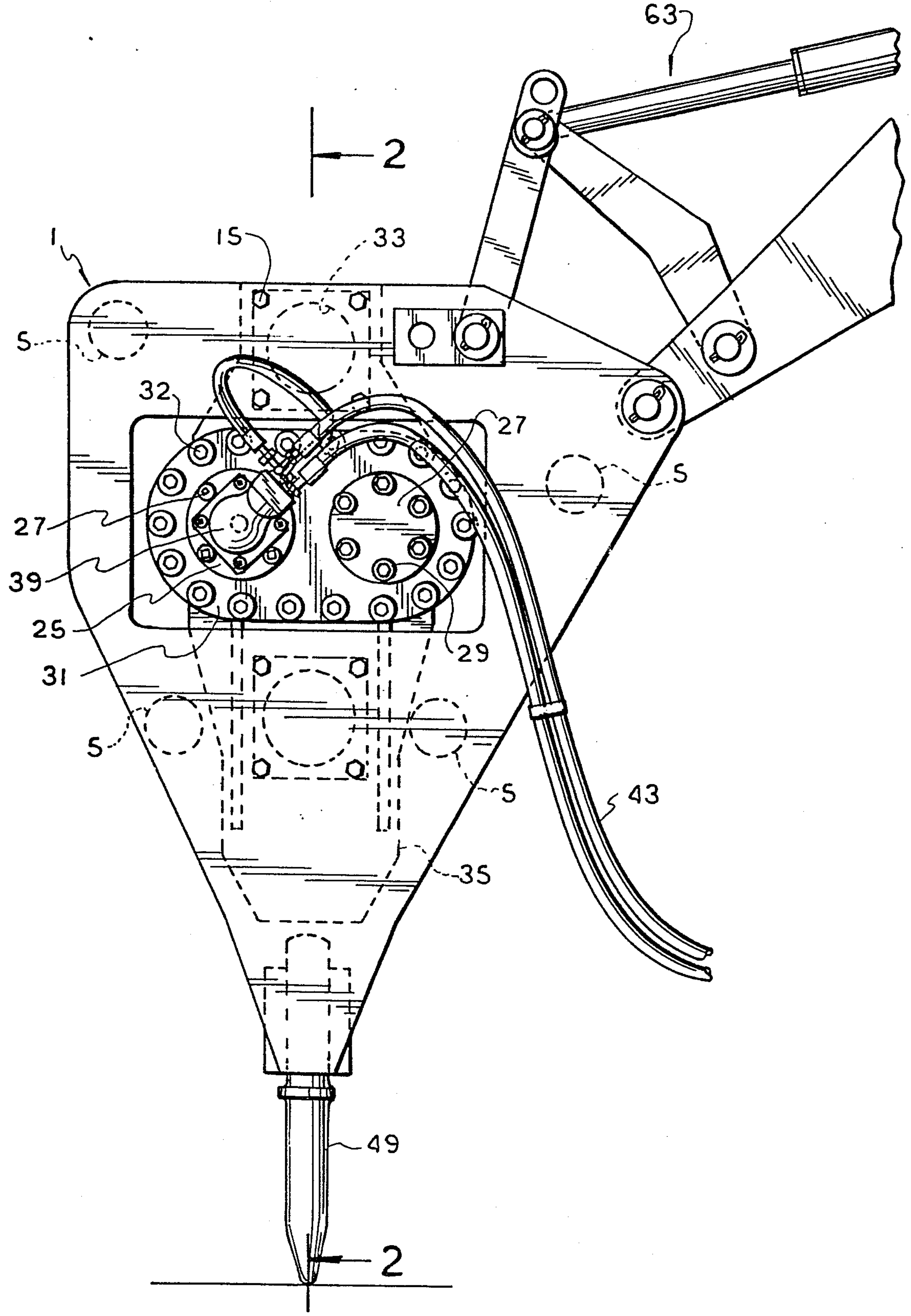
wr = 106.59 in. - lbs.

wr^2 = 166.76 lbs. - in^2

K spring rate

(item 11) = 3400 #/in. (239 Kgsm/cm^2)

FIG. 1



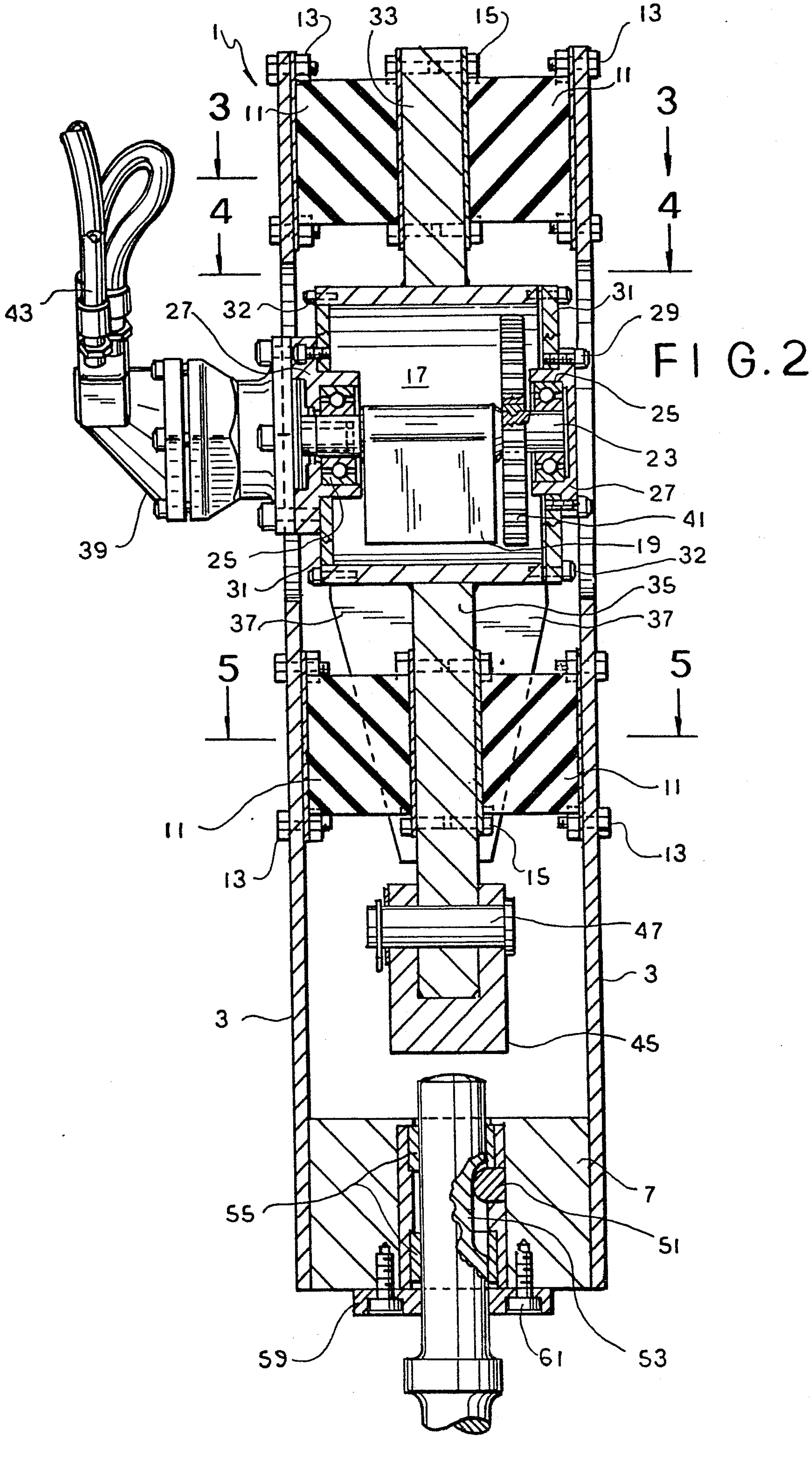

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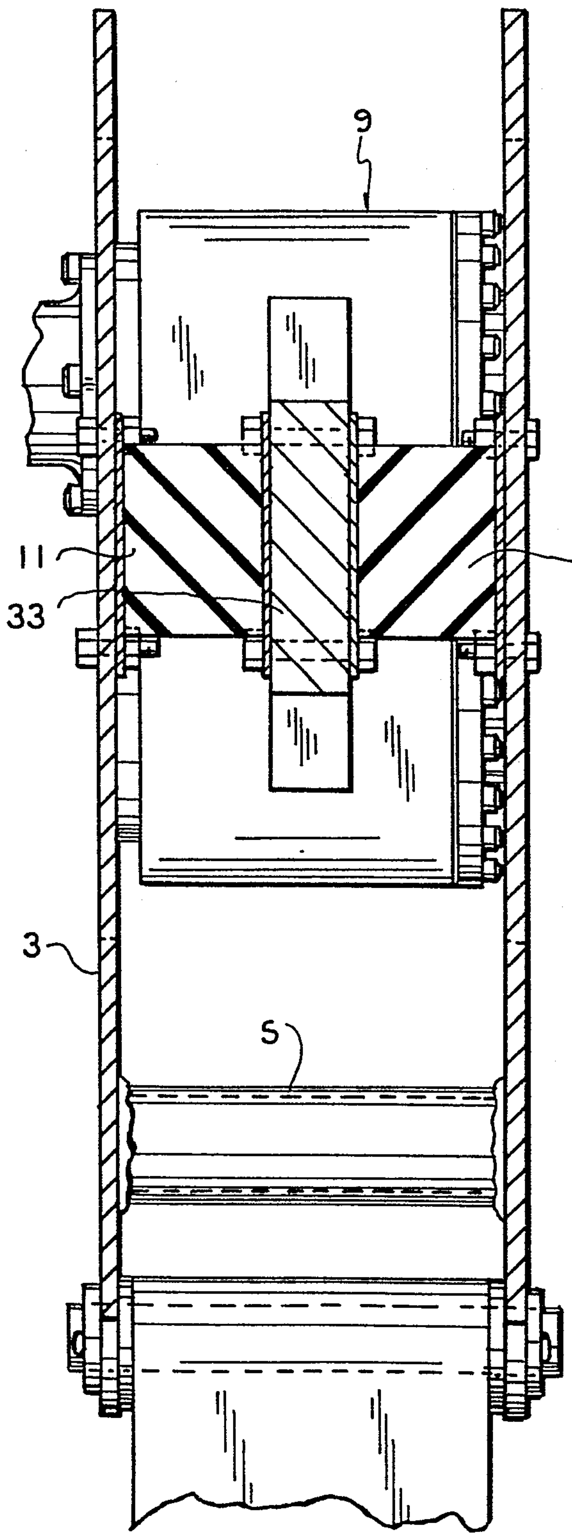
FIG. 2

FIG. 3



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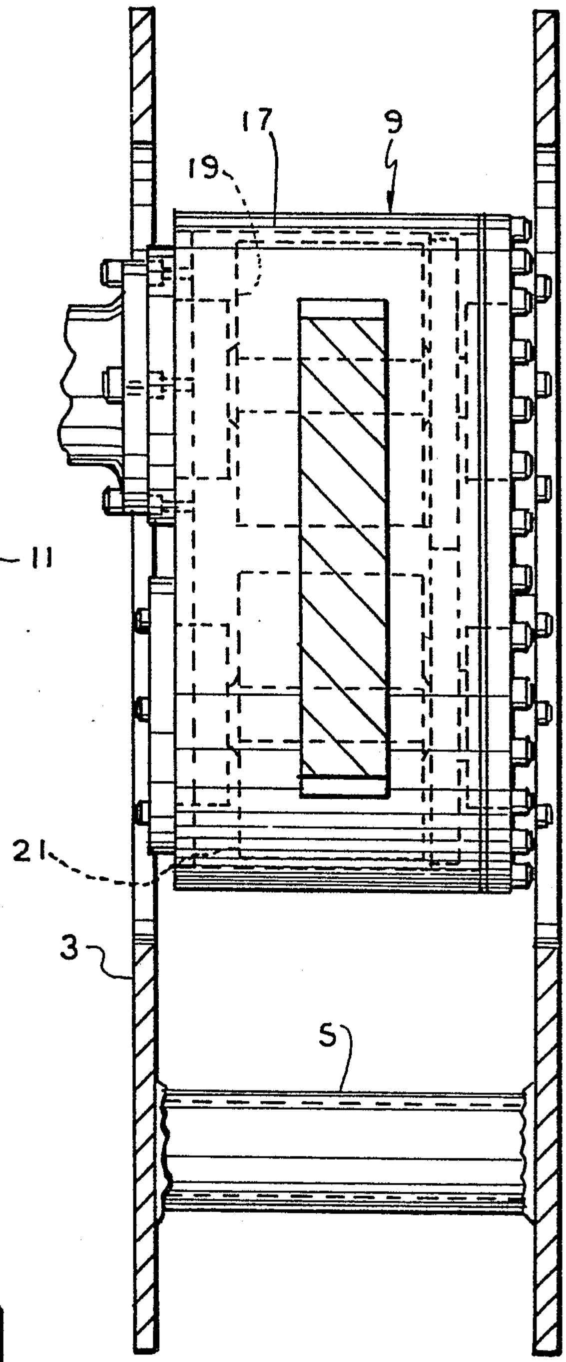

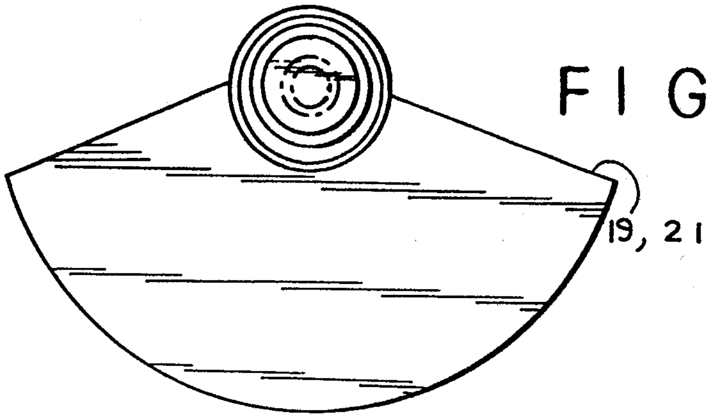

FIG.7

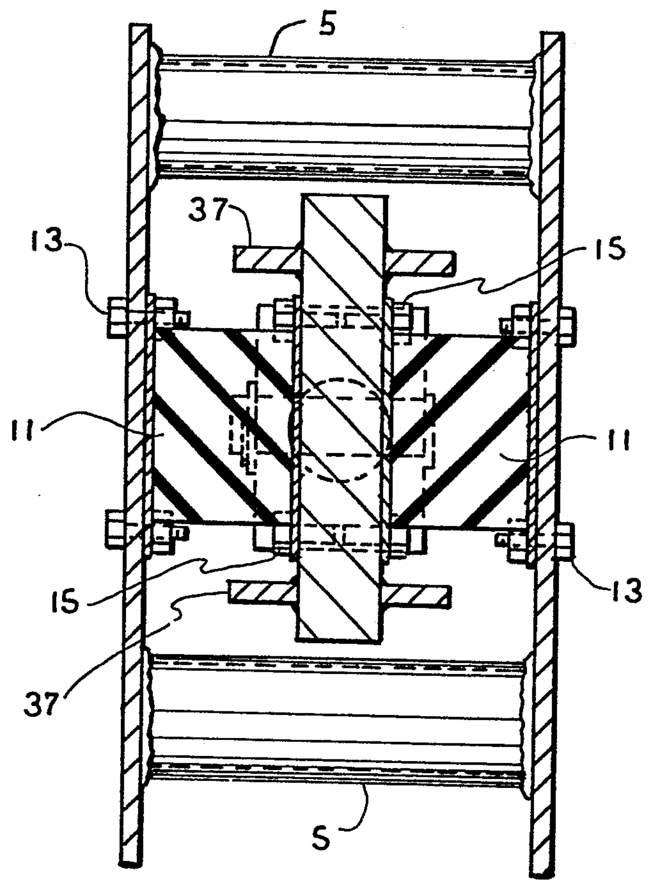

FIG.5

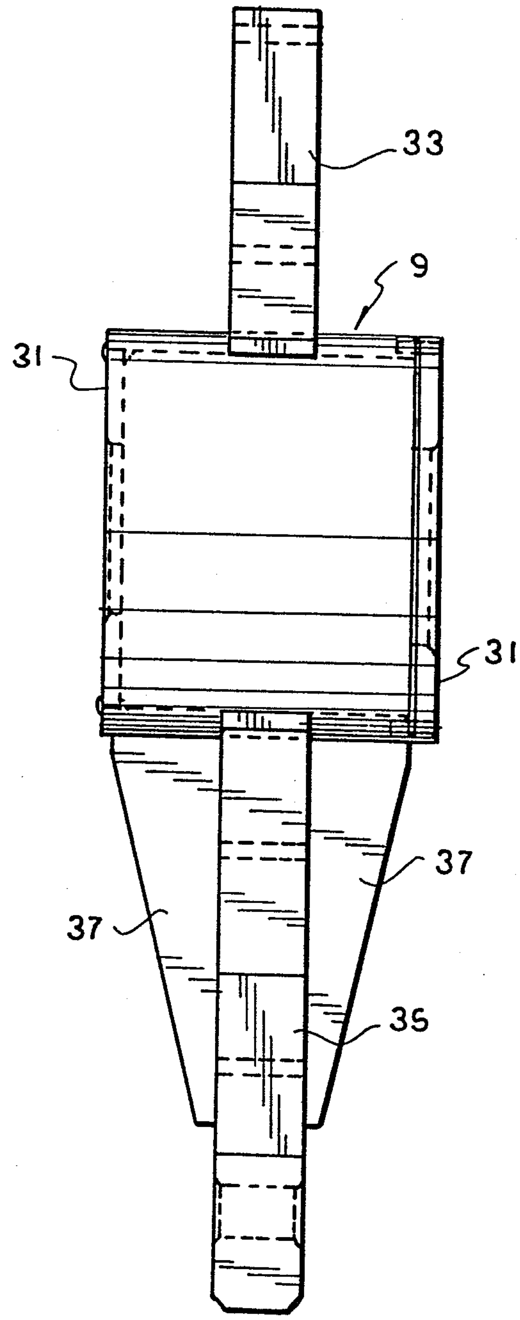

FIG.6