United States Patent [19]

Spencer

[11] E **Re. 30,598**

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[54] **METHOD FOR TRANSVERSE CUTTING**

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[73] Assignee: **Paper Converting Machine Company**, Green Bay, Wis.

[21] Appl. No.: **12,034**

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Related U.S. Patent Documents

Reissue of:

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Issued: **Aug. 16, 1977**
Appl. No.: **658,181**
Filed: **Feb. 17, 1976**

[51] **Int. Cl.**[3] **B26D 1/58;** B23D 25/02

[52] **U.S. Cl.** **83/14;** 83/37; 83/329; 83/327; 83/174

[58] **Field of Search** 83/14, 37, 38, 174, 83/327, 329, 337, 315

[56] **References Cited**

U.S. PATENT DOCUMENTS

1,630,132	5/1927	Molins	83/315
1,801,169	4/1931	Molins et al.	83/327
1,846,942	2/1932	Bronander et al.	83/329
3,213,731	10/1965	Renard	83/329

FOREIGN PATENT DOCUMENTS

253666	4/1967	Austria	83/329

Primary Examiner—Donald R. Schran
Attorney, Agent, or Firm—Tilton, Fallon, Lungmus

[57] **ABSTRACT**

A method and apparatus for the transverse cutting of elongated material such as superposed web plies (for example, a convolutely wound log) wherein the material is uniformly advanced along an axial path and a cutting disc blade is moved through an orbit in planetary fashion which intersects the path and is skewed relative thereto while the disc blade is rotating in planes perpendicular to the path.

5 Claims, 33 Drawing Figures

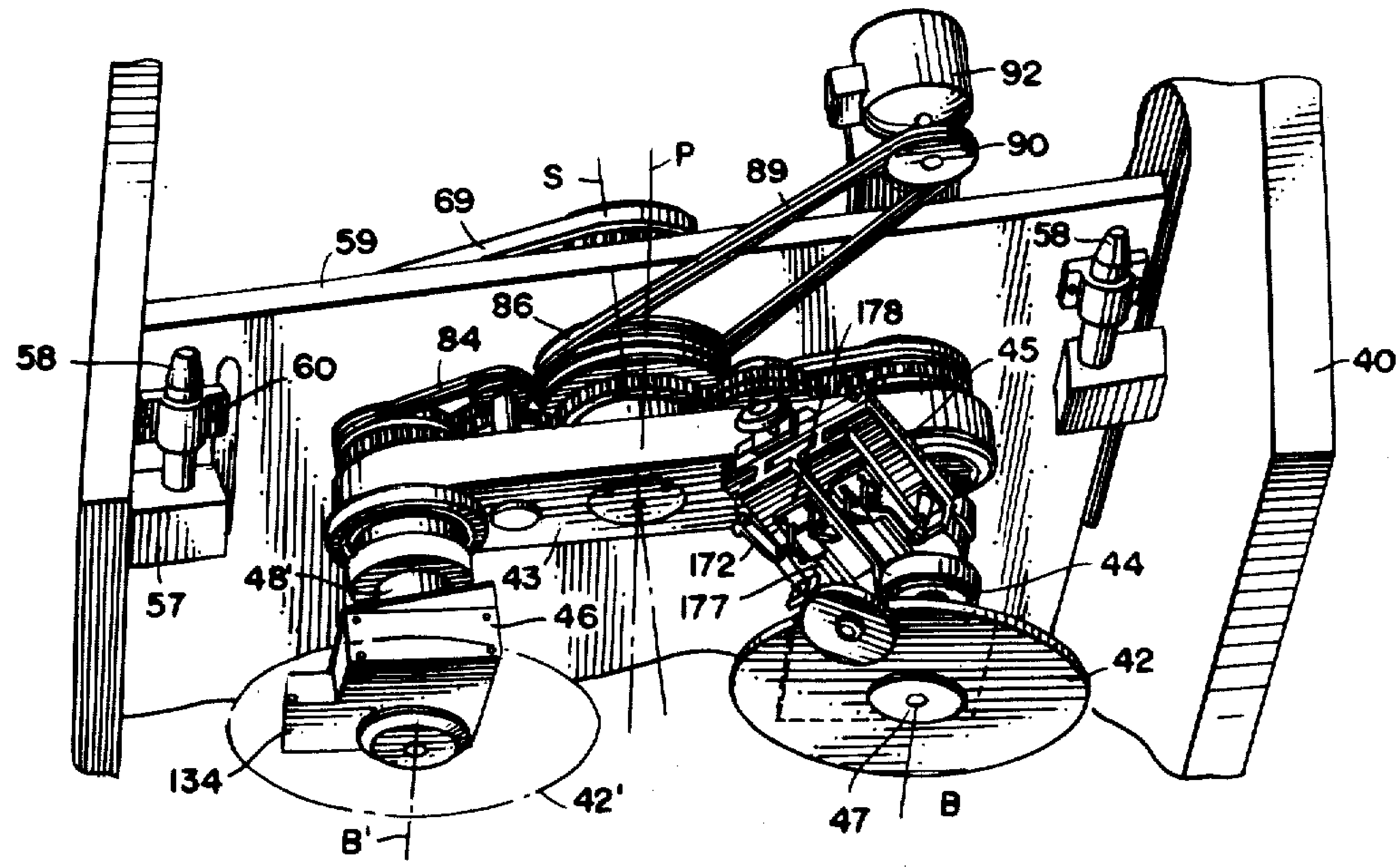


FIG. 1

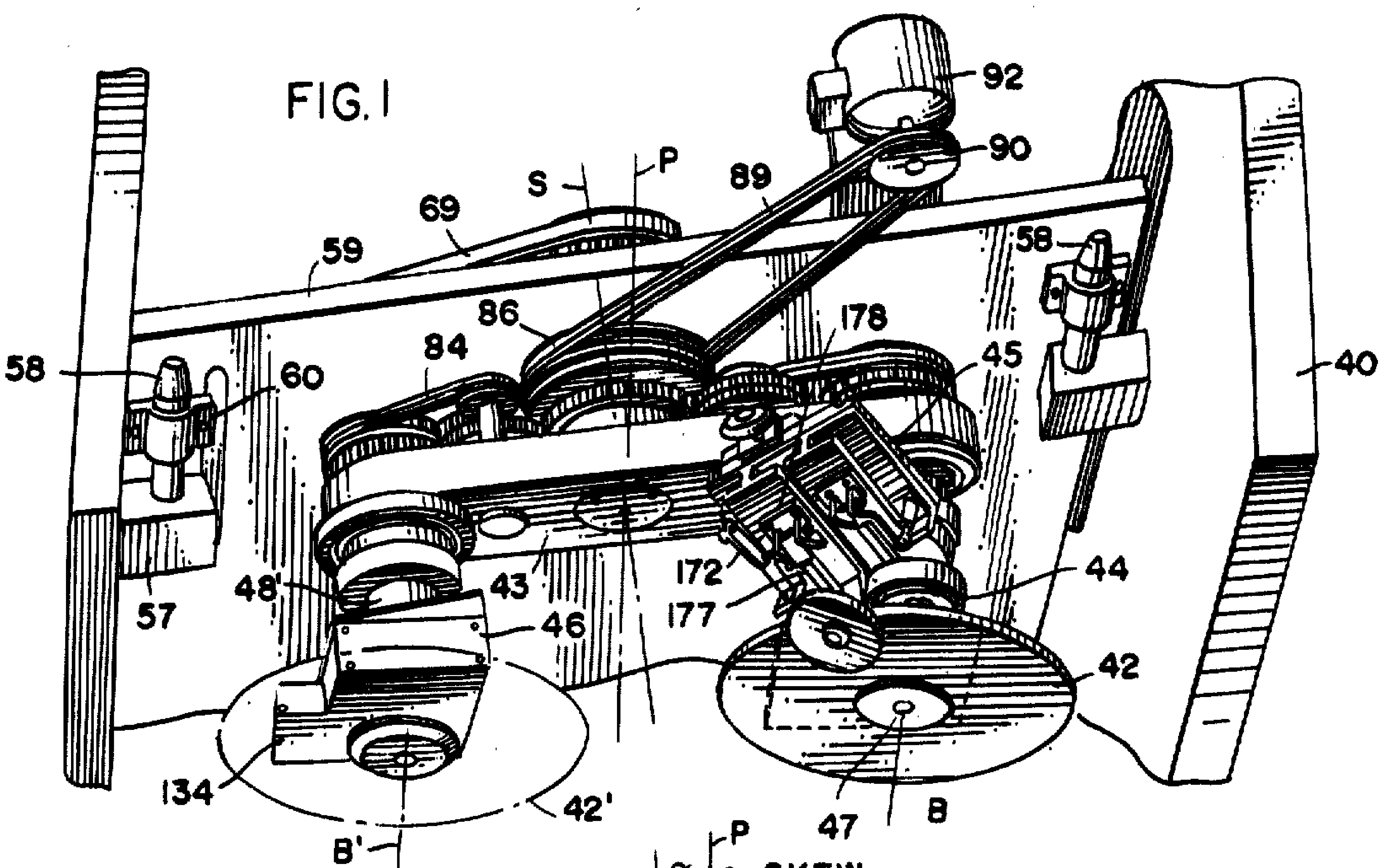


FIG. 2

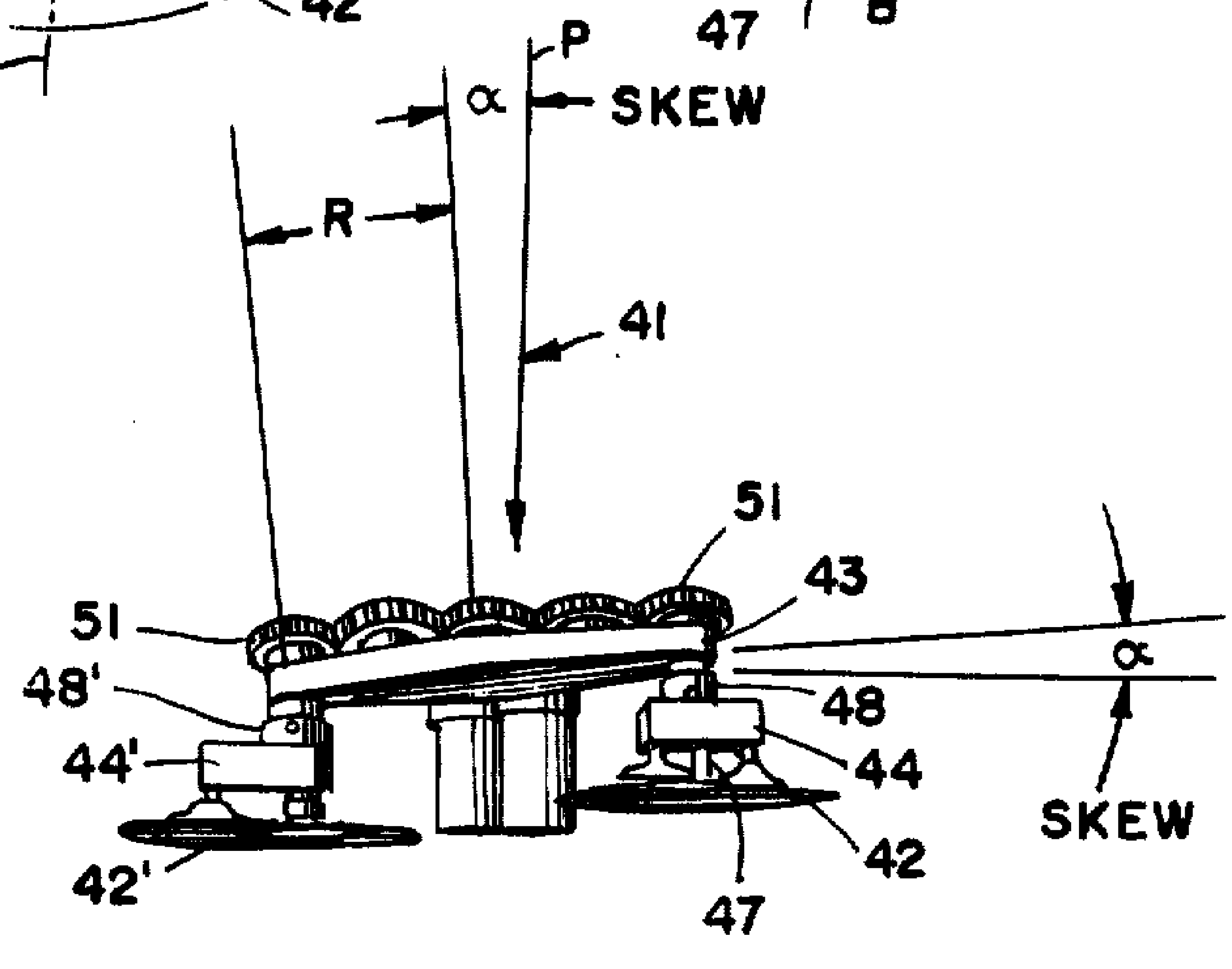


FIG. 3

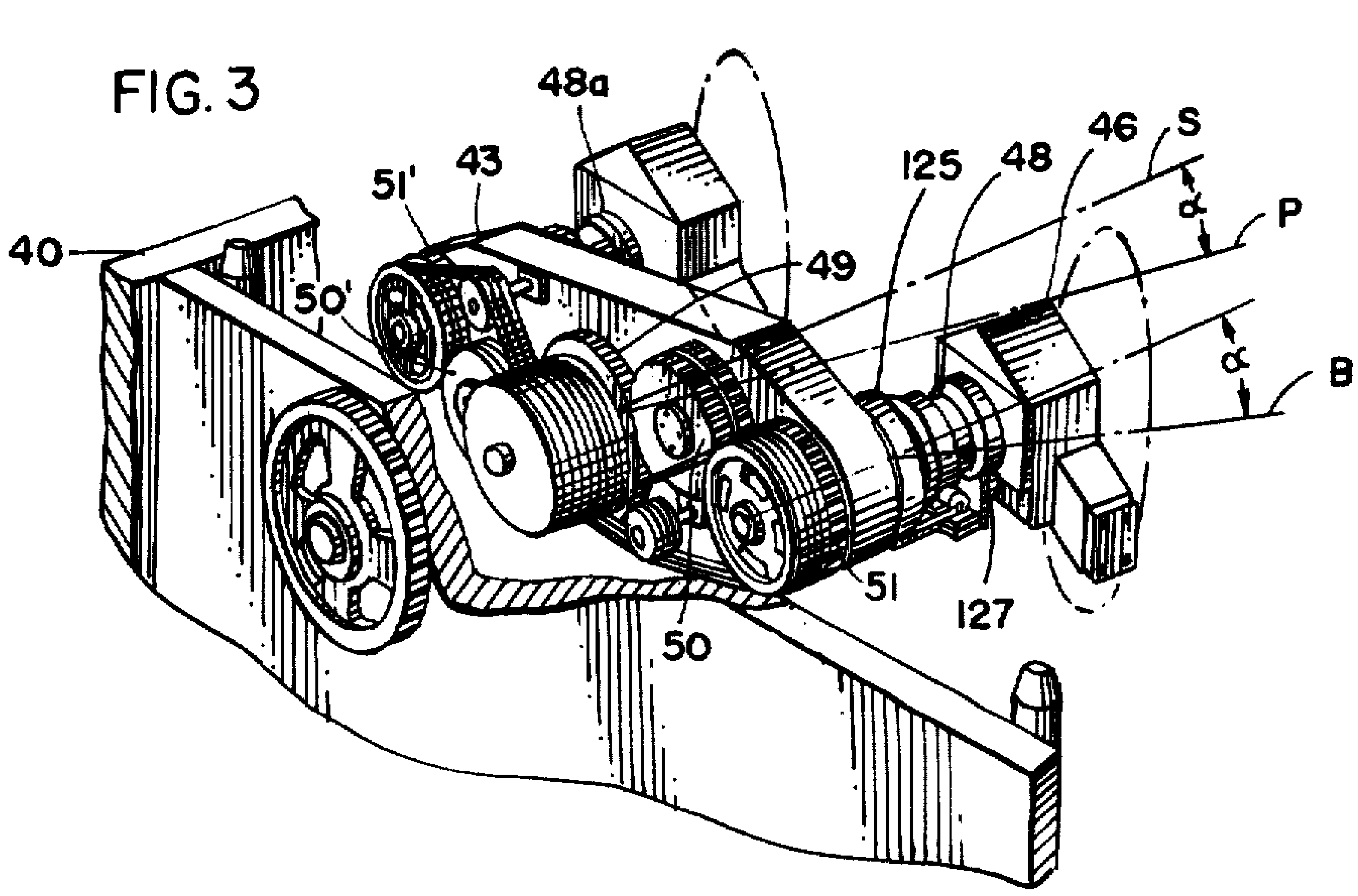
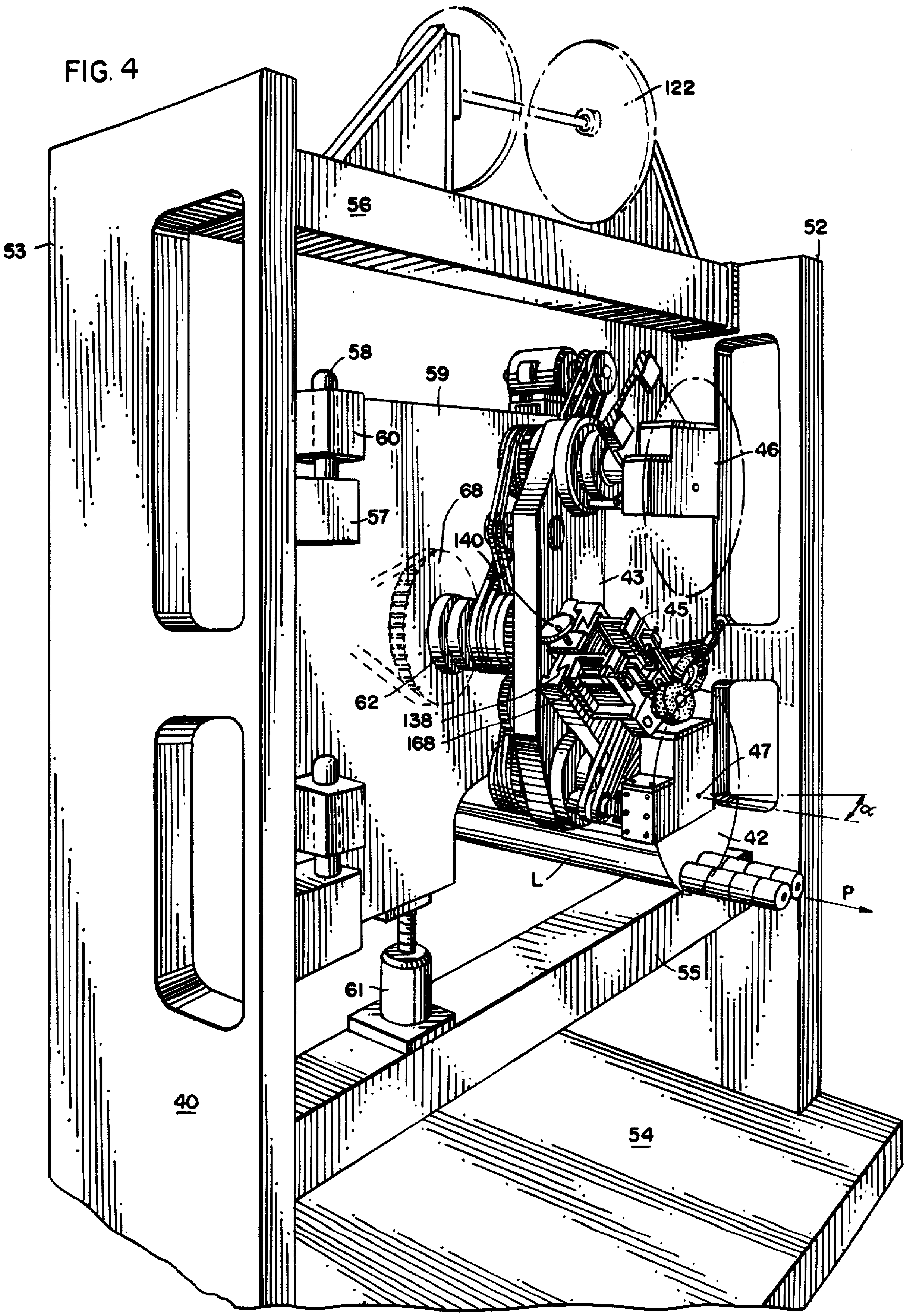


FIG. 4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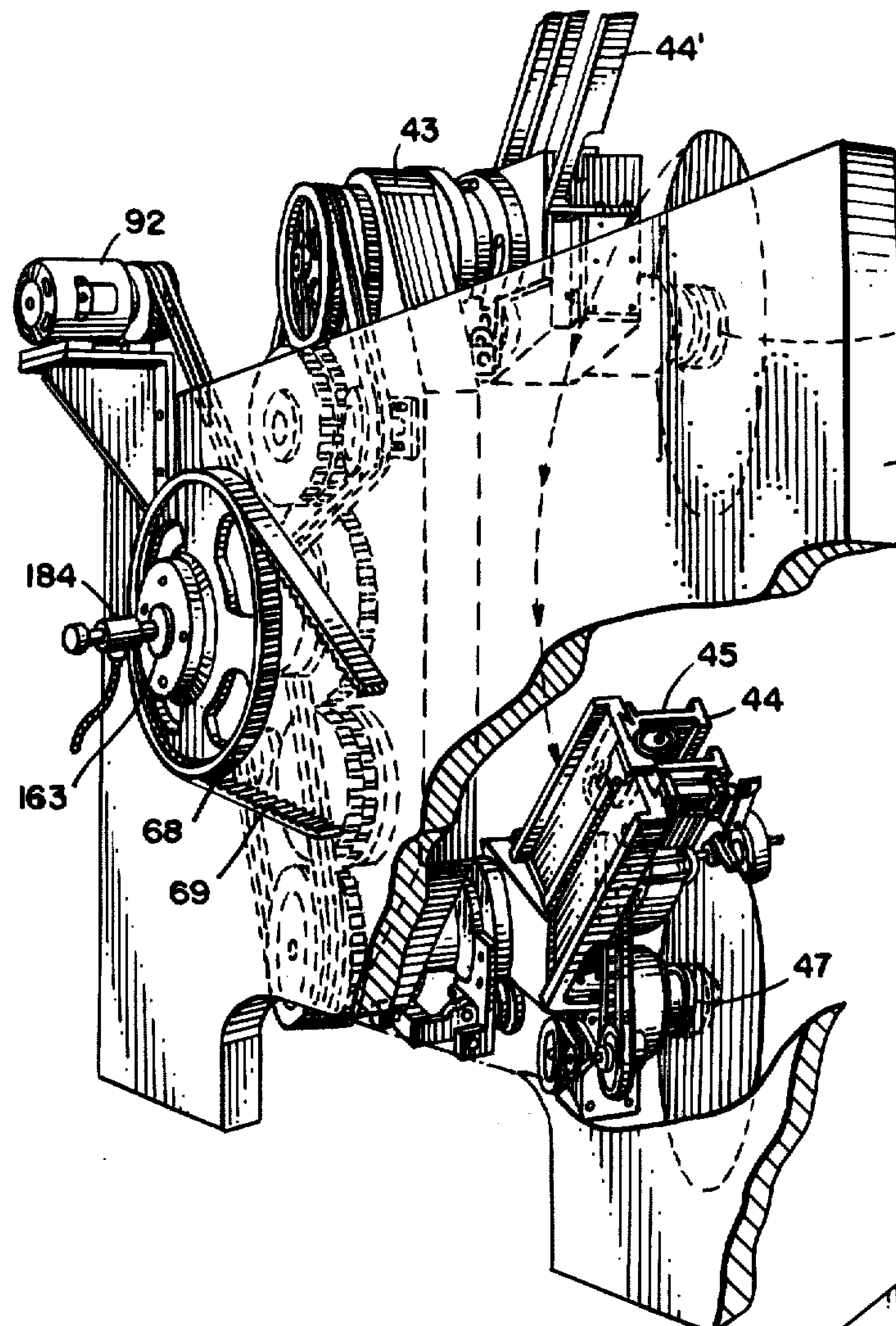

FIG. 5

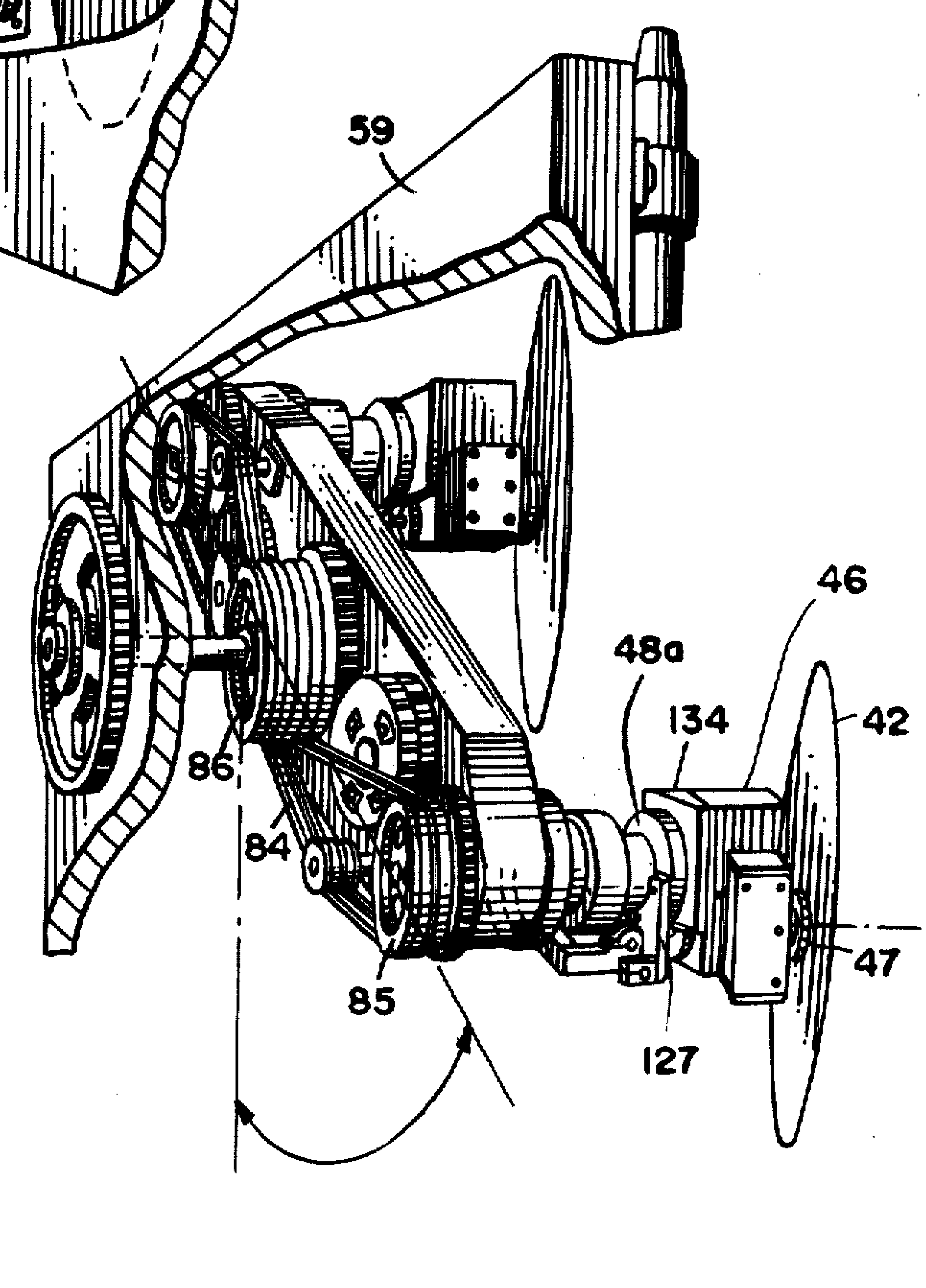

FIG. 6

FIG. 7

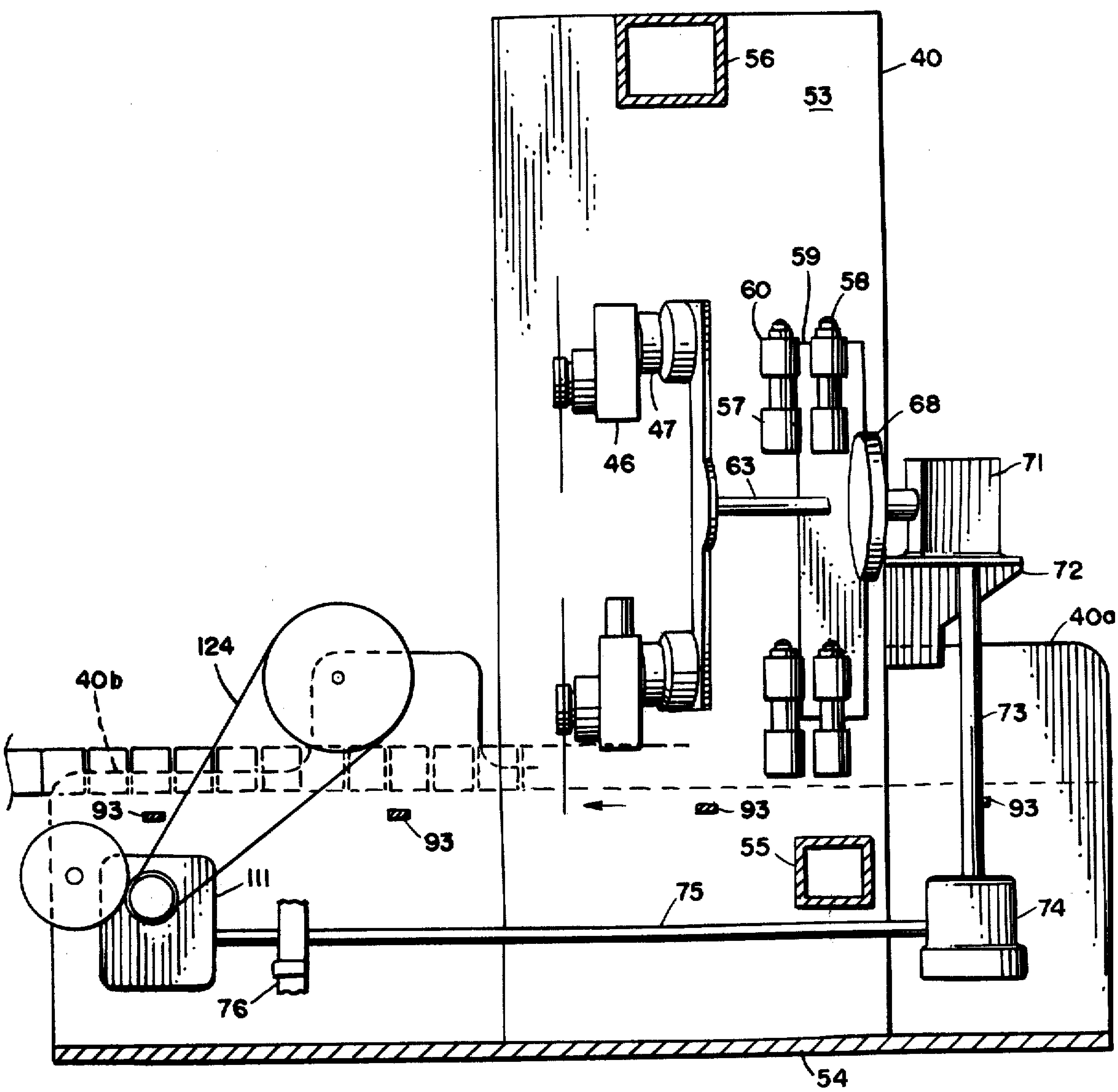
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FIG. 9

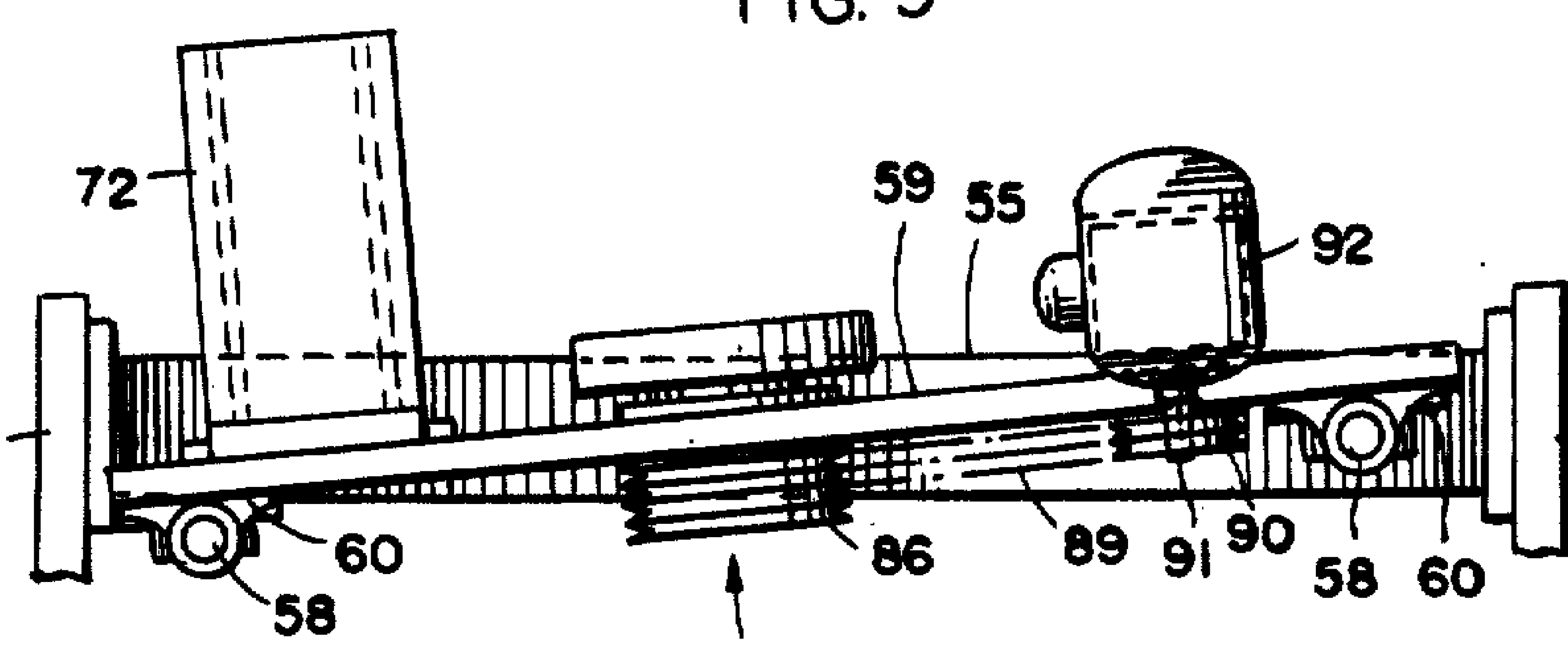
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FIG. 8

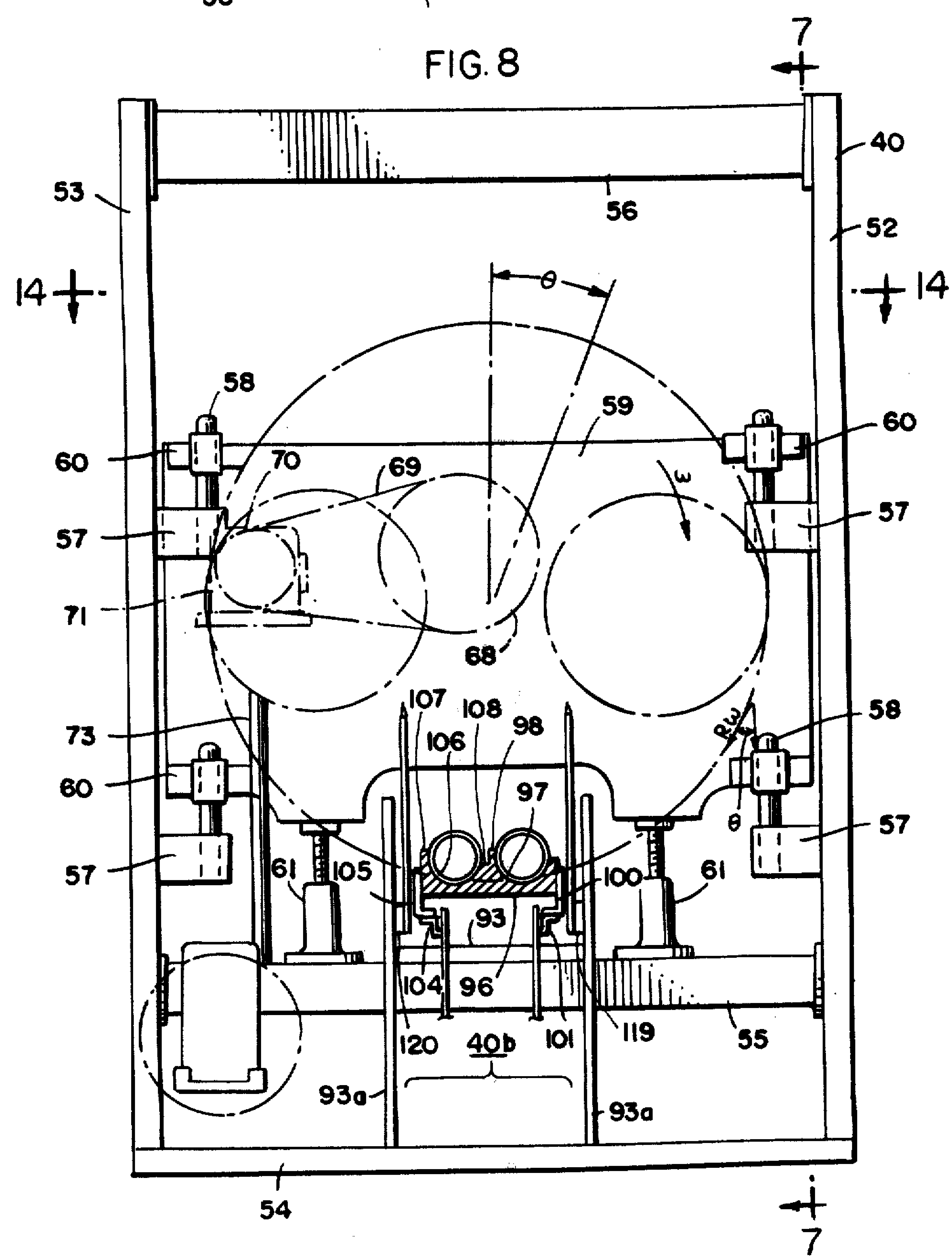
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FIG. 10

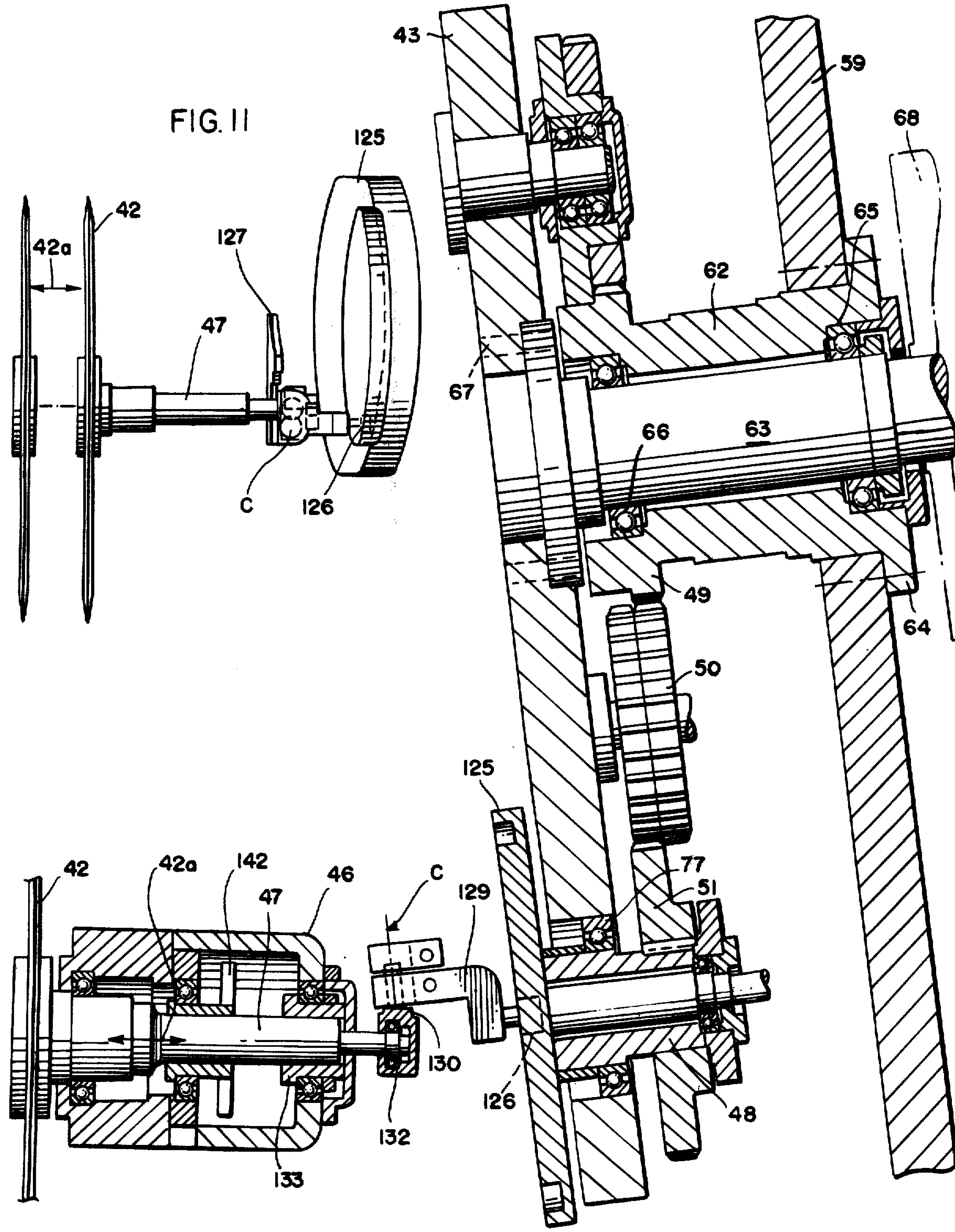
FIG. 11
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42a
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FIG. 12

FIG. 13

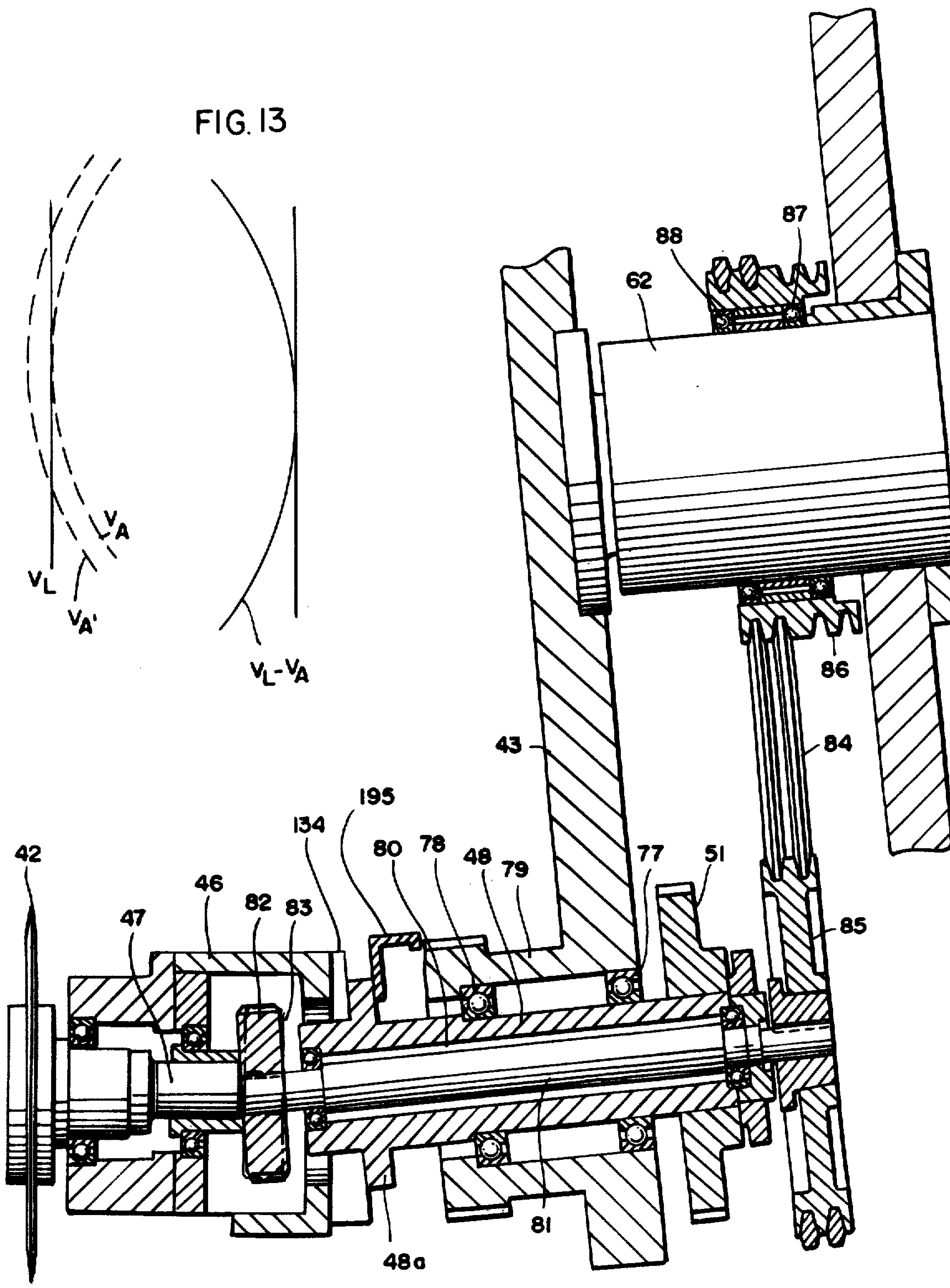
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FIG. 14

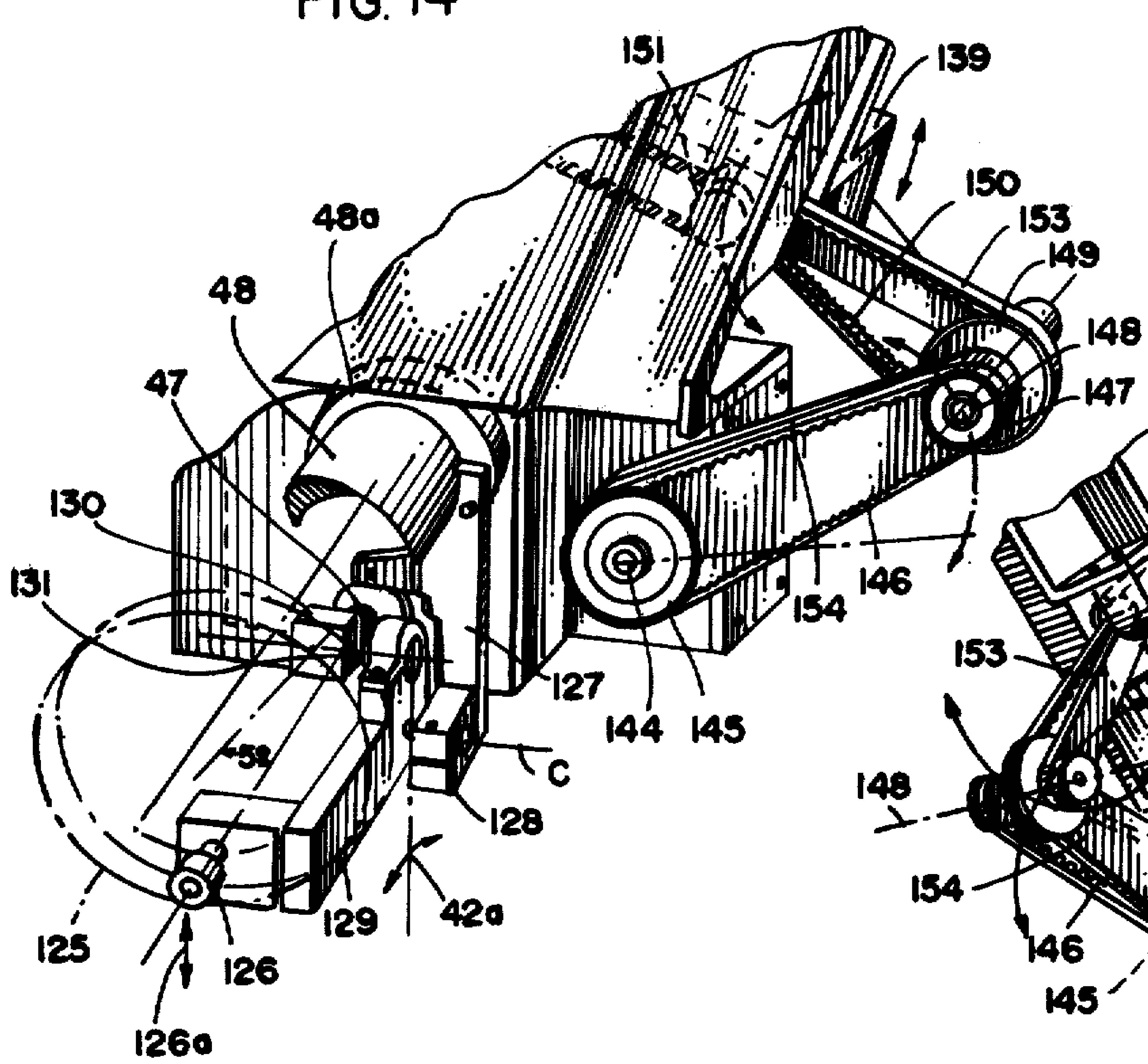
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FIG. 15

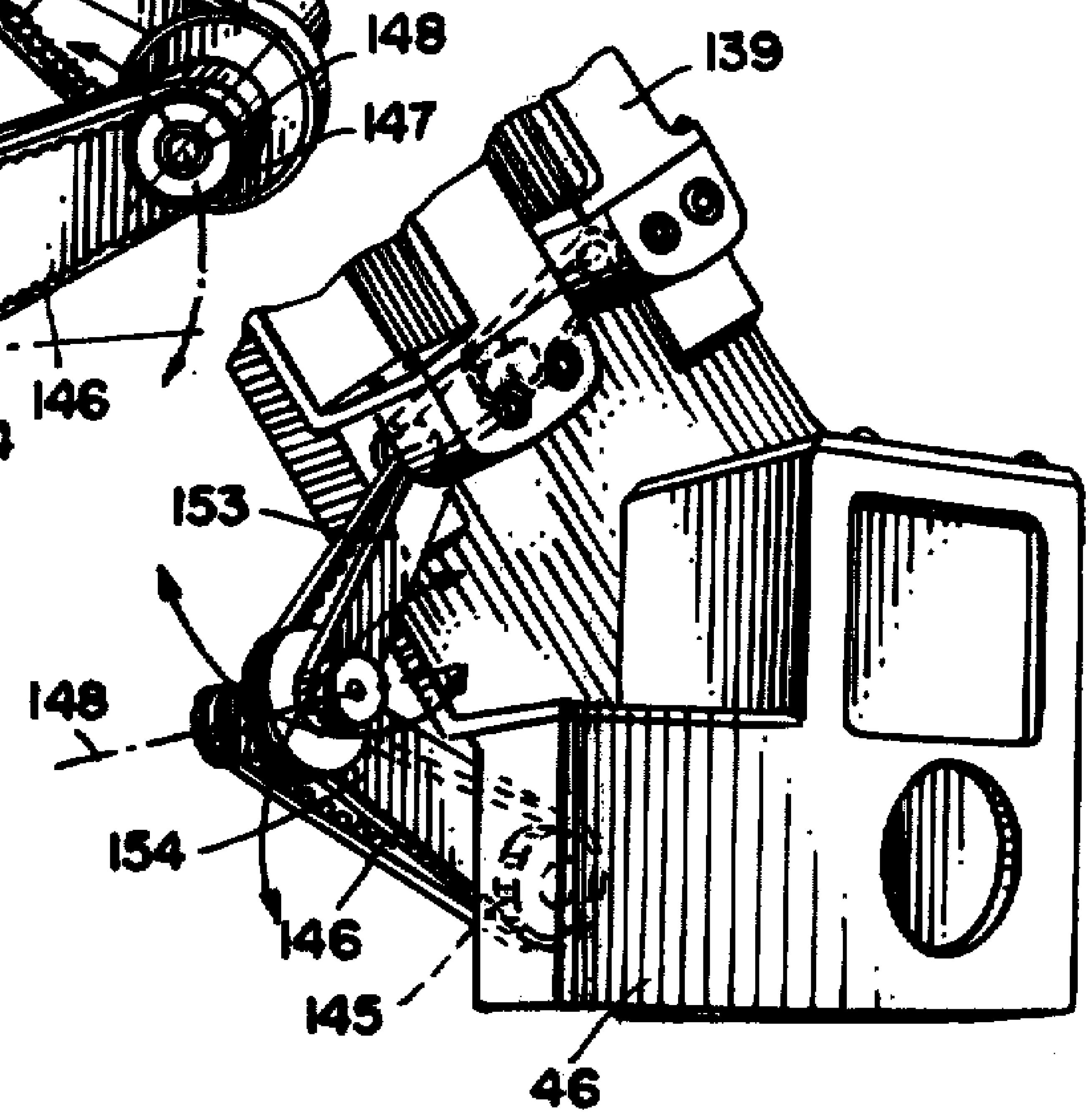
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FIG. 16

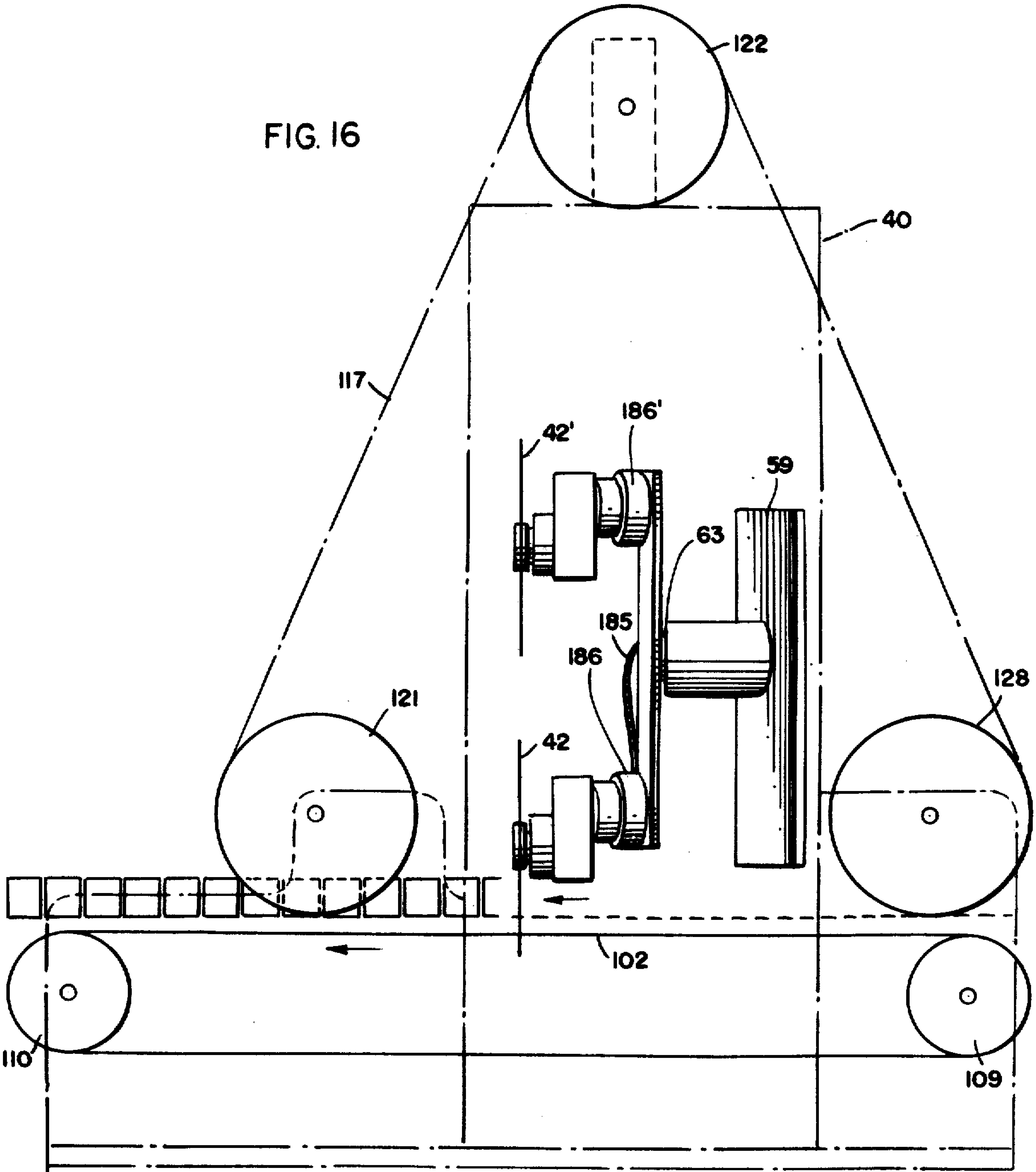


FIG. 17

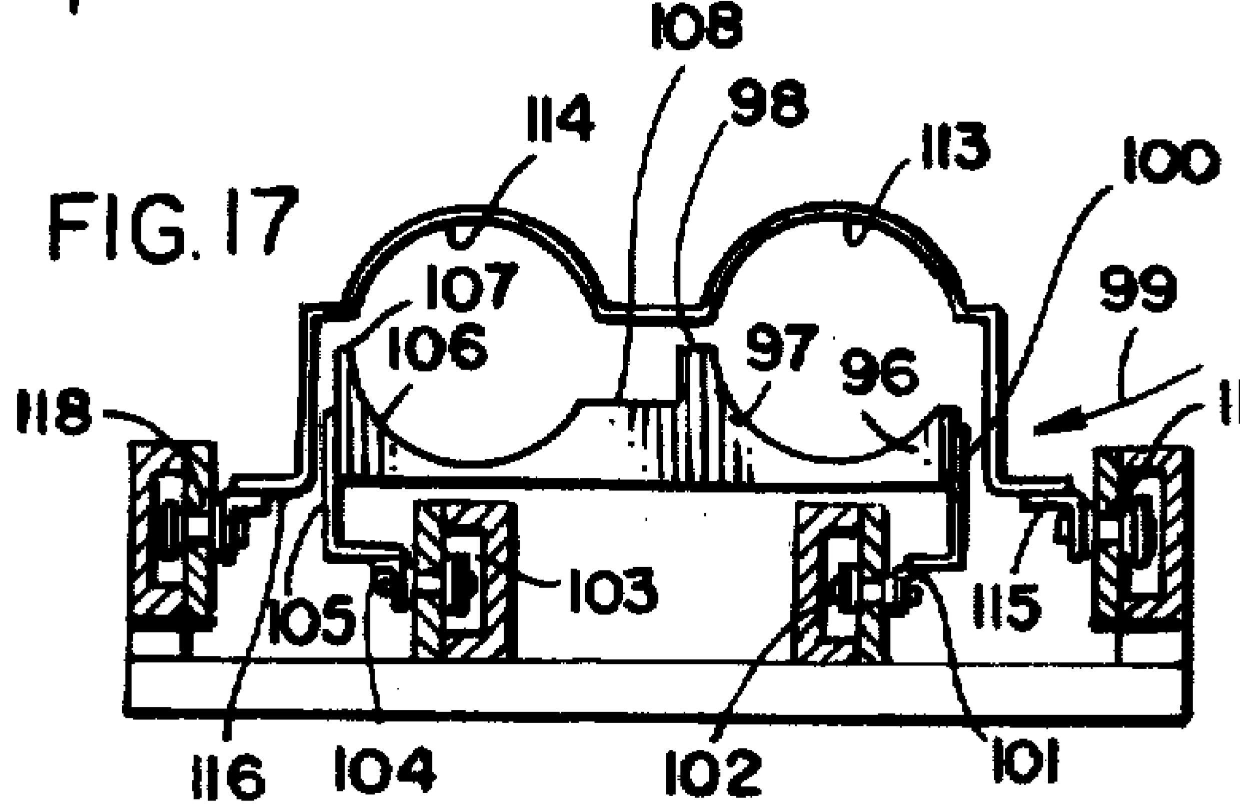


FIG. 18

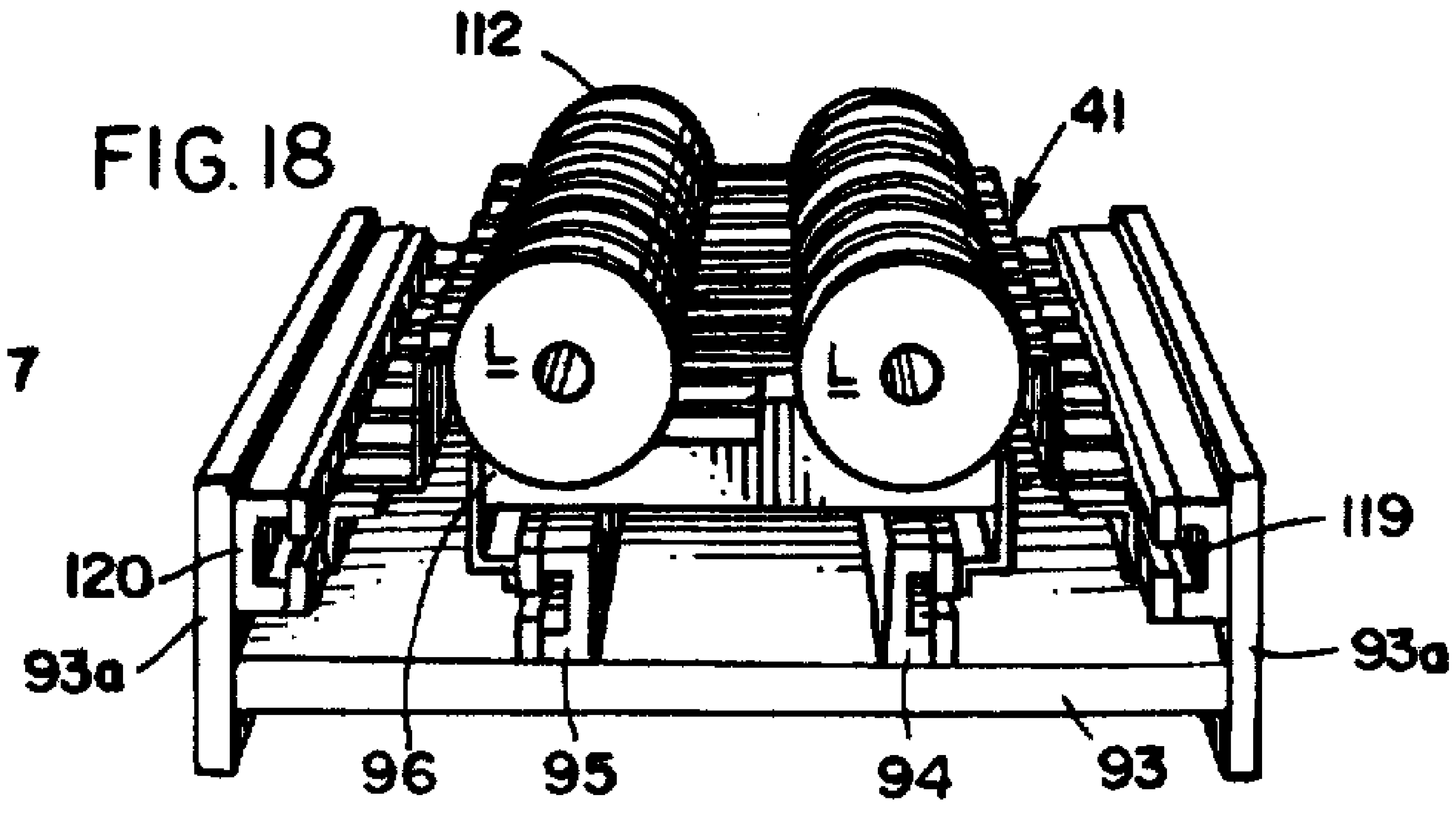
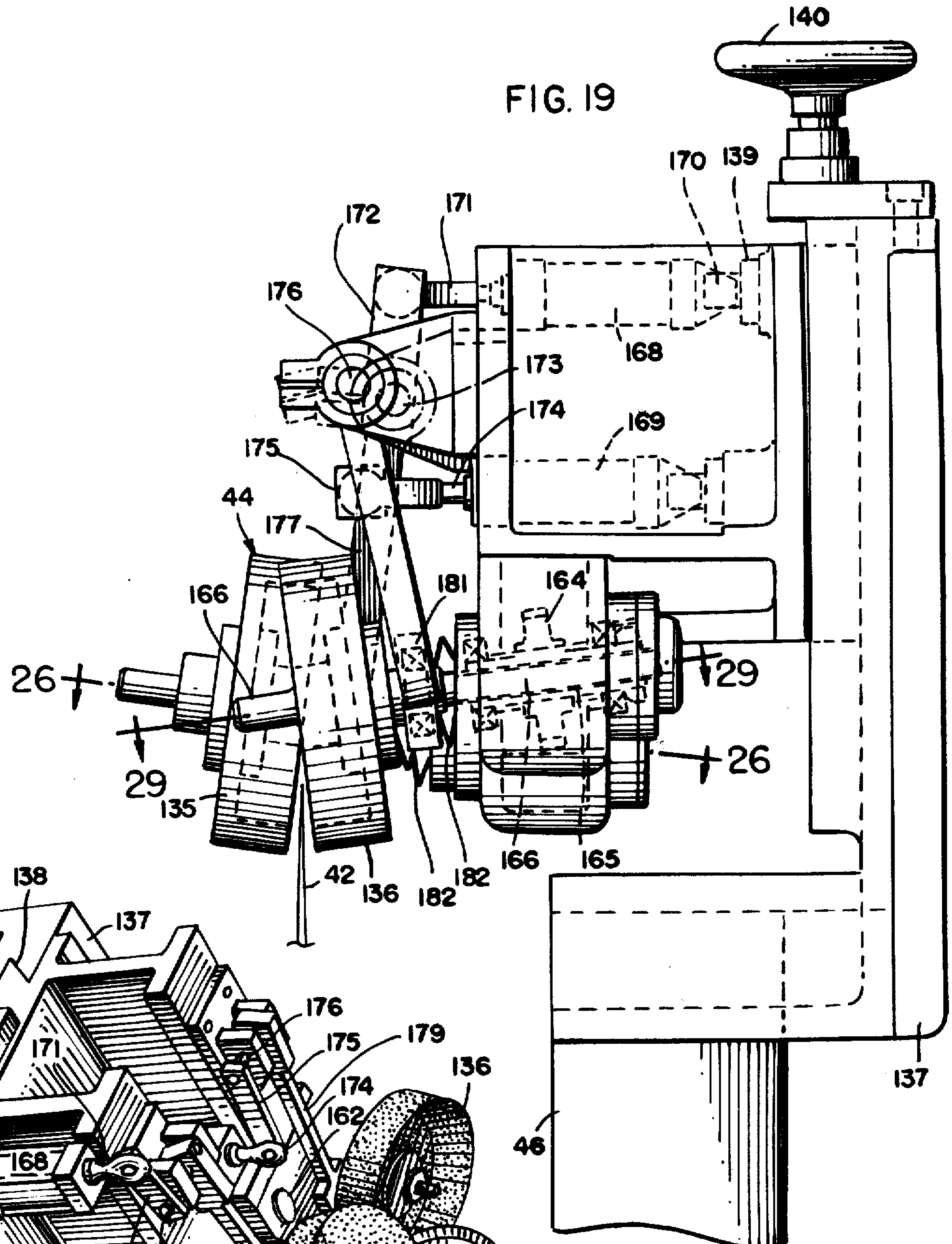


FIG. 19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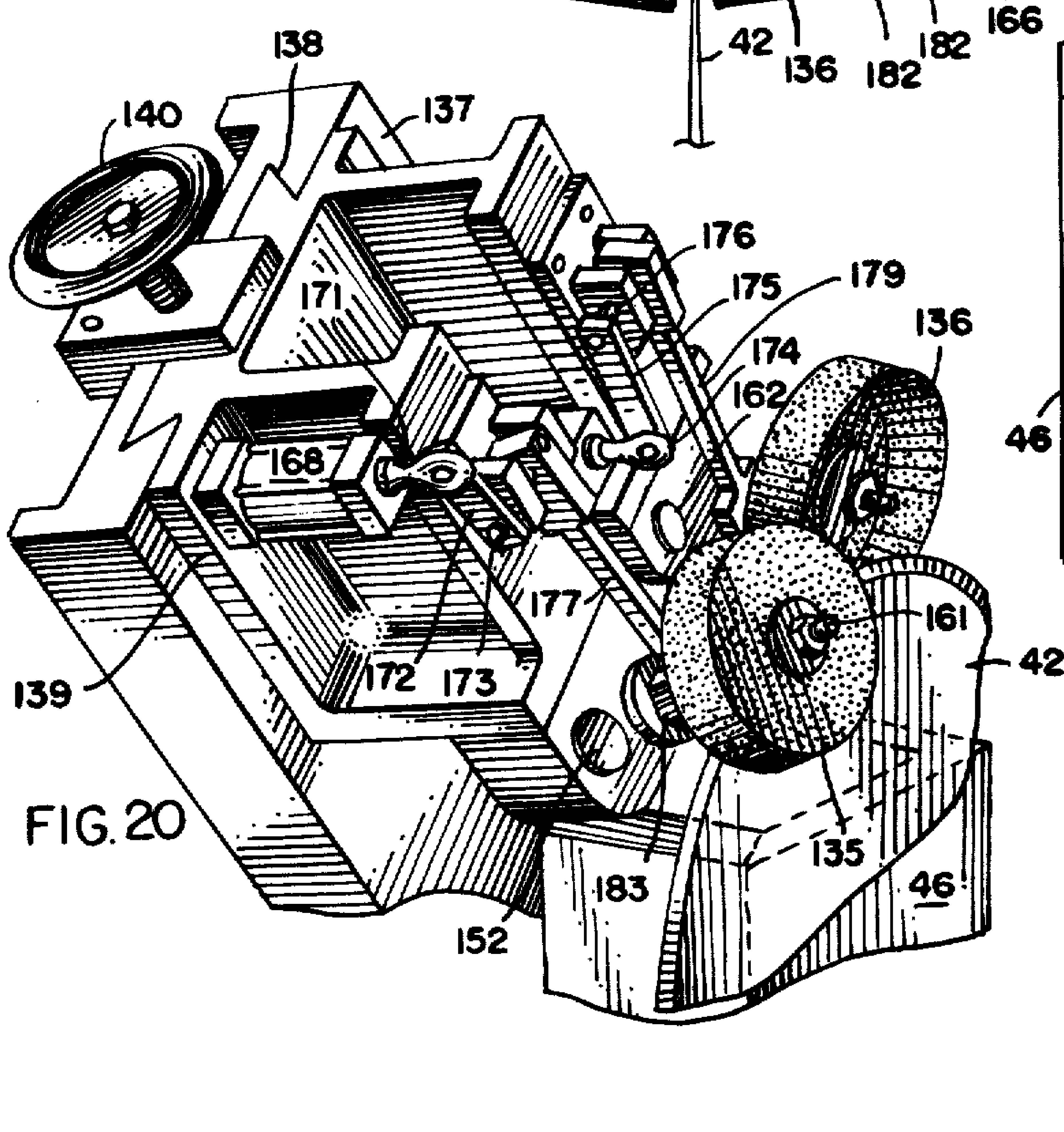
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FIG. 20

FIG. 22

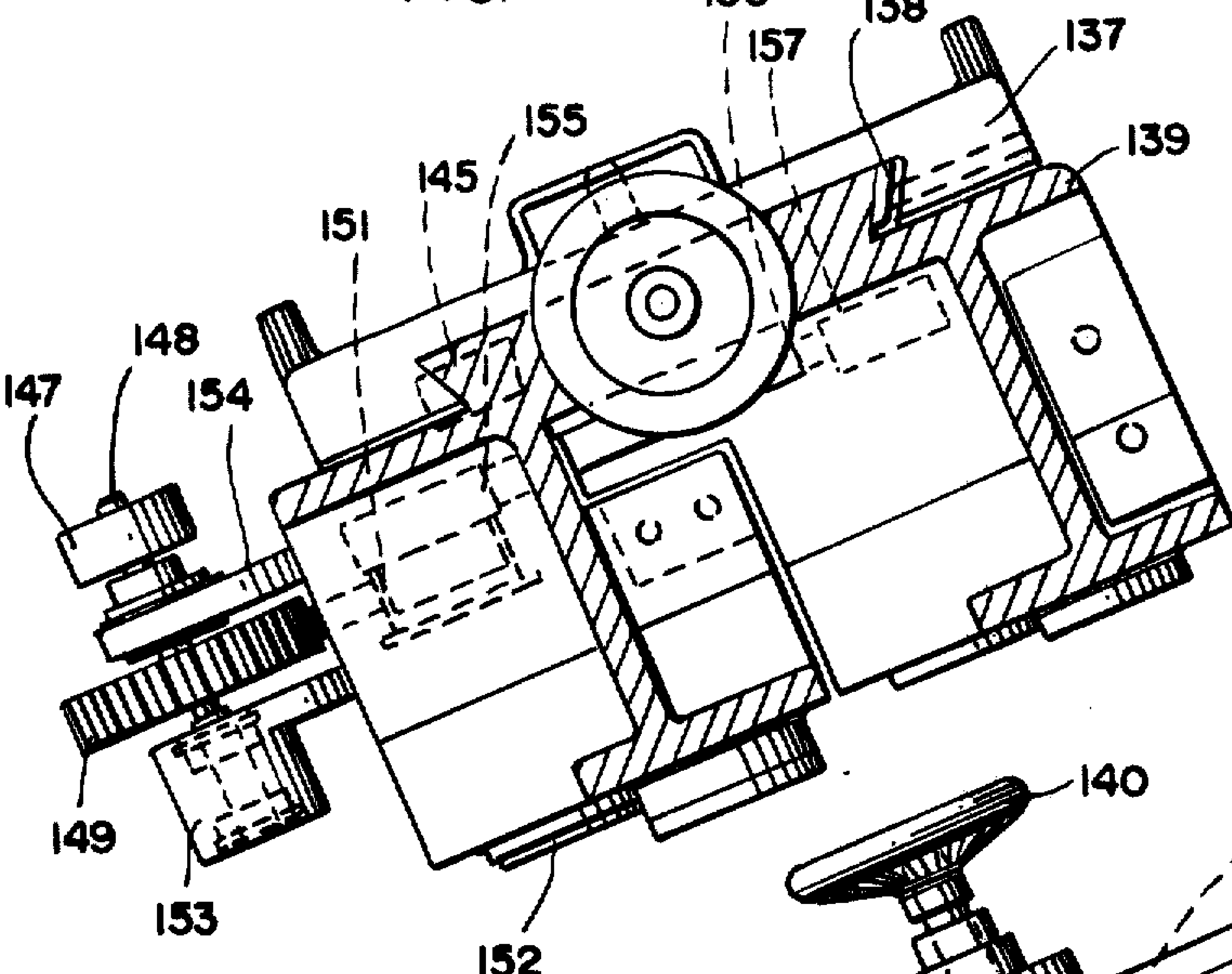
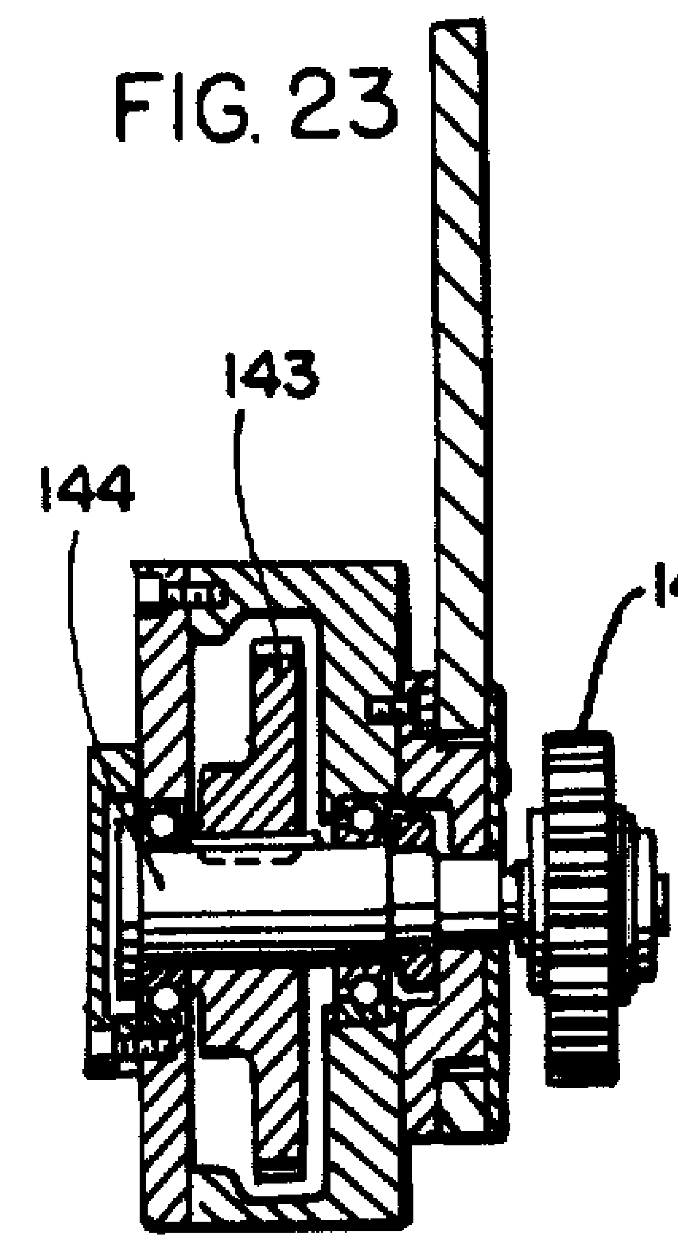


FIG. 23



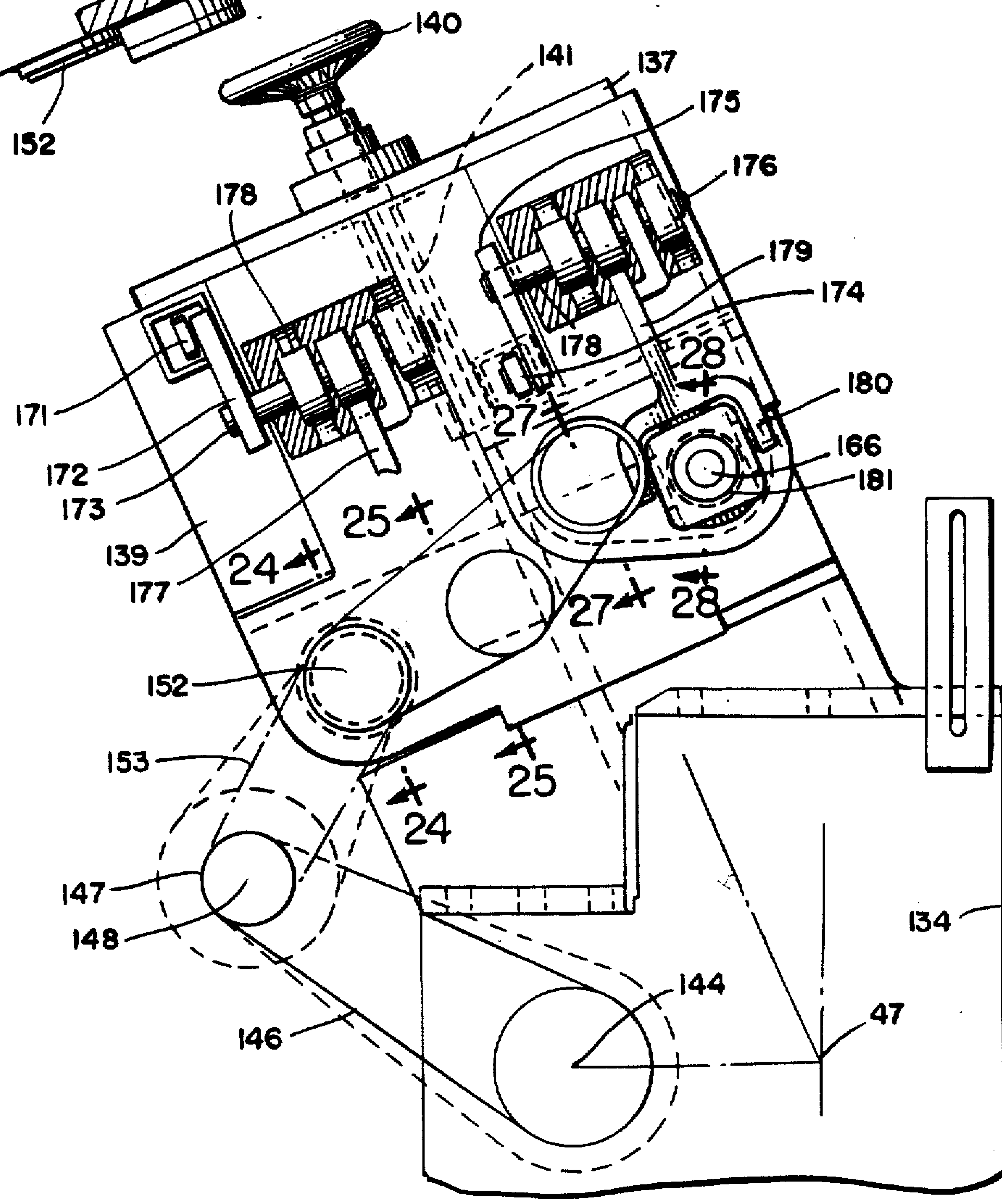


FIG. 21

FIG. 24

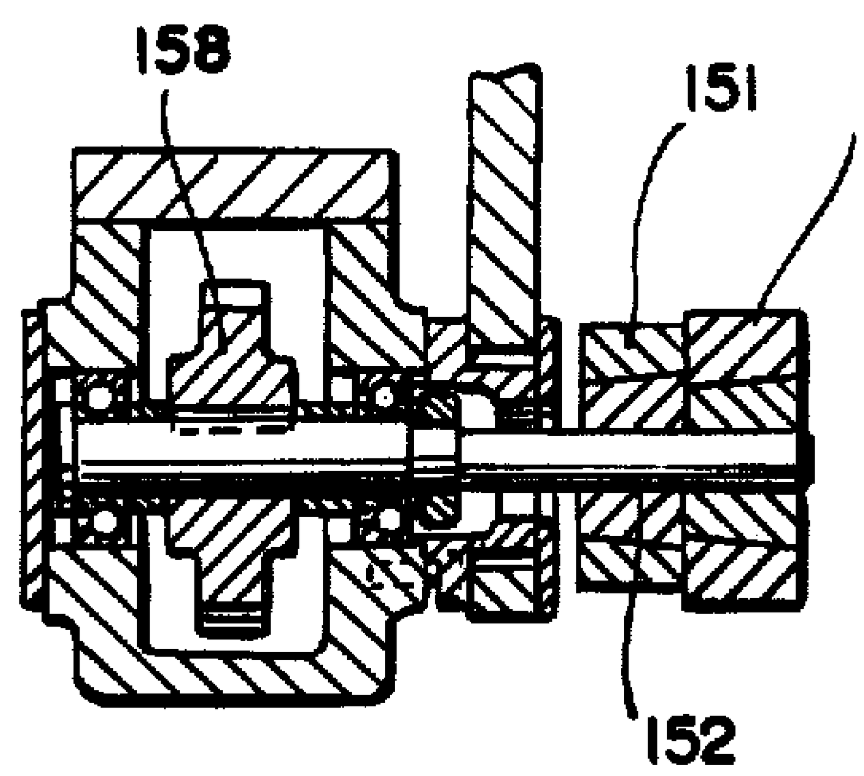


FIG. 27

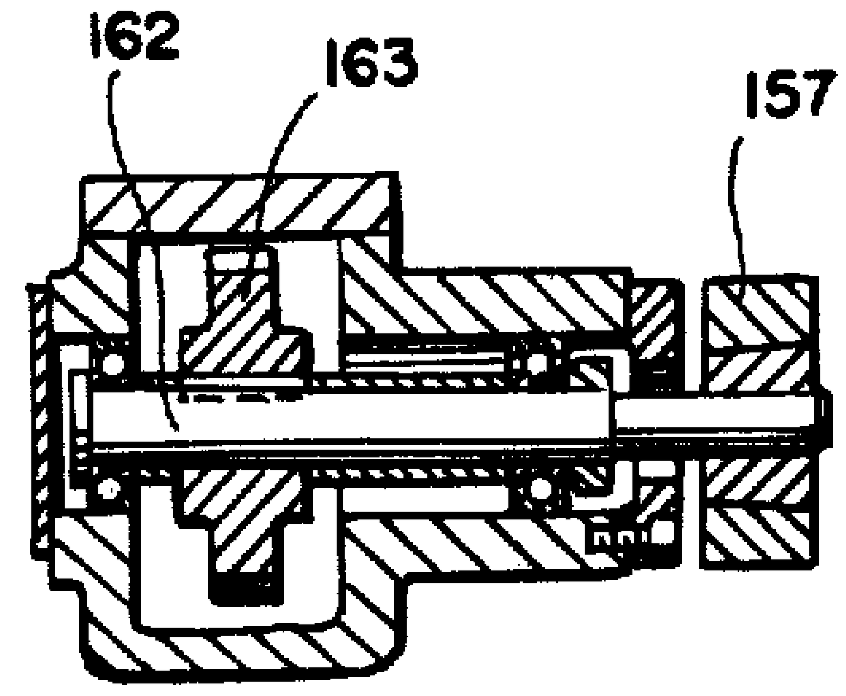


FIG. 25

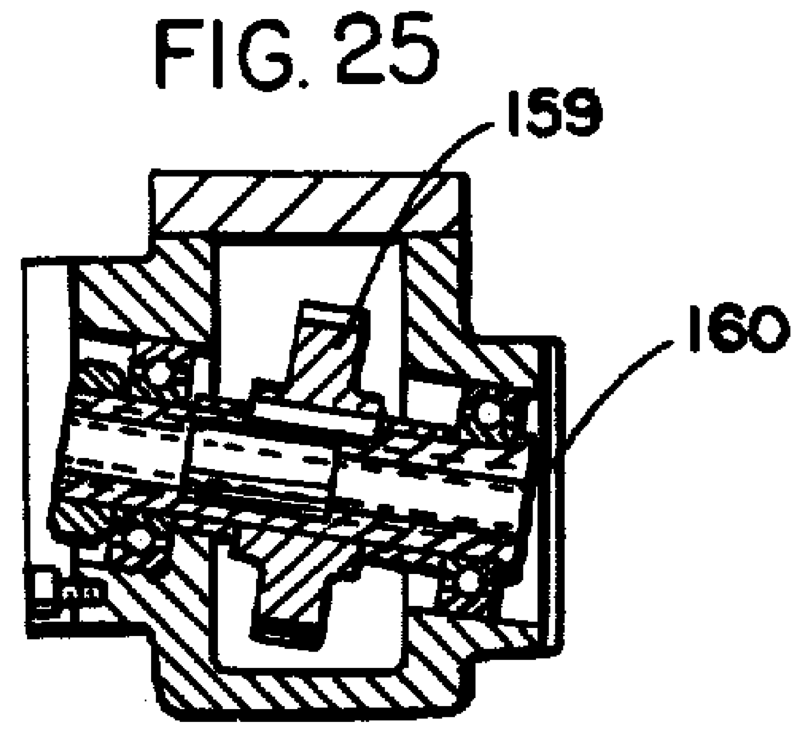


FIG. 28

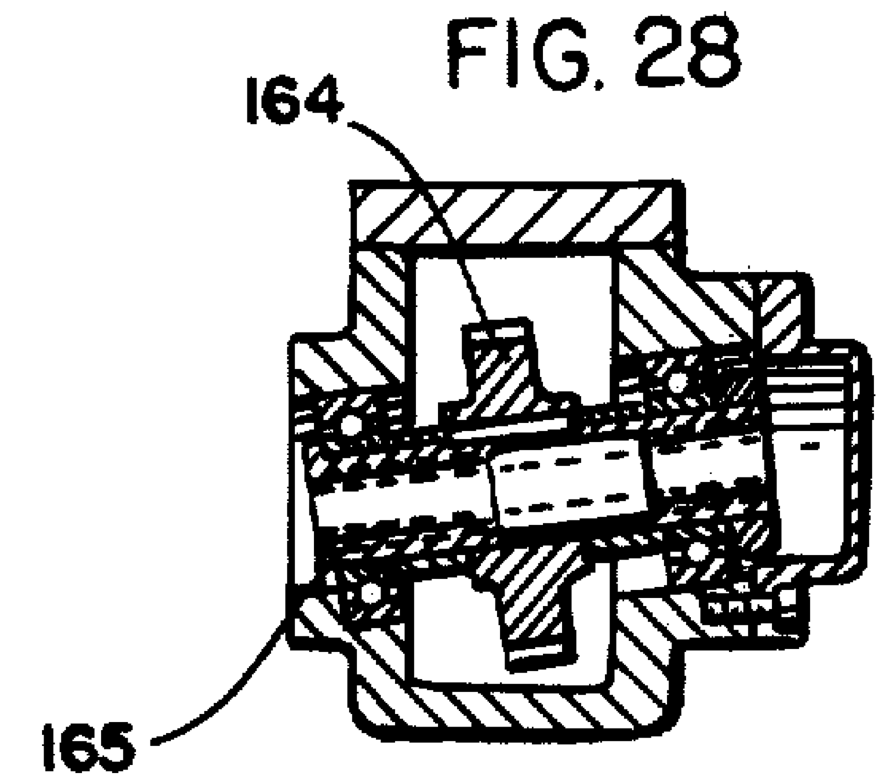


FIG. 26

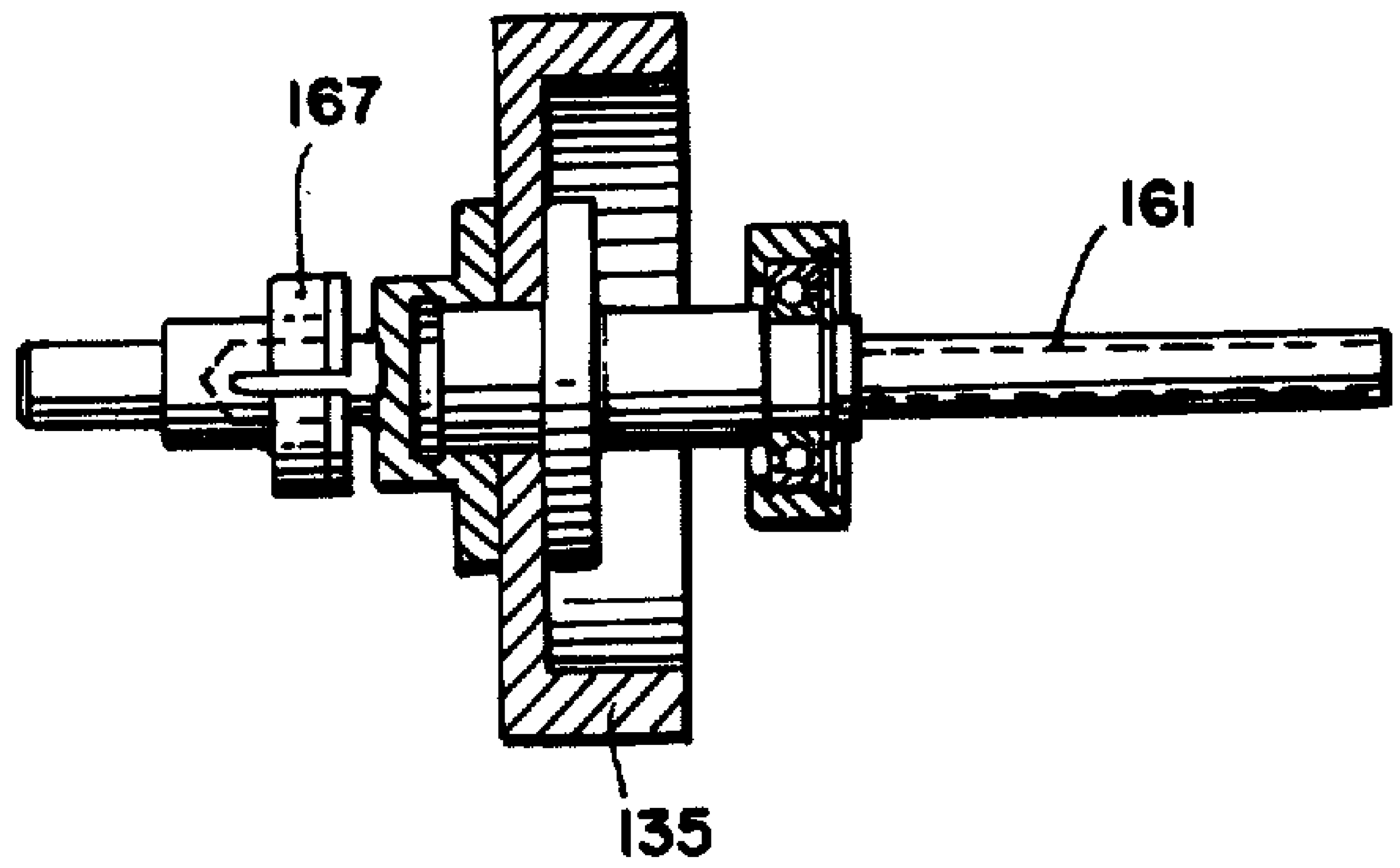


FIG. 29

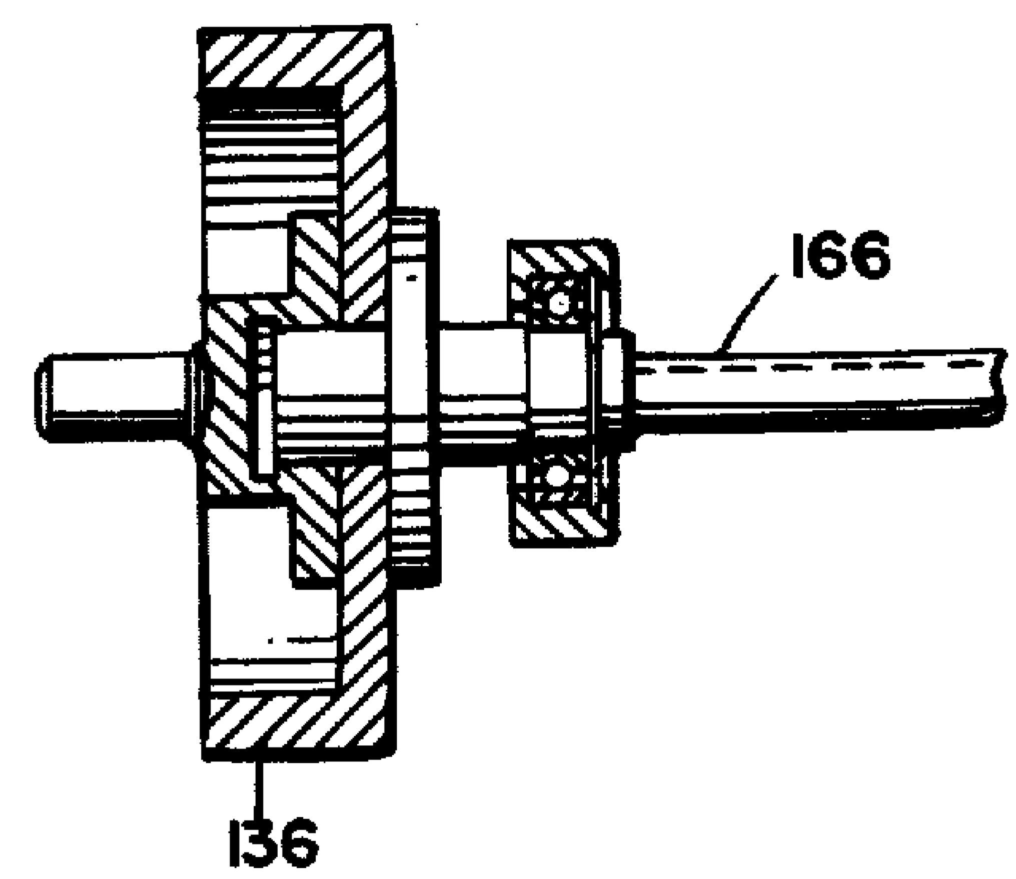


FIG. 30

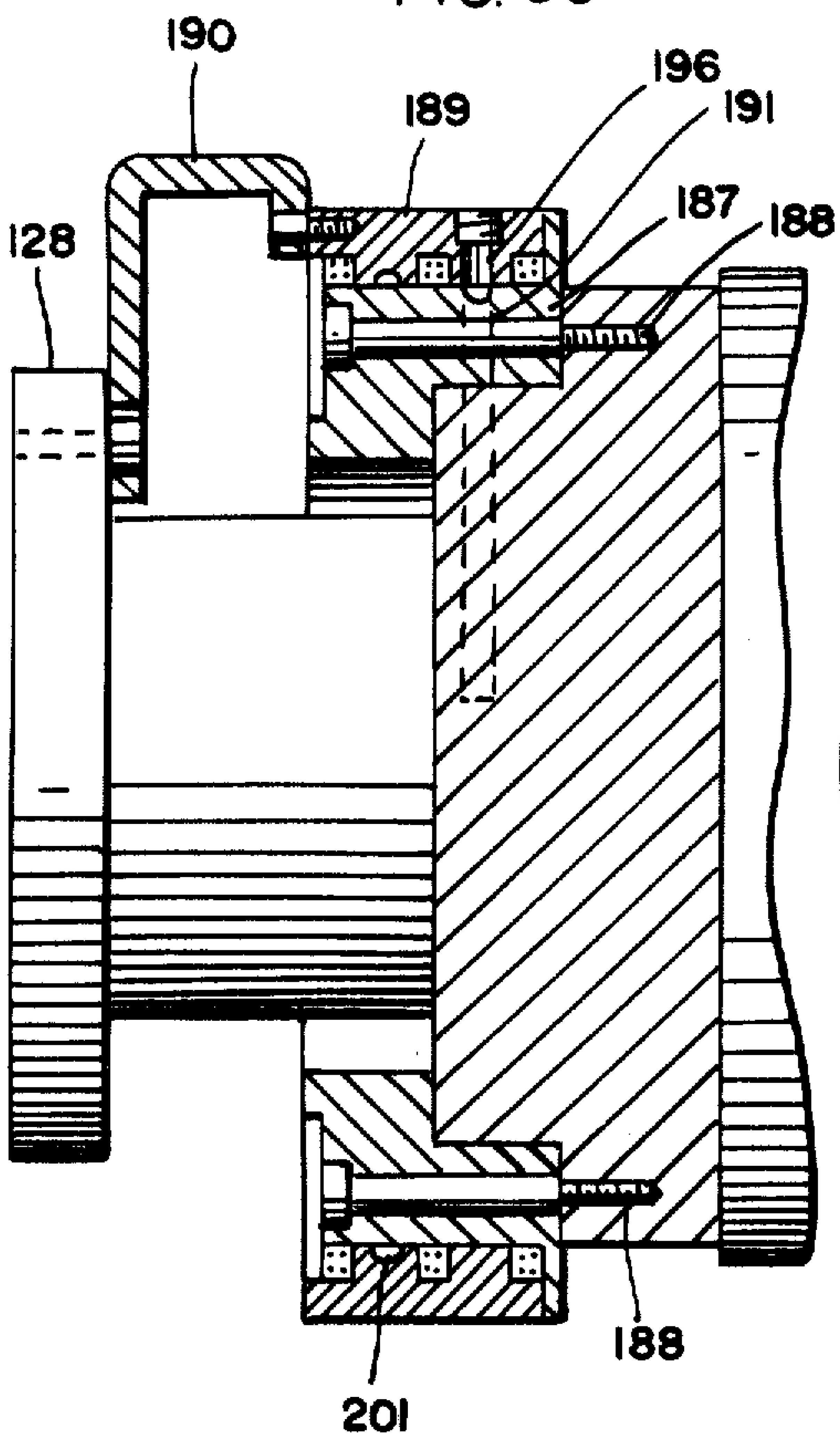


FIG. 31

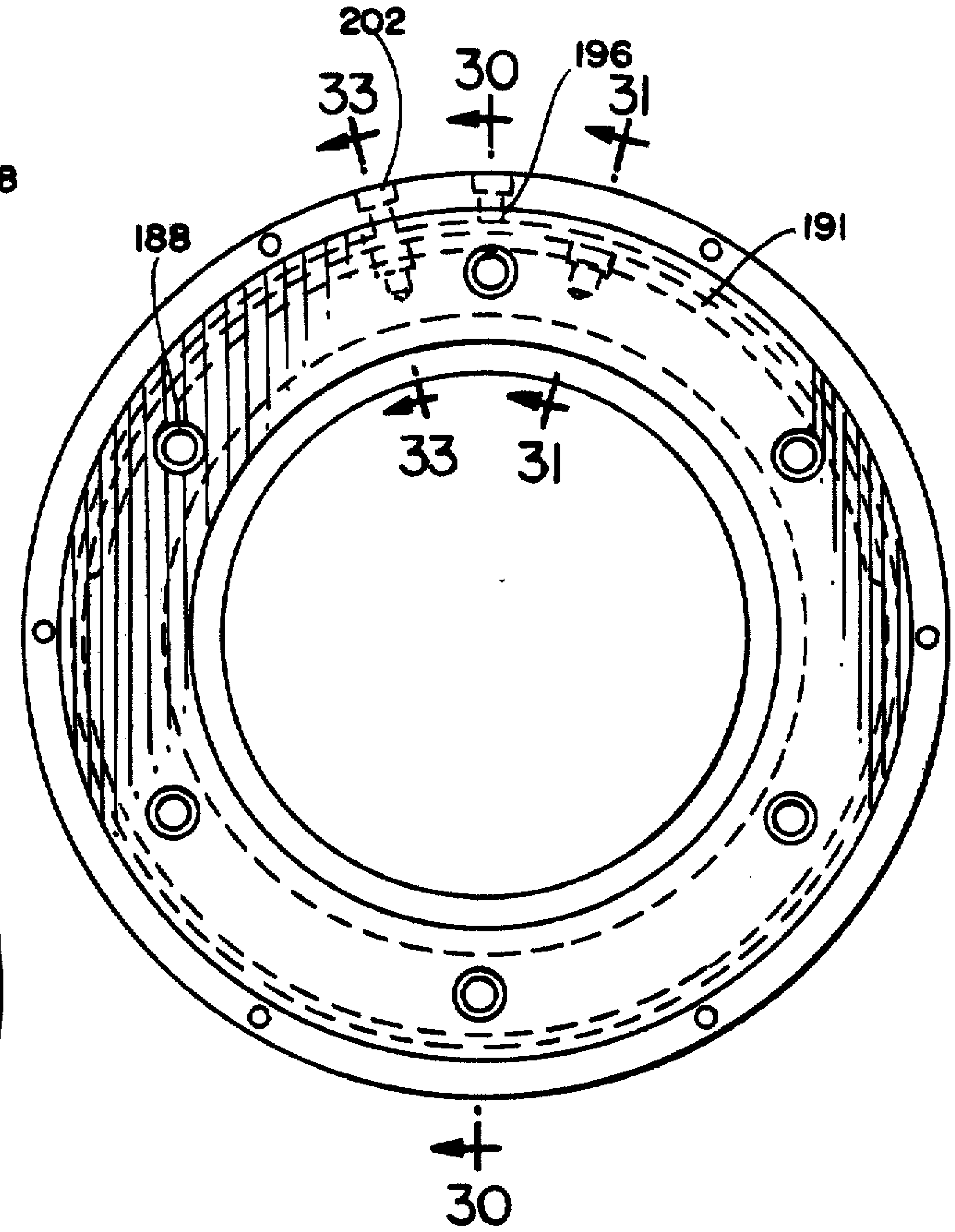


FIG. 32

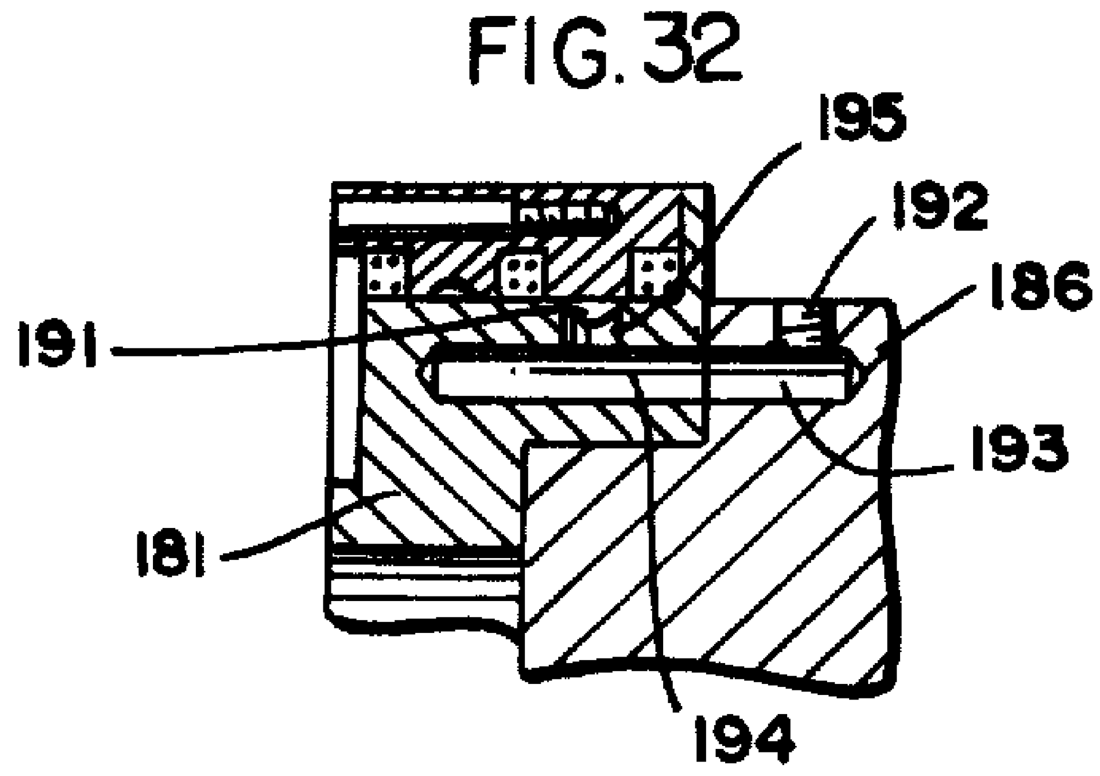
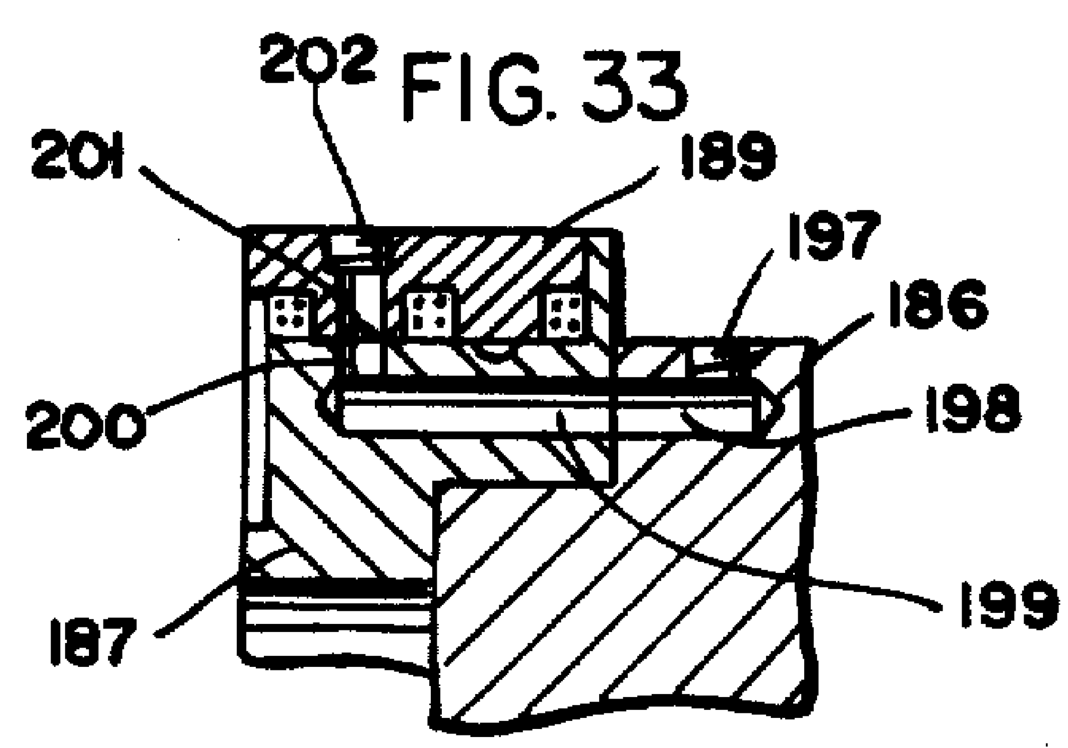


FIG. 33



METHOD FOR TRANSVERSE CUTTING

Matter enclosed in heavy brackets [ ] appears in the original patent but forms no part of this reissue specification; matter printed in italics indicates the additions made by reissue.

BACKGROUND OF THE INVENTION

This invention relates to a method and apparatus for transversely cutting elongated material such as superposed web plies and, more particularly, to the use of a unique orbiting saw for cutting through elongated web material such as toilet tissue and toweling logs or stacked superposed plies such as folded tissues and towels.

In the production of web rolls such as toilet paper and toweling, a jumbo sized roll (often 4 to 6 feet in diameter and upwards of 6 feet in length) was unwound, transversely perforated, longitudinally slit and rewound to provide the well known retail size rolls. This was done continuously and automatically in "rewinders" such as can be seen in U.S. Pat. No. 2,769,600. In the late 1950's and early 1960's, it was realized that problems could arise because of the longitudinal slitting, i.e., slicing the relatively wide web into 4½ inch widths (a square of toilet paper conventionally being 4½ by 4½ inches). If the wide sheet were imperfect in spots (sometimes characterized as "fish-eyes" and referring to large holes in the paper) the entire rewinder had to be shut down because of discontinuity in the winding. Therefore, the longitudinal slitting during the rewinding was eliminated and the rewound, transversely perforated roll or log transferred from the rewinder to a log saw. Because automation was important to maintain high production rates and low economy, the log saws were integrated into the rewinder line.

Among the first log saws were the so-called "Gilbertville" saws employing a circular disc having a peripheral cutting edge which oscillated back and forth across the path of the wound log emanating from the rewinder. The Gilbertville saw can be seen in U.S. Pat. Nos. 2,752,999 and 2,766,566. As the rewinder speeds increased, a new variety of log saw appeared and gained popularity, this being the "orbiting" saw—see Renard U.S. Pat. No. 3,213,721.

As rewinder speeds became greater and greater—thereby requiring more logs to be sawed per unit of time, various improvements were made in the orbiting saws such as the variable-speed orbit saw of U.S. Pat. Nos. 3,213,734 and 3,292,470. In an effort to keep up with the high volume output of rewinders, the art even went to transversely cutting four logs simultaneously, i.e., four lanes of logs being advanced through the sawing station—see U.S. Pat. No. 3,905,260. However, this involved expensive and complicated indexing machinery, i.e., apparatus for stepping the logs or stacks into the sawing station for each cut.

With the increased demand for paper rolls and stacks and the ability to deliver the same at high volume from rewinders, it became clear that the existing orbital saws could not maintain the pace. Of course, additional saws could be placed in parallel but this created material handling problems, occupied valuable space, and was generally inefficient.

SUMMARY OF INVENTION

The above severe time problem has been overcome through the use of a unique orbiting saw wherein the logs or stacks are advanced uniformly and continuously along an axial path. At the same time, the disc blade is moved through an orbit which is skewed relative to the patch of advance but the cutting disc blade is mounted on the skewed member so as to rotate always in planes perpendicular to the path of advance, thereby developing the necessary square cut. More particularly, the support for the disc blade, although orbiting, remains in a fixed attitude with respect to the ground, i.e., it moves in "planetary" fashion about the orbit axis. Here, I employ the term planetary in its engineering sense—and not as a synomym for orbit. Additionally, the cutting disc blade (or blades) can be mounted for slight axial movement (in a direction parallel to the path of advance) so as to compensate for or "match" the component of velocity of the blade in the direction of the path with the velocity of a log or stack moving in the path. Novel and advantageous log moving means are employed to coact with the orbiting blades to assure high speed, reliable operation. Still further, disc blade sharpening devices are mounted on the planetary member so as to maintain high speed, efficient operation as well as avoid the need for stopping th orbiting for sharpening. Other details and facets of the invention, as well as objects and advantages thereof may be seen in the ensuing specification.

DETAILED DESCRIPTION

The invention is described in conjunction with an illustrative embodiment in the accompanying drawing, in which

FIG. **1** is a fragmentary perspective view of the cutting portion of the inventive apparatus;

FIG. **2** is a schematic top plan view of the apparatus of FIG. **1** and which is appropriately labeled to show the skew principle;

FIG. **3** is another schematic perspective view similar to FIG. **1** but taken from the rear, i.e., from the upstream side of the cutting blades;

FIG.**4** is a fragmentary perspective view of the apparatus, differing from FIG. **1** in showing more of the frame and showing the cutting blades aligned vertically;

FIG. **5** is another fragmentary perspective view also viewed from the rear but with the skew member vertical as in FIG. **4**;

FIG. **6** is still another fragmentary rear perspective view, this view showing the skew arm at about 45°;

FIG. **7** is a fragmentary longitudinal sectional view of the inventive log saw as would be seen along the line **7**—**7** of FIG. **8**;

FIG. **8** is a fragmentary end elevational view of the apparatus of FIG. **7**—as seen from the front or downstream side of the apparatus;

FIG. **9** is a fragmentary top plan view of the plate employed for mounting the skew arm and blade drive;

FIG. **10** is a fragmentary top plan view of the skew arm, the mounting and drive therefor and a portion of the planetary housing;

FIG. **11** is a fragmentary side elevational view of the cam and associated mechanism for velocity compensation;

FIG. **12** is a view similar to FIG. **10** but showing the drive for the disc blade;

FIG. 13 is a chart illustrating the velocity compensation achieved according to the mechanism of FIG. 11;

FIGS. 14 and 15 are fragmentary perspective views (from rear and front, respectively) of the velocity compensation mechanism and drive for the sharpening means;

FIG. 16 is a side elevational view similar to FIG. 7 but showing details of the log advancing mechanism;

FIG. 17 is an end elevational view of a portion of the log conveyor;

FIG. 18 is a perspective view of a fragment of the log advancing mechanism;

FIG. 19 is a fragmentary side elevational view of a portion of the planetary housing and featuring the sharpening means for the cutting blades;

FIG. 20 is a fragmentary perspective view of the apparatus portion seen in FIG. 19;

FIG. 21 is a front elevational view of the sharpening means of FIG. 19;

FIG. 22 is a top plan view of the sharpening means of FIG. 21;

FIG. 23 is a fragmentary side elevational view of a portion of the drive for the sharpening means of FIGS. 19-22;

FIGS. 24 and 25 are sectional views taken along the lines 24—24 and 25≧25, respectively of FIG. 21;

FIG. 26 is a sectional view taken along the line 26—26 as applied to FIG. 19;

FIGS. 27 and 28 are sectional views taken along the line 27—27 and 28—28, respectively, of FIG. 21;

FIG. 29 is a sectional view taken along the line 29—29 of FIG. 19;

FIG. 30 is a sectional view taken along the line 30—30 of FIG. 31 and showing a portion of the rotary connection for delivering fluid power to the actuating cylinders for the sharpening means and for delivering coolant to the cutting blades;

FIG. 31 is an end elevational of the fluid rotary connection described in conjunction with FIG. 30; and

FIGS. 32 and 33 are sectional views taken along the lines 32—32 and 33—33, respectively of FIG. 31.

OPERATION GENERALLY

In the illustration given, and with reference to FIG. 1, the numeral 40 designates generally the frame of the inventive saw apparatus. The frame 40 includes a log conveyor 41 (see FIGS. 2 and 18) constituting means for advancing logs L at a constant speed along a path P which can be considered axial relative to the generally cylindrical logs L. Advantageously, two lanes of logs are advanced simultaneously by the conveyor 41. Alternatively stacks of elongated web material can be advanced along the path P such as the C-folded toweling illustrated in U.S. Pat. No. 3,288,009.

The cutting is performed by means of a pair of cutting disc blades 42 and 42′ although it is possible to use only one disc and still realize the benefits of the invention. For example, if the apparatus is set up for toilet paper rolls (having an axial length of 4½ inches) with each blade 42 and 42′ making a cut each cycle or orbit, the machine can be readily altered to toweling (having an axial length of 9 inches) merely by replacing one of the blades 42 or 42′ with a suitable counterweight. Inasmuch as the disc blade mounting and drive mechanism is the same for both blades (the orbit-providing member being essentially symmetrical), only one blade arrangement need be considered.

The novel orbiting is provided by an elongated skew arm 43 (see particularly FIG. 2). The skew angle α is a small acute angle—normally about 5° and can be considered either as the angle the skew arm 43 makes with a plane perpendicular to the path P as designated in FIG. 2, or the angle the axis of rotation S of the skew member 43 makes with the path P: this definition of the skew angle being illustrated and labeled in FIG. 3.

The axes B and B′ of the blades 42 and 42′, however, are parallel to the path P so that the blades are always positioned in planes perpendicular to the path P. The angle of the skew in conjunction with the rotational velocity of the skew arm 43 causes the translation of the blades (in the longitudinal direction parallel to the path P) to match the velocity of the logs at two points during each cycle as will be explained hereinafter with respect to FIG. 13.

The numerals 44 and 44′ (see FIG. 2) designate sharpening means and, by comparing FIGS. 1 and 5, it will be seen that the sharpening means always maintain the same attitude with respect to the ground, i.e., the frame 40. The sharpening means 44 and 44′ are carried by supports 45 always above the casings 46 for the spindles or shafts 47 and 47′ (see FIG. 5) of the two blades 42 and 42′—and travel therewith. Thus, it is possible to sharpen each blade 42 and 42′ each cycle—during that portion of the orbit when the blade is not cutting through the log L. This action is made possible by mounting the casings 46 on planetary shafts or supports 48 and 48′ (see FIG. 3).

The planetry shafts 48 and 48′ are also housings for the drives to the cutting blades—in particular, the spindles 47 and 47′. The details of this arrangement will be described in detail hereinafter with respect to FIG. 12.

The planetry motion i.e., orbiting without rotating of the planetary shaft 48 and 48′ is achieved through a gear train which can be readily appreciated from a consideration of FIG. 3. In FIGS. 3, the numeral 49 designates the stationary main gear of a planetary drive, mounted stationary relative to the frame (see FIG. 10) and is coaxial with the axis of rotation of the skew arm 43. The skew arm 43 carries idler gears 50 and 50′ and planetary gears 51 and 51′ which are associated, respectively, with the planetary shafts 48 and 48′. The orbiting motion imparted to the idler gears 50 and 50′ is canceled by the planetary gears 51 and 51′ so that the planetary shafts 48 and 48′ remain in a fixed attitude relative to the ground. This fixed attitude, in addition to making possible the provision of sharpening means that travel with the cutting blades, also makes possible a relatively simple drive for the blades—even taking into account the reverse "skew" introduced into the system to align the blade spindles 47 and 47′ with the path P.

In the operation of the apparatus as presented schematically in FIGS. 1-3, for example, the blade 42 is shown in a position shortly before it engages the logs L. As the skew member 43 rotates, the blade 42 moves downwardly and forwardly and, after a half cycle of operation, comes to the position occupied in FIG. 2 by the blade 42′, passing through the orientation depicted in FIG. 6. In FIGS. 1—3, the offset of the blades when they are horizontally aligned can be readily appreciated. When the blades are vertically aligned, as in FIGS. 4-5, the blades 42 and 42′ lie in the same plane. Thus, when a blade is in its cutting mode, it is in the lower half of the orbit and is moving forwardly, i.e., in the direction of log flow. During the upper half of the orbit of each blade, the blade is moving rearwardly.

However, at all times, each blade is rotating in a series of planes all perpendicular to the path P.

The skew angle α is a function of the log velocity and the geometry of the system—for a log speed of 45 inches per second, (yielding 1200 toilet paper rolls per minute) and for an offset of the blade spindle **47** and **47'** of 18 inches, the skew angle α is approximately 5°.

DETAILS OF SKEW ARM SUPPORT AND DRIVE

Reference is now made to FIGS. **4**, **7** and **8**. Again, the numeral **40** designates the frame which includes upright members **52** and **53** (see FIG. **8**) secured together by a base plate **54**, an intermediate cross tie member **55** and an upper cross tie member **56**. The frame **40** has extensions **40**a and **40**b to support the log conveyor head and tail shafts.

The upright members **52** and **53** have bolted thereto brackets **57**. Each of the brackets **57** carries an upstanding pin **58** rigidly fixed therein (see also FIG. **1**). The pins **58** serve as guides for the skew plate **59** which carries the skew arm **43**. In other words, the skew plate **59** is disposed at the same angle relative to the path P as is the skew arm **43**. This can be appreciated from a consideration of FIGS. **1** and **9** where the skew plate **59** is seen to be equipped with slide members **60** which are apertured to receive the pins **58**. The plate **59** is supported in a desired position by means of screw jacks **61** (see also FIG. **4**) which are in turn supported by the cross tie member **55**. By adjusting the screw jacks **61**, the location of the skew plate **59** on the pin **58** is adjusted accordingly—this being needed because the position of the skew plate **59** determines the position of the skew arm **43** and hence the position of the blades **42** and **42'**. With high speed operation, the cutting disc blades wear down fairly rapidly (from a 24 inch diameter to an 18 inch diameter), so the screw jacks **61** lower the plate **59** and hence the blades **42** and **42'** to properly engage the logs L.

The skew plate **59** (see the upper right hand portion of FIG. **10** and the central portion of FIG. **4**) carries a housing **62** in which the skew arm shaft **63** is rotatably supported. Here it should be noted that for the purpose of clarity and ease of understanding, the skew plate **59** and the skew arm **43** are presented in the drawings at an angle so as to properly orient the reader. In consequence, the blade spindle **47** is depicted parallel with the edge of the drawing—see the lower left hand portion of FIG. **10**.

The housing **62** is suitably bolted to the skew plate **59** as at **64** and is equipped with internal bearings as at **65** and **66** to rotatably support the skew arm shaft **63**. The skew arm shaft **63** in turn is bolted as at **67** to the skew arm **43**.

Still referring to FIG. **10**, the skew arm shaft **63** has fixed to it a pulley **68** on the upstream side of the skew plate **59**. Referring now to FIG. **8**, the pulley **68** is seen to be connected by means of a timing belt **69** to a drive pulley **70** supported on the shaft of a gear box **71** (see also FIG. **7**). The gear box **71** is supported on a pedestal **72** extending rearwardly, i.e., upstream, from the skew plate **59** (see also FIG. **9**). Thus, the gear box **71** moves vertically with the plate **59** to adjust for wear on the blades **42** and **42'**.

Rotational power is delivered to the gear box **71** by means of a vertically extending shaft **73**. The lower portion of the shaft **73** is splined to accommodate the desired vertical movement of the skew plate **59** and is received within another right angle drive gear box **74** (see FIG. **7**). Rotational power to the gear box **74** is derived from a line shaft **75** (still referring to FIG. **7**) which is suitably supported on the frame **40**. The line shaft **75** is equipped with a V-belt drive **76** which derives rotational power from a motor (not shown).

PLANETARY SHAFT MOUNTING

Referring now to FIG. **10**, the planetary shaft **48** is seen in the lower central portion thereof. This shaft **48** is suitably journalled in the skew arm **43** by means of bearings **77** and **78** (designated only in FIG. **12**). The shaft **48** has fixed to it the planetary gear **51** while the skew arm **43** rotatably supports the idler gear **50**. In the illustration given, the idler gear **50** is a split gear in order to phase the planetary gear **51** in proper relation to the main gear **49**. The main gear **49** is fixed to the housing **62** and hence is rigid with the skew plate **59**. Thus, as the skew arm shaft **63** is rotated, the skew arm **43** rotates to move the idler gears **50** and **50'** around the main gear **49**. By interposing the planetary gear **51** between the planetary shaft **48** and the idler gears **50**, the planetary shaft **48** is maintained in a given attitude with respect to the frame notwithstanding the rotation of the skew arm **43**.

BLADE DRIVE

Reference is now made to FIG. **12** wherein once again, the skew arm **43** is shown inclined to the edge of the paper. The skew arm **43**, as seen in FIG. **12** is equipped with a boss **79** (omitted from the showing in FIG. **10**) which accommodates the second bearing **78** for the planetary shaft **48**. The planetary shaft **48** is hollow, i.e., has a throughbore **80** which accommodates a drive shaft **81** for the disc blade **42**. As can be seen in the lower left hand portion of FIG. **12**, the drive shaft **81** is connected to the blade spindle **47** by means of helical gears **82** and **83** which permit the positioning of the blade spindle **47** with its axis parallel to the path P—and therefore introduces a reverse skew angle.

Rotational power for the drive shaft **81** is derived from a belt drive **84** which includes a pulley **85** fixed to the drive shaft **81** and a drive pulley **86** journalled on the housing **62** by means of bearings **87** and **88**. The pulley **86** also can be seen in FIGS. **1** and **9** and there is seen to be driven through a belt system **89** from a pulley **90** mounted on the output shaft **91** of a motor **92**. The motor **92** is also mounted on the skew plate **59** so that the drive for the blades **42** and **42'** moves with them as the skew plate **59** is adjusted to compensate for blade wear. The pulley **86** (still referring to FIG. **9**) is a double pulley with the downstream portion delivering rotational power to the blade pulleys while the upstream portion receives power from the motor **92**.

LOG CONVEYOR

FIG. **18** shows in perspective a section of the log conveyor **41**. The conveyor **41** includes a series of spacers **93** (see FIG. **7**) supported by upstanding frames **93**a (compare FIGS. **8** and **18**). The spacers **93** have chain guides **94** and **95** which support and guide runs of endless chain for supporting the lower carriers **96**.

A more detailed showing o the lower carrier **96** can be seen in FIG. **17**. The carrier **96** includes a unitary casting suitably contoured to support the logs or rolls. The support for the right hand roll (as seen in FIG. **17**) has an arcuate portion **97** curving upwardly to a point **98** which is essentially horizontally aligned with the axis of a log supported by the carrier **96**. Thus, the left hand portion of the part **97** constitutes an abutment restraining any sideways movement of a log under the influence of the movement of the disc blades in the direction indicated by the arrow **99**.

Secured to the right hand portion of the part **97** is an L-shaped clip **100** which is secured to another clip **101** provided as part of one of the lower chain runs **102**. The other lower chain run **103** (which travels in the guide **95**) also is equipped with a plurality of clips **104**—one for each of the lower carriers **96**. Secured to the clip **104** is another L-shaped clip **105** which is connected (as by welding) to the casting **96**. The casting **96** is another arcuate part **106** for supporting the left hand roll. Again, the left hand portion of he arcuate recess **106** rises to the point **107** approximately on the horizontal center line of the log supported thereby to stabilize the same against the force exerted by the orbiting dics blade during a cut. The right hand portion as at **108** of the part **106** is contoured upwardly and connected to the arcuate part **97** as seen in FIG. **17**.

Referring to FIG. **16**, the chain **102** is designated in the lower center portion of the view and is seen to be entrained about sprockets **109** and **110** which are rotatably supported on the frame—more particularly, on the portions **40a** and **40b**, respectively. Power for the chains **102** and **103** is derived from the line shaft **75** (see FIG. 7) via the gear box **111**.

A plurality of upper carriers **112** (see FIGS. **17** and **18**) are also provided as part of the log conveyor **41**. The upper carriers **112** are also deformed strap-like members, providing arcuate portions as at **113** and **114** (see FIG. **17**), corresponding, respectively, to the acruate portions of the lower carriers as at **97** and **106**. The upper carriers **112** are secured to clips **115** and **116** associated, respectively, with endless chain runs **117** and **118**. The chain runs **117** and **118** are guided in guides **119** and **120** (see FIG. **18**). The guides **119** and **120** are mounted on the auxiliary side frames **93a** (see FIG. **18**). The chain **117** (for example) is designated in FIG. **16** and is seen to be entrained on three sprockets **121**, **122** and **123**. Each of the sprockets (along with their companions for the chain **118**) are mounted on cross shafts suitably journalled in the frame. The triangular configuration of the chains **117** and **118** avoids any interference of the return run of the upper chains with the cutting mechanism—which is supported within the frame **40** immediately above the log conveyor **41**. Power for the chains associated with the lower carriers **96** is provided from the gear box **111** through spur gears and for the upper carriers **112** is also provided from the gear box **111** but via a timing belt drive **124**—see the lower left hand end portion of FIG. 7.

In the operation of the log conveyor **41**, each set of upper and lower carriers **96** and **112** is suitably spaced longitudinally from adjacent sets of carriers (see FIG. **18**) to permit passage between adjacent sets of the disc blades **42** and **42'**, as the case may be. As indicated previously, the log conveyor **41** operates continuously so as to advance the log L at a uniform speed through the cutting station.

VELOCITY MATCHING MECHANISM

From the foregoing it will be appreciated that the logs L are being advanced by the log conveyor **41** at constant velocity through the cutting station. In the system illustrated, i.e., one to produce 1200 rolls of 4½ inches long toilet tissue each minute from two logs and with two cutting disc blades **42** and **42'**, each log must be advanced at a speed of 45 inches per second. The log velocity V_L is determined as follows:

$$V_L = \frac{\text{cuts/minutes} \times \text{cut length}}{60 \text{ sec/min.}} \qquad (1)$$

Here there are 600 cuts per minute and the cut length is 4½ inches.

However, the blade velocity in the direction of log travel, i.e., the path P is not linear. Instead, the blade velocity in the axial direction (V_A) is:

$$V_A - R\omega \sin\alpha \sin \Theta \qquad (2)$$

wherein R is the radius of the cutting arm—see FIG. **2**, ω is the rotational velocity of the skew arm **43**, and Θ is the angular disposition of the skew arm. For simplicity, Θ (see FIG. **8**) is zero when the saws are on the vertical axis. Cutting, in the system illustrated, occurs between $\Theta = 135°$ and $\Theta = 225°$. The cutting arm radius R is 18 inches and the angular velocity ω is 10π radians per second, viz., $2\pi/60 \times \text{RPM}$ of skew arm **43**. Solving equation (2) for α (assuming $V_A = V_L$), a value of 4°-34' for the skew angle results.

By increasing the skew angle slightly, to 5°, the velocity of the blade in the axial direction V_A' is greater than the log velocity V_L at the middle of the cut, but lower at the beginning and end of the cut—see the left hand part of FIG. **13**. In contrast, by selecting the lower skew angle, the exact match is at mid-cut with V_A being lower than V_L at all other parts of the cut—see the curve designated V_A.

The middle segment of the FIG. **13** curve illustrates graphically the difference between V_L-V_A or the velocity compensation required to match the axial component of the blade velocity with the log velocity ($V_A = V_L$) throughout the entire cut, viz., $135° \leqq \Theta \leqq 225°$. Thus, the equation for velocity match is

$$V_L - R\omega \sin\alpha \cos \Theta = 0 \qquad (3)$$

In the illustrated embodiment, the velocity matching is achieved by a cam **125**—see FIG. **10**. The axial movement of the disc blade **42** (or **42'** as the case may be) to achieve this velocity match is determined by integrating equation (3), between values of Θ from $\pi/4$ to $-\pi/4$, resulting in

$$x = V_L \frac{\Theta}{\omega} - R \sin\alpha \sin\Theta \qquad (4)$$

With the parameters employed, the movement axially of the disc blade is approximately ±1/32 inch—see the right hand segment of FIG. **13**. However, substantial g forces are involved which can be appreciated by differentiating equation (3) with respect to time, resulting in the following:

$$\text{accel.} = R\omega^2 \sin\alpha \sin\Theta \qquad (5)$$

With the parameters selected, the acceleration is almost 3 g.

As indicated, the axial movement of the blade **42** (see FIG. **10**) is achieved through the use of a cam **125**—fixed to the skew arm **43** (see the lower central portion of FIG. **10** and also the right hand portion of FIG. **11**).

The numeral **126** designates a cam follower which is connected via a bell crank linkage to the blade spindle **47**. Thus, as the cam follower **126** is moved in a plane parallel to the skew arm **43** (as designated by the double ended arrow **126**a in FIG. **14**), the disc blade **42** is moved axially as indicated by the double ended arrow designated **42**a (see the lower left hand portion of FIG. **10**). Here it should be appreciated that cam follower movement is achieved because the cam **125** is fixed to the skew arm **43** (and therefore assumes different attitudes relative to the ground) whereas the cam follower **126** is supported by the planetary shaft **48** (see FIG. **14**) and thus maintains a constant attitude relative to the ground. Still referring to FIG.**14**, it is seen that a bracket **127** is fixed (as by bolting) to shoulder **48**a provided on the planetary shaft **48**. For the purose of mounting the cam follower **126**, a block **128** is fixed to the bracket **127**. An L-shaped arm **129** is pivotally mounted on the block **128** for movement about the axis C. The L-shaped arm **129** carries the cam follower **126** at one end and at the other end supports a push-pull block **130**. The vertical movement of the cam follower **126** is depicted by the double ended arrow **126**a in FIG. **14**. This converted to rotary motion (of push-pull on the spindle **47**) by the pivot mounting—see the curved double-ended arrow **42**a of FIG. **14**.

More particularly, the cam follower arm **129** clamps a rod **131** eccentric to the pivot axis C, the rod **131** carrying the block **130**. The block **130** is connected to the blade spindle **47**—see FIG. **10**. Thus, as the cam follower **126** moves up and down (as seen in FIG. **14**), the spindle **47** moves in or out. The push-pull block **130** retains the enlarged end of the spindle **47** and rotatably supports the spindle **47** by means of a bearing **132**. Additionally, the spindle **47** is splined and rides in splined blocks **133** to accomodate the axial or push-pull movement. By contouring the cam as a function of equation (4), the advantageous velocity match can be achieved. For different log speeds or different saw geometries, the apparatus can be adapted readily by changing the tapered bracket **134** (see FIGS. **6** and **12**) and shimming the brackets **57**—to change the skew angle. Also the cam **125** can be replaced to achieve a different velocity match.

BLADE SHARPENING MEANS

The blade sharpening means **44** and **44'** are depicted schematically in FIG. **2**. These are illustrated as two wheels or "stones" which are mounted on the casings **46** and thus also maintain a constant attitude relative to ground. More particularly (and now with reference briefly to FIGS. **1**, **7** and **15**), each planetary shaft supports a casing **46** for the spiral gears **82** and **83** previously designated with respect to FIG. **12**. The casings **46** provide the support for the blade sharpening supports **45**.

In FIG. **19**, the sharpening means **44** includes a pair of stones **135** and **136** inclined so as to give an approximately 14° angle of bevel to the cutting edge of the disc blade **42**. Thus, each stone **135** and **136** is inclined at approximately 7° to the plane of the disc blade **42**, but in opposite directions. Means to be described provide (1) for rotating the stones **135** and **136** about their own axes, (2) for moving the stones **135** and **136** axially relative to the disc blade **42**—to limit the sharpening to the half-cycle when the blade is not cutting, and (3) for moving the stones **135** and **136** radially relative to the blade **42** to compensate for blade wear.

The casing **46** carries the support **45** which includes a superstructure **137**. The superstructure **137** is equipped with a dovetail slot **138**. Slidably received for essentially vertical movement within the dovetail slot **138** is a portion of the subframe **139**—the subframe **139** carries the sharpening means **44**. Vertical adjustment or positioning of the subframe **139** is achieved by means of the hand wheel **140** which is rotatably mounted within the superstructure **137** and is threadably received at **141** (see FIG. **21**) within the subframe **139**. By turning the hand wheel **140**, the subframe **139** is moved radially relative to the disc blade **42** to adjust for wear of the sharpened periphery.

Each of the stones **135** and **136** is rotatably mounted within the subframe **139** but because the drives for these stones, in the illustrated embodiment, determine in large measure the structure of the mounting, the drive system for delivering rotational power to the stones **135** and **136** will now be described.

All of the rotational power for both stones **135** and **136** is derived from a spur gear **142** (see the lower left hand portion of FIG. **10**) which is driven in synchronism with the blade spindle **47**. Not shown in FIG. **10** but immediately behind the spur gear **142** is a second spur gear **143** (see FIG. **23**). The spur gear **143** is mounted on a shaft **144** (still referring to FIG. **23**) which carries a pulley **145**—see also FIGS. **14** and **15**. The pulley **145** is seen in dotted line in FIG. **22** and the axis of the shaft **144** is so designated in FIG. **21**.

The pulley **145** is connected by a belt **146** to a pulley **147** with the pulley **147** being mounted on a shaft **148**. The shaft **148** carries a drive pulley **149** which is connected by means of a belt **150** to a driven pulley **151** (see FIG. **14**) mounted on a shaft **152** journaled in the subframe **139** (see FIG. **24**). The axis of the shaft **152** is designated in FIG. **21**. The shaft **148** is journalled in a bearing arm **153** roatably mounted on the shaft **152** and is also rotatably mounted on a pivot arm **154** rotatably mounted on the shaft **144**. Thus, as the subframe **139** is moved downwardly (as viewed in FIGS. **14** and **15**), the shaft **152** likewise moves downwardly and to accommodate this while still driving the same, the shaft **148** moves outwardly, i.e., to the left in FIG. **21** under the elbow-type action achieved by the cooperation of arms **153** and **154**—see the curved arrows about the shaft axis **148** in FIG. **15**.

The second pulley **155** on the shaft **152** (see FIGS. **22** and **24**) delivers power by a belt **156** to a pulley **157** associated with the stone **136**.

Also mounted on the shaft **152** (see FIG. **24**) is a spiral gear **158**. This mates with a spiral gear **159** (see FIG. **25**) which is fixed to a hollow shaft **160**. The hollow shaft **160** is equipped with internal splines to carry the splined shaft **161** (see FIG. **26**) which in turn carries the stone **135**. The splined connection between the shafts **160** and **161** permits the axial movement of the stones to bring them into engagement with the edge of the disc blade **42** when sharpening is indicated.

Each of the spiral gears **158** and **159** is arranged at approximately 3½° angle so that the total offset of the shaft **160** (or the shaft **161**) relative to the shaft **152** is approximately 7°—this being one-half of the bevel desired for the periphery of the disc blade **42**.

The drive for the other stone **136** (see FIGS. **27-29**) is essentially the same. For example, there is an intermediate shaft **162** which carries a spiral gear **163**—much the same as the shaft **152** carries the gear **158**. As mentioned previously there is a pulley **157** which receives rotational power from the shaft **152** via the pulley **155**. The spiral gear **163**, also fixed to the shaft **162**, mates with a spiral gear **164** (see FIG. **28**) mounted on the exterior of a hollow shaft **165**. The shaft **165** is equipped with splines in its hollow interior which receive the splined shaft **166** (see FIG. **29**) and which carries the stone **136**. One difference between the drive connection to the stone **136** (as seen in FIGS. **27–29**) from the drive to stone **135** (shown in FIGS. **24–26**) is that the spiral gears **163** and **164** (FIGS. **27** and **28**) are each arranged at an angle of about 3¾° which develops not only the desired 7° offset referred to in connection with the stone **135** but also develops an additional 3° offset in a direction orthogonal to the 7° direction. This insures that the edge of the stone moving into the blade performs the sharpening contact. In other words, if the stone **136** were not additionally canted as just described, the edge of the stone moving away from the axis of the blade would be doing the sharpening with the possibility of a poor bevel, flying debris, etc.

Before describing the means for moving the stones radially relative to the disc, the stone dressing feature will be described. From time to time it is advantageous to "dress" the stones, i.e., refurbish the grinding surfaces thereof. Rather than do this by hand, a laborious procedure and since it would be dangerous to do this while the disc blade **42** is rotating, a separate dresser drive is provided. For this purpose, a dresser drive nut **167** (see FIG. **26**) is provided on the shaft **161** (which also carries the stone **135**). This permits the attachment of a portable motor or the like for turning the stone **135**. When the stone **135** is rotated, rotational power is transmitted through the spiral gears **159** and **158** to the shaft **152** (see FIGS. **25** and **24**). From there, rotational power is delivered via the pulley **155**, the belt **156** to the pulley **157** which rotates the shaft **162**. Then, through the spiral gear connection of gears **163** and **164**, the shaft **165** is turned so as to rotate the stone **136**. Inasmuch as the pulley **151** (see FIG. **24**) is also turned along with shaft **152**, this normally would result in rotating the disc blade **42**. To avoid that, the pulley **145** (see FIGS. **22** and **23**) is equipped with a one-way clutch. Thus, rotational power transmitted during stone dressing from the pulley **151** merely turns the pulleys **148** and **145** but does nothing further to the disc blade **42**.

The means for moving the stones **135** and **136** axially relative to the blade **42** include air cylinders **168** and **169** (see FIGS. **4** and **19**) which are mounted on the subframe **139**. The air cylinder **168** is pivotally mounted as at **170** on the subframe **139**. The piston rod **171** of the cylinder **168** is pivotally connected to a link **172**—see also FIG. **20**. This can be appreciated further from a consideration of the upper left hand corner of FIG. **21** where it is seen that the generally vertically extending link **172** is fixed to a cross shaft **173** (see also FIG. **20**). In like fashion, the piston rod **174** of the cylinder **169** is connected by means of a link **175** to a cross shaft **176**. Fixed on the cross shaft **176**, is a fork member **177** which is also seen in FIGS. **1** and **20**. The cross shafts **173** and **176** are rotatably supported in brackets **178** (compare FIGS. **1** and **21**). Interposed between each of the cross shafts **173** and **176** and their associated brackets **178** are rubber bushings. Each rubber bushing acts like a torsion spring mounting resisting rotary movement and thus urges the associated stone out of contact with the blade when the air pressure is released from the associated cylinder **168** or **169**, as the case may be.

Inasmuch as the connection of the fork **179** (see the upper central portion of FIG. **21**) to its associated stone shaft **166** and stone **136** is the same as the connection of the fork **177** to the splined shaft **161** and stone **135**, only the latter will be described—the fork **179** being seen in FIG. **19**.

The fork **179** is bifurcated at its lower end to provide arms **180** (see FIG. **21**). Each arm **180** slottedly engages a side of a ball bearing collar **181** mounted on the shaft **66**. Thus, as the piston rod **174** is extended from the cylinder **169**, the link **175** is pivoted about the axis of the shaft **176**. The rubber torsion spring mount of the shaft **176** tends to resist this movement and in so doing stores energy tending to return to its original condition. However, when the shaft **176** is turned under the urging of the air cylinder **169**, the fork **179** is also pivoted about the axis of shaft **176** and moves its associated ball bearing collar **181** slightly to the left in FIG. **19**, thereby urging the stone **136** into engagement with the periphery of the blade **42**. The vertical slotted engagement of the collar **181** with the fork arms **180** permit a slight vertical movement so as to move along an arcuate pth closely approximating but not exactly linear with the axis of the shaft **166**. The axial movement of the shaft **166** while the same is still delivering rotational power to the stone **136** from the gear **164** is achieved through the use of the splines connecting the shaft **166** with the gear carrying hollow shaft **165**. The gearing and splines are protected against stone dust by the accordion sleeves **182** (see FIG. **19**).

From the foregoing, it will be appreciated that the sharpening means **44**, more particularly the cylindrical stones wheels **135** and **136** are driven from the same drive as that which drives the associated blade, i.e., the blade spindle **47**. This substantially reduces both the weight and the space requirements while achieving high efficiency not only as far as blade sharpness is concerned but also the fact that the blades are sharpenable during the portion of each orbit when they are not cutting, and without the need for stopping the blade in a particular section of the orbit. Through the use of air pressure, illustrated, the stones are "air loaded" against mechanical stops **183** (see FIG. **20**) for equal pressure on the disc blade **42**, not for the purpose for positioning the grinding wheels as was characteristic of the prior art.

To deliver pressure fluid, i.e., air pressure to the air cylinders **168** and **169**, and also to deliver lubricant or cooling fluid to the edges of the disc blades **42** and **42'**, a rotary connection **184** (see FIG. **5**) is employed in conjunction with the skew arm shaft **63**. The skew arm shaft is equipped with suitable longitudinal bores (not shown) which communicate with flexible hoses as at **185** (see FIG. **16**) leading to a boss **186** integral with the skew arm **43**. Thus, there is no rotary motion between the noses **185** and the skew arm **43**. It should be noted that if only one disc blade **42** is employed, there would be no need for rotary connection inasmuch as the flexible hose would then run from the frame to the portion of the planetary shaft which retains a fixed attitude relative to the frame.

The integral boss **186** is shown in fragmentary form in FIG. **30** and is seen to be equipped with a collar **187** bolted thereto be means of bolts **188**. The collar **187** forms one half of the second rotary connection in conjunction with an internally grooved ring **189**. The outer collar **189** is fixed to the planetary shaft **48** by means of a J-shaped clip **190** which extends between the collar

189 and the previously mentioned ring **128** fixed with respect to the planetary shaft **48**. Thus, as the skew arm **43** rotates, the inner collar **187** rotates within the outer collar **190** to provide the second rotary connection. Still referring to FIG. **30**, the inner collar **187** is equipped with a peripheral groove **191** which extends over about 180°—see also FIG. **31**. Compressed air is delivered to the partial perimetric groove through an inlet **192** (see FIG. **32**) coupled to one of the hoses **185**. The inlet **192** in the boss **186** communicates with an axial bore **193** in the boss **186** which is in permanent communication with an aligned axial bore **194** in the collar **187**. A short radial bore **195** communicates the bore **194** with the previously mentioned 180° perimeter groove **191**.

The outer collar **189** is equipped with a radially extending bore **196** which is aligned with the 180° perimeter bore **191** and serves as an outlet for compressed air. Thus, over somewhat less than about one-half the cycle compressed air is delivered through the bore **196** to the air cylinders **168** and **169**.

A second inlet is provided as at **197** (see FIG. **33**) in boss **186**. This communicates with an axial bore **198** in the boss **186** which in turn is aligned with another axial bore **199** in the inner collar **187**. The bore **199** has a radial branch **200** which is in continuous communication with a perimeter groove **201** in the innerface of the outer collar **189**. An outlet **202** (see FIGS. **31** and **33**) communicates the branch bore **202** through hoses (not shown) to the rotating edge of the disc blade **42**. It will be appreciated that a similar arrangement is provided relative to the boss **186**′ (see FIG. **16**) for use in conjunction with the disc blade **42**′. By this arrangement, coolant or lubricating fluid is delivered continuous to the cutting blades while sharpening is performed only during the half cycle when the blade is not in contact with the work.

I claim:

1. A method of cutting logs of convolutely wound web material *such as toilet paper and toweling,* comprising:

uniformly advancing a log along a path into a cutting station having an arm above said path,

rotating said arm about an axis skewed with respect to said path, said arm having a disc blade *support* mounted thereon *and rotatably carrying a disc blade* with the blade axis spaced from the arm axis to cause said blade to intersect said path and cut a log *adjacent the bottom of the blade orbit,* maintaining said blade axis at an angle skewed relative to said arm axis to position said disc blade continually perpendicular to said path whereby said disc blade makes a square cut through a log, [and] *said support carrying sharpening stones,*

continuously planetarily rotating said disc blade *support* about the blade axis from said arm axis *to maintain said sharpening stones above the axis of said disc blade and in a fixed relation to said blade axis whereby said stones are adapted to sharpen said blade at any point during the orbit thereof, and*

continuously rotating said disc blade about said blade axis independently of said arm.

2. The method of claim **1** in which a pair of logs are advanced in side-by-side relation in said path whereby said cutting is performed while said blade axis moves through an angle of about 90°.

3. The method of claim **1** in which said [arm is equipped with cylindrical stones for sharpening said disc blade, rotating] said stones *are* continuously *rotated* and selectively [moving said stones] *moved* into sharpening engagement with said disc blade when the same is out of cutting engagement with said log.

4. The method of claim **1** in which said arm is mounted for vertical movement in said station, and moving the axis of said arm vertically toward said path to compensate for wear on said disc blade.

5. A method of cutting logs of convolutely wound web material, comprising:

uniformly advancing a log along a path into a cutting station having an arm above said path,

rotating said arm about an axis skewed with respect to said path, said arm having a disc blade mounted thereon with the blade axis spaced from the arm axis to cause said blade to intersect said path and cut a log,

maintaining said blade axis at an angle skewed relative to said arm axis to position said disc blade continually perpendicular to said path whereby said disc blade makes a square cut through a log,

continuously rotating said disc blade about the blade axis from said arm axis, said arm being equipped with cylindrical stones for sharpening said disc blade, and rotating said stones continuously and selectively moving said stones into sharpening engagement with said disc blade when the same is out of cutting engagement with said log.

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