Oct. 29, 1968 J. M. STEINKE 3,407,678

MECHANISM FOR PRODUCING ROTARY OUTPUT MOTION WITH HARMONIC DISPLACEMENT CHARACTERISTICS

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FIG. 1

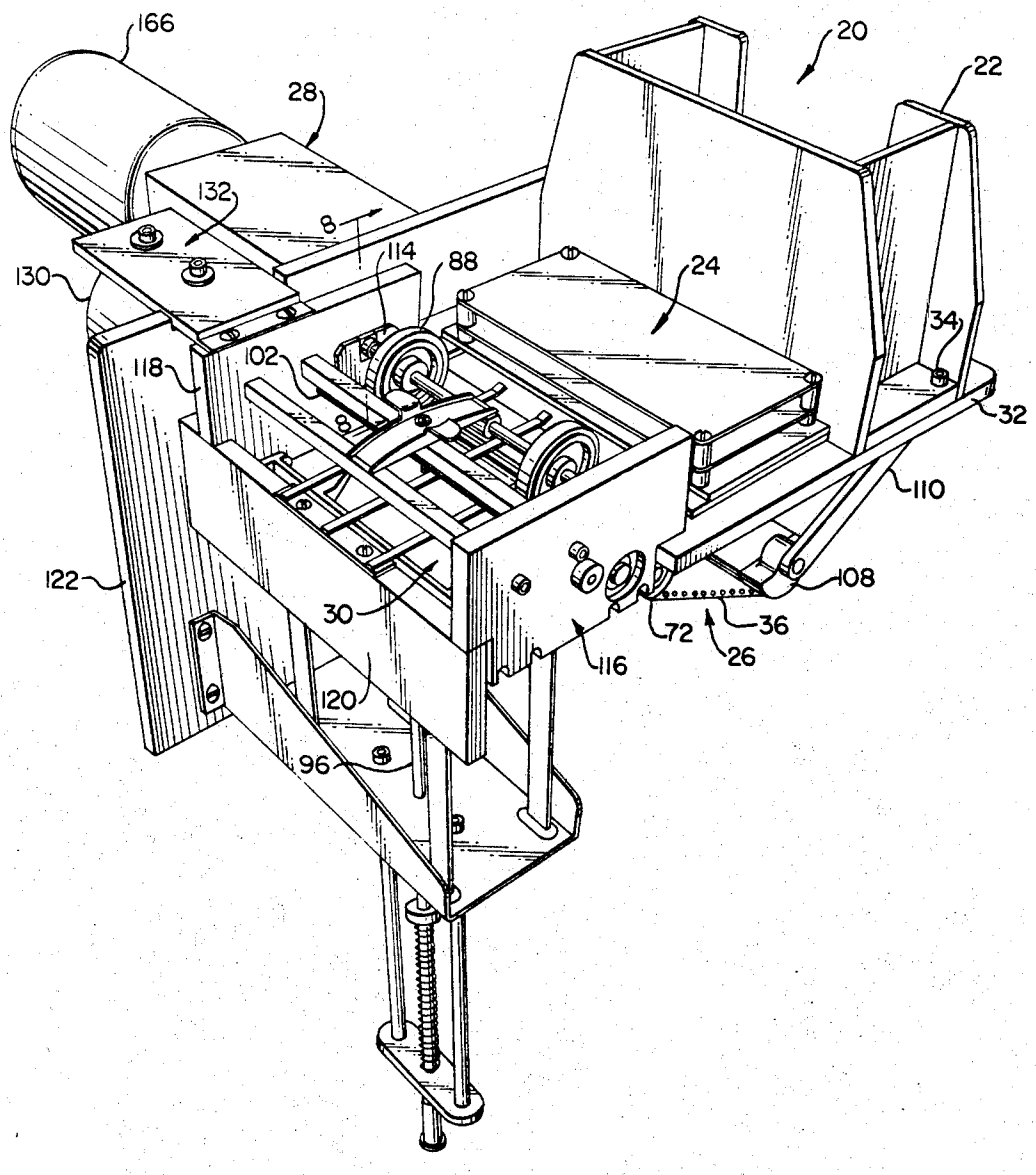


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FIG. 2

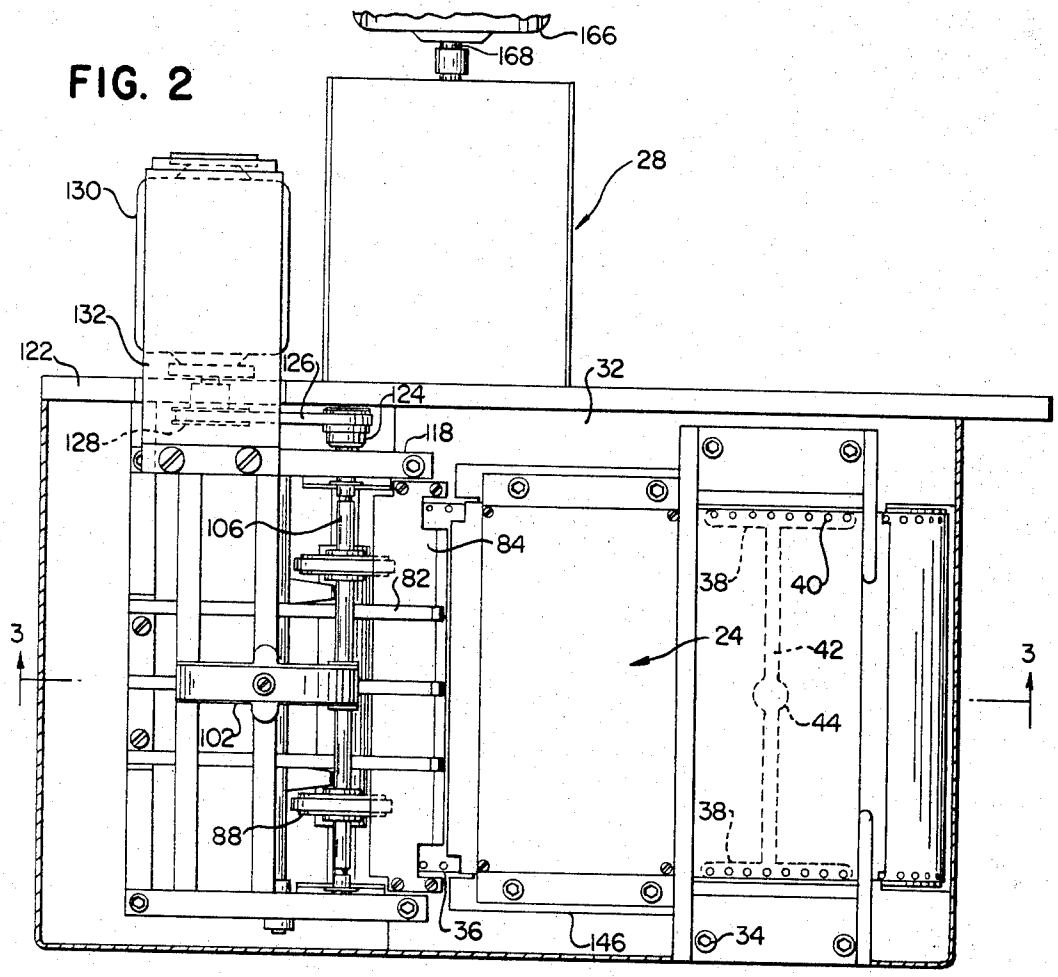


FIG. 6

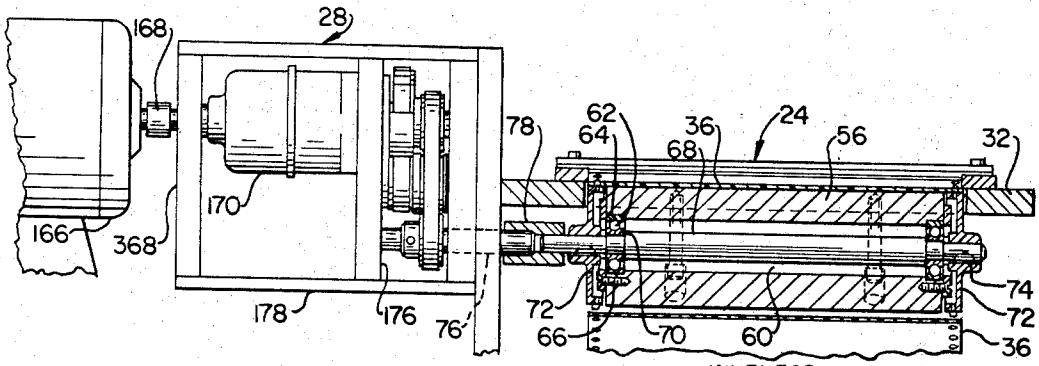


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FIG. 3

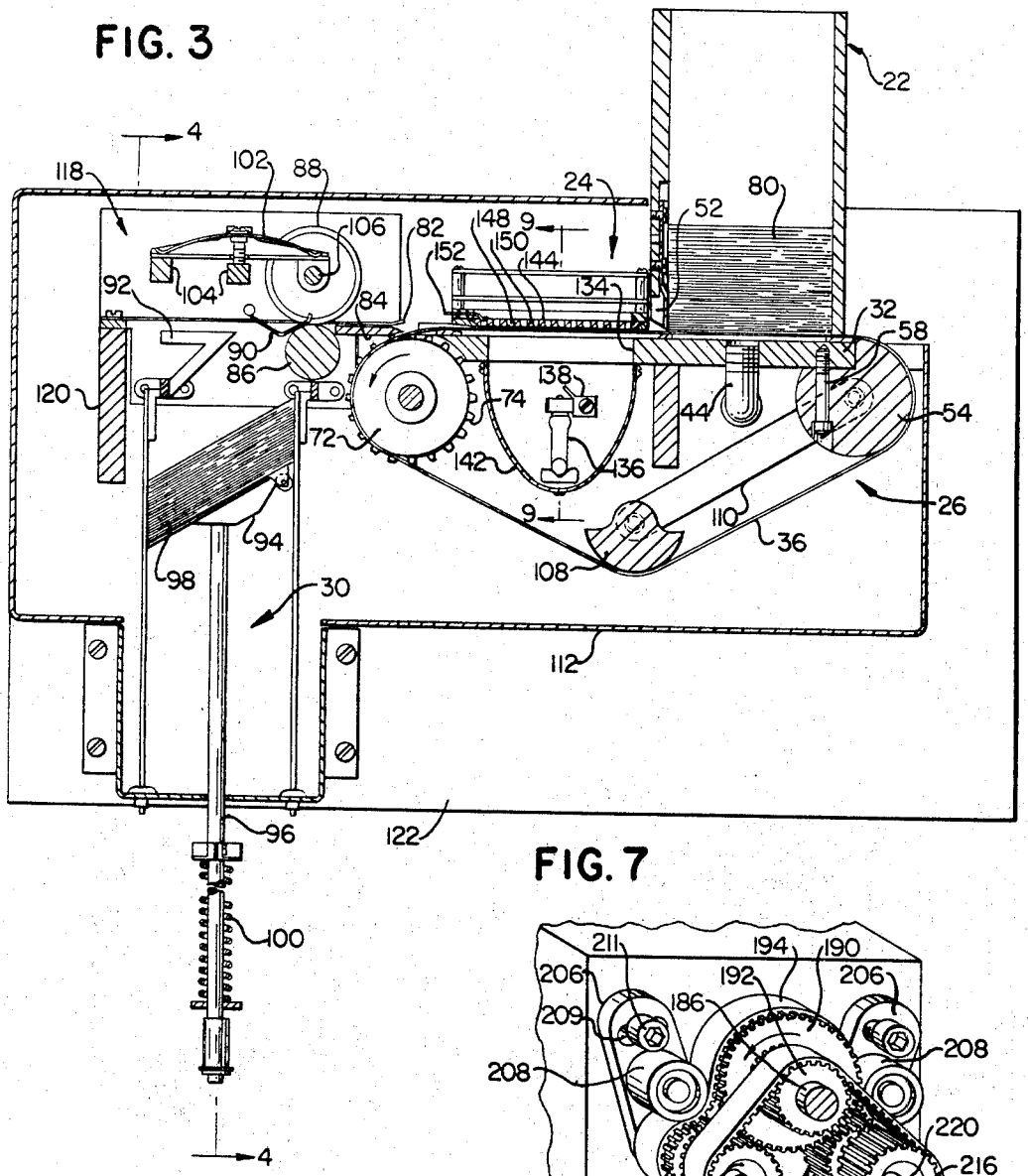
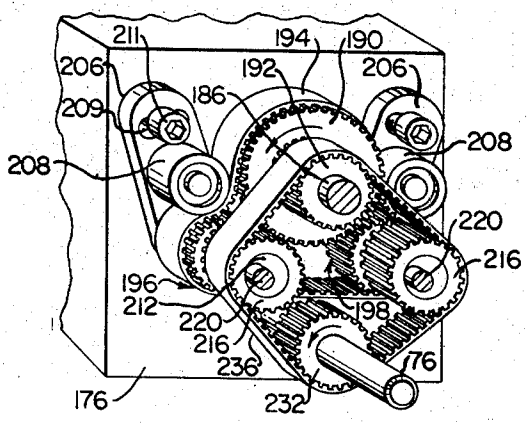



FIG. 7

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MECHANISM FOR PRODUCING ROTARY OUTPUT MOTION WITH HARMONIC DISPLACEMENT CHARACTERISTICS

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FIG. 4

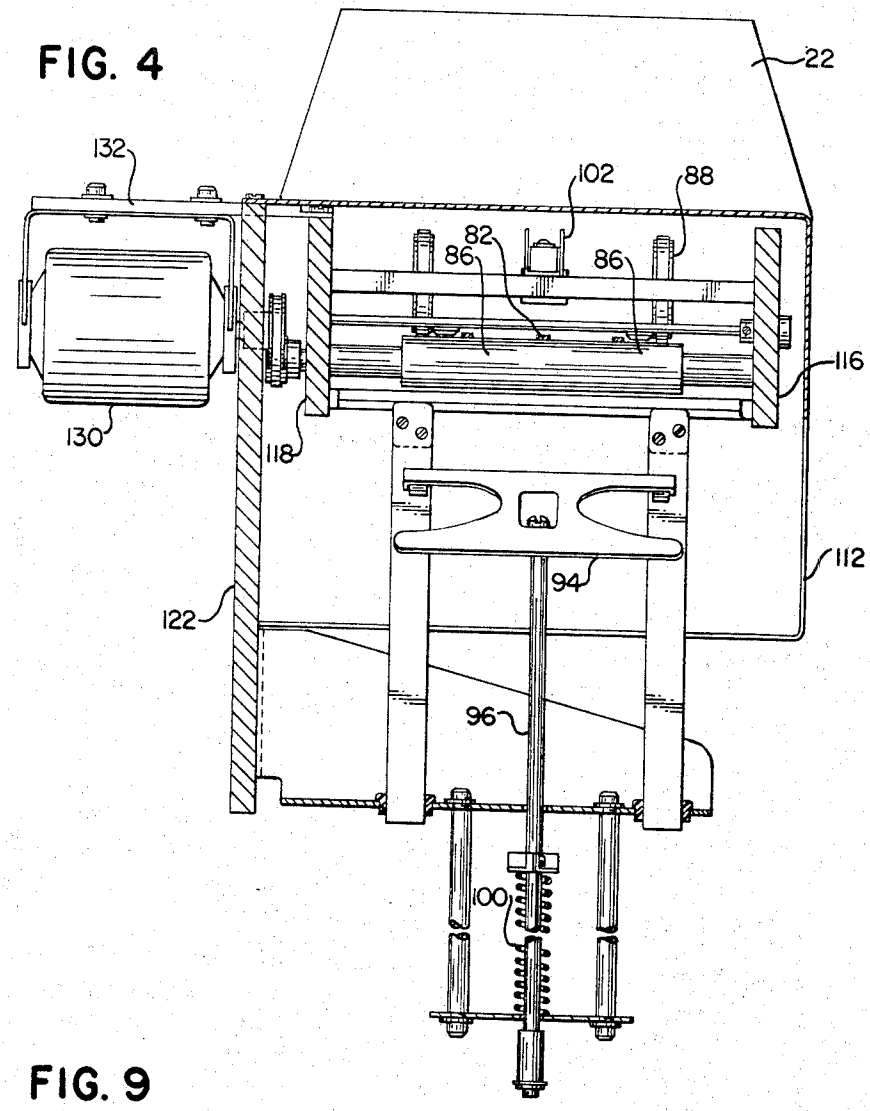


FIG. 9

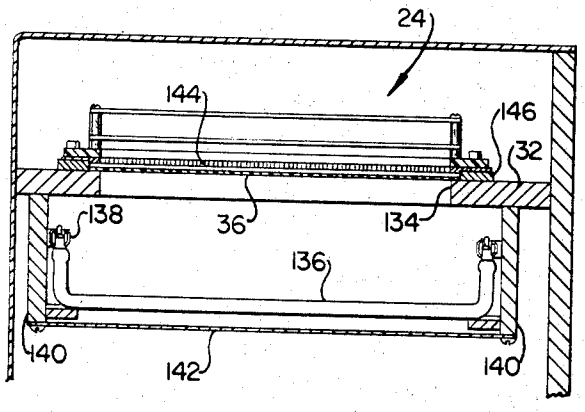


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FIG. 5

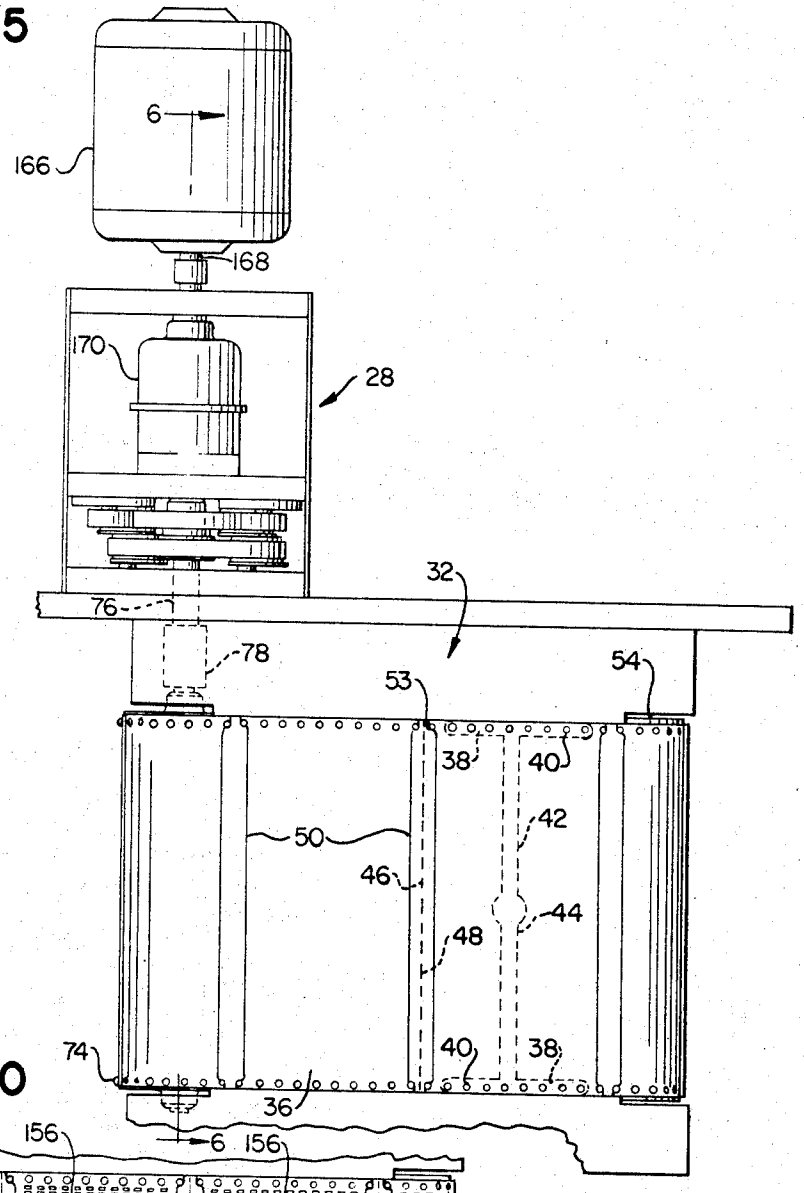


FIG. 10

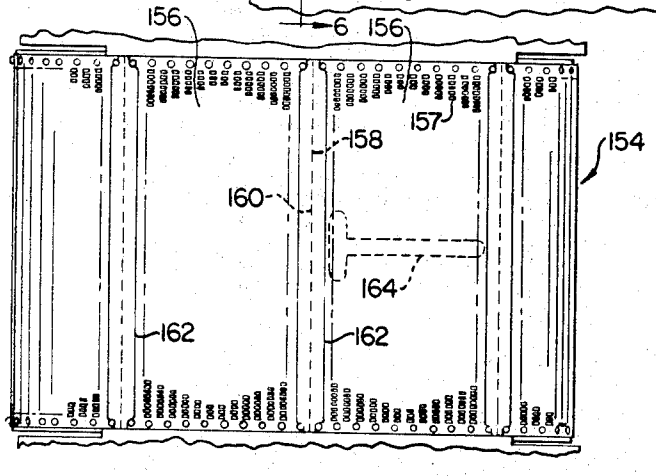


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FIG. 8

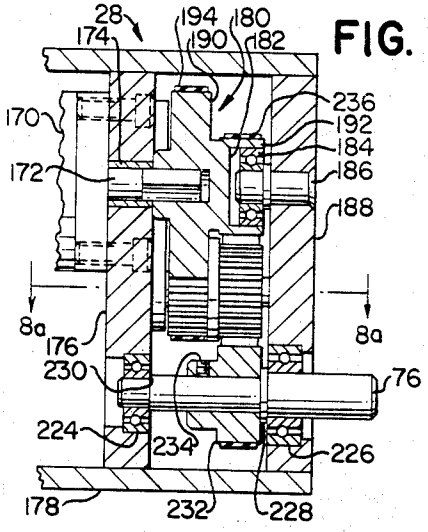


FIG. 8a

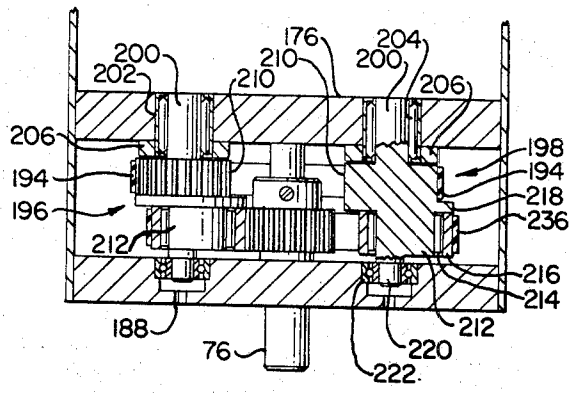


FIG. 12

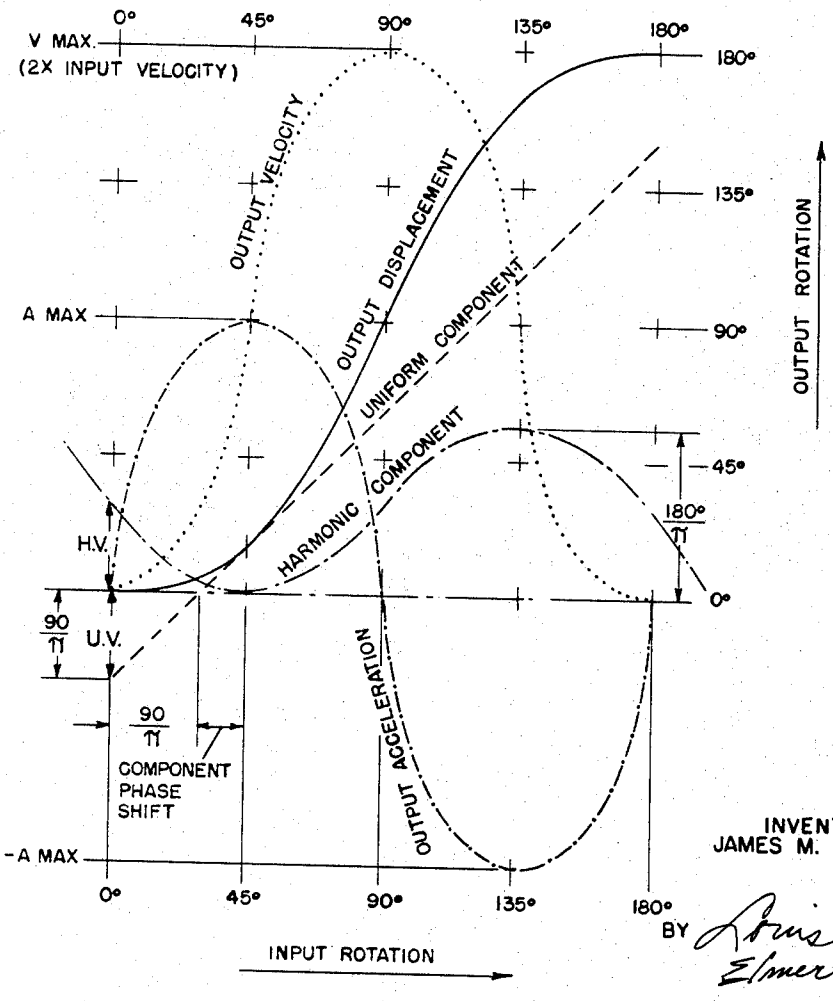


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FIG. 11

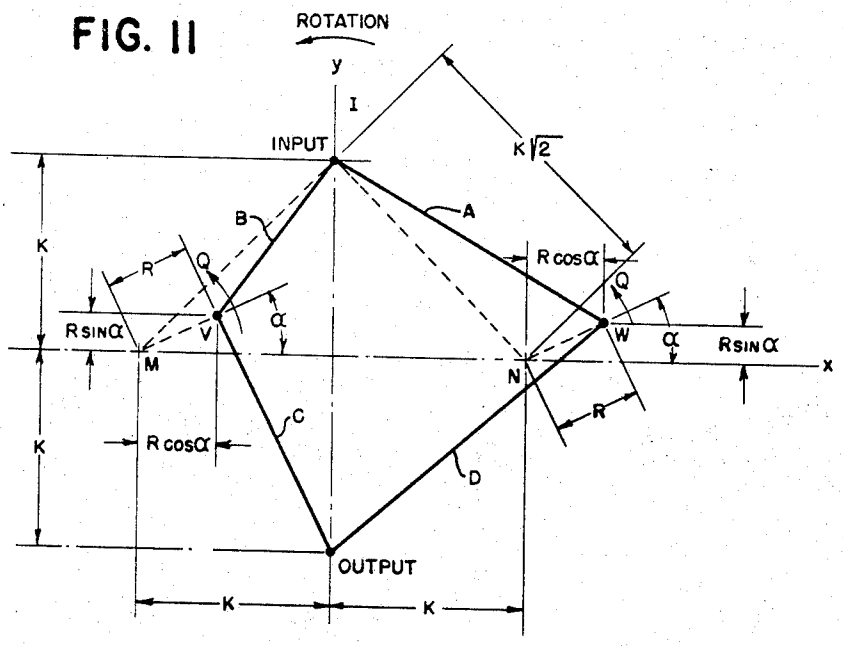


FIG. 18

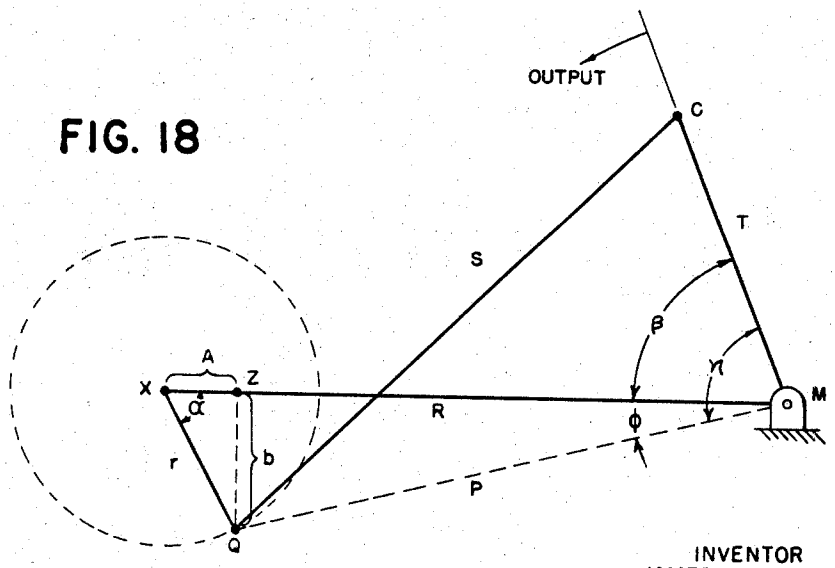


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FIG. 13

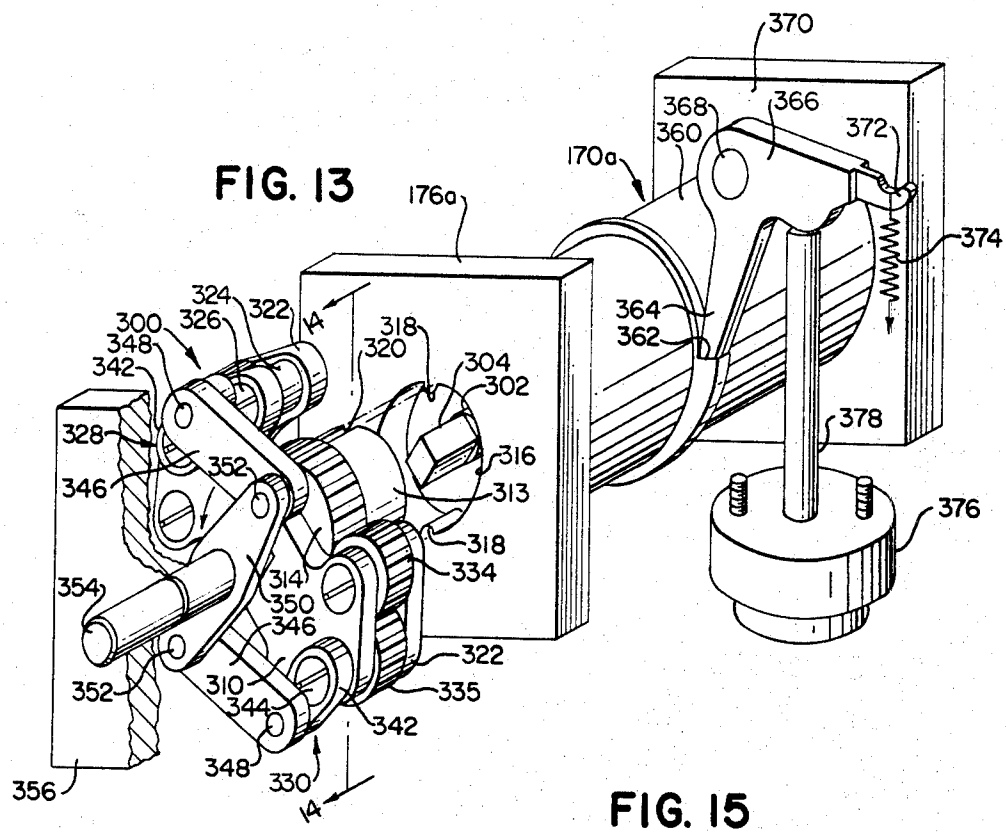


FIG. 14

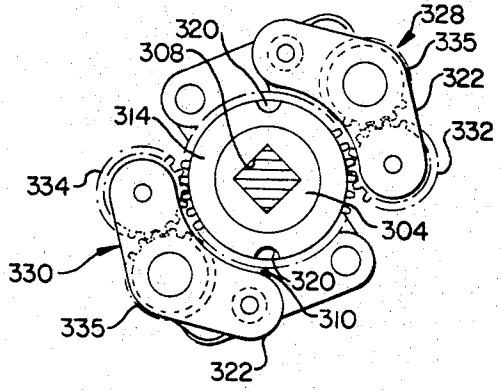


FIG. 15

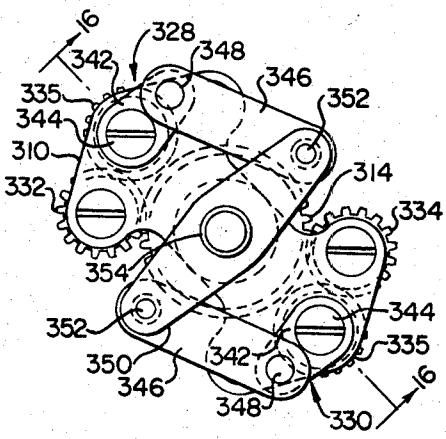


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FIG. 16

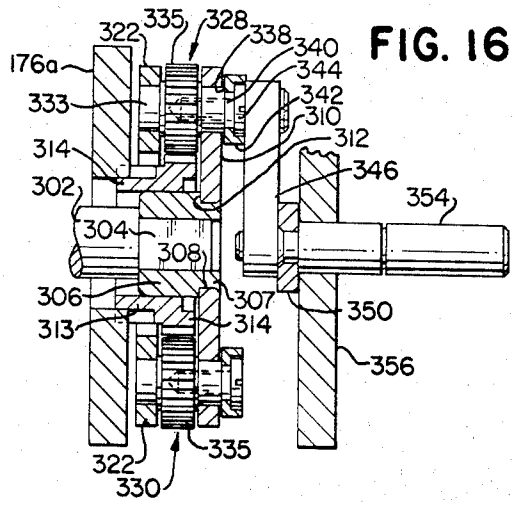


FIG. 17

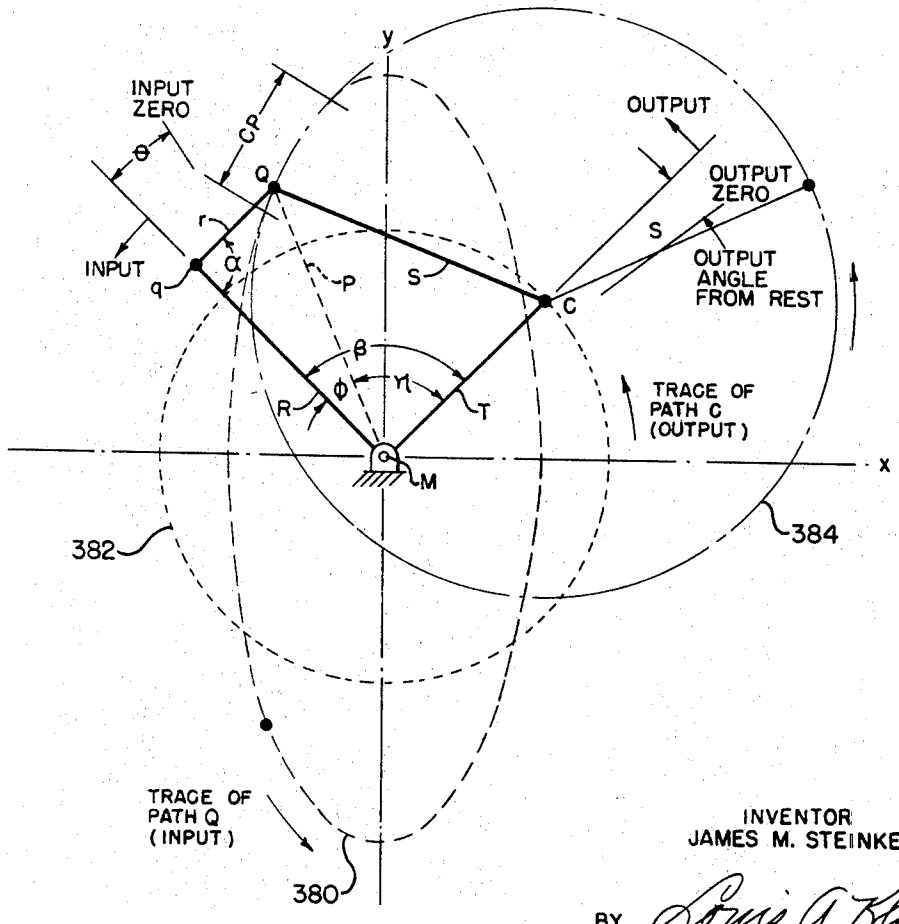


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FIG. 19

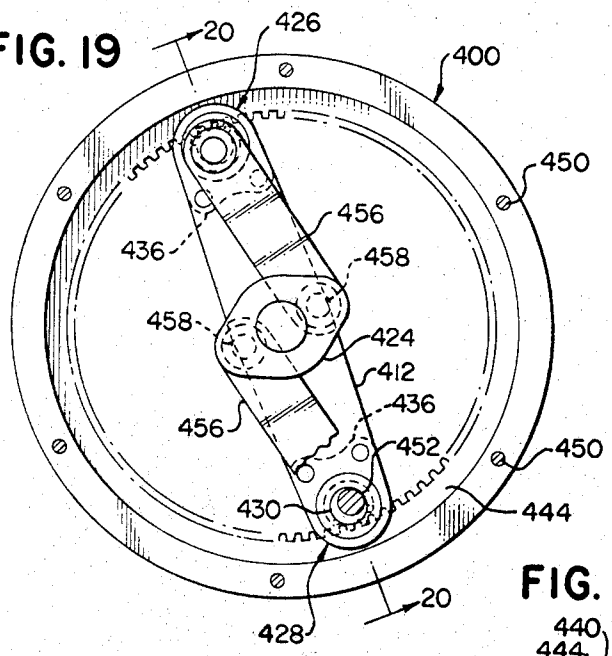



FIG. 20

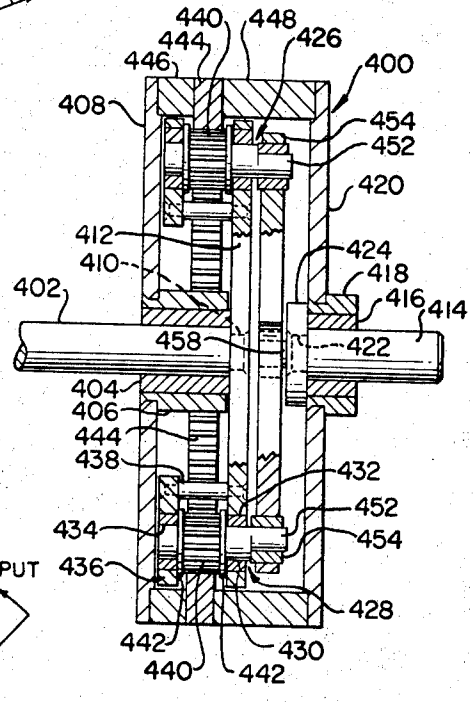
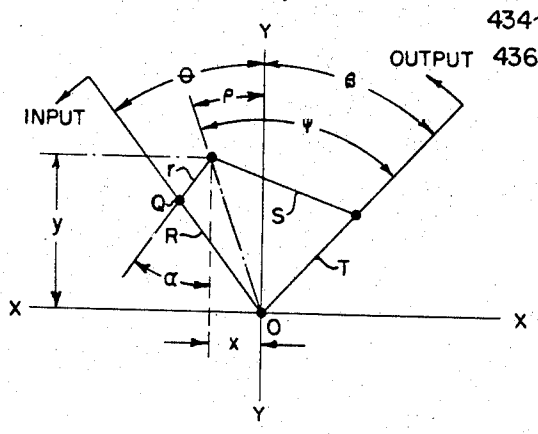



FIG. 21

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United States Patent Office 3,407,678
Patented Oct. 29, 1968



3,407,678

# MECHANISM FOR PRODUCING ROTARY OUTPUT MOTION WITH HARMONIC DISPLACEMENT CHARACTERISTICS

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Filed Dec. 19, 1966, Ser. No. 602,846
13 Claims. (Cl. 74—394)

## ABSTRACT OF THE DISCLOSURE

A drive mechanism which converts an input of constant rotary motion to an output of intermittent rotary motion is shown in three different embodiments, all using crank members. The crank members are rotated by belt, epicyclic, and hypocyclic connections to alter the constant rotary input to produce a dwell in the output shaft of the mechanism.

This invention relates to input data equipment, and more particularly it relates to a drive mechanism for a card feed mechanism employed in an input data machine such as a high-speed punched card reader which provides input data to a computer system.

Recent developments in electronic digital computers have greatly increased the speeds at which the computers perform their internal operations. Often, the total computation time is governed and limited by the rate at which input data is fed into the computer. While magnetic tapes and drum inputs have facilitated the entry of data into the computer system, much of the data fed into such systems is still in the form of punched tabulating cards.

In an effort to reduce the time during which the computer must wait for input data to be fed to it from punched cards, the card feed mechanism shown herein was developed, so that cards could most effectively be fed to a read station where, upon proper command from the computer, all of the data in the card could be simultaneously read and fed to the computer. By this construction, the time for feeding the input data to the computer is greatly reduced over the existing punched card readers which employ sensing means which serially read rows of information from the punched cards as they are moved thereby. However, when all the data in the card is being read at once, special problems arise in the handling of the cards because each card must be fed to a reading station where all of the data in the card is in reading relationship therewith, and then the card is momentarily stopped during the actual reading period. Reading all the data in the card at once enables the data to be rearranged, if necessary, at the time it is fed into the computer, without the use of complicated intermediate storage devices such as buffer registers.

The drive mechanism of the present invention is used with a card feed mechanism which in turn utilizes a conveyor-like endless feed band which is movably supported and intermittently driven in one direction under a card feed hopper. The feed band is provided with picker knives which are dimensioned so as to engage only the bottom card in the hopper and are spaced along the feed band so as to receive the card between two adjacent picker knives and move it to the read station.

The feed band is driven by the drive means of this invention, which subject each card to a gradual acceleration when the card is removed from the card feed hopper and a gradual deceleration as the card is momentarily brought to a stop at the read station where the data contained therein is read. The drive means produce varying intermittent rotary motion output from a constant rotary motion input to drive the conveyor as mentioned above.



Accordingly, a primary object of this invention is to produce a drive mechanism for a high-speed, card-feed mechanism which is especially adaptable for transporting punched cards from a feed hopper to an operative station, such as a read station, at which station the cards are precisely and momentarily stopped to be read and from which station the cards are subsequently moved to a stacker pocket.

Another object of this invention is to provide a drive mechanism for a high-speed card-feed mechanism which is especially adaptable for transporting punched cards from a hopper to a read station which provides input data to a computer system, the read station being such as to simultaneously read all the information in the card so as to minimize the time during which the computer is detained in receiving such information.

Another object of this invention is to provide a drive mechanism for a high-speed card-feed mechanism which transports the punched cards from a feed hopper to an operative station while subjecting the cards to a minimum of extreme accelerating and decelerating forces.

A further object of this invention is to provide an improved harmonic-motion-type drive mechanism which is especially adaptable for use in a high-speed card feed mechanism.

Another object of this invention is to provide an improved mechanical summing device which produces intermittent rotary motion from a continuous constant rotary input.

These and other objects and advantages of this invention will become more readily understood in connection with the following description and the drawings, in which:

FIG. 1 is a perspective view of the card feed mechanism with which the drive mechanism of this invention is used, showing, generally, the card feed hopper, the read station, the conveyor for transporting cards to the read station, the drive mechanism for driving the conveyor, and the stacker pocket for receiving cards which have been read;

FIG. 2 is a plan view of the top of the card feed mechanism shown in FIG. 1;

FIG. 3 is a cross-sectional view in elevation taken along the line 3—3 of FIG. 2, showing details of the feed hopper, the read station, and the conveyor;

FIG. 4 is a cross-sectional view in elevation, taken along the line 4—4 of FIG. 3, showing details of the stacker pocket for receiving the cards from the read station, the reading head at the read station, and the conveyor belt;

FIG. 5 is a plan view of the top of the card feed mechanism showing one embodiment of the driving means for driving the conveyor, and also showing the read station; the card feed hopper being omitted from this drawing to show the picker blades on the conveyor;

FIG. 6 is an elevational view, partly in section and taken along the line 6—6 of FIG. 5, showing details of the means for mounting and driving the feed band;

FIG. 7 is a perspective view of one embodiment of the driving means for producing intermittent rotary motion for driving the conveyor belt;

FIG. 8 is an elevational view, partly in section and taken along the line 8—8 of FIG. 1, showing more details of the driving means shown in FIG. 7;

FIG. 8*a* is a plan view, partly in section and taken along the line 8*a*—8*a* of FIG. 8, showing more details of the drive mechanism;

FIG. 9 is an elevational view, partly in section and taken along the line 9—9 of FIG. 3, showing details of the read station;

FIG. 10 is a plan view, similar to FIG. 5, of another modification of the conveyor belt of this invention, showing a belt formed of a plurality of sections joined to-

gether, each section having therein an array of holes which are in registration with the holes of a fully punched tabulating card when the card is placed thereon;

FIG. 11 is an elevational view of a geometrical model similar to the embodiment of the intermittent rotary drive mechanism shown principally in FIG. 7;

FIG. 12 is a graph showing the relationship of velocity, acceleration, and output with regard to input and output rotations of the drive mechanism shown principally in FIG. 7;

FIG. 13 is a perspective view of a second embodiment of the driving means for producing intermittent rotary motion for driving the conveyor belt;

FIG. 14 is an elevational view of the second embodiment of the drive mechanism taken along the line **14—14** of FIG. 13;

FIG. 15 is an elevational view of the front of the drive mechanism shown in FIG. 13;

FIG. 16 is an elevational view, partly in cross section, taken along the line **16—16** of FIG. 15, showing more details of the drive mechanism;

FIG. 17 is a geometrical model of the second embodiment of the driving means shown principally in FIG. 13;

FIG. 18 is a geometrical model similar to FIG. 17 but somewhat enlarged and showing the links in a different position;

FIG. 19 is an elevational view of a third embodiment of the driving means for producing intermittent rotary motion for driving the conveyor belt;

FIG. 20 is a cross-sectional view of the driving means shown in FIG. 19 and is taken along the line **20—20** thereof; and

FIG. 21 is a geometrical model of the third embodiment of the driving means shown in FIGS. 19 and 20.

FIG. 1 is a perspective view of the card feed mechanism **20** with which the drive mechanism **28** of this invention is used. The card feed mechanism is composed of several basic elements, which are the card input feed hopper **22**, the read station **24**, the conveyor means **26** for delivering the cards to be read from the hopper **22** to and from the read station **24**, the drive mechanism **28** for driving the conveyor, and the card-receiving pocket **30** for receiving the cards from the read station **24**.

The card feed hopper **22** is of standard size to receive standard punched tabulating cards, and the hopper is secured to the feed table **32** by fasteners **34**, as seen in FIGS. 1 and 2. There is sufficient clearance between the bottom of the hopper **22** and the feed table **32** to permit the feed band **36** to pass therebetween.

The feed table **32** is provided with a pair of spaced parallel grooves **38** (FIG. 2), which are aligned with the sprocket driving holes **40** in the feed band **36**, so that they will pass thereover. Another groove **42** in the feed table **32** interconnects the parallel grooves **38** and is connected to a vacuum line **44**, shown in FIG. 3, which in turn is connected to a source of vacuum (not shown). When cards are placed in the hopper **22**, the lowest card in the hopper is brought into close contact with the feed band **36** due to the air being withdrawn through holes **40**, which provides contact-producing forces in addition to the weight of the cards above the lowest one.

The feed band **36** is best shown in FIG. 5, in which the read station **24** and the feed hopper **22** are omitted to facilitate the showing of the feed band **36**. In one embodiment, the feed band **36** itself is made of a transparent, flexible, durable plastic, such as Mylar, which is perforated along its edges to provide aligned spaced driving sprocket holes **40**. The feed band **36** may be made from a strip having its ends **46** and **48** abutting under a picker blade **50**.

The picker blades **50** are mounted in spaced parallel relationship on the feed band **36** at right angles to its lateral edges, so as to receive the width of a standard tabulating card between any two adjacent picker blades **50**. These blades also may be made of plastic or Mylar when the belt itself is made of such material, and they are secured to the feed band **36** by suitable adhesives.

The blades **50** have a thickness which is less than the thickness of a tabulating card to insure that only one card will be taken from the lower side of the feed hopper **22** when the feed band **36** passes thereunder. The side of the hopper **22** which is adjacent to the read station **24** is provided with a suitable, adjustable, throat-knife mechanism, designated generally as **52** (FIG. 3), which permits only one card at a time to be taken from the hopper via the feed band **36**. The picker blades **50** for the feed band **36** are notched at **53** (FIG. 5) to provide clearance for the tines **74** (FIG. 3) of the driving sprocket **72**, which pass through the holes **40**.

FIGS. 2, 3, 5, and 6 show the means for mounting the feed band **36** in the card feed mechanism. In order to eliminate some of the inertia of the mechanism, the feed band **36** is mounted to slide over stationary cylinders **54** (FIG. 3) and **56** (FIG. 6), which are secured to the feed table **32** by screws **58**, as shown in FIG. 3. The cylinder **54** extends for the full width of the feed band **36**, as shown in FIG. 5; however, the cylinder **56**, as shown in FIG. 6, does not extend across the full width of the feed band **36** but is made shorter to accommodate the driving sprockets **72**, as will be explained later.

The cylinder **56**, shown in FIG. 6, has a hole **60** extending axially therethrough, and at each of the extremities of the hole **60** the cylinder **56** is provided with an annular shoulder **62**, against which a suitable bearing and support member **64** abuts. The member **64** is secured to the cylinder **56** by screws **66** to concentrically support a shaft **68** for rotation in the hole **60**. The outer extremities of the shaft **68** are smaller in diameter than its central portion to provide shoulders **70**, which abut against the pertaining bearing and support members **64** to thereby restrain the shaft **68** against axial movement in the cylinder **56**.

The driving sprockets **72** are secured to the reduced diameter portions of the shaft **68** outwardly of the bearing and support members **64** by suitable keys **73**. The sprockets **72** are spaced apart on the shaft **68** to enable the tines **74** on the sprockets **72** to enter the sprocket driving holes **40** on the feed band **36** in driving engagement therewith.

One end of the shaft **68** is detachably secured to an output shaft **76** (FIGS. 5 and 6) by a connector **78**. The output shaft **76** is driven by the drive mechanism **28**, which will be discussed in detail later. The drive mechanism **28** is effective to deliver intermittent rotary motion to the output shaft **76**, which in turn drives the feed band **36**.

The driving sprockets **72** rotate counter-clockwise (as viewed in FIG. 3) to move the feed band **36** under the feed hopper **22**, where the lowest card of a stack of cards **80** is forced onto the band between a pair of adjacent picker blades **50**. As the feed band **36** moves, the card just removed from the hopper **22** is moved to the read station **24**. The drive mechanism **28**, which rotates the sprockets **72** to move the feed band **36**, is also effective to move the feed band **36** at a variable speed and also to cause the band to dwell, so that the card removed from the hopper **22** onto the feed band **36** stops momentarily under the read station **24**, where it is read. The drive mechanism **28**, in connection with the feed band **36**, is effective to move the card to and stop it at the read station in proper registration with the read station **24** without the aid of additional stops.

The read station **24** utilizes photoelectric means to simultaneously read all columns of data in the punched card which is waiting momentarily under the read station to be read upon a signal from the computer or other device which is to receive the data. Usually, the computer or other device which receives the data operates at such high speeds that it is always ready to receive the data from the cards, and, therefore, the feed band **36** can operate continuously, repeating the process of positioning and momentarily stopping the card under the read station

24. The card feed mechanism is effective to feed cards at rates up to approximately twelve hundred cards per minute. After being read, the card is moved from the read station 24 by the feed band 36 to the card-receiving pocket 30, where the cards which have been read are collected.

The receiving pocket 30, shown principally in FIG. 3, is of standard construction and includes the usual deflector fingers 82 and deflector plate 84, which guide the card between a feed roller 86 and discs 88. Upon leaving the feed roller 86 and the discs 88, the card passes a deflector 90 and a pocket card guide 92, which direct the cards downwardly, where they come to rest upon a card reception plate 94, which is supported on a plate support tube 96. As the load of cards 98 on the plate 94 increases, the plate support tube 96 descends to compress a spring 100. A pressure bridge 102, mounted on cross bars 104 and a shaft 106, on which the discs 88 are mounted, varies the contact pressure of the discs 88 on the feed roller 86, between which the cards are fed.

Tension on the feed band 36 is obtained through use of a weight 108 (FIG. 3), which has an arcuate surface in sliding contact with substantially the entire width of the feed band 36. The weight 108 is supported by two arms 110, only one of which arms is shown in FIG. 3 and which arms are pivotally secured at their lower ends to opposed sides of the weight 108. The upper ends of the arms 110 are pivotally secured to opposed sides of the stationary cylinder 54.

The shaft 106 and the feed roller 86, shown in FIG. 3, are rotatably supported in suitable bearings 114 (FIG. 1), which are secured to front and rear frame support plates 116 and 118, respectively, which in turn are secured to a side plate 120 (FIG. 1), which is secured to a rear wall plate 122.

The discs 88 are fixed to rotate with the shaft 106, which has a drive pulley 124 fixed to one end thereof, as shown in FIG. 2. A suitable belt 126 drivingly connects the pulley of a motor 130 with the drive pulley 124 to rotate the discs 88. A mounting plate 132 is used to secure the motor 130 to the rear wall plate 122 and the frame rear plate 118.

The details of the read station 24 using photoelectric means are shown in FIGS. 3 and 9. The feed table 32 is provided with an opening 134, through which light from a flash tube 136 may pass when the flash tube is energized. The flash tube 136 is secured in position by mounting brackets 138, which are fastened to supports 140 (FIG. 9) depending from the feed table 32. A suitable light shield 142 is used to confine the light within the opening 134. The flash tube 136 is of the xenon type, which is similar to a strobe light, and is energized by suitable external circuitry connected with the computer or device to which the data in the cards is fed.

At the time that the flash tube 136 is energized, a punched card bearing the information to be read is positioned over the opening 134 in proper registration with the read head 144. Suitable guides 146 (FIG. 9) are secured to the feed table 32 to maintain the card to be read and the feed band 36 in proper registration with the read head 144 in the direction of the width of the feed band 36. The card being read is maintained on the feed band 36 between two adjacent picker blades 50, and the drive mechanism is effective to position the card to be read in proper registration with the read head 144 along the length of the feed band 36.

With the card on the feed band 36 in proper registration with the read head 144, and with a signal from the computer, the flash tube 136 is energized, and light therefrom passes up through the opening 134 in the feed table, through the punched holes in the card being read, and to the read head 144.

The read head 144 is made of a layer of glass which is opaque to light except for a plurality of window areas areas 148 are arranged in the read head 144 in an array which permit light to pass therethrough. These window which is identical to the array of rows and columns of areas available for punching in a standard tabulating card. When a punched card is positioned in proper registration under the read head 144, and the flash tube is energized, light passes through the feed band 36, through the holes in the punched card, and through the pertaining window areas 148 in the read head 144. Each window is provided with a photo-responsive means which can generate a signal when light is received thereby.

In the embodiment shown, the side of the read head 144 which is opposite to the card being read has a thin layer of photoconductive material thereon. This photoconductive material is formed into discrete areas 150, FIG. 3, with one such area convering each window area 148 of the read head 144. Contact leads such as 152 are used to individually connect each such discrete area 150 with the computer or external device to which the information is fed. Thus, when a hole is present in the card being read, the light passes through the feed band 36, the pertaining hole in the card, and the pertaining window area 148 to the particular discrete area 150 of photoconductive material to generate an electrical signal therein. By this arrangement, all the holes in the card are simultaneously read to produce individual signals for each hole appearing therein.

FIG. 10 shows another embodiment of the feed band used in this invention, in which the belt, designated generally as 154, is made from metal, such as stainless steel which is approximately .003 inch thick. The belt 154 is made of a plurality of sections 156, which have therein holes 157, which are formed in the pattern of a fully punched tabulating card with which the belt 154 is used. The joining edges 158 and 160 of the individual adjacent sections 156 are secured to a common picker blade 162, which performs the dual function of joining the sections 156 and also of acting as a picker blade for transporting the punched cards.

When the belt 154 is used, the feed table 32 is provided with a groove 164 (FIG. 10), which is located centrally of the belt passing thereover. The groove 164 communicates with the vacuum line 44 (shown in FIG. 3) to assist in the transfer of cards from the feed hopper 22 to the belt 154.

*Intermittent rotary drive mechanism*

FIGS. 5 to 8 and 8*a* show one embodiment of the drive mechanism of this invention, which is designated generally as 28. The drive mechanism 28 is essentially a mechanical summing-up device which utilizes a continuous uniform-speed input to produce variable-speed intermittent rotary output. This input is used to provide two separate motion components; one component remains a constant rotational motion, and the second generates a motion very similar to harmonic motion. Both of these motions are algebraically added together by the drive mechanism 28 to produce usable, variable-speed, intermittent output motion having specific characteristics at the output shaft 76. The intermittent rotary drive mechanism 28 is:

(a) A novel harmonic motion generator whose displacement, velocity, and accelerations are coincident at a zero point.

(b) A low-cost, mechanical device comprising rotating components producing intermittent rotary motion from an input of constant, uniform rotation.

(c) A motion-generating device whose output is instantaneously at rest, increases in speed according to harmonic motion characteristics, achieves an instantaneous value twice that of the input velocity, and returns to the instantaneous zero velocity according to harmonic motion characteristics.

Referring to FIGS. 5 to 8 and 8*a*, the drive mechanism 28 comprises an input motor 166, whose output shaft 168 (FIG. 5) is connected to a clutch 170. The

output shaft **172** (FIG. 8) of the clutch **170** is rotatably supported in a sleeve-type bearing **174**, which is supported in a planar support member **176**, which is part of the housing **178** of the drive mechanism **28**.

The output shaft **172** (FIG. 8) of the clutch **170** drives a dual input gear **180**, which is fixed at one side to rotate therewith. The other side of the input gear **180** is provided with an annular recess **182**, which is concentric with the shaft **172**. A bearing **184** (which is rotatably supported on a stub shaft **186**) is inserted in the recess **182** to rotatably support the input gear **180**. The stub shaft **186** is secured to the planar support member **188** of the housing **178**.

The dual input gear **180** has a large spur gear **190** and a small spur gear **192** formed thereon, as shown most clearly in FIGS. 7, 8, and 8*a*. An endless, timing, chain-type belt **194** passes over the large spur gear **190** to drive the cranks **196** and **198**. These cranks **196** and **198** are identical, and each has a shaft **200**, which is rotatably mounted in bearings **202** and **204**, respectively, which bearings are mounted in the planar support member **176**. Each shaft **200** of the cranks **196** and **198** passes through an aperture adjacent one end of the belt-tensioning brackets **206**, which are located next to the planar support member **176** (FIG. 7). Each bracket **206** is provided with a belt-tensioning roller **208**, whose axis of rotation is parallel to the axis of rotation of the dual input gear **180**. The bracket **206** is also provided with an arcuate slot **209** (FIG. 7), through which an adjusting screw **211** passes. By pivoting the brackets **206** about their lower ends, through which the shafts **200** pass, the tensioning rollers **208** provide for an adjustment in the tension of the belt **194**, and the brackets are locked in the adjusted positioning by tightening the adjusting screws **211**.

The belt **194** drives the gear **210** (FIG. 8*a*) of each of the cranks **196** and **198**, and each gear **210** is concentric with its shaft **200**. Each crank **196** and **198** is provided with an eccentrically-positioned crank shaft **212** (FIG. 8*a*), on which a bearing **214** is mounted. A gear **216** is then rotatably mounted on the bearing **214** and is separated from the gear **210** by a flange **218**, which is integrally formed on the respective crank (**196** and **198**).

Each of the cranks **196** and **198** is provided with a second shaft **220** (FIG. 8*a*), which is concentric with the shaft **200** and which is rotatably mounted in a bearing **222**, which in turn is mounted in an annular recess in the planar support member **188** to thereby support the crank.

The output shaft **76** of the drive mechanism **28** is rotatably supported in bearings **224** and **226** (FIG. 8) and is provided with a flange **228**, which abuts against the bearing **226** to restrain axial movement of the shaft in one direction. The shaft **76** is restrained against axial movement in the opposite direction by a shoulder **230**, on the shaft **76**, which abuts against the bearing **224**. An output gear **232** is positioned against the flange **228** and is secured to the shaft **76**, to rotate therewith, by a screw **234**.

The axes of the shafts **172**, **200**, and **76** (FIGS. 8 and 8*a*) are parallel to one another, and the axes of the shafts **200** for the cranks **196** and **198** are equidistantly spaced from the axes of the output shafts **172** and **76**. In the specific embodiment shown, the input gear **192**, the gears **216** on the cranks **196** and **198**, and the output gear **232** all have the same number of teeth and pitch diameter (FIGS. 7 and 8*a*). A timing, chain-type belt **236** is used to interconnect the input gear **192**, the gears **216**, and the output gear **232**, as shown in FIG. 7.

When the drive mechanism **28** is driven, the output shaft **172** (FIG. 8) of the clutch **170** rotates both the large input gear **190** and the small input gear **192** at a constant r.p.m. The belt **194**, in turn, rotates the cranks **196** and **198** at a constant r.p.m. As these cranks rotate the crank shafts **212** (on which the gears **216** are rotatably mounted), the gears **216** are also rotated at a constant rate with respect to the axis of rotation of the shafts **200**.

As previously stated, the small input gear **192** is rotated at a constant rate and is used to drive the belt **236**. The belt **236** engages the teeth of the input gear **192**, the output gear **232**, and the gears **216**, as shown in FIG. 7. The cranks **196** and **198** are aligned so that identical points on the cranks will bear the same angular relationship to an imaginary line joining the axes of rotation of the shafts **220** as these cranks are rotated by the belt **194**. The gears **216** rotate on the crank shafts **212** as the cranks **196** and **198** are rotated.

If the cranks **196** and **198** (FIG. 7) were to remain stationary and only the belt **236** were to be driven, then the gears **216** would simply rotate on the eccentric shafts **212**, and the output delivered to the output gear **232** would be simply constant rotary motion, the same as the input motion. Also, if the input gear **192** were uncoupled from the large input gear **190**, so that the input gear **192** could rotate freely with respect to the shaft **172**, and if the cranks **196** and **198** alone were driven, then the motion imparted to the output gear **232** by the belt **236** would cause an oscillation of the output shaft **76**. The oscillation would be simple harmonic motion, and one complete rotation of the cranks **196** and **198** would cause one oscillation of the output shaft **76**. The number of oscillations per complete revolution of the large input gear **190** would be dependent upon the ratio of the number of teeth in the large input gear **190** compared to the number of teeth in the gears **210** on the cranks **196** and **198**. The angular displacement of the output shaft **76** in turn would be dependent upon the throw of the cranks **196** and **198**, which for each crank is the distance which the axis of the shaft **212** is offset from the axis of rotation of the shaft **200**.

However, when the drive mechanism is treated as a composite unit, the output shaft **76** (FIG. 7) senses the algebraic sum of the constant rotary motion supplied by the belt **236** from the small input gear **192** and the simple harmonic motion generated by the cranks **196** and **198** via the belt **194**. By a proper combination of dimensions and spacing (to be described later) for the various elements in the drive mechanism, the desired motion characteristics are obtained at the output shaft **76** to drive the feed band **36**, which feeds the tabulating cards to the read station **24**.

When a card is being fed to the read station **24**, it is desirable to subject the card to a gradual acceleration from the rest position in the feed hopper **22** to maximum velocity while it is traveling on the feed band **36**. The deceleration of the feed band **36** should also be gradual when the card carried thereby is brought to a stop under the read station **24**, where the card is read upon proper signal from the computer or other utilization device with which the read station is operatively connected. The drive mechanism **28** of this application is designed to drive the feed band **36** so as to produce such desired motion in transporting the cards while still performing the feeding operation at rates up to approximately 1,200 cards per minute.

The drive mechanism **28** itself is utilized to position and stop the feed band **36** without the aid of additional stops, so that the cards carried by the feed band **36** will be properly positioned under the read station **24**. In order to produce a dwell in the rotation of the output shaft **76** of the drive mechanism, it is necessary at some point in its cycle to have the components of constant velocity motion and harmonic motion cancel or offset each other, so as to produce the dwell or zero velocity in the output shaft **76**.

This offsetting of motions is shown in the graph in FIG. 12, which shows a family of curves relating to the operating characteristics of the drive mechanism **28**. In this graph, the input rotation of the dual input gear **180** is plotted along the horizontal axis, and the output rotation of the output shaft **76** is plotted along the vertical axis. The value of the component of constant uniform velocity

(marked U. V. on the curve marked UNIFORM COMPONENT) is equal and opposite to the value of the component of harmonic motion (marked H. V. on the curve marked HARMONIC COMPONENT) at 0 degrees of input rotation, so that both components offset each other to produce 0 degrees of output rotation of the output shaft **76**. Note, too, on FIG. 12, that the curves marked OUTPUT VELOCITY, OUTPUT DISPLACEMENT, and OUTPUT ACCELERATION of the output shaft **76** are all zero when the offsetting of motions, previously mentioned, occurs at 0 degrees of output rotation. This graph will be later discussed in more detail in connection with the geometrical model shown in FIG. 11, from which certain formulas were obtained in designing the drive mechanism **28**.

The formulas for obtaining the relationships of the various elements used in the drive mechanism **28** shown principally in FIG. 7 were derived in connection with the geometrical model shown in FIG. 11.

In deriving the formulas, the following stipulations were made with regard to FIG. 11:

(1) The points M and N, which represent the rotational axes of the cranks, are equidistantly spaced from the axes of rotation of the INPUT and OUTPUT members.

(2) The crank arms R are equal in length and travel at the same synchronized angular rate about their respective axes M and N relative to a fixed line; line X, for example.

(3) The gears **192, 232,** and **216,** shown in FIG. 7, all have the same pitch diameter; the geometrical model does not shown such gears but uses points marked as INPUT, OUTPUT, and M and N, respectively, as equivalents.

(4) The lengths A, B, C, D, and E of FIG. 11 represent changing lengths of the belt drive means **236** between adjacent gears **192, 232,** and **216,** as shown in FIG. 7, as the crank arms R rotate counter-clockwise, as viewed in FIG. 11.

(5) The geometrical model of FIG. 11 is shown with the crank arms R rotating in a counter-clockwise direction, marked Q, and being displaced at an angle $\alpha$ from line X.

From the geometrical model shown in FIG. 11, A, B, C, and D represent changing lengths of belt as the crank arms R rotate counter-clockwise about their centers M and N. When the crank arms R have rotated through an angle $\alpha$, the lengths of A, B, C, and D are derived from the geometry of FIG. 11 as follows:

$$A=\sqrt{(K-R\sin\alpha)^2+(K+R\cos\alpha)^2}$$
$$B=\sqrt{(K-R\sin\alpha)^2+(K-R\cos\alpha)^2}$$
$$C=\sqrt{(K+R\sin\alpha)^2+(K-R\cos\alpha)^2}$$
$$D=\sqrt{(K+R\sin\alpha)^2+(K+R\cos\alpha)^2}$$

Comparing the geometrical model to the mechanism shown in FIG. 7, the points M and N represent the axes of rotation of the cranks **196** and **198**, which axes are the shafts **200** of the respective cranks **196** and **198**. The shown in FIG. 11 represent the distances which the centers of the shafts **212** are displaced from the centers of the shafts **200** of the respective cranks **196** and **198**. The INPUT, OUTPUT, and points V, U are shown merely as points on the geometrical model; however, in the embodiment shown in FIG. 7, these elements are shown as gears **192, 232, 216,** and **216,** respectively, all having the same number of teeth and pitch diameter. It follows, then, that the length of the belt **236** shown in FIG. 7 is equal to the various lengths of the belt between the last-named gears plus a length of belt necessary to travel around portions of the same four gears. As all of the four gears named above are of the same size, and as the belt engages one fourth of the circumference of each gear, the length of the belt portion engaging all four gears is equal to the circumference of one of the gears.

In the geometrical model shown in FIG. 11, the belt length (L) is obtained as follows:

$$L=S+A+B+C+D;$$

in which S=the circumference of a gear or sprocket having a radius (r).

Continuing with the geometrical model, and rearranging the equation for the belt length (L), the following expression evolved:

$$L-S=A+B+C+D.$$

From the geometrical model of FIG. 11, an approximation of K in terms of L and S was found by letting $R=0$.

For example, when R=0 in FIG. 11, the belt length A decreases until the point W on the extremity of the radius R coincides with the point N. When the coincidence of the points W and N occurs, the length of A becomes:

$$A=K\sqrt{2}$$

When R=0, the remaining belt lengths B, C, and D also equal $K\sqrt{2}$.

Substituting the approximate values for A, B, C, and D in the formula below:

$$L-S=A+B+C+D$$
$$L-S=K\sqrt{2}+K\sqrt{2}+K\sqrt{2}+K\sqrt{2}$$

yields

$$L-S=4K\sqrt{2}$$

and, rearranging,

$$K=\frac{L-S}{4\sqrt{2}}$$

yields

$$K=.176777\ (L-S)$$

By the above formula, an approximation of K is obtained when L and S are known. In actual practice, the above approximation of K proved satisfactory in determining the crank radius (R) from the following derived formula:

$$R=\frac{r}{4}\sqrt{\frac{32K^2-r^2}{16K^2-r^2}}$$

in which $r$=the gear radius which is equal to the gear radius of the gears **192, 232,** and **216** of FIG. 7.

The following calculation illustrates how the dimensions of the various elements of the driving mechanism shown in FIG. 7 are determined:

(1) Assume a belt length of (L)=10.000 inches.
(2) Assume large input gear **190** having 34 teeth.
(3) Assume gears **192, 232,** and **216** as having:
  (a) radius (r)=.541 inch
  (b) circumference $S=2\pi r=3.400$ inches
  (c) 17 teeth, 1/5 pitch.
(4) Approximate the value of K from the formula:
  K=.17677 (L—S)
  $K=.17677\ (10.000-3.400)$
  $K=1.1667$ inches.
(5) Solve for R (the crank arm) from the formula:

$$R=\frac{r}{4}\sqrt{\frac{32K^2-r^2}{16K^2-r^2}}$$

$$R=\frac{.541}{4}\sqrt{\frac{32(1.1667)^2-(.541)^2}{16(1.1667)^2-(.541)^2}}$$

$$R=.1919\cong.192.$$

(6) Determine the phase shift of the crank arm R for an angle $\beta$ (not shown) with respect to the output shaft; this will be described later. The output shaft of the mechanism delivers inermittent rotary motion with harmonic motion characteristics to transport the cards on the feed belt as previously mentioned. To produce this output motion, the constant rotary motion which is delivered to the input shaft of the drive mechanism is divided into two components. One component produces a motion of constant velocity; the other component produces a motion having harmonic motion characteristics.

In order to produce a dwell in the output shaft which will momentarily position and stop the conveyor so that a card carried thereby will be positioned at the read station as previously explained, it is necessary at some point to have the value of the constant uniform velocity motion (H. V. of FIG. 12) and the value of the harmonic velocity motion (U. V.) cancel or offset each other, as shown in FIG. 12.

In the geometrical model shown in FIG. 11, the offsetting of motions mentioned in the previous paragraph occurs when $\alpha$ is equal to ninety degrees. In the physical embodiment shown in FIG. 7, the dwell in the output shaft **76** occurs when the cranks **196** and **198** are each rotated ninety degrees clockwise from the position shown, so that the axis of each shaft **212** lies directly above the axis of the respective crank **196** and **198**, as viewed in FIG. 7.

When the dwell in the output shaft **76** occurs, the card carried by the feed band **36** is positioned at the read station **24**. Upon proper command from the computer or other utilization device with which the card reader is used, all the data in the card will be read simultaneously, as explained previously. Upon completion of the reading operation, the clutch **170** is actuated, and the card feed mechanism positions the next card to be read at the read station **24**. By the time this next card is positioned at the read station **24**, the computer is ready for reading it, so that the clutch **170**, in essence, remains in the actuated state, and the card feed mechanism alone continues to position the cards to be read at the read station **24**. The means for actuating the clutch **170** is identical to the means for actuating the clutch **170***a*, which is shown in FIG. 13 and which will be described later.

The OUTPUT VELOCITY curve (also shown in FIG. 12) for the output shaft **76**, which drives the feed band **36** in the card feed mechanism, indicates that the card being fed will be gradually accelerated from zero velocity at the card hopper **22** to a maximum OUTPUT VELOCITY of two times input velocity at ninety degrees of INPUT ROTATION, and then it will be gradually decelerated until it comes to a stop under the read station **24**. By this construction, the cards being moved are subjected to a minimum of abrupt changes in velocity while still being fed at rates up to approximately 1,200 cards per minute.

The clutch **170** (FIG. 5) is provided with means (to be later described) to disconnect the input motor **166** from the drive mechanism **28** when the drive mechanism is in the dwell portion of its cycle. Once the drive mechanism **28** is energized, it alone will accurately position the cards being carried by the feed band **36** under the read station **24**, one after another, to be read there.

FIG. 13 shows another embodiment of the drive mechanism of this invention, which is designated generally as **300**, and the clutch **170***a*, which will be later described in more detail.

The drive mechanism **300**, in general terms, comprises frame means for rotatably mounting the input means, which are rotated at a constant rate. There are first connecting means, which operatively connect the input means with the output means, which in turn are also rotatably mounted in said frame means. The first connecting means also include crank members which are rotatably mounted thereon. There are also second connecting means, which operatively connect the crank members with the first connecting means. As the crank members are rotated, they are effective to alter the motion delivered to the output means, so as to produce zero velocity or a dwell at said output means at least once for a predetermined rotation of said input means. The output means are operatively connected to the feed band **36** to drive it, and the dwell produced is effective to position a card, being moved by the feed band, under the read station **24**.

The drive mechanism **300**, shown in FIGS. 13 to 16 inclusive, receives its constant rotational input from the output shaft **302** of the clutch **170***a*, which is similar to the clutch **170** except for the output shaft **302**. The shaft **302** is provided with a portion **304** (square in cross-section), which fits into a mating square recess on a tubular bearing member **306** (FIG. 16).

The bearing member **306** has a reduced portion **307** (square in cross-section), which fits into a mating recess **308** in a carrier **310** and abuts therein against a shoulder **312**, as shown in FIG. 16. The end of the square portion **307** is flattened out to secure the carrier **310** to the bearing **306**, which are both rotated by the shaft **302**.

As the shaft **302** turns, it rotates the carrier **310** and the tubular bearing **306** at a constant rate; the bearing **306** rotates and is supported in a hub **313** of a fixed gear **314**. The hub of the fixed gear **314** is inserted in an opening **316** (FIG. 13) in a planar support member **176***a* and is fixed against rotation therein by a pair of projections **318** on the support member **176***a*, which projections mate with a pair of complementary grooves **320** diametrically positioned on the periphery of the hub **313** of the gear **314**.

The carrier **310** has secured thereto a pair of generally arcuate, planar members **322**, which are maintained in spaced parallel relation to one side of the carrier **310** by spacers **324** and fasteners **326** (FIGS. 13 and 15). The planar members **322** provide support for cranks **328** and **330** and for gears **332** and **334** (FIG. 14), which operatively connect the cranks **328** and **330**, respectively, to the fixed gear **314**.

The cranks **328** and **330** are identical, and each is provided with a shaft **333**, which is rotatably mounted in suitable openings in the carrier **310** and the arcuate planar member **322**, as particularly shown in FIG. 16. A gear **335** is fixed to rotate with the shaft **333**, which is also provided with suitable spacers to provide running clearance for the gear **335** between the carrier **310** and the arcuate planar member **322**.

Each gear **335** is so positioned on the carrier **310** as to be out of mesh with the teeth of the fixed gear **314**. As mentioned earlier, each of the gears **335** for the cranks **328** and **330** utilizes a separate gear **332** and **334**, respectively, to operatively connect it with the fixed gear **314**, the gears **335** being rotatably supported between the carrier **310** and the arcuate planar members **322**.

When the clutch **170***a* is actuated, its shaft **302** rotates the carrier **310** counter-clockwise (as viewed in FIG. 13) about the fixed gear **314**. As the carrier **310** rotates, the gears **332** and **334**, carried thereby, are rotated and, in turn, drive the gears **335** of the cranks **328** and **330**, respectively. Each of the shafts **333** of the cranks **328** and **330** is provided with a clearance shoulder **338** (FIG. 16) and a reduced end portion **340** (square in cross-section), which is inserted into a complementary square opening in one end of a crank arm **342** to thereby rotate the crank arm. Each of the crank arms **342** has an annular recess, into which a screw **344** is positioned to secure one end of the crank arm **342** to the pertaining shaft **333**.

The remaining end of each crank arm **342** is pivotally joined to one end of a separate drag link **346** by a pin **348**, as shown in FIGS. 13 and 15. The remaining ends of the drag links **346** are pivotally secured to opposed ends of an output drive plate **350** by a pin **352**. The drive plate **350** is secured at a right angle to an output shaft **354**, and its axis lies on a line midway between the pins **352**. The output shaft **354** is supported for rotation in a support member **356**, as shown in FIGS. 13 and 16, and it is in axial alignment with the shaft **302** of the clutch **170***a*.

The clutch **170***a* may be of a conventional type or of a type which has a rotating shell **360**, which rotates counter-clockwise, as viewed in FIG. 13, to drive the clutch output shaft **302** when the clutch is energized. The clutch **170***a* is provided with a pair of equally-spaced abutment stops **362** on its periphery, which are engaged by one leg **364** of a pawl **366**, which is pivotally supported on a shaft **368** secured to a planar support member **370**.

The remaining leg, **372**, of the pawl **366** is spring-urged clockwise (FIG. 13) by a spring **374**.

Whenever a group of cards is to be read, a solenoid **376** is energized to drive an operating plunger **378** upwardly, as viewed in FIG. 13, to move the pawl **366** counter-clockwise, thereby disengaging the leg **364** from the abutment stop **362**, which in turn permits the input motor **166** (FIG. 1) to drive the clutch **170***a*. As long as the pawl **366** is disengaged from the abutment stops **362**, the clutch is effective to rotate the clutch shaft **302** at a constant rate counter-clockwise, as viewed in FIG. 13, and the drive mechanisms disclosed herein are effective to convert constant rotational input to intermittent variable-speed rotary motion output to drive the feed band **36**, as previously explained.

The drive mecahnism **300** is a mechanical motion summing device of two separate motions, which are derived from a common source of constant rotational input (shaft **302**, FIG. 13). One of the motions is a constant rotary motion supplied by the carrier **310** and linkage comprising the cranks **328** and **330**, the links **346**, and the output drive plate **350**. As long as the cranks are not rotated relative to the carrier **310**, the output shaft **354** will be driven at a constant rate; however, when the cranks **328** and **330** are driven by the coupling of the gears **335** to the fixed gear **314**, the motion at the output shaft **354** becomes a variable-speed intermittent rotary motion with the necessary dwells for intermittently driving the feed band **36**, which positions the cards carried thereby under the read station **24**.

A geometrical description of the drive mechanism **300** is illustrated in the geometrical model shown in FIG. 17. While the drive mechanism shown in FIGS. 13 to 16 inclusive is shown as having a stationary gear with external teeth, the mechanism could employ a stationary gear having internal teeth, with the gears driving the cranks in mesh with the internal teeth.

There are certain relationships which exist among the various elements shown in the geometrical model in FIG. 17 and the embodiment shown FIGS. 13 to 16 inclusive. They are:

(a) The pitch diameter of the fixed gear **314** (FIGS. 13 to 16 inclusive) is twice that of the planet gears **332** and **334**, which revolve around the fixed gear **314**.

(b) The distance R (FIG. 17) may be any selected length and represents the distance between the axes of rotation of the carrier **310** and the crank arms **342** (FIGS. 13 to 16 inclusive).

(c) The distance *r* (FIG. 17) is less than R but greater than zero and represents the length of the crank arms **342** (FIGS. 13 to 16 inclusive).

(d) The length T (FIG. 17) is equal to 2r and represents the length of the drive plate **350** (FIGS. 13 to 16 inclusive) as measured between the axes of the ontput shaft **354** and the pins **352**.

(e) The length S (FIG. 17) is equal to $\sqrt{R^2-r^2}$ and represents the length of the drag links **346** (FIGS. 13 to 16 inclusive).

f) The input and output shafts of FIG. 17 are located at M, and the input shaft **302** and the output shaft **354** in the embodiment of FIGS. 13 to 16 inclusive are on a common center line of rotation.

(g) The input carrier **310** (FIGS. 13 to 16 inclusive) is rotated at a constant speed.

FIG. 17 is a geometrical model of the second embodiment of the drive mechanism **300**, which is shown principally in FIG. 13, and, as previously stated, the elements shown in FIG. 17 have counterparts in the physical embodiment shown in FIG. 13. For example: as the input carrier R (carrier **310**) is rotated counter-clockwise, as viewed in FIG. 17, the crank arm *r* (arm **342**) is rotated clockwise about the outer end *q* of R; and the point Q, the input, will trace the elliptical path shown by the long dash lines **380**. After a certain angular rotation of the input carrier R, the model shown in FIG. 17 will assume the general configuration shown in FIG. 18.

The output of the drive mechanism (FIG. 17) is represented by the point C, which travels in the circular path shown by the short dash lines **382**. As the input carrier R an dthe smaller crank arm r are rotated, a condition will be reached during which the clockwise rotation of the crank arm *r* will oppose or offset the counter-clockwise rotational effect on the point C caused by the input carrier R. This offseting of motions will cause the point C to become stationary or to dwell during a portion of a revolution of the input carrier R. This dwell is utilized to stop the feed band **36**, so that a card carried thereby will be accurately positioned under the read head **24**.

During the dwell or the time that the point C is stationary, the drag link S may be considered as rotating about the point C, as represented by the circular path **384** in FIG. 17. The extent of the dwell is determined by that succession of points defining a substantially common path (C-P) or arc for the point Q as the point Q travels along the elliptical path **380** and the circular path **384**, as shown in FIG. 17. This common path C-P in FIG. 17 occurs for an angle of about thirty degrees of input rotation, and, during this time, the common portions of both the elliptical and circular paths mentioned match each other to within .001 inch, which is well within the normal variables induced by practical machining tolerances of the parts involved in the actual embodiment shown in FIGS. 13 to 16 inclusive.

The continuation of the elliptical and circular curves above and below the range shown by the common path C-P of FIG. 17 indicates a smooth advance into and away from the dwell position with a minimum of abrupt changes in force or direction.

From a graphical solution, the optimum values of the relationship among R, *r*, S, and T (FIG. 17) are believed to be the following:

When

$$R=\text{any value}$$
$$r=(\sqrt{2}-1)R$$
$$T=(2\sqrt{2}-2)R$$
$$S=\sqrt{R^2-r^2}$$

When

$$R=1$$
$$r=.4142$$
$$T=.8284$$
$$S=.91018$$

From the above, the values of *r*, T, and S develop as proportionality constants, so that, for a given radius R, these values can be readily determined.

The values of *r*, T, and S are not necessarily restricted to the above proportionality constants; however, when they are changed, the output displacement of the drive mechanism will also change. In order to increase the versatility of the drive mechanism, displacement prediction formulas were developed, so as to predict hte output displacement with different values of R, *r*, T, and S.

The formulas used for obtaining output displacement of the drive mechanism with different combinations of values for the elements R, *r*, T, and S are derived in connection with the geometrical model shown in FIGS. 17 and 18, with the point Q of FIG. 17 being rotated clockwise (when viewed as being on the crank arm *r*) to the position shown in FIG. 18. The angle $\beta$ is the included angle between R and T. The angle $\alpha$ is the included angle between *r* and R. The angle $\phi$ is the included angle between R and a line P drawn from the center of rotation M to the point Q on the end of *r*, and the angle $\eta$ is the included angle between T and the line P.

From the point Q in FIG. 18, a line *b* is drawn perpendicular to R at Z to produce the short segment A,

which is equal to $r \cos \alpha$, while the line $b$ is equal to $r \sin \alpha$. Referring to the triangle formed by Q, Z, and M,

Equation 1 $P^2=b^2+(R-A)^2$

Equation 2 $P^2=(r \sin \alpha)^2+(R-r \cos \alpha)^2$

Referring to the triangle formed by X, Z, and Q,

Equation 3 $r^2=A^2+b^2$

Equation 4 $b^2=r^2-A^2$

From Equation 1, $P^2=b^2+(R-A)^2$, and substituting Equation 4 above in Equation 1,

$$P^2=(r^2-A^2)+(R-A)^2$$

Equation 5 yields: $P^2=R^2-2AR+r^2$.

The value of $A=r \cos \alpha$ when substituted in Equation 5 yields:

Equation 6 $P^2=R^2+r^2-2rR \cos \alpha$

From the law of cosines,

Equation 7 $$\cos \eta=\frac{P^2+T^2-S^2}{2PT}$$

and

Equation 8 $$\cos \phi=\frac{R^2+P^2-r^2}{2PR}$$

Substituting the value of $P^2$ determined in Equation 6, and $P=\sqrt{P^2}$, into Equation 7 yields:

Equation 9

$$\cos \eta=\frac{R^2+r^2+T^2-S^2-2Rr \cos \alpha}{2T\sqrt{R^2+r^2-2Rr \cos \alpha}}$$

Similarly, substituting the value of $P^2$ determined in Equation 6, and the value of $P=\sqrt{P^2}$, into Equation 8, yields after simplification:

Equation 10

$$\cos \phi=\frac{R-r \cos \alpha}{\sqrt{R^2+r^2-2Rr \cos \alpha}}$$

Referring to FIG. 17, the following relationships exist:

(a) R may be any value not zero.

(b) The crank arm $r$ is greater than zero but less than R.

(c) T is equal to $2r$.

(d) S is equal to $\sqrt{R^2-r^2}$.

(e) The output is equal to $(\theta-B)$.

(f) The angle $\beta$ is equal to $(\phi+\eta)$.

The values of R, $r$, T, and S selected for a specific physical embodiment would naturally be dependent upon the particular application in which the drive mechanism is used. In the embodiment shown in FIGS. 13 to 16 inclusive, the values of R and $r$ were selected to be one inch and .4142 inch, respectively, with the values of T and S determined to be .8284 inch and .91018 inch, respectively. With these values, smooth accelerations and decelerations were obtained. While the drive mechanism **300** technically has one instantaneous zero point, the mechanism has a practical or effective dwell for about thirty degrees of input rotation.

The method of securing the drive mechanism **300** to the clutch **170***a* (FIG. 13) so as to obtain the desired dwell is as follows. The square end **304** of the clutch shaft **302** is inserted into the mating recess **308** of the carrier **310**, and the carrier is rotated counter-clockwise (as viewed in FIG. 13) until an abutment stop **362** on the rotating shell **360** of the clutch **170***a* engages the leg **364** of the pawl **366** to stop the carrier **310**. Each crank arm **342** is then positioned with respect to an imaginary line joining the axes of the cranks **328** and **330**, so that the included angle therebetween is equal to 122.4 degrees. In the geometrical model shown in FIG. 17, the included angle of the previous sentence is equal to $\alpha$, which is the included angle between $r$ and R. In the drive mechanism **300**, the crank arms **342** are positioned on opposite sides of the imaginary line previously mentioned as shown in FIG. 13. Once the angle $\alpha$ is set for each crank **328** and **330**, the carrier **310** is pushed towards the clutch **170***a*, so that the gears **332** and **334** engage the teeth on the ring gear **314** and thereby operatively connect the cranks **328** and **330** respectively to the fixed gear **314**.

After the drive mechanism **300** is connected to the clutch **170***a* in this manner, the output shaft **354** is then connected to the shaft **68** of the conveyor **26** via the connector **78** (FIG. 6), so that a card carried by the feed belt **36** will be positioned at the reading station **24** in reading relationship therewith. Upon command from the computer or other utilization device with which the card read mechanism is associated, the solenoid **376** (FIG. 13) will be energized to begin the cycle for the drive mechanism **300**, and the card will be read while at the read station **24**. At the completion of the read operation, the drive mechanism **300** moves the feed band **36** to deliver another card at the read station **24**. In general, the computer can receive the data faster than the card feed mechanism can feed the cards to the read station **24**, so that the computer will be waiting to receive the data in the next card. If the computer is not ready to receive the data in the next card to be read, an abutment stop **362** (FIG. 13) on the rotating shell **360** of the clutch **170***a* engages the leg **364** on the pawl **366** to stop the drive mechanism **300**, and at this time the card to be read will be positioned at the read station **24**. If the computer is ready to receive the data, the solenoid **376** of the pawl **366** will be energized to clear the leg **364** from the stop **362**, and the rotating shell **360** of the clutch **170***a* will not be stopped thereby. The dwells produced by the drive mechanism **300** will then alone be effective to position the card to be read at the read station **24**.

A third embodiment of the drive mechanism, designated generally as **400**, is shown in FIGS. 19 and 20 and is similar in construction to the drive mechanism **300** shown in FIGS. 13 to 16 inclusive; however, the drive mechanism **400** is generally hypocyclic, whereas the drive mechanism **300** is epicyclic.

The drive mechanism **400** includes an input shaft **402** (FIG. 20), which is rotatably mounted in a bearing **404**, which is retained in a shouldered bearing sleeve **406**, which in turn is mounted in an opening in a circular housing plate **408**. The end **410** of the shaft **402** is square in cross-section and fits into a complementary square opening in an input carrier **412**, and a fastener (not shown) secures the carrier **412** to the shaft **402** to rotate therewith.

The output shaft **414** of the drive mechanism **400** is rotatably mounted in a bearing **416**, which is mounted in a shouldered bearing sleeve **418**, which in turn is mounted in an opening in a circular housing plate **420**. The end portion **422** of the shaft **414** is square in cross section and fits into a complementary square opening in an output drive plate **424**, and a fastener (not shown) secures the output drive plate **424** to the shaft **414** to rotate therewith.

The input shaft **402** of the drive mechanism **400** of FIG. 20 is driven by a clutch (not shown) which is identical to the clutch **170***a* shown in FIG. 13 in connection with the second drive mechanism **300**. As the input shaft **402** is rotated at a constant rate, the carrier **412** is also rotated and carries with it the cranks designated generally as **426** and **428**. The cranks **426** and **428** are identical, and their axes of rotation are equidistantly spaced from the axis of rotation of the shaft **402**. Each crank **426** and **428** (FIGS. 19 and 20) has a shaft **430**, which is rotatably mounted in bearings **432** and **434**, which are secured in aligned apertures in the input carrier **412** and the respective planar support member **436**, respectively. Each support member **436** is secured to the input carrier **412** and is maintained in spaced parallel relationship therewith by spacer studs **438**.

Each crank **426** and **428** has a gear **440** fixed to rotate with its respective shaft **430**, and the gear is positioned between spacer discs **442**, which in turn are positioned be-

tween the carrier **412** and the respective support member **436**, as shown in FIG. 20. Each gear **440** is in mesh with an internally toothed ring gear **444**, which has circular spacer rings **446** and **448** (FIG. 20) (square in cross-section) positioned on opposed sides thereof. The circular housing plates **408** and **420**, the spacer rings **446** and **448**, and the ring gear **444** all have aligned holes spaced along their perimeters, through which screws **450** (FIG. 19) are inserted to secure the named units together and form the housing for the drive mechanism **400**.

The shafts **430** (FIG. 20) of the cranks **426** and **428** are each provided with a cam shaft **452** extending from one end thereof, the axis of which is offset from the axis of its respective shaft **430** by a distance $r$ (not shown), which represents the length of the crank arm for the crank members. The geometry of the drive mechanism **400** will be later discussed in relation to the geometrical model shown in FIG. 21.

The cam shafts **452** mentioned above are operatively connected to the output shaft **414** as follows. Each cam shaft **452** is provided with a bearing **454**, which is inserted over the end thereof and abuts against a portion of the end of the shaft **430**, as shown in FIG. 20. A drag link **456** is provided for each crank **426** and **428**, each link having in one end thereof a hole which fits over the bearing **454** of the respective cranks **426** and **428**. The remaining ends of the drag links **456** are pivotally secured to the drive plate **424** by screws **458** (FIG. 19) at points which are equidistantly spaced from the axis of rotation of the drive plate **424** and which are positioned along a diametral line thereof.

When the input shaft **402** of the drive mechanism **400** (FIGS. 19 and 20) is rotated at a constant rate, intermittent rotary motion is produced at the output shaft **414** in the same general manner as was produced by the drive mechanism **300** shown in FIGS. 13 to 16 inclusive. As the input shaft **402** is rotated, the input carrier **412** also rotates counter-clockwise, as viewed in FIG. 19. While the cranks **426** and **428** are being carried counter-clockwise with the carrier **412**, the gears **440** thereof which are in mesh with the ring gear **444** are rotated clockwise about the axis of their respective shafts **430**, thereby rotating the cranks **426** and **428** clockwise.

If the gears **440** of the cranks **426** and **428** were temporarily disconnected from the ring gear **444**, and if the input shaft **402** (FIG. 20) were rotated at a constant rate, then the output drive plate **424** would also be rotated at a constant rate by the carrier **412** and the drag links **456**. However, when the gears **440** of the cranks **426** and **428** are in mesh with the ring gear **444**, as shown in FIGS. 19 and 20, the clockwise rotation of the cranks **426** and **428**, as mentioned in the previous paragraph, will oppose or offset the counter-clockwise rotation imparted to the output drive plate **424** by the input carrier **412** and cause a dwell in the output drive plate **424** and the output shaft **414**. As previously explained, a dwell in the output shaft such as the shaft **414** is effective to stop the feed band **36**, so that a card carried thereby will be momentarily positioned at the reading station **24**, where the data in the card will be read.

FIG. 21 represents a geometrical model of the drive mechanism **400** shown in FIGS. 19 and 20, which model was used for determining certain mathematical relationships among the various elements of the drive mechanism **400** as follows. The length R of FIG. 21 represents the length of the input carrier **412** (FIG. 20) as measured between the axes of rotation of the input shaft **402** and the shaft **430** of one of the cranks **426** and **428**. The length $r$ of FIG. 21 is the crank member and represents the distance which the axis of the cam shaft **452** is offset from the axis of the shaft **430**, as shown in FIG. 20. The length S of FIG. 21 represents the length of a drag link **456** (FIG. 19) as measured between the centers of its mounting holes on opposed ends thereof. The length T of FIG. 21 represents the length of the output drive plate **424** (FIG. 19) as measured from its axis of rotation to the axis of the screw **458**, which pivotally secures the last-named drag link **456** to the drive plate **424**. The INPUT to the geometrical model rotates R counter-clockwise, as viewed in FIG. 21, and for an input angle of $\theta$, the OUTPUT will be rotated counter-clockwise for an angle of $\beta$. Both R and T are rotated about O (FIG. 21), and $r$ is rotated clockwise about Q from the position shown in FIG. 21 as R is rotated counter-clockwise. One end of the link S is pivotally joined to one end of $r$, and the other end of the link S is pivotally joined to one end of T, as shown in FIG. 21.

To simplify the construction, the means for rotating about point Q on R (FIG. 21) are not shown; however, in the drive mechanism **400** (FIGS. 19 to 20) these means primarily include the ring gear **444** and the gears **440** of the cranks **426** and **428**, which are carried on the input carrier **412**. The number of cycles per input revolution in the drive mechanism **400** is dependent upon the number of teeth in the gears **400** relative to the number of teeth in the ring gear **444**.

In the geometrical model (FIG. 21), N equals the number of cycles per input revolution; also, R is greater than $r$, which is greater than zero. As a practical relationship, $r$ should be less than $R/N$.

From the geometry of the model of FIG. 21,

Equation 11 $$x=R\sin\theta-r\sin\alpha$$

and

Equation 12 $$y=R\cos\theta+r\cos\alpha$$

Given an input rotation of $\theta$ and an output of $\beta$ and from $\beta=\rho-x$; the following values for $x$ and $\rho$ of FIG. 21 can be determined from the following equations:

$$\psi=\arccos\frac{R^2+r^2+T^2-S^2+2Rr\cos(N\theta)}{2T\sqrt{R^2+r^2+2Rr\cos(N\theta)}}$$

and

$$\rho=\arccos\frac{R\cos\theta+R\cos\alpha}{\sqrt{R^2+r^2+2Rr\cos(N\theta)}}$$

where:

$$\alpha=(N-1)\theta$$

$$T=Nr$$

and

$$S=\sqrt{R^2-(T-r)^2}$$

The Equations 11 to 14 inclusive mentioned previously can be used for either epicyclic-type crank members, as shown in the drive mechanism **300**, or hypocyclic-type crank members, as shown in the drive mechanism **400**. These equations are especially adaptable for computer programming to determine output rotation, velocity, acceleration, and torque on the output shaft of the drive mechanisms **300** and **400** with respect to a given input rotation $\theta$.

The drive mechanism **400** may be connected to the clutch **170***a* shown in FIG. 13 by the technique previously discussed in relation to the drive mechanism **300**.

What is claimed is:

**1.** A mechanism for producing intermittent rotary motion comprising:

frame means;

input means having constant rotation and being rotatably mounted in said frame means;

output means rotatably mounted in said frame means;

first connecting means to operatively connect said input means with said output means so as to supply a constant rotational motion to said output means, and including crank means;

and second connecting means to operatively connect said crank means with said input means to thereby rotate said crank means;

said crank means, when rotated, being effective to alter the motion of said first connecting means so as to produce zero velocity at said output means relative to said input means at least once for a predetermined rotation of said input means;

said second connecting means including belt means.

2. A mechanism for producing intermittent rotary motion comprising:

frame means;

input means having constant rotation and being rotatably mounted in said frame means;

output means rotatably mounted in said frame means;

first connecting means to operatively connect said input means with said output means so as to supply a constant rotational motion to said output means, and including crank means;

and second connecting means to operatively connect said crank means with said input means to thereby rotate said crank means;

said crank means, when rotated, being effective to alter the motion of said first connecting means so as to produce zero velocity at said output means relative to said input means at least once for a predetermined rotation of said input means;

said first connecting means including belt means, and said crank means comprising two crank members with each said crank member having an axis of rotation which is equidistantly spaced from the axis of rotation of said input means and said output means.

3. The mechanism as claimed in claim 2 in which said crank members are operatively connected to said belt means so as to be rotated at the same angular velocity and displacement relative to a line joining the axes of rotation of said crank members.

4. A mechanism for producing intermittent rotary motion comprising:

frame means;

input means having constant rotation and being rotatably mounted in said frame means;

output means rotatably mounted in said frame means;

first connecting means to operatively connect said input means with said output means so as to supply a constant rotational motion to said output means, and including crank means;

and second connecting means to operatively connect said crank means with said input means to thereby rotate said crank means;

said crank means, when rotated, being effective to alter the motion of said first connecting means so as to produce zero velocity at said output means relative to said input means at least once for a predetermined rotation of said input means;

said input means having constant rotation comprising first and second input members rotatably mounted in said frame means and fixed to rotate together on a common axis;

said output means comprising a shaft rotatably mounted in said frame means and having an axis of rotation spaced from and parallel to said common axis, and an output member fixed to rotate with said shaft;

said crank means of said first connecting means comprising first and second crank members rotatably mounted in said frame means with each crank member having an axis of rotation which is equidistantly spaced from said common axis and said axis of said shaft of said output means;

each said crank member having a crank shaft eccentrically positioned thereon and a rotatable member rotatably mounted on said crank shaft;

said first connecting means further comprising first belt means operatively connecting said input member, said rotatable members of said first and second crank members, and said output member to form a closed loop therearound;

said second connecting means comprising second belt means operatively connecting said second input member with said first and second crank members so as to rotate said crank members in timed relation with said first input member.

5. A mechanism for producing intermittent rotary motion comprising:

frame means;

input means having constant rotation and being rotatably mounted in said frame means;

output means rotatably mounted in said frame means;

first connecting means to operatively connect said input means with said output means so as to supply a constant rotational motion to said output means, and including crank means;

and second connecting means to operatively connect said crank means with said input means to thereby rotate said crank means;

said crank means, when rotated, being effective to alter the motion of said first connecting means so as to produce zero velocity at said output means relative to said input means at least once for a predetermined rotation of said input means;

said first connecting means comprising a carrier means fixed to said input means to be rotated thereby and link means operatively connecting said carrier means with said output means and including said crank means rotatably mounted on said carrier means.

6. A mechanism for producing intermittent rotary motion comprising:

frame means;

constant rotational input shaft means, and output shaft means rotatably mounted in said frame means;

first connecting means to connect said input shaft means with said output shaft means so as to supply a constant rotational motion to said output shaft means and comprising:

carrier means secured to said input shaft means to be rotated thereby,

link connecting means operatively connecting said carrier means with said output shaft means so as to rotate the latter at a constant speed and including crank means rotatably mounted on said carrier means,

second connecting means operatively connecting said crank means with said input shaft means and comprising:

a stationary gear fixed to said frame means and having the axis thereof concentric with the axis of said input shaft means,

gear means carried by said carrier means and operatively connected to said crank means so as to rotate said crank means as said carrier means is rotated relative to said stationary gear,

said second connecting means being effective to alter the motion of said link connecting means so as to produce zero velocity at said output shaft means relative to said input shaft means at least once for a predetermined rotation of said input shaft means.

7. The mechanism as claimed in claim 6 in which said input shaft means and output shaft means are axially aligned in said frame means.

8. The mechanism as claimed in claim 6 in which the gear means carried by said carrier means and said stationary gear form an epicyclic connection; said stationary gear having external teeth operatively connected to said gear means.

9. The mechanism as claimed in claim 6 in which the gear means carried by said carrier means and said stationary gear form a hypocyclic connection; said stationary gear having internal teeth operatively connected to said gear means.

10. A mechanism for producing intermittent rotary motion comprising:

frame means,

constant rotational input shaft means, and output shaft means rotatably mounted in said frame means,

carrier means perpendicular to the axis of said input shaft means and fixed thereto to be rotated thereby,

a pair of crank means diametrically positioned on opposed ends of said carrier means and rotatably mounted thereon,

each said crank means having a crank shaft whose axis is parallel to and is displaced from the axis of its respective said crank means,

said output shaft means having an output shaft and a drive plate perpendicular to the axis of rotation of said output shaft and fixed thereto to be rotated therewith,

a drag link for each said crank means, said drag link having one end pivotally joined to said crank shaft of its respective crank means, the remaining ends of said drag links being pivotally joined to said drive plate at opposed points of a diametral line thereon,

a stationary gear fixed to said frame means and having an axis concentric with the axis of said input shaft means,

connecting gear means carried by said carrier means and operatively connecting said stationary gear with said crank means so as to rotate said crank means as said carrier means is rotated, said crank means, when rotated, being effective to alter the constant rotary motion imparted by said carrier means so as to produce zero velocity at said output shaft means at least once for a predetermined rotation of said input shaft means.



**11.** The mechanism as claimed in claim **10** in which:

each said crank means has a rotating axis which is displaced from the axis of said input shaft means by a distance R,

each said crank shaft has an axis which is displaced from the axis of its respective crank means by a distance *r*,

the length of said drag links is equal to S,

said remaining ends of said drag links are secured to said drive plate at a distance T from the axis of rotation of said output shaft means,

and R is any length not zero, *r* is greater than zero but less than R, and S [T] is equal to $\sqrt{R^2-r^2}$.

**12.** The mechanism as claimed in claim **11** in which said stationary gear and said connecting gear means form an epicyclic connection; said stationary gear having external teeth operatively connected to said gear means.

**13.** The mechanism as claimed in claim **11** in which said stationary gear and said connecting gear means form a hypocyclic connection; said stationary gear having internal teeth operatively connected to said gear means.


**References Cited**

UNITED STATES PATENTS

| | | | |
|---|---|---|---|
| 2,399,493 | 4/1946 | Luehrs et al. | 74—394 |
| 2,852,954 | 9/1958 | Hobbs | 74—394 |
| 3,015,236 | 1/1962 | Toensing | 74—84 |
| 3,287,984 | 11/1966 | Neubauer | 74—84 |


MILTON KAUFMAN, *Primary Examiner.*

# U.S. DEPARTMENT OF COMMERCE

## PATENT OFFICE

## Washington, D.C. 20231

# UNITED STATES PATENT OFFICE

# CERTIFICATE OF CORRECTION

Patent No. 3,407,678 October 29, 1968

James M. Steinke

It is certified that error appears in the above identified patent and that said Letters Patent are hereby corrected as shown below:

Column 5, lines 74 and 75, "areas 148 are arranged in the read head 144 in an array which permit light to pass therethrough. These window" should read -- which permit light to pass therethrough. These window areas 148 are arranged in the read head 144 in an array --. Column 9, line 55, cancel "of the respective cranks 196 and 198. The" and insert -- , as shown in FIG. 8a. The crank arms R --. Column 18, line 36, to the left of the equation insert -- Equation 13 --; line 40, to the left of the equation insert -- Equation 14 --.

Signed and sealed this 10th day of March 1970.

(SEAL)

Attest:

Edward M. Fletcher, Jr.
Attesting Officer

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Commissioner of Patents