United States Patent

Gladow et al.

[15] **3,691,871**

[45] **Sept. 19, 1972**

[54] **ROTARY MOTION TRANSMITTING APPARATUS**

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[52] **U.S. Cl.** **74/798,** 33/172, 74/89.2

[51] **Int. Cl.** **F16h 13/00,** G01b 3/22, F16h 25/16

[58] **Field of Search**.........74/798, 89.2, 89.21, 89.22, 74/89; 308/6 R

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[57] **ABSTRACT**

Apparatus for transmitting rotary motion between two relatively movable members. The apparatus includes an elongated thin flexible element in the form of a band arranged in a pair of adjoining alternate loops with a roller confined within each of the loops. The rollers roll along a circular path as the band progressively passes around the rollers. The rollers and the band are supported on a cylindrical guide surface and the rate of motion of the components of the apparatus relative to each other permits a high degree of amplification or reduction between input and output elements. As an alternative usage, the apparatus readily lends itself to two inputs in a manner equivalent to a geared differential.

19 Claims, 16 Drawing Figures

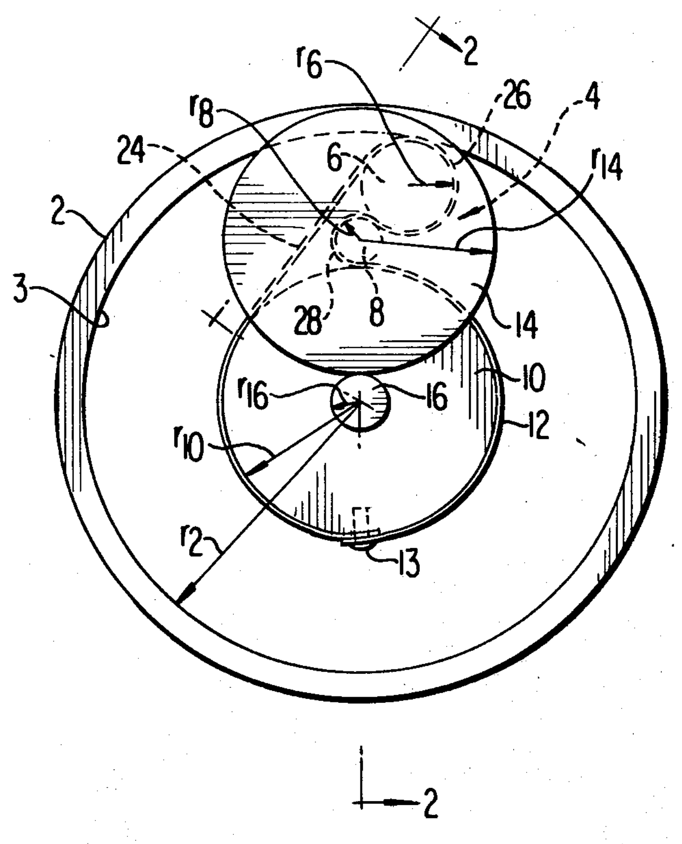


FIG. 1

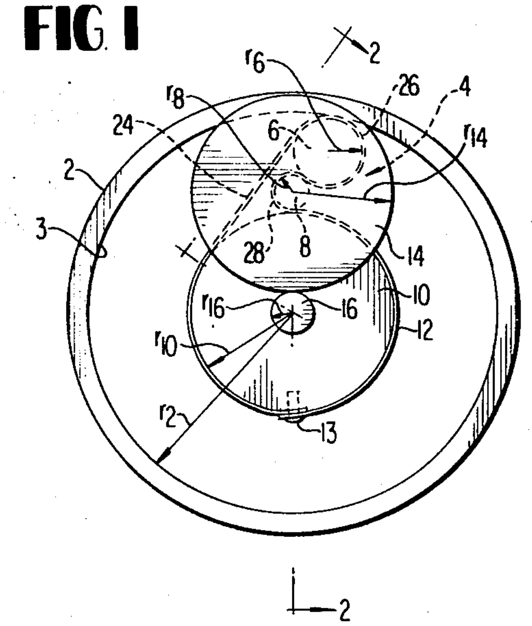
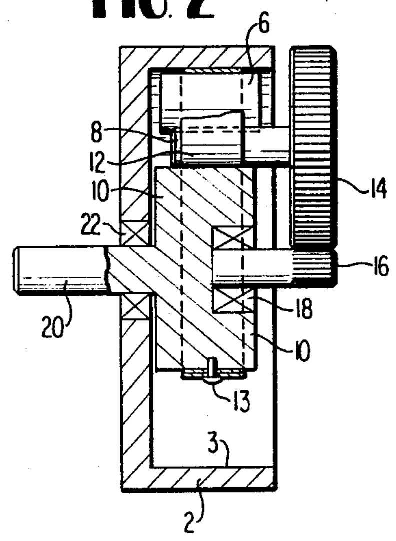


FIG. 2



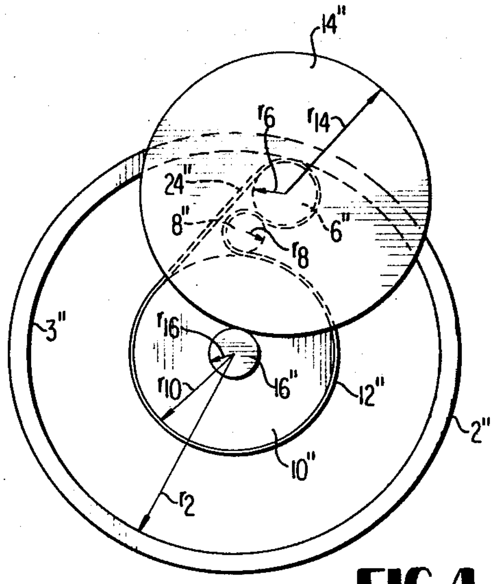
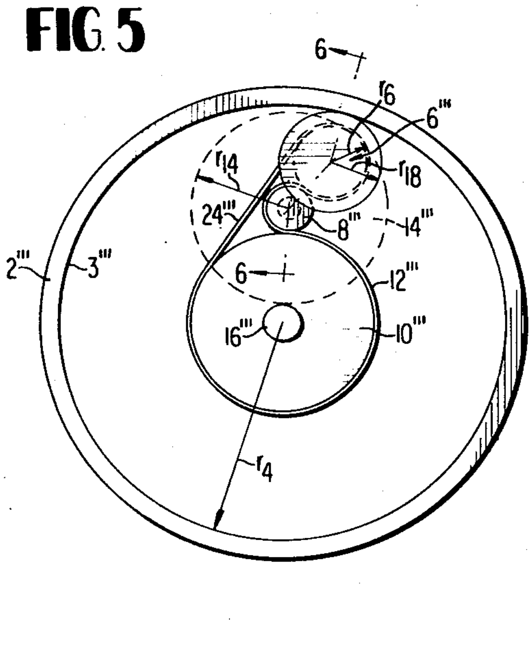


FIG. 4

FIG. 5



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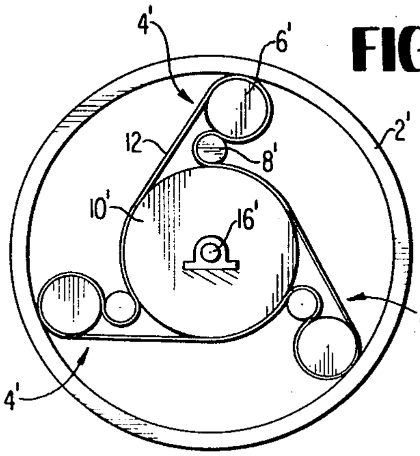


FIG. 3

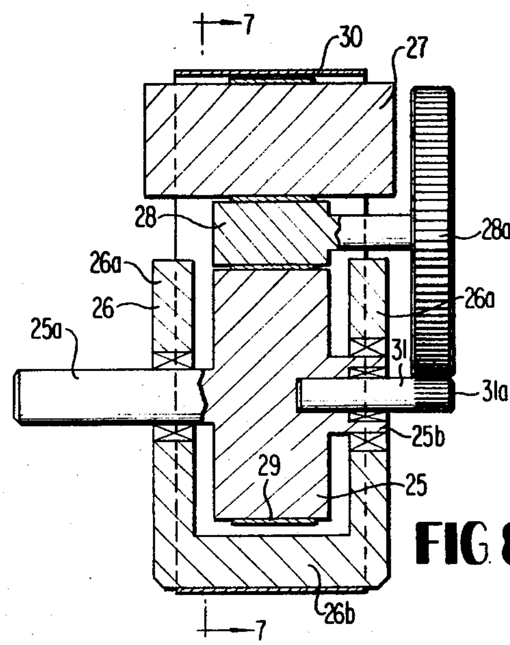


FIG. 8

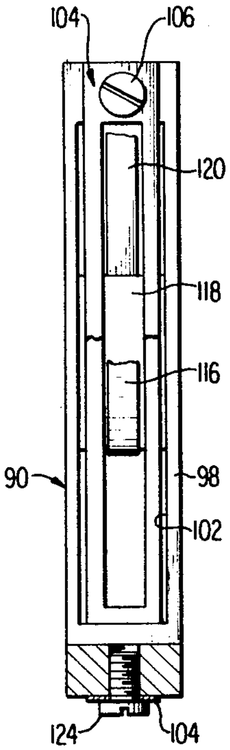


FIG. 13

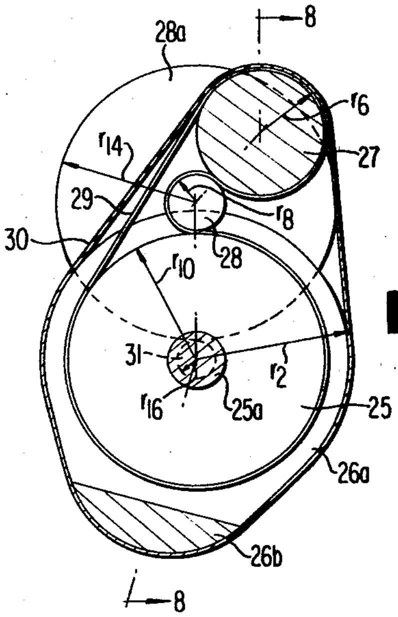


FIG. 7

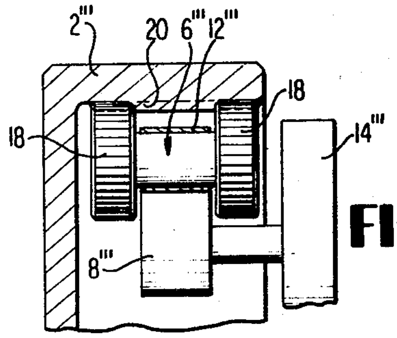
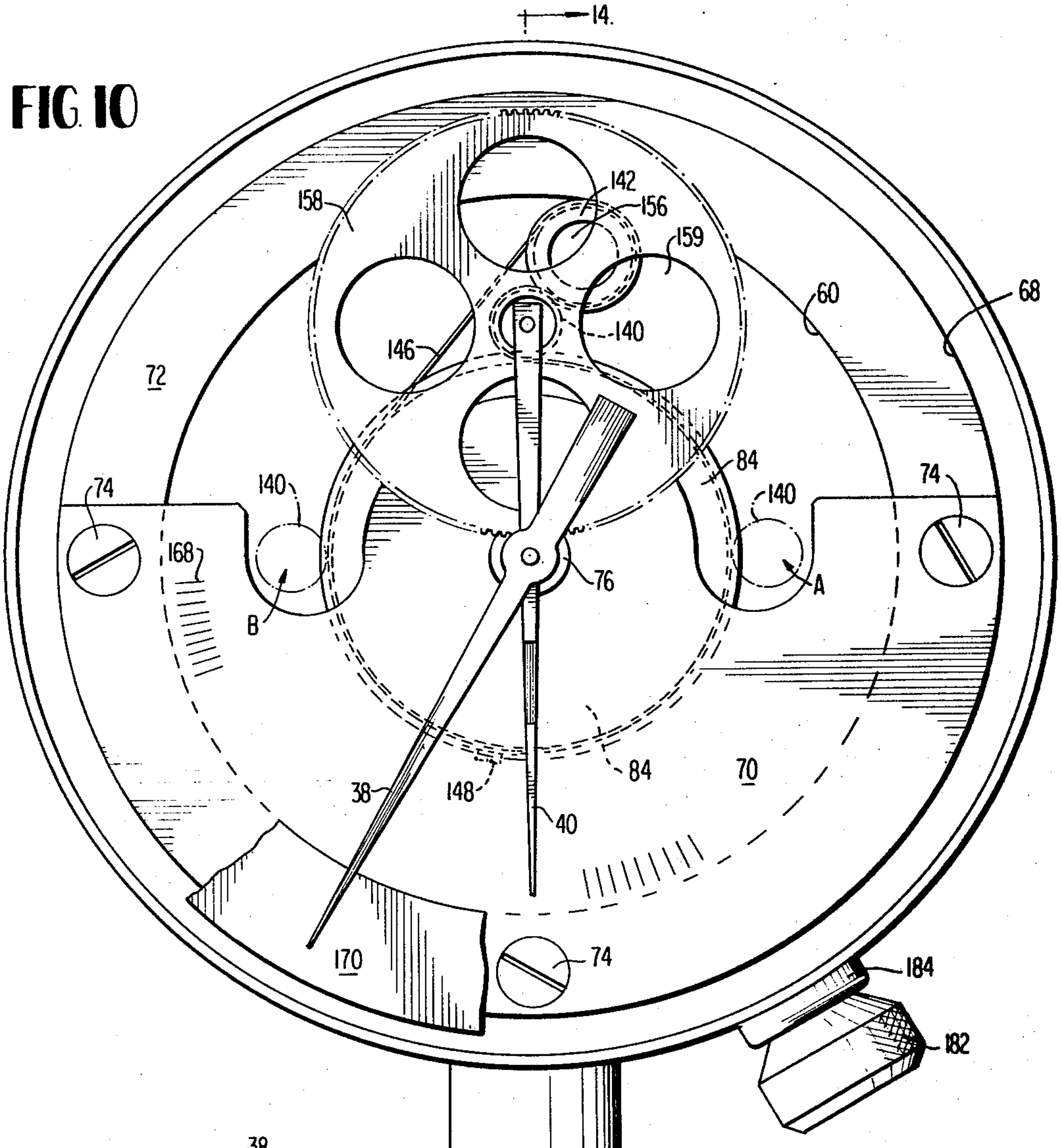


FIG. 6

FIG 10



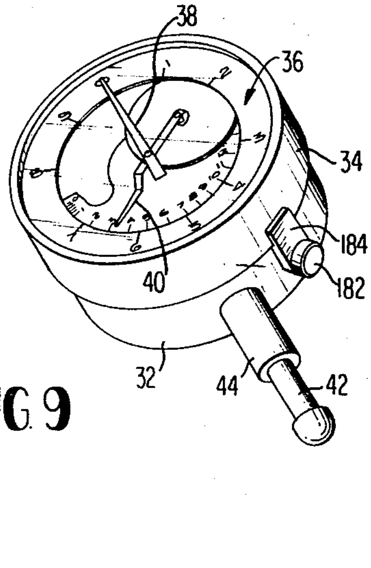


FIG 9

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FIG 12

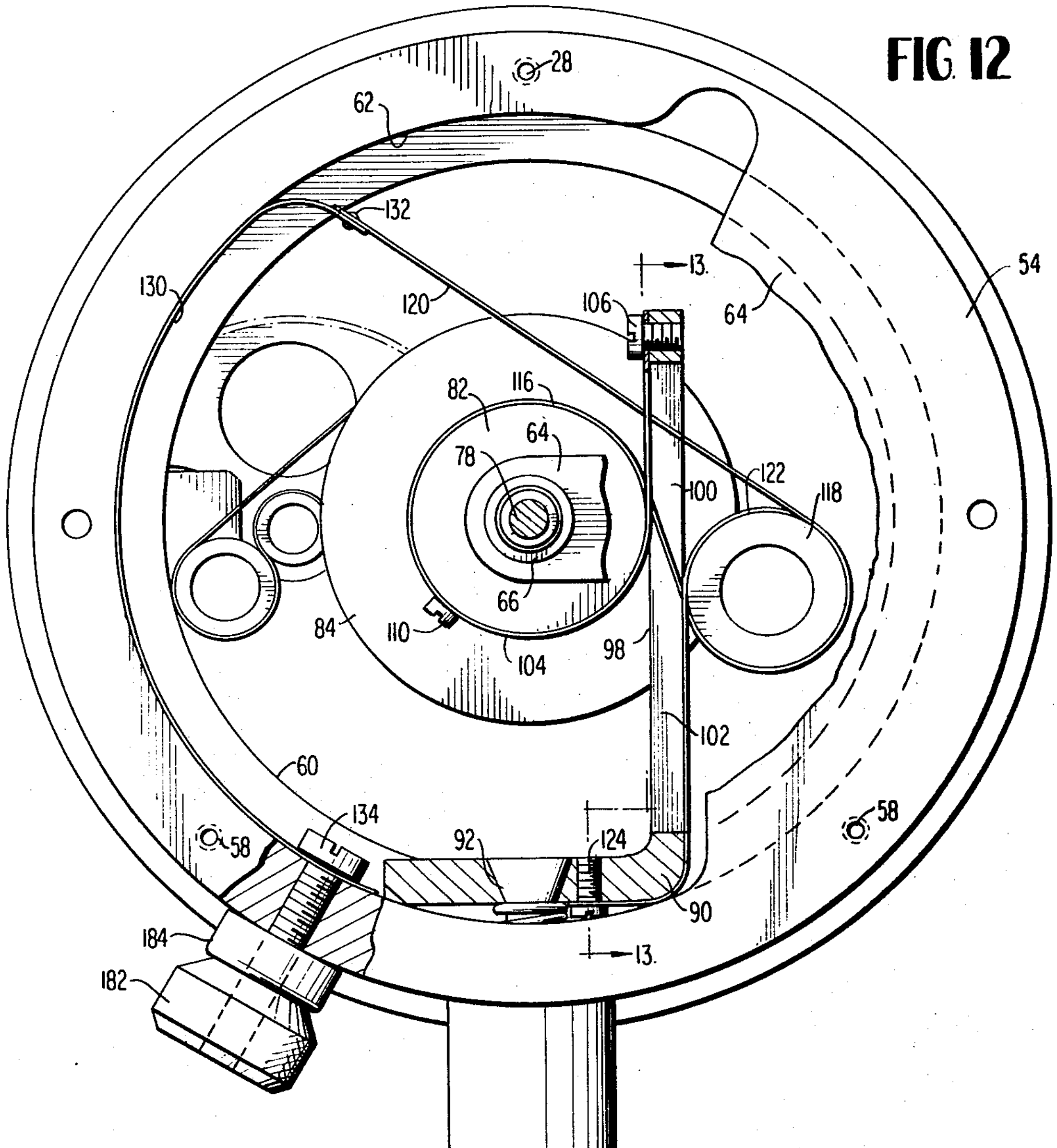
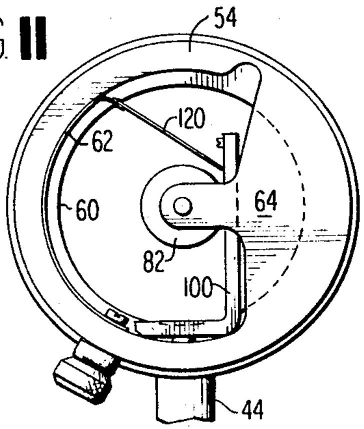


FIG 11



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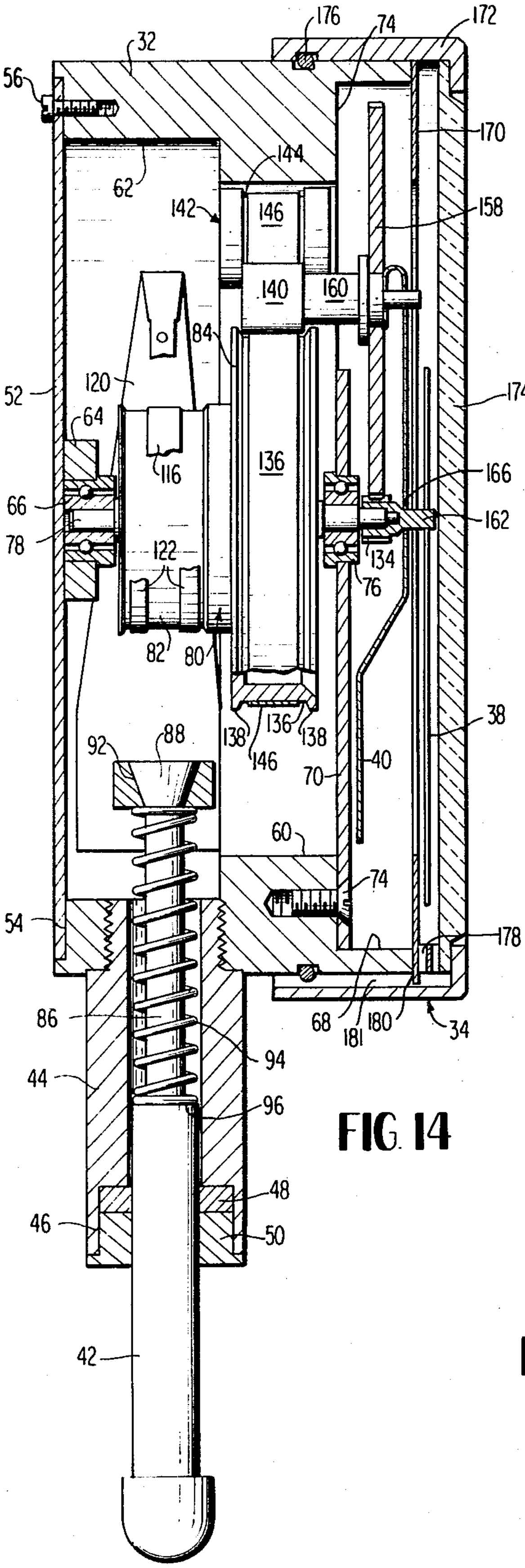

FIG. 14

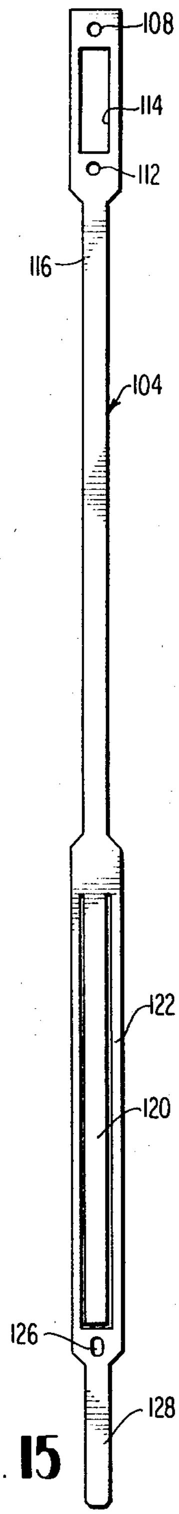

FIG. 15

FIG. 16

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ROTARY MOTION TRANSMITTING APPARATUS

BACKGROUND OF THE INVENTION

This invention relates to rotary apparatus, and more particularly to apparatus for transmitting rotary motion between relatively movable rotary elements.

Rotary motion transmission devices typically employ gears for providing an increased or reduced rate of rotation of an output shaft in relation to the rate of rotation of an input shaft. The gears ensure that slippage does not occur between the input and output shafts, as it does in belt drives or hydraulic couplings, for example. In a speed reduction transmission, for example, a small diameter pinion drives a gear at a rate of rotation that is proportional to the ratio of the pitch diameters of the gears. The degree of speed rotation that can be obtained using a pinion of a certain size is limited by the size of the output gear which will fit within the transmission case. Reducing the size of the pinion may improve the degree of speed reduction, but if the pinion is made too small, it can not be adequately supported on a shaft for transmitting the required torque.

Also, the teeth of the gears must be machined accurately in order to minimize frictional resistance and to provide an accurate angular relationship between the input and output shafts, if that is necessary. Machining of gears, particularly those having a small diameter, is a slow and costly operation. As a result, gear transmissions are relatively expensive.

A single gear mesh consisting of one gear and one pinion requires two shafts and, typically, four bearings. If a larger degree of speed reduction is desired than that readily obtainable with a single gear mesh, multiple gear meshes are often used. This, in turn, requires a corresponding increase in the number of shafts and bearings. The large number of components required adversely affects the total manufacturing cost. The large number of bearings means that more energy is lost through accommodate machining inaccuracies. When a normal geared system is reversed, all backlash must be removed before motion can occur at the output. In an amplification system, any backlash occurring in an early gear mesh is greatly amplified. Therefore, anti-backlash methods, such as torsion springs or spring loaded split gears, are often required.

SUMMARY OF THE INVENTION

In view of these disadvantages in prior rotary motion transmitting apparatus, it is an object of this invention to provide improved rotary motion transmitting apparatus.

It is a further object of this invention to provide rotary drive apparatus that is capable of producing a high ratio of angular motion between input and output rotary shafts. This high ratio is to be provided without requiring an excessive number of components or components which are of such minute size as to be troublesome in manufacturing or assembly.

A further object of this invention is to provide a rotary drive apparatus which has fewer bearings, a minimum number of sliding friction points, and lower energy loss.

A still further object of this invention is to provide a rotary drive apparatus which eliminates much of the required manufacturing precision, particularly that of early gear meshes in amplification systems.

Another object of this invention is to provide a rotary drive apparatus in which backlash is minimized or eliminated.

Another object of this invention is to provide a rotary drive apparatus having friction. Also, all components must typically be located rather precisely with respect to each other; this also affects the total manufacturing cost.

For some specialized applications, such as extremely high ratio speed reduction or double input devices (differentials), planetary gear systems are sometimes used. Planetary systems are marked by the fact that one or more gears or gear clusters have a planetary (revolving) motion in addition to the more normal rotational motion. Planetary gear systems have the same disadvantages as conventional gear systems and usually have even more components.

Gear systems which are used as rotary motion amplification devices are particularly subject to the disadvantages of gears. For example, the large number of friction losses (several bearings and multiple gear meshes) coupled with progressively diminishing torque severely restricts the degree of amplification possible. Few geared systems exhibit amplification ratios larger than 60 to 1.

Another disadvantage of geared amplification systems is that any machining inaccuracies are progressively multiplied. Therefore, any inaccuracy occurring in an early gear mesh shows up in a greatly amplified form at the output.

Another disadvantage of geared amplification systems is backlash. Backlash is the space between adjacent mating gear teeth and must normally be provided to planetary action and the advantages thereof (extremely high speed ratios and/or double inputs) using fewer components than conventional gear planetaries.

A still further object of this invention is to provide a rotary drive apparatus in which slippage can be minimized or eliminated if so desired.

A still further object of this invention is to provide a rotary drive apparatus wherein the cost of manufacturing and assembly is minimized.

These objects are accomplished in accordance with one embodiment of the invention by apparatus including a rotary input element connected with a circular guide surface and a thin flexible band connected with the guide surface and arranged in adjoining alternate loops spaced outwardly from the guide surface. A roller is retained within each of the loops and additional guide means keeps the centers of the rollers at approximately a uniform distance from the center of the guide surface as the rollers progress along the guide surface. The band frictionally engages the surface of the rollers thereby imparting rotation to the rollers at a predetermined rate as both rollers progress along the guide surface. As an alternative, the band can be secured to the rollers in order to positively prevent slippage. While this does restrict total motion of the device, it may become desirable in cases where extreme repeatability is required. A rotary output means is connected with one of the rollers to provide amplified rotation of the output means in response to rotation of the input means.

DESCRIPTION OF THE DRAWINGS

These preferred embodiments of the invention are illustrated in the accompanying drawings in which:

FIG. 1 is a side elevational view of rotary motion transmitting apparatus in accordance with this invention;

FIG. 2 is a cross sectional view of the rotary apparatus along the line 2—2 in FIG. 1;

FIG. 3 is a side elevational view of one modified form of the rotary apparatus of FIG. 1;

FIG. 4 is a side elevational view of another modified form of the apparatus of FIG. 1;

FIG. 5 is a side elevational view of a third modified form of the apparatus of FIG. 1;

FIG. 6 is a cross sectional view of the apparatus along the line 6—6 in FIG. 5;

FIG. 7 is a cross sectional view of another embodiment of this invention along the line 7—7 in FIG. 8;

FIG. 8 is a cross sectional view of the apparatus along the line 8—8 in FIG. 7;

FIG. 9 is a perspective view of a dial indicator embodying this invention;

FIG. 10 is an enlarged front elevational view, partially in cross section, of the dial indicator;

FIG. 11 is a rear elevational view of the dial indicator with the back cover removed;

FIG. 12 is an enlarged elevational view, partially in cross section, of the rear of the dial indicator;

FIG. 13 is a cross sectional view of the dial indicator along the line 13—13 in FIG. 12;

FIG. 14 is a cross sectional view of the dial indicator along the line 14—14 in FIG. 10;

FIG. 15 is a plan view of the spindle band; and

FIG. 16 is a plan view of the drum band.

DETAILED DESCRIPTION OF PREFERRED EMBODIMENTS

A preferred embodiment of rotary apparatus for transmitting motion in accordance with this invention is illustrated in FIGS. 1 and 2. The apparatus includes a ring 2 which has a substantially cylindrical internal guide surface 3 for supporting a roller cluster 4 for progressive movement around the circumference of the ring. The cluster includes an outer roller 6 and an inner roller 8. The inner roller 8 is supported by a drive wheel 10 which has a smaller external diamter than the internal diameter of the ring 2. The rollers are retained in the space between the ring 2 and the wheel 10 by a thin flexible band 12 which extends around the rollers 6 and 8 in adjoining alternate loops. The ends of the band overlap on the surface of the wheel 10 and are secured by a rivet or screw 13.

A gear 14 is fixed on the end of the roller 8 and the teeth on the gear mesh with corresponding gear teeth on a shaft 16 which is journaled in bearings 18. A shaft 20 is journaled in bearings 22 and secured to the wheel 10, so that rotation of the shaft 20 also turns the wheel 10.

The band 12 in the rotary apparatus of FIG. 1 extends from the peripheral surface of the wheel 10 outwardly toward the internal guide surface 3 of the ring 2 along a free path indicated at 24, and then passes around the periphery of the outer roller 6, encircling the roller in an outer band loop 26. Immediately adjoining the loop 26 is another loop 28 which projects in the opposite direction around the peripheral surface of the roller 8. The band 12 passes between the outer roller 6 and the guide surface of the ring 2, between the outer roller 6 and the inner roller 8, and between the inner roller 8 and the periphery of the wheel 10.

The outer roller 6 and the inner roller 8 are retained within the loops 26 and 28, respectively, because the sum of the diameters of the rollers is greater than the radial distance separating the internal guide surface 3 and the peripheral surface of the wheel 10. Also, the diameter of each roller must be less than the radial distance between the surface 3 and the wheel surface. It is also essential that the diameter of the roller 8 be sufficiently small in relation to the wheel 10 and the roller 6 that the band 12 does not engage the roller 8 along the free path 24. The tension in the band is sufficiently great to draw the roller 6 firmly against the guide surface 3 and to draw the inner roller 8 firmly against the surface of the wheel 10. The frictional forces between the rollers and the band, and between the band and guide surface 3 prevent the band 12 from slipping relative to these surfaces. Thus, relative rotation between the ring 2 and the wheel 10 causes the roller 6 and 8 to rotate as the band passes progressively around the rollers and the cluster 4 moves as a unit along a circular path about the center of the wheel 10, without sliding.

The apparatus illustrated in FIGS. 1 and 2 is one preferred embodiment of the invention and it is apparent that the apparatus can be altered to meet specific requirements. For example, the rivet or screw 13 which holds the ends of the band 12 against the surface of the wheel 10 restricts the degree of angular rotation of the cluster 4 about the center of the wheel 10. If it is desirable for the cluster to be capable of moving around the entire circumference of the wheel 10, a continuous band can be substituted for the band 12 and screw 13 is then omitted. One advantage of using a band of finite length and overlapping the ends, as in FIG. 1, is that the band can be produced economically and tension in the band can be imposed merely by sliding ends of the bands relative to each other before tightening the screw 13.

Another alternative is to replace the gear 14 with a wheel having a smooth peripheral surface that engages a corresponding smooth surface on the shaft 16. In order to minimize the possibility of slippage between such a friction wheel and the shaft 16, the sum of the radii of the wheel 14 and shaft 16 should be slightly larger than the center distance separating the axes of the roller 8 and the wheel 10. For certain applications, such as in a measuring instrument, high resistance to slippage between the wheel 14 and the shaft 16 is necessary, so that the instrument will remain accurate throughout many repeated measurements. Additional resistance to slippage can be accomplished by wrapping flexible bands around the wheel 14 and the shaft 16 to provide a greater surface of engagement than can be obtained with the cooperating cylindrical surfaces of the wheel 14 and the shaft 16. The band or belt could have either a crossed belt configuration in which the shaft 16 rotates in a direction opposite to that of the wheel 14, or an open belt configuration in which the wheel 14 and the shaft 16 rotate in the same direction. In both the crossed band and open band type drives, the sum of the radii of the wheel 14 and shaft 16 is substantially less than the center distance between the axes of the wheel 10 and the roller 8.

Since the apparatus illustrated in FIGS. 1 and 2 is intended merely to be schematic, only the essential bearings are disclosed. It may be desirable to include bearings or other supports for the various components of the apparatus. Also, the rollers 6 and 8 are not restrained axially in the embodiment of the invention illustrated in FIGS. 1 and 2. Various means of guidance can be employed to maintain the rollers in alignment with the desired path. One example of such guidance features includes the use of flanges on the inner roller 8, which engage the outer roller 6 and the wheel 10 to prevent axial misalignment. If the cluster 4 moves only through a small arc with respect to the axis of the wheel 10, alignment of the rollers 6 and 8 can be maintained by attaching the band 12 to the respective rollers by screws, or by welding or soldering, or by other suitable means.

An important feature of this invention is that the output shaft 16 is on a common center with the wheel 10, as contrasted with a conventional gear system in which the various gears are arranged on different axes of rotation and the center distance between the gear axes must be accurately located and maintained.

It is apparent that the apparatus of FIGS. 1 and 2 is reversible to produce a speed reduction in angular displacement between the input element and the output element. For example, if the ring 2 is held stationary, torque that is applied through the shaft 16 turns the wheel 10 at a speed that is the inverse of the ratios listed in table I for variations I and II. Similarly, if the wheel 10 is held stationary and torque is applied to shaft 16, the speed of rotation of the ring 2 relative to the shaft 16 is given by the inverse of the speed ratio for variation V. The output and input elements listed in table I can readily be reversed for speed reduction without frictionally binding the movalbe components, as occurs in conventional gear type transmissions.

A modified form of the apparatus shown in FIGS. 1 to 4 is illustrated in FIG. 3. In this modified form, the ring 2' is arranged concentrically of the wheel 10' for rotation by a plurality of clusters 4'. Each cluster includes a pair of rollers 6' and 8' which are held against the internal surface of the ring 2' and the periphery of the wheel 10' by a thin flexible band 12' in the manner illustrated in FIGS. 1 and 2. The band extends continuously around each of the clusters 4' and is maintained in tension. A bearing 15 is shown supporting the wheel 10' for rotation. As the wheel 10' or the ring 2' rotates, the clusters 4' progress around the internal surface of the ring 2' while maintaining the separation from each other, and thereby support the ring 2' concentrically on the wheel 10'. The structure of FIG. 3 can also be used simply as a bearing for the wheel 10' by mounting the ring 2' in a support. Also, one or more of the clusters 4' can be used as part of a motion amplification or reduction system in accordance with this invention.

One important advantage of the apparatus of Gis. 1 and 2 is that a variety of speed ratios can be achieved between the various input and output elements. The following table shows six variations (I–VI) of the apparatus of FIGS. 1 and 2. In each of the variations, the inner roller 8 supports the wheel or gear 14 and the outer roller 6 serves as an idler.

TABLE I

Variation 1

Wheel 10: Input
Ring 2: Stationary
Shaft 16: Output
Drive: Friction, geared, crossed belt
Speed ratio:

$$\frac{\theta_{16}}{\theta_{10}}=\frac{1-\frac{r_{14}r_2}{r_8r_{16}}}{1-\frac{r_2}{r_{10}}}$$

Variation II

Wheel 10: Stationary
Ring 2: Input
Shaft 16: Output
Drive: Friction, geared, crossed belt
Speed ratio:

$$\frac{\theta_{16}}{\theta_2}=\frac{1-\frac{r_{10}r_{14}}{r_8r_{16}}}{1-\frac{r_{10}}{r_2}}$$

Variation III

Wheel 10: Input
Ring 2: Output
Shaft 16: Stationary
Drive: Friction, geared, crossed belt
Speed ratio:

$$\frac{\theta_2}{\theta_{10}}=\frac{1-\frac{r_8r_{16}}{r_{14}r_2}}{1-\frac{r_8r_{16}}{r_{10}r_{14}}}$$

Variation IV

Wheel 10: Input
Ring 2: Stationary
Shaft 16: Output
Drive: Open belt
Speed ratio:

$$\frac{\theta_{16}}{\theta_{10}}=\frac{1+\frac{r_{14}r_2}{r_8r_{16}}}{1-\frac{r_2}{r_{10}}}$$

Variation V

Wheel 10: Stationary
Ring 2: Input
Shaft 16: Output
Drive: Open belt
Speed ratio:

$$\frac{\theta_{16}}{\theta_2}=\frac{1+\frac{r_{10}r_{14}}{r_8r_{16}}}{1-\frac{r_{10}}{r_2}}$$

Variation VI

Wheel 10: Input
Ring 2: Output
Shaft 16: Stationary
Drive: Open belt
Speed ratio:

$$\frac{\theta_2}{\theta_{10}}=\frac{1+\frac{r_8r_{16}}{r_{14}r_2}}{1+\frac{r_8r_{16}}{r_{10}r_{14}}}$$

The drive between the wheel or gear 14 and the shaft 16 can be selected to produce a direct engagement drive, such as gearing or frictional engagement between the peripheral surfaces of the wheel 14 and the shaft 16. The shaft 16 can be driven also by a belt that crosses between the wheel 14 and the shaft 16. The gear drive, friction drive, and crossed belt drive each causes the shaft 16 to rotate in the opposite direction to that of the wheel 14 (variations I–IV), as noted in the above table. The shaft 16 can be driven in the same direction as the wheel 14 by an open belt that passes around the wheel 14 and the shaft 16 (variations IV–VI). Of course, the periphery of the respective rotary elements must be spaced apart to allow relative rotation.

In the above table, and in describing this invention the radius that is referred to is the effective radius, rather than the actual roller radius. The band that is wrapped on the roller surface flexes as it passes around the respective rollers and the neutral axis in flexure is approximately midway through the thickness of the band. The effective radius is measured from the center of the roller to the neutral axis. Therefore, the effective radius is larger than the actual roller radius by a distance that is half of the band thickness. Conversely, the effective radius of an internal surface, such as the guide surface 3 in FIG. 1, is smaller than the actual radius of the surface by half of the band thickness. Where the radius refers to a gear, the radius is measured to the pitch circle of the gear.

A second preferred embodiment of the invention is illustrated schematically in FIG. 4. The apparatus of this embodiment includes a ring 2' which is mounted concentrically with a rotary wheel 10' on which is supported a rotary shaft 16'. A roller cluster is interposed between the internal surface 3' of the ring 2' and the peripheral surface of the wheel 10'. An endless band 12' passes around the periphery of the wheel 10' and extends over the outer roller 6' and in the inner roller 8' in alternate loops, as shown in FIG. 4. A free portion 24' of the band extends between the outer roller 6' and the surface of the wheel 10'. A gear 14' is fixed on the end of the roller 6' and engages a corresponding gear formed in the surface of the shaft 16'. The apparatus of FIG. 4 has substantially the same structure and arrangement of the apparatus illustrated in FIGS. 1 and 2, except that the gear 14' is mounted on the outer roller 6', rather than the inner roller. Also, the band 12' is a continuous band which has sufficient tension to prevent slippage between the band and the various surfaces supporting the band. Since the band is continuous, the cluster of rollers is capable of progressing completely around the circumference of the wheel 10'. Several variations of the operation of the apparatus are listed in Table II.

TABLE II

Variation VII

Wheel 10: Input
Ring 2: Stationary
Shaft 16: Output
Drive: Friction, geared, crossed belt

Speed ratio:

$$\frac{\theta_{16}}{\theta_{10}}=\frac{1+\frac{r_{14}r_2}{r_6r_{16}}}{1-\frac{r_2}{r_{10}}}$$

Variation VIII

Wheel 10: Stationary
Ring 2: Input
Shaft 16: Output
Drive: Friction, geared, crossed belt

Speed ratio:

$$\frac{\theta_{16}}{\theta_2}=\frac{1+\frac{r_{10}r_{14}}{r_6r_{16}}}{1-\frac{r_{10}}{r_2}}$$

Variation IX

Wheel 10: Input
Ring 2: Output
Shaft 16: Stationary
Drive: Friction, geared, crossed belt

Speed ratio:

$$\frac{\theta_2}{\theta_{10}}=\frac{1+\frac{r_6r_{16}}{r_{14}r_2}}{1+\frac{r_6r_{16}}{r_{10}r_{14}}}$$

Variation X

Wheel 10: Input
Ring 2: Stationary
Shaft 16: Output
Drive: Open belt

Speed ratio:

$$\frac{\theta_{16}}{\theta_{10}}=\frac{1-\frac{r_{14}r_2}{r_6r_{16}}}{1-\frac{r_2}{r_{10}}}$$

Variation XI

Wheel 10: Stationary
Ring 2: Input
Shaft 16: Output
Drive: Open belt

Speed ratio:

$$\frac{\theta_{16}}{\theta_2}=\frac{1-\frac{r_{10}r_{14}}{r_6r_{16}}}{1-\frac{r_{10}}{r_2}}$$

Variation XII

Wheel 10: Input
Ring 2: Output
Shaft 16: Stationary
Drive: Open belt

Speed ratio:

$$\frac{\theta_2}{\theta_{10}}=\frac{1-\frac{r_6r_{16}}{r_{14}r_2}}{1-\frac{r_6r_{16}}{r_{10}r_{14}}}$$

The drive between the wheel or gear 14' and the shaft 16' can, of course, be either by direct frictional engagement or by gearing, or crossed-belt to produce counterrotation of the shaft 16' relative to the wheel or gear 14'. Also, an open belt can be used to transmit rotary motion between the gear or wheel 14' and the shaft 16'.

A third modified form of the apparatus of FIG. 1 is illustrated in FIGS. 5 and 6. In describing this embodiment, those elements that correspond to components of the apparatus of FIGS. 1 and 2 are designated by the same reference numeral followed by a double prime superscript. The roller 6'' includes a pair of gears 18 which engage a ring gear 20 formed in the internal circumference 3'' of the ring 2''. The intermediate portion of the roller 6'' rigidly connects together the gears 18 and serves as a roller surface supporting the band 12''. The cooperating roller 8'' is aligned with the intermediate portion of the roller 6'' and is shown in FIG. 5, the free portion 24'' of the band passes between the peripheral surface of the wheel 10″ and the intermediate portion of the roller 6″. The rotation of the inner roller 8″ is transmitted to the shaft 16″ through a wheel or gear 14″ in the same manner as in the apparatus of FIG. 1. The fact that the gears 18 roll on a radius that is greater than the intermediate portion of the roller 6″ produces amplified motion between the wheel 10″ and the shaft 16″. In table III, the variations of motions designated XIII to XVIII correspond to variations I to VI of table I, except for the use of a stepped roller 6″ for one of the rollers in the cluster, and the advantage of providing the stepped roller to achieve higher speed ratios is apparent from comparison of the speed ratio formulas in table III with those of Table I.

TABLE III

Variation XIII

Wheel 10: Input
Ring 2: Stationary
Shaft 16: Output
Drive: Friction, geared, crossed belt

Speed ratio:

$$\frac{\theta_{16}}{\theta_{10}}=\frac{1-\frac{r_{14}r_6r_2}{r_8r_{18}r_{16}}}{1-\frac{r_6r_2}{r_{10}r_{18}}}$$

Variation XIV

Wheel 10: Stationary
Ring 2: Input
Shaft 16: Output
Drive: Friction, geared, crossed belt

Speed ratio:

$$\frac{\theta_{16}}{\theta_2}=\frac{1-\frac{r_{10}r_{14}}{r_8r_{16}}}{1-\frac{r_{10}r_{18}}{r_6r_2}}$$

Variation XV

Wheel 10: Input
Ring 2: Output
Shaft 16: Stationary
Drive: Friction, geared, crossed belt

Speed ratio:

$$\frac{\theta_2}{\theta_{10}}=\frac{1-\frac{r_8r_{18}r_{16}}{r_{14}r_6r_2}}{1-\frac{r_8r_{16}}{r_{10}r_{14}}}$$

Variation XVI

Wheel 10: Input
Ring 2: Active
Shaft 16: Output
Drive: Open belt

Speed ratio:

$$\frac{\theta_{16}}{\theta_{10}}=\frac{1+\frac{r_{14}r_6r_2}{r_8r_{18}r_{16}}}{1-\frac{r_6r_2}{r_{10}r_{18}}}$$

Variation XVII

Wheel 10: Stationary
Ring 2: Active
Shaft 16: Output
Drive: Open belt

Speed ratio:

$$\frac{\theta_{16}}{\theta_2}=\frac{1+\frac{r_{10}r_{14}}{r_8r_{16}}}{1-\frac{r_{10}r_{18}}{r_6r_2}}$$

Variation XVIII

Wheel 10: Input
Ring 2: Active
Shaft 16: Stationary
Drive: Open belt

Speed ratio:

$$\frac{\theta_2}{\theta_{10}}=\frac{1+\frac{r_8r_{18}r_{16}}{r_{14}r_6r_2}}{1+\frac{r_8r_{16}}{r_{10}r_{14}}}$$

As an example of the advantage obtained by the use of the stepped roller, it has been determined that comparable apparatus having the structure of FIGS. 1 and 5, respectively, with the dimensions selected as follows, the apparatus of FIG. 5 produces a speed ratio of 289 to 1 between the shaft 16″ and the wheel 10″, while the apparatus of FIG. 1 produces a speed ratio of only 49 to 1.

Variation I	Variation XIII
$r_{10}=1.0$	$r_{10}=1.0$
$r_8=0.2$	$r_8=0.2$
$r_{14}=1.0$	$r_{14}=1.0$
$r_6=0.3$	$r_6=0.3$
$r_2=2.0$	$r_{18}=0.55$
$r_{16}=0.2$	$r_2=2.0$
	$r_{16}=0.2$

As a modification of the apparatus of FIG. 5, the gear 14″ may be mounted on the outer roller 6″ instead of the inner roller 8″, as shown in FIG. 5. If the outer roller 6″ has the stepped structure shown in FIGS. 5 and 5 and the remaining structure is substantially identical with that of FIGS. 5 and 6, then the following table gives the speed ratios for various input and element arrangements.

TABLE IV

Variation XIX

Wheel 10: Input
Ring 2: Stationary
Shaft 16: Output
Drive: Friction, geared, crossed belt

Speed ratio:

$$\frac{\theta_{16}}{\theta_{10}}=\frac{1+\frac{r_{14}r_2}{r_{18}r_{16}}}{1-\frac{r_6r_2}{r_{10}r_6}}$$

Variation XX

Wheel 10: Stationary
Ring 2: Input
Shaft 16: Output
Drive: Friction, geared, crossed belt

Speed ratio:

$$\frac{\theta_{16}}{\theta^2}=\frac{1+\frac{r_{10}r_{14}}{r_6r_{16}}}{1-\frac{r_{10}r_{18}}{r_6r_2}}$$

Variation XXI

Wheel 10: Input
Ring 2: Output
Shaft 16: Stationary
Drive: Friction, geared, crossed belt

Speed ratio:

$$\frac{\theta_2}{\theta_{10}}=\frac{1+\frac{r_{18}r_{16}}{r_{14}r_2}}{1+\frac{r_6r_{16}}{r_{10}r_{14}}}$$

Variation XXII

Wheel 10: Input
Ring 2: Stationary

Shaft 16: Output
Drive: Open belt
Speed ratio:

$$\frac{\theta_{16}}{\theta_{10}}=\frac{1-\frac{r_{14}r_2}{r_{18}r_{16}}}{1-\frac{r_6r_2}{r_{10}r_{18}}}$$

Variation XXIII

Wheel 10: Stationary
Ring 2: Input
Shaft 16: Output
Drive: Open belt
Speed ratio:

$$\frac{\theta_{16}}{\theta_2}=\frac{1-\frac{r_{10}r_{14}}{r_6r_{16}}}{1-\frac{r_{10}r_{18}}{r_6r_2}}$$

Variation XXIV

Wheel 10: Input
Ring 2: Output
Shaft 16: Stationary
Drive: Open belt
Speed ratio:

$$\frac{\theta_2}{\theta_{10}}=\frac{1-\frac{r_{18}r_{16}}{r_{14}r_2}}{1-\frac{r_6r_{16}}{r_{10}r_{14}}}$$

In the description of the embodiments of FIGS. 1 to 6 and in Tables I to IV, elements have been described as input elements and output elements. The apparatus is reversible and torque can be applied to those elements that have been designated output elements and the resultant effect on the elements designated input elements would be the inverse of the speed ratio given in the Tables. It is also possible to utilize the apparatus of FIGS. 1 to 6 as a form of differential mechanism in which two elements serves as input elements and a third element serves as an output element. For example, in the embodiment of FIG. 1, the wheel 10 and the ring 2 might serve as the two input elements, with the shaft 16 serving as the output element. Due to the geometry of the apparatus, the angular displacement of the wheel 10 would produce a greater effect on the ultimate rotation of the shaft 16 than would the same angular displacement of the ring 2, if appropriate sizes of the outer and inner rollers 6 and 8, the internal diameter 3 of the ring, and the diameter of the wheel 10 were selected. It is also possible to use the apparatus of this invention as a differential mechanism in which a single input motion produces two output motions.

In operation, assuming that torque is applied in a clockwise direction as viewed in FIG. 1 to the wheel 10, while the ring 2 is held stationary, motion of the wheel 10 advances the band sequentially around the outer roller 6 and around the inner roller 8. At the same time, the cluster 4 moves in a counterclockwise direction, with the gear 14 also rotating counterclockwise. Since the gear 14 meshes with the gear on the shaft 16, the shaft turns clockwise. The apparatus of FIGS. 4 and 5 operate in the same manner, except that the gear 14′ rotates in the same direction as the wheel 10′.

A second embodiment of the invention is illustrated in FIGS. 7 and 8. In this embodiment, a wheel 25 has shafts 25*a* and 25*b* projecting on opposite sides. A body 26 if journaled in suitable bearings for rotation about the shafts 25*a* and 25*b*. The body 26 includes a pair of flanges 26*a* spaced on opposite sides of the wheel 25 and a web 26*b* rigidly connecting the flanges together. Each flange 26a has a circular edge which is concentric with the central axis of the wheel 25. The radius of the wheel 25 is indicated by r_{10} in FIG. 7 and the radius of the flanges is indicated by r_2 in FIG. 7.

The apparatus also includes a roller cluster having an outer roller 27 and an inner roller 28. These rollers have the radii r_6 and r_8, respectively, as shown in FIG. 7. A thin flexible band 29 passes around the wheel 25 and is looped around the outer roller 27 and the inner roller 28 in the same manner as described with respect to the embodiment of FIG. 1. Another band 30 having sufficient width to span the distance between the flanges 26*a*, as shown in FIG. 8, is mounted as an open belt between the outer roller 27 and the body 28. The band is supported on the circular edges of the flanges 26*a*, as shown in FIG. 7. The band 30 is illustrated as a continuous band, but the band may be formed of a strip of finite length and the ends of the band may be overlapped and secured to the web 26*b*, if desired. Both of the bands 29 and 30 have sufficient tension to maintain the rollers of the cluster in the relative positions shown in FIG. 7 as the cluster progresses relative to the wheel 25 and the flanges 26*a*, and the tension is sufficient to prevent sliding between the bands 29 and 30 and the surfaces cooperating with the bands.

The roller 28 has an axial extension which is rigidly joined with a gear 28*a* that is concentric with the central axis of the roller 28. A shaft 31 is supported for rotation in bearing in the shaft 26*b* for rotation about an axis parallel to the central axis of the roller 28. A gear 31*a* on the shaft 31 meshes with the gear 28*a*. The pitch radius of the gear 28*a* is indicated as r_{14} in FIG. 7 and the pitch radius of the gear 31 is indicated at r_{16} in FIG. 7.

The band 30 provides the external guiding function corresponding to the ring 2 in the embodiments of FIGS. 1 and 2. Instead of being rigid, however, the guide surface is flexible. This becomes particularly significant in providing maximum amplification or reduction of rotary motion between the shaft 25*a* and the shaft 31. Referring to Table I above, the equation for speed ratio of variation I indicates that the maximum amplification or reduction occurs when the radius of the outer guide surface r_2 is substantially equal to the radius of the inner guide surface r_{10}. In the embodiment of FIGS. 7 and 8, the radius of the flanges 26*a* is sufficiently larger than the radius of the wheel 25 to allow the apparatus to operate without requiring the application of excessively high torque. The various radii are indicated in FIG. 7 by subscript numbers which correspond to the radii bearing the same subscript numbers in FIG. 1 and the speed ratios listed in Table I are appropriate for determining the speed ratios for various sizes of elements in the apparatus illustrated in FIGS. 7 and 8. Furthermore, the variations described for the modified apparatus of FIGS. 4 to 6 are also appropriate for the apparatus of FIGS. 7 and 8 to produce the desired motions.

In operation, when a clockwise torque is applied to the shaft 25a, as viewed in FIG. 7, the wheel 25 rotates and this angular rotation is transmitted to the band 29, which advances the band around the outer roller 27 and the inner roller 28. At the same time, the band 30 advances without slipping, producing a clockwise motion of the cluster of rollers 27 and 28. Rotation of the outer roller 27 is transmitted to the inner roller 28 through the band 29, which in turn rotates the gear 28*a*, which ultimately drives the shaft 31. The various sizes of the rollers and gears determine the degree of amplification achieved by the apparatus of FIGS. 7 and 8, but it is apparent that very high amplifications are possible. The apparatus may be operated in a reverse direction by applying an input torque through the shaft 31 to achieve a reduced rate of rotation of the shaft 25*a*. The presence of the web 26*b* limits the degree of rotation of the cluster to an arc of about 180° with respect to the body 26. The apparatus of FIGS. 7 and 8 can be modified to eliminate the web 26*b* by providing adequate support for the flanges that does not interfere with movement of the band 30. Also, the flanges 26*a* can be modified to have a circular edge around the entire circumference. When modified in this way, the cluster of the apparatus of FIGS. 7 and 8 would be capable of revolving around the entire circumference of the flanges 26*a*.

In another embodiment of this invention, the rotary apparatus is incorporated in a dial indicator, which provides an amplified indication of the displacement of a movable spindle in the instrument. The dial indicator is illustrated in FIGS. 9 to 16.

Referring to FIG. 9, the dial indicator includes a case 32 with a cover 34 on the front of the case. A dial face indicated generally at 36 has pointers 38 and 40 which provide coarse and fine indications of movement of a spindle 42 which projects outwardly from the side of the case 32.

The spindle is supported for longitudinal movement in a stem 44 which is secured in the wall of the case 32 by screw threads (FIG. 14), or by other suitable means, such as welding or integral casting. At its outer end, the stem has a counterbore 46 for receiving a packing ring 48 and a bushing 50. The spindle 42 is freely sidable through the bushing 50 and the packing ring 48.

The back of the dial indicator has a removable cover plate 52. The rear of the case 32 has a recessed shoulder 54 for receiving the circular cover plate 52 and the plate is held securely against the shoulder 54 by screws 56 which are received in tapped holes 58 in the case 32. The cover plate may be secured on the case by any other suitable means. The back of the indicator is shown in FIG. 11 with the cover plate 52 removed.

The case 32 has a central bore 60 and a counterbore 62 at the back side of the case. The counterbore 62 is milled, cast, or otherwise formed in such a way that a portion of the rear wall 64 of the case remains to support a rotary bearing assembly 66. As shown in FIG. 11, the remainder of the rear wall is milled out to provide an opening for assembly of the internal components of the dial indicator.

There is another counterbore 68 at the front side of the case 32 and a bearing support plate 70 is secured against the radial shoulder 72 at the bottom of the counterbore 68 by means of screws 74. The bearing plate 70 has a hole in which a bearing assembly 76 is received. A shaft 78 is supported in the bearings 66 and 76 for rotation. A unitary wheel and drum element 80 is fixed on the shaft 78. The element 80 includes a wheel 82 and a drum 84.

The spindle 42 has a portion 86 (FIG. 14) of reduced diameter and the end of the spindle has a tapered head 88. A guide element 90 is connected with the spindle portion 86 by a tapered socket 92 in which the head 88 is received. A coil spring 94 is compressed between a shoulder 96 on the spindle 42 and the lower surface of the element 90. The spring 94 preferably exerts sufficient force inwardly on the element 90 to maintain the head 88 in the socket 92, but if excessive axial force is applied inwardly to the spindle 42, the spring 94 yields, thereby preventing damage to the components of the dial indicator.

The guide element 90, as shown in FIGS. 12 and 13 has a pair of opposed guide surfaces 98 and 100 which are parallel to the central axis of the spindle 42. The element 90 has a longitudinal slot 102. A thin flexible band having the shape illustrated in FIG. 15 is secured at one end to the element 90 by a screw 106. The band 104 has a hole 108 adjacent one end for receiving the screw 106. The band 104 extends along the guide surface 98 and passes between the surface of the wheel 82 and the guide surface 98. The band is secured to the wheel by a screw 110, as shown in FIG. 9. The band has a hole 112 for receiving the screw 110. A longitudinal slot 114 is provided in the band. The band extends around the wheel 82 and through the slot 114. As shown in FIG. 13, the mid-portion 116 of the band has a narrower width than the width of the slot 114.

A roller 118 is positioned in the case 32 adjacent the guide surface 90. The roller 118 is preferably hollow to minimize inertia of the roller. The intermediate portion of the band passea round the roller 108 and a tonque 110 extends outwardly from the surface of the roller 118 and through the slot 102 in the element 90. The remainder 122 of the band passes between the roller 118 and the guide surface 100 and extends along the surface 100. The end of the band is secured to the element by a screw 124. As shown in FIG. 15, the band has a slot 126 for receiving the screw 124 and a tab 128 allows the band to be tensioned before the screw 124 is tightened. After the band has been secured in place, the tab 124 is removed. Although screws are used in this preferred embodiment for fastening the bands to cooperating elements, the band can also be attached by welding, soldering, or other suitable means.

Tension is applied to the end of the tongue 120 by means of a leaf spring 130 which is mounted in the counterbore 62. The spring 130 is secured at one end to the tongue 120 by a rivet 132 or other suitable means. The opposite end of the spring 130 is secured in place by a screw 134 which is received in a tapped hole in the case 32. As shown in FIG. 14, the width of the spring progressively increases from the end connected with the tongue 120 toward the opposite, stationary end. As the spindle 42 is moved upward, the roller 118 rotates, and pulls the tongue toward the right, as viewed in FIG. 12, thereby progressively pulling the end of the spring away from the wall of the counterbore 62. The spring force exerted by the spring 130 on the tongue 120 is substantially constant as the roller 188 rotates in response to displacement of the spindle 42. The shape of the spring 130 is such that the end of the spring moves almost in a straight line in the direction of the length of the tonque 120 and when the spindle 42 is at its innermost position, the spring 130 assumes a circular shape, which minimizes stresses in the spring. The spring 130 provides the force for urging the spindle outwardly toward the position shown in FIG. 12. Since the force of the spring 130 is substantially constant, the force required to displace the spindle remains substantially constant throughout its range of movement.

Sufficient tension is applied to the band 104 to prevent slippage between the roller 82 and the guide surface 98 and between the roller 118 and the guide surface 100. Inward displacement of the element 90, as viewed in FIG. 12, causes the wheel 82 to rotate counterclockwise and for the roller 118 to rotate clockwise. The tension in the band 104 would normally urge the roller 118 to move away from the point of engagement between the wheel 82 and the surface 98 lengthwise of the surface 100 in order to decrease the included angle between the band portion in the slot 102 and the guide surface 100. The tongue 120 which is maintained under tension by the spring 130 counteracts this tendency of increasing the center distance between the roller and the wheel and yet allows rotation of the roller in response to longitudinal movement of the guide surface 100. The wheel 82 and the roller 118 serve as guide rollers tightly clamping the member 90 between them for movement along a stright path. Both the wheel 82 and the roller 118 have flanges to ensure that the components retain their correct position.

Since the wheel 82 is integral with the drum 84, motion is transmitted from the member 90 to the drum 84 through the wheel 82. The drum 85 has a cylindrical surface 136 and a pair of flanges 138 along opposite edges of the surface 136. A roller 140 is supported on the surface 136 of the drum and a tubular roller 142 is supported on the wall of the central bore 60 which forms a guide surface for the roller 142. The roller 142 has a cylindrical recess 144 which is aligned with the roller 140.

A thin flexible band 146, as shown in FIG. 16, extends around the surface 136 of the drum and is wrapped on the outer roller 142 in the recess 144. As shown in FIG. 10, the band passes between the inner and outer rollers 140 and 142 and is looped over the inner roller 140 before passing between the roller 140 and the surface 136 of the drum. The opposite ends of the band are secured on the surface of the drum by a screw 148. The band has a hole 150 at one end through which the screw 148 extends and has a slot 152 at the opposite end for receiving the screw. The band 146 also has a tab 154 at one end for applying tension in the band during assembly.

The tension in the band 146 urges the center of the outer roller 142 to move in a counterclockwise direction, as viewed in FIG. 10, and urges the center of the inner roller 140 to move in a clockwise direction with respect to the central axis of the shaft 78. The sum of the diamter of the inner roller 140 and the radius of the recess 144 and the radius of the roller 142 is greater than the radial distance separating the surface 136 and the guide surface of the bore 60. Therefore it is impossible for the rollers to move past each other and the tension in the band merely serves to wedge the rollers tightly into engagement with each other and into engagement with the respective guide surfaces, so that slippage between the band, the rollers and the guide surfaces does not occur.

As a means of adjusting the diameter of the recess 144 of the roller 142, a tapered plug 156 is inserted in the hollow roller 142 after the assembly of the band and rollers. The plug 156 causes expansion of the roller 142 to the extent necessary to provide accurate calibration.

Rotation of the drum 84 in a clockwise direction as viewed in FIG. 10, feeds the band 146 off of the surface 136 and onto the surface of the recess 144. The drum draws an equal length of band onto the surface 136 between the inner roller 140 and the drum 84. This lengthwise movement of the band causes the outer roller 142 to rotate about its central axis in a clockwise direction and the inner roller 140 to rotate about its central axis in a counterclockwise direction. Since the surface of the roller 142 directly engages the wall of the bore 60, the center of the roller 142 is displaced along an arcuate path in a counterclockwise direction. Similarly the center of the inner roller 140 moves along an arcuate path in a counterclockwise direction. The rollers 140 and 142 remain in the same relative positions as they move along the respective guide surfaces. The pair of rollers 140 and 142 that move together in this manner is known as a cluster.

The motion of the inner roller 140 is transmitted to the pointer 38 through a gear 158. The gear 158 is supported on an extension 160 of the roller 140, and is secured to the extension by any suitable means. The gear 158 has openings 159 to reduce the inertia of the gear and the openings can be aligned with the center of the roller 142 to allow the insertion of the tapered plug 156. The shaft 78 projects outwardly from the bearing 76 and a sleeve 162 is slidably supported on the shaft 78. A pinion 134 secured to the sleeve 162 meshes with the gear 158. As the gear 158 rotates, it causes rotation of the sleeve 162 and the pointer 38 which is fixed on the sleeve 162 moves in response to rotation of the roller 140. A secondary pointer 40 is mounted on the sleeve 162 by means of a hole 166 in the pointer 40, so that the pointer freely rotates about the center of the sleeve 162.

The roller cluster is movable from a position in which the center of the inner roller is located at the positioned marked A in FIG. 10 when the spindle 42 is at its outermost position. The maximum extent of movement of the cluster is shown by the position of the inner roller 140 at the location marked B in FIG. 10. At this position, the spindle 42 is displaced inwardly to the maximum extent of its travel. In FIG. 9, the primary pointer 38 and the secondary pointer 40 are shown at approximately the zero position corresponding to the position of the inner roller at A. A scale 168 is provided on the surface of the bearing plate 70 for indicating the position of the secondary pointer 40. A circular scale 170 abuts against the end of the case 32 and a cover 34 is applied over the end of the case. The scale 170 is marked with an appropriate scale for the pointer 38. The cover 34 includes a bezel 172 and a crystal 174 for viewing the movement of the pointers. The cover 34 is held in place on the case 32 by a garter spring 176 which is received in aligned grooves in the cover 34 and in the case 32. A wavy washed 178 is interposed between the crystal 174 and the scale 170. The wavy washer has a sinosoidal shape and the amplitude of the curvature is slightly greater than the distance between the inner surface of the crystal **174** and the outer surface of the scale **170**, so that the washer **178** urges the cover **34** outwardly.

The cover **34** is free to rotate relative to the case **32**, being restrained by the spring **176** only in an axial direction. The scale **170** turns with the cover, since a tab **180** on the ring is received in a slot **181** formed in the bezel **172**. As shown in FIGS. **9**, **10** and **12**, a locknut **182** is applied to the outer end of the screw **134**. A clamping washer **184** projects over the edge of the bezel **172** and by tightening the nut **182**, the cover **34** is maintained in any selected rotational position. This locking feature allows the zero position marked on the scale **170** to be aligned with the pointer **38** when the cluster is at any described initial position.

In operation, the dial indicator is set up in accordance with conventional practice to measure longitudinal displacement of the spindle **42**. As the spindle moves inwardly with respect to the case **32**, motion is transmitted from the element **90** to the wheel **82** and the roller **118** by means of the band **104**. As the roller **118** rotates clockwise, the tongue portion **120** of the band progressively pulls the leaf spring **130** away from the wall of the counterbore **62**. The spring force applied to the element **90** through the band **104** provides a constant bias urging the spindle outwardly.

The counterclockwise rotation of the wheel **82** causes the drum **84** also to rotate in a counterclockwise direction as viewed in FIG. **9**. Referring to FIGS. **7** and **11**, rotation of the drum **84** is transmitted to the cluster rollers **140** and **142** by means of the band **146**. The inner roller **140** is initially in the position indicated at A in FIG. **10** and rotation of the drum **84** in a clockwise direction causes the inner roller **140** to rotate in a counterclockwise direction and to move progressively along an arcuate path from the position A toward the position indicated at B in FIG. **10**. As the inner roller **140** rotates, the gear **158** which is connected with the roller also rotates in a counterclockwise direction. This rotation is transmitted to the sleeve **162** causing the pointer **38** to rotate in a clockwise direction. Displacement of the central axis of the inner roller **140** along the arcuate path also causes the secondary pointer **40** to rotate about the sleeve **162** in a counterclockwise direction.

Since there are no gears, except for the pointer gears **158** and **134**, the problem of accuracy of machining the gear teeth has been eliminated. Also, the difficulty of transmitting torque through the gear chain no longer is a limiting factor on the amplification that can be obtained in a dial indicator. Not only can a large range of readings be obtained, but the accuracy of the readings remains substantially uniform throughout the measuring range. It has been found that amplification of as much as 266 to 1 can readily be obtained by the mechanism of this invention. This amplification is far greater than that previously obtainable with gear type devices of the same size. Furthermore, any inaccuracies in machining the various cylindrical surfaces of the components are not amplified. This means that a 1 percent change in the diameter of one of these cylindrical surfaces produces approximately a 1 percent change in the overall ratio. Therefore tolerances on the various parts are not as critical as in the gear type mechanisms.

An important advantage of the apparatus of this invention as exemplified in the several embodiments is that a relatively simple structure employing easily manufactured shapes is capable of providing a high degree of amplification or reduction of rotary motion. Furthermore, the rolling action between components allows the apparatus to function in environments where dirt particles may be present on the rolling surfaces. The band and rollers are capable of rolling over any dirt or dust particles, without interferring with the operation of the apparatus.

While this invention has been illustrated and described in accordance with several preferred embodiments, it is recognized that variations and changes may be made therein without departing from the invention as set forth in the claims.

What is claimed is:

1. Rotary apparatus comprising:

an elongated, thin flexible element arranged in a pair of adjoining alternate loops,

an outer roller in one of said loops and an inner roller in the other of said loops, said element in said loops being wrapped on more than one half of the circumference of the surface of the respective rollers,

guide means maintaining the centers of said rollers on concentric paths about a common center while said rollers roll relative to said guide means, said guide means including an inner guide surface concentric with said common center at a radius less than the radius of both of said paths,

said element extending between said one loop at the surface of said outer roller and said inner guide surface at a first location and being interposed between said inner roller and said inner guide surface at a second location, said locations being spaced circumferentially on said inner guide surface, and

tension means applying tension in said band to clamp said element between said inner and outer rollers and between said inner roller and said inner guide surface, whereby said rollers rotate relative to each other and relative to said guide means at a predetermined rate.

2. The rotary apparatus according to claim **1** wherein said guide means includes an outer guide surface extending continuously around said common center and cooperating with said outer roller to maintain said roller in said path.

3. The rotary apparatus according to claim **2** wherein said outer guide surface is substantially circular and concentric with said common center, said outer guide surface having a circumference greater than the circumference of said inner guide surface.

4. The rotary apparatus according to claim **1** wherein said guide means includes an outer guide surface concentric with said common center, the radial distance between said inner guide surface and said outer guide surface being less than the sum of the diameters of said rollers.

5. The rotary apparatus according to claim **4** wherein said outer guide means includes a thin flexible band, means supporting said band on a cylindrical surface concentric with said common center, said band extending continuously around said outer roller and said band supporting surface.

6. The rotary apparatus according to claim 4 including means for rotating said inner guide surface relative to said outer guide surface.

7. A mechanism comprising support means forming an arcuate support surface, guide means having a substantially arcuate surface, said guide means surface and said support means surface having a common center of curvature and at least one of said surfaces being movable relative to the other about said center, a first roller and a second roller interposed between said support surface and said guide surface, the sum of the diameters of said rollers being greater than the radial distance separating said guide surface and said support surface, drive means wedging said rollers in opposition to each other and each in opposition to one of said surfaces, said drive means including a thin elongated flexible element cooperating with said rollers and said arcuate surfaces to cause predetermined rotation of said rollers relative to each other and relative to said surfaces while progressing as a unitary cluster along an arcuate path between said surfaces.

8. The mechanism according to claim 7 wherein said element extends between one of said rollers and said guide surface in an outer loop and between the other of said rollers and said support surface in an inner loop.

9. The mechanism according to claim 7 wherein said support surface is provided on the periphery of a rotary drum, and including means for rotating said drum relative to said guide surface.

10. The mechanism according to claim 9 wherein said element is secured on said support surface; said element extending around one of said rollers in an outer loop, and around the other of said rollers in an inner loop, said element also extending between said guide surface and said outer roller and extending between said support surface and said inner roller, said drum being journaled for rotation about an axis passing through said center of curvature, whereby the drum rotates about said axis to produce translation of said rollers along said path.

11. Motion transmitting mechanism comprising:

bearing means having inner and outer curved bearing surfaces, said bearing surfaces being arranged in opposed relation,

a pair of rollers interposed between said bearing surfaces in a cluster, the sum of the diameters of said cluster rollers being greater than the radial distance separating said bearing surfaces, the diameter of each roller being less than the radial distance separating said bearing surfaces,

a thin flexible band arranged in a pair of alternate adjoining loops between said outer and inner bearing surfaces, said cluster rollers each being positioned within one of the band loops and supporting said band on their respective surfaces, portions of said band passing between said outer bearing surface and one roller and between the other roller and the inner bearing surface and extending away from said roller cluster and being fixed against displacement relative to said inner bearing surface, means for rotating one of said bearing surfaces relative to the other bearing surface, whereby rotary motion is transmitted to said cluster rollers.

12. The motion transmitting mechanism according to claim 11 wherein said band is under sufficient tension to prevent relative rotational motion between said cluster rollers and said band.

13. The motion transmitting mechanism according to claim 11 wherein said inner and outer bearing surfaces have a fixed curvature and said bearing surfaces are concentric.

14. The motion transmitting mechanism according to claim 11 wherein said outer bearing surface has a variable curvature and includes a flexible guide.

15. The motion transmitting mechanism according to claim 11 wherein said bearing means include a wheel and a ring, said inner bearing surface being on the periphery of said wheel, said outer bearing surface being on the interior of said ring.

16. The motion transmitting mechanism according to claim 15 including at least one additional roller cluster interposed between said bearing surfaces, said clusters being spaced apart around said wheel surface, said band being arranged in a pair of alternate adjoining loops at each of said roller clusters, whereby said clusters progress relative to said wheel surface while maintaining the spacing between the clusters.

17. Rotary motion apparatus comprising a first rotary shaft and a second rotary shaft, said first and second shafts being mounted for rotation about a common axis, said first shaft having a cylindrical support surface concentric with said axis, inner and outer rollers associated with said support surface, a thin flexible endless drive band intertwined around said inner and outer rollers in alternate adjoining loops, said drive band maintaining said outer roller spaced from said support surface, means mounting a guide surface for rotation coaxially of said support surface, an endless restraining band encircling said guide surface and said outer roller and restraining said inner and outer rollers for translation along circular paths concentric with said axis, means preventing slippage of said bands relative to said rollers and said support surface thereby causing predetermined relative rotation between said rollers and said support surface, and means for transmitting rotary motion between said inner roller and said second shaft, whereby a differential in speed is transmitted between said first and second shafts.

18. The apparatus according to claim 17 wherein said restraining band has a length different from said drive band, said restraining band and said drive band being maintained in tension, whereby slippage between said rollers and said bands does not occur.

19. The apparatus according to claim 17 wherein said support surface includes a wheel, said restraining band being supported on the periphery of said guide surface, said guide surface having a larger diameter than said wheel, and said inner roller having a smaller diameter than said outer roller, whereby amplified motion is transmitted between said first and said second shafts.

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