United States Patent [19]

Darnell

[11] **3,728,912**

[45] **Apr. 24, 1973**

[54] BICYCLE TRANSMISSION

[76] Inventor: **Rex C. Darnell,** 8960 Jackson Road, Dexter, Mich.

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[21] Appl. No.: **87,838**

[52] **U.S. Cl.** **74/769,** 74/332, 74/740, 280/236

[51] **Int. Cl.**.....**F16h 57/10,** F16h 5/36, B62m 11/14

[58] **Field of Search**.....74/768, 769, 764, 74/765, 753

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Assistant Examiner—Thomas C. Perry
Attorney—Beaman & Beaman

[57] **ABSTRACT**

A bicycle ratio changing transmission of improved efficiency and cost of construction disposed within the crank hanger housing and characterized by the use of a minimum of inexpensive, sturdy components.

4 Claims, 14 Drawing Figures

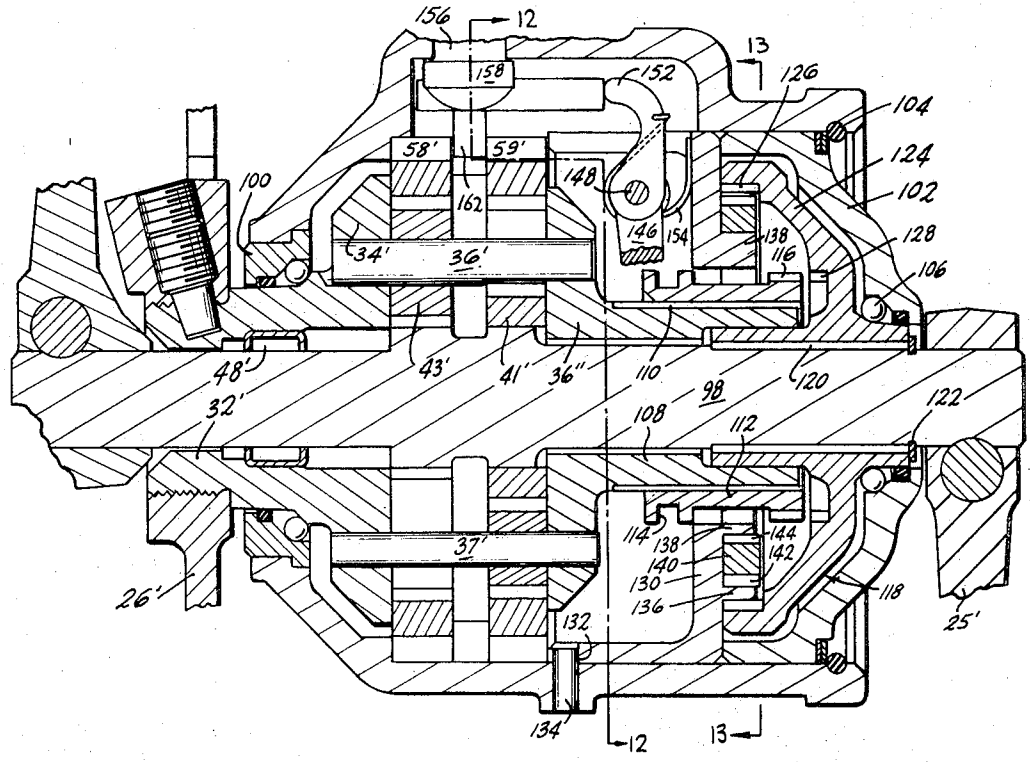
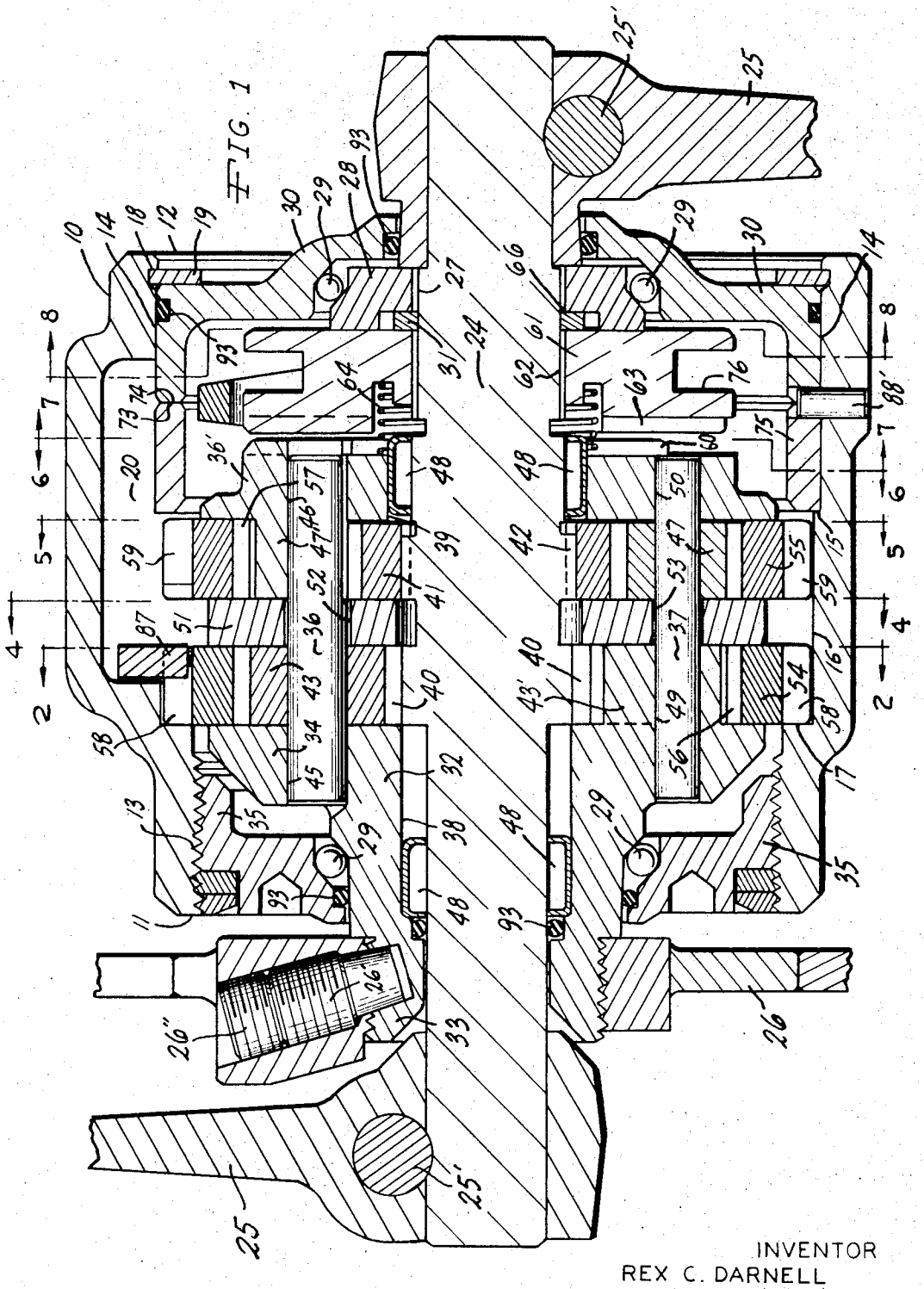


FIG. 1



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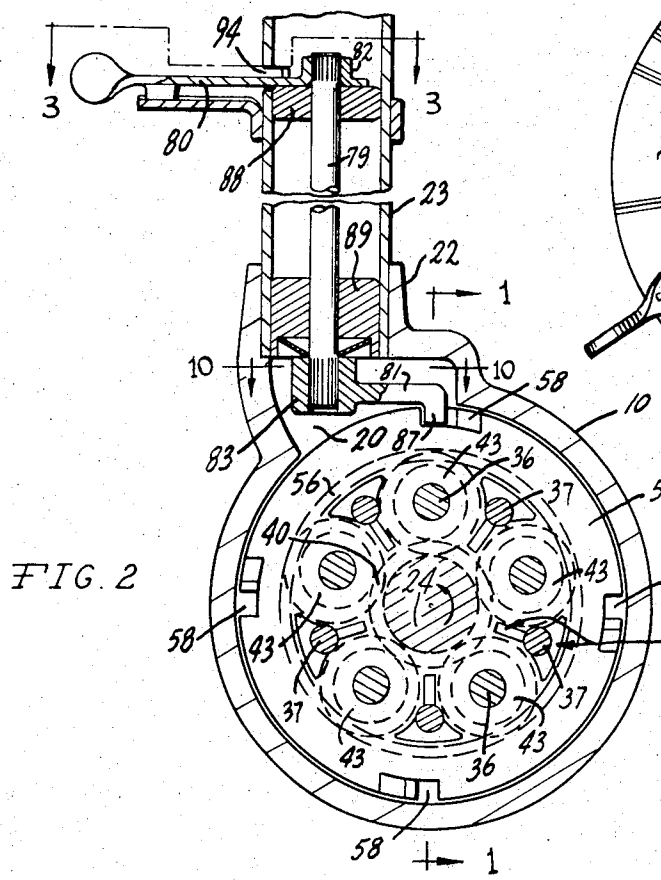


FIG. 2

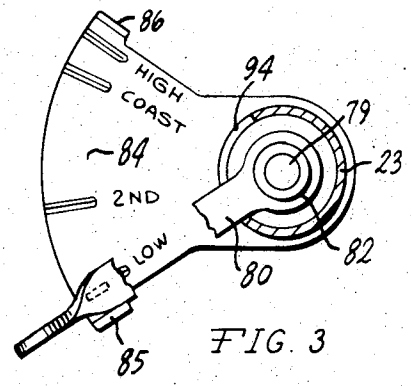


FIG. 3

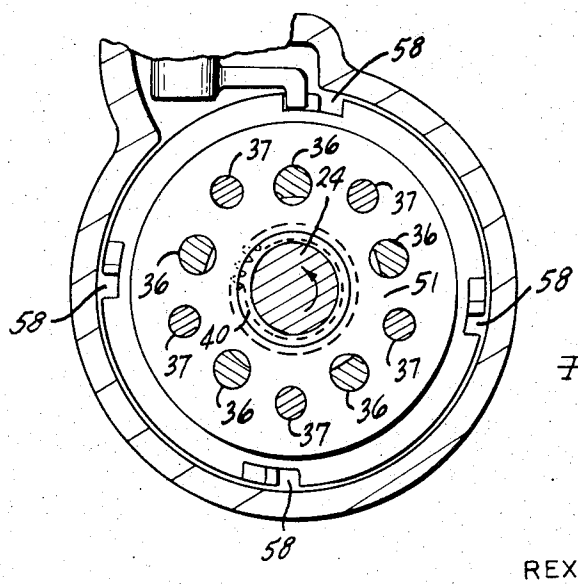


FIG. 4

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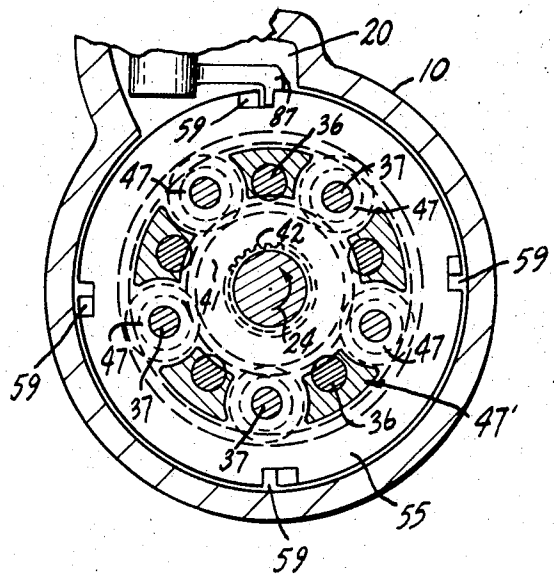


FIG. 5

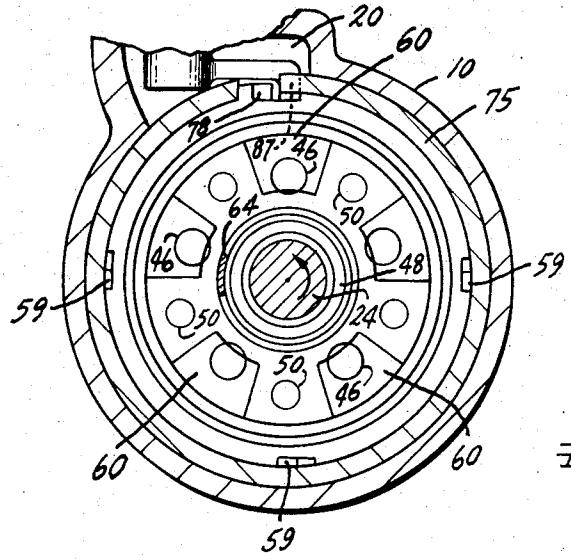


FIG. 6

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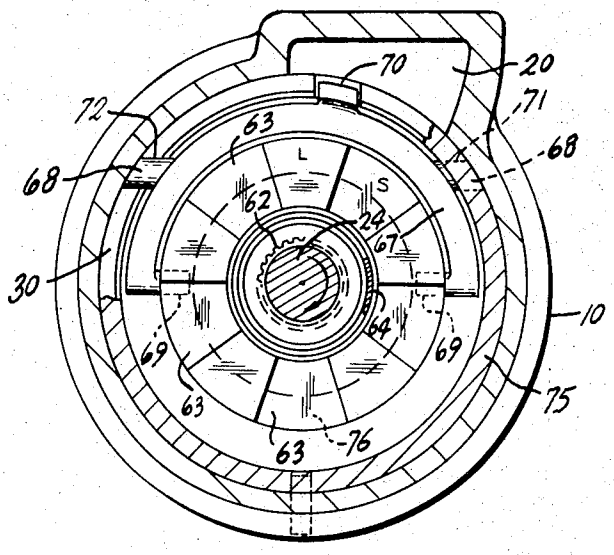


FIG. 7

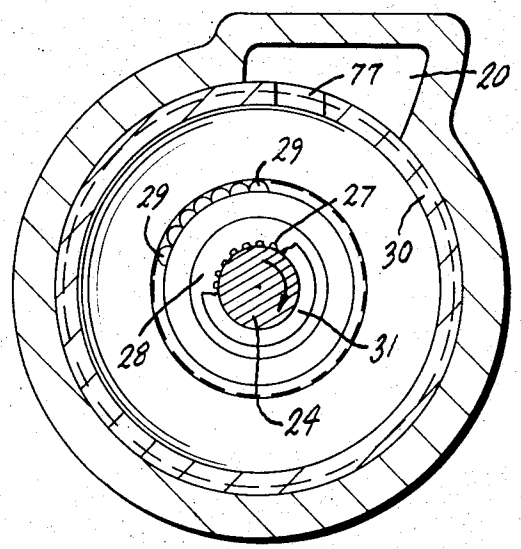


FIG. 8

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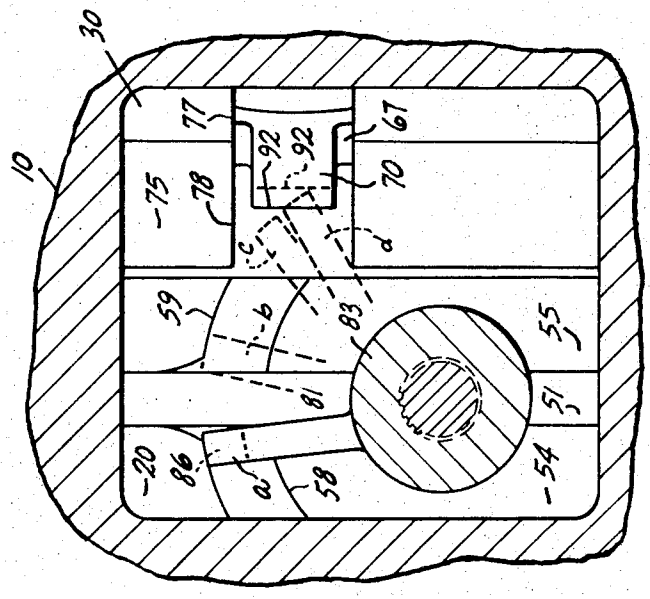


FIG. 10

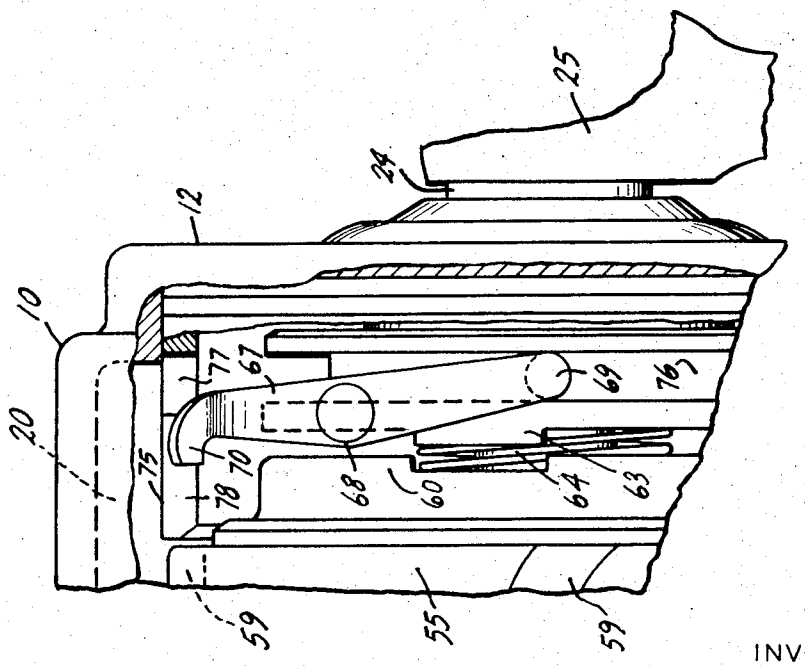


FIG. 9

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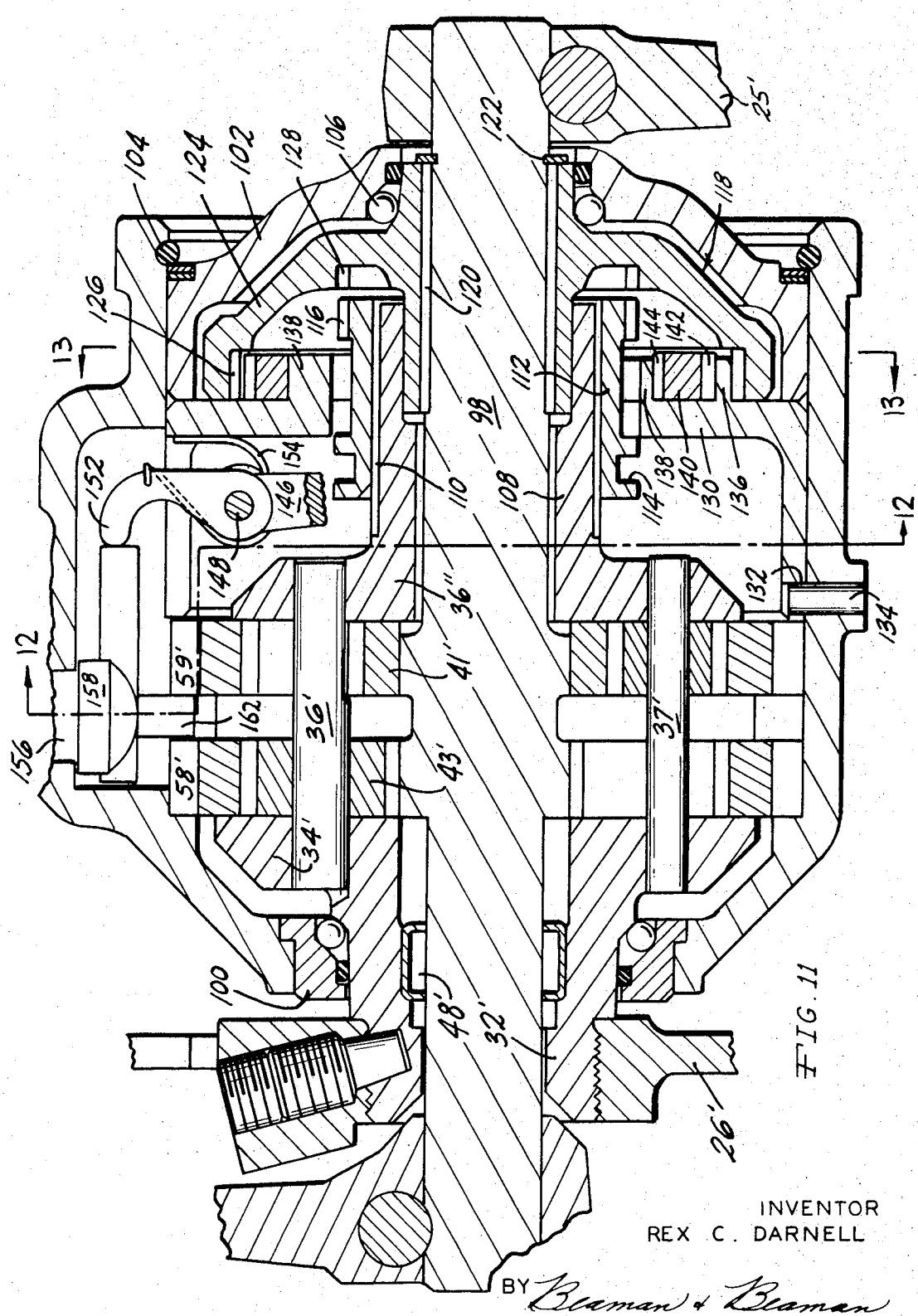


FIG. 11

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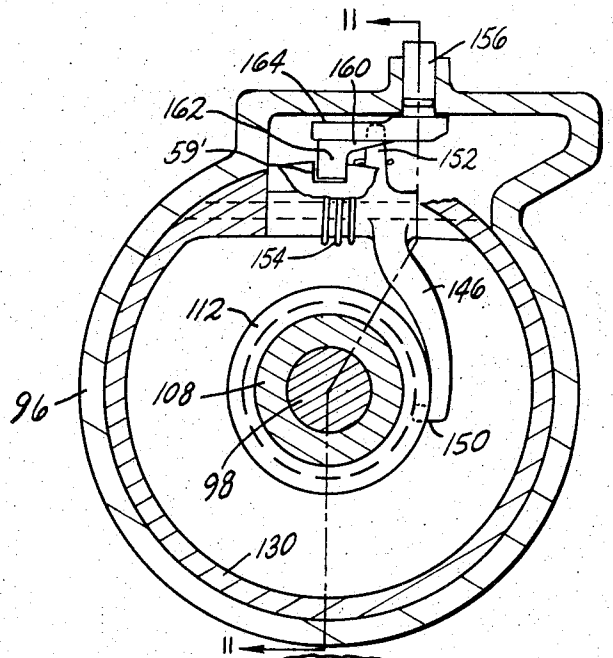


FIG. 12

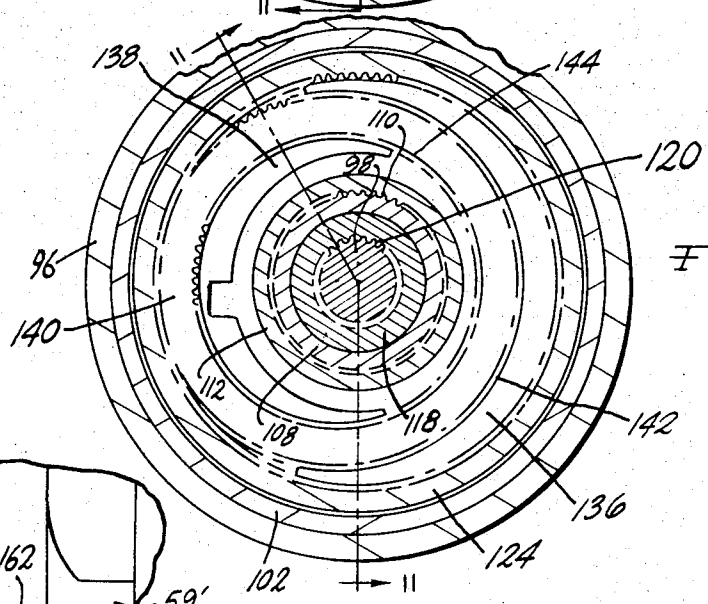


FIG. 13

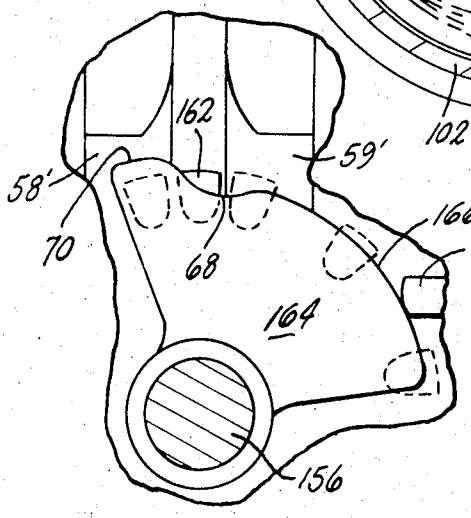


FIG. 14

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BICYCLE TRANSMISSION

BACKGROUND OF THE INVENTION

The desirability of equipping a manpowered bicycle and the like with a ratio changing transmission has long been recognized. With the introduction of the so-called safety bicycle having a crank hanger housing supported on the frame between the wheels, there were early proposals involving the location of ratio changing transmissions between the crank and crank sprocket driving the chain to the rear wheel. See U. S. Pat. Nos. 504,168, 516,933 and 2,392,250.

Notwithstanding the obvious advantages of having the ratio changing transmission disposed in the crank hanger housing, such an arrangement has lacked commercial acceptance because of limited ratio selection, among other things. In lieu thereof a much more cumbersome and hazardous form of transmission has come into general use involving a series of sprocket sets with selective chain shifting mechanism.

While there has been limited commercial acceptance of rear wheel hub ratio changing devices, a survey of bicycle servicing shops indicates that small, delicate and expensive parts of such mechanisms are highly prone to fracture and excessive wear. The multiplicity of parts in this form of ratio changing transmission appears to be the principle course of trouble. Also, the control means must extend from adjacent the hub of the rear wheel forward to a point within the reach of the rider. As such control mechanism is exposed to dirt and moisture, sticking and malfunction are frequently experienced.

SUMMARY OF THE INVENTION

The basic concept herein involved is found within the disclosures of the aforesaid patents showing ratio changing mechanism disposed with the crank hanger housing. According to the invention this early concept is being presented in a more practical form with the contribution to the art being deemed to reside in the provision of adequate ratio selection through a rugged and novel twin planetary gearing system coupled with a positive drive clutch. The entire ratio changing mechanism is encased in an oil bath to insure smooth, low friction operation contributing highly to the mechanical efficiency and reduced manual effort to propel the vehicle. At the same time it has been possible to avoid the difficulties connected with the control mechanism extending to the hub of the rear wheel through the use of a simple, strong torque rod protected against damage and corrosion by being located within a frame tube opening into the crank hanger housing.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a vertical cross section taken through the crank hanger housing on the center line of the pedal shaft, on line I—I of FIG. 2,

FIG. 2 is a vertical cross section taken on line 2—2 of FIG. 1 and along the vertical axis of a frame tube opening into the crank hanger housing in which tube the control mechanism is disposed,

FIG. 3 is a sectional view taken on line 3—3 of FIG. 2 showing the quadrant shift,

FIGS. 4, 5 and 6 are vertical cross sections taken on lines 4—4, 5—5 and 6—6, respectively, of FIG. 1 showing the planetary gearing system,

FIG. 7 is a vertical cross-sectional view taken on the same plane as line 6—6 of FIG. 1 but looking in the opposite direction showing the clutch mechanism,

FIG. 8 is a vertical cross section taken on line 8—8 of FIG. 1,

FIG. 9 is a fragmentary plan view of the clutch mechanism with the crank hanger housing shown broken away,

FIG. 10 is a horizontal cross-sectional view taken on line 10—10 showing the lower end of the shaft lever,

FIG. 11 is a vertical, diametrical cross section of a four speed embodiment of the invention,

FIG. 12 is an elevational cross section view of the cam operated shift mechanism taken along line 12—12 of FIG. 11,

FIG. 13 is an elevational view of the overdrive ring gear arrangement taken along line 13—13 of FIG. 11, and

FIG. 14 is a plan view of the shift mechanism and control cam employed with the embodiment of FIG. 11.

DESCRIPTION OF THE ILLUSTRATED EMBODIMENTS

Referring to the illustrated form of the invention shown in FIGS. 1–10, the crank hanger housing 10 is of cast metal with open opposite sides 11 and 12 for the assembly of the transmission mechanism therein. As shown, the side 11 has a threaded bore 13. Side 12 has a bore 14 terminating at the shoulder 15 of the bore 16 of less diameter. Bore 16 terminates at the shoulder 17 which defines the inner end of the threaded bore 13. A groove 18 is provided in the bore 14 to receive the snap-in retaining ring 19. The cylindrical configuration of the bores 14 and 16 are interrupted by the cavity 20 into which the neck portion 22 of the housing 10 opens to accommodate the lower end of the tubular frame member 23 extending upwardly and forwardly to the front wheel fork portion of the bicycle frame. The means of attaching other frame members to the housing 10 are not shown and may take many forms.

A drive shaft 24 is supported in the housing 10 for manual rotation through the pedal arms 25 removably attached at opposite squared ends of the shaft 24 by members 25' in a well known manner.

Ball bearing support structure is indirectly provided through the planetary gearing system for that end of the shaft 24 embraced by the sprocket drive 26. The other end of the shaft 24 has a spline portion 27 which carries the inner ball race 28 for the balls 29. The outer ball bearing support 30 has a close sliding fit in the bore 14, being held in place by the retaining ring 19. The clutch stop 31 determines the inner position of the inner ball race 28. The inner ball race 32 structure has a threaded hub 33 carrying the sprocket 26 and a flange portion 34 which is a component of the planetary gearing system. An adjustable threaded outer ball bearing support 35 is carried in the threaded bore 11. A set screw 26' and lock screw 26'' lock the sprocket 26 to the hub 33.

Pins 36 and 37, which carry the star gears of the planetary gearing system, are supported at one end in the flange 34 and at the opposite end in the support

36'. Needle bearings 48 are provided between the shaft 24 and the inner bores 38 and 39 of the race 32 and support 36', respectively, to complete the support of the shaft 24 in the ball bearing structure. It will be noted that the inner ball race 28 is directly driven by the shaft 24 while the inner ball race 32 is indirectly driven by the shaft 24 through the planetary gearing system or the positive clutch as will be hereinafter described.

The planetary gearing system provides low and second gear of the transmission while the positive clutch provides high gear.

Low gear sun gear 40 is preferably machined integral with the shaft 24 while second gear sun gear 41 is carried on the spline 42. Low gear star gears 43 mesh with sun gear 40 and are carried on pins 36 supported and press fitted at opposed ends in the bores 45 and 46 in the flange 34 and support 36', respectively. Star gears 47 are similarly carried on pins 37 supported and press fitted at opposed ends in the bores 49 and 50. Lugs 43' and 47' shown in FIGS. 1, 2 and 5 are of slightly greater axial projection than the width of the associated gears 43 and 47 and thus providing the latter with running side clearance. A spacer 51 is disposed between the gears 40 and 41 having bores 52 and 53 to provide clearance for the pins 36 and 37, respectively.

To complete the planetary gear system, ring gears 54 and 55 are provided with internal teeth 56 and 57, respectively, with which the star gears 43 and 47 mesh. The ring gears 54 and 55 are provided with a series of slots 58 and 59, respectively, disposed about their periphery.

The positive clutch structure comprises jaws 60 defined on the outer face of the support 36'. High gear clutch 61 is carried for axial movement on the spline 62 on shaft 24 and has jaws 63 complimentary to the jaws 60. A spring 64 normally holds the jaws 60 and 63 out of mesh and urges the clutch 61 against the clutch stop 31 supported against axial displacement in a groove 66 defined between the spline 27 and 62. The lever 67 for shifting the clutch 61 against the tension of the spring 64 has trunnions 68, fingers 69 and a projection 70. Bearings for the trunnions 68 are provided by complimentary semicylindrical depressions 71 and 72 provided in the opposed faces 73 and 74 of the spacer support ring 75 and support 30. Fingers 69 are disposed in the groove 76 of the clutch 61 to move the same into high gear position. The lever 67 is rocked by a force applied to the projection 70; the support 30 being relieved at 77 to provide clearance for the projection 70 as is the spacer support ring 75 at 78.

The mechanism for selectively controlling the relationship between the shaft 24 and the sprocket 26 comprises a control shaft 79 longitudinally disposed preferably within the frame member 23 to dispose the upper shaft lever 80 within the convenient reach of the bicycle rider. Movement of the lever 80 is transmitted by the shaft 79 to the lower shaft lever 81; levers 80 and 81 having hubs 82 and 83, respectively, into which the knurled ends of the shaft 79 are press fitted.

A shift quadrant 84 is fixed to the frame member 23 and provided with stops 85 and 86 to determine the arc through which the lever 80 may be moved. When the lever 80 is adjacent the stop 85, the transmission is in low gear. When it is adjacent the stop 86 the transmission is in high gear. Between the stops 85 and 86 second gear and "coast" positions are indicated. An actuator nose 87 is provided on the lower shift lever 81 which has an arcuate movement across the plane of rotation of each of the elements 54, 55 and 61, the nose 87 moving into and through the slots 58 and 59 of the ring gears 54 and 55 to selectively arrest the latter. The pin 88' holds the support 30 and spacer 75 against rotation with their clearance slots 77 and 78, disposed in the arcuate path of movement of the nose 87. Suitable bearing members 88 and 89, press fitted in the frame member 23, support and center the control shaft 79. A spring washer 90, in the recess 91 of the member 89, tensions the shaft 79 and holds the same in the selected positions.

OPERATION

When the upper shift lever 80 is in the position shown in FIGS. 1 and 3, the nose 87 is disposed in one of the four slots 58 of the ring gear 54 to arrest rotation of the gear 54. In this "low speed" position the star gears 43 will bodily move within the ring gear 54 to rotate the inner ball race 32 and sprocket 26 through the pins 36 with maximum gear reduction.

Star gears 47 and ring gear 55 rotate idly. Rotation of the lever 80 from the position shown in FIG. 3 from "low" to "2nd" position will shift the nose 87 from the position shown in FIG. 1 to a position in one of the slots 59 of the ring gear 55 to arrest the rotation of the latter. Now the star gears 47 will bodily move within the ring gear 55 to rotate the race 32 and sprocket 26 through the pins 37 with lesser gear reduction with star gears 43 and ring gear 54 rotating idly.

During the shifting of the nose 87 back and forth between the slots 58 and 59 of the ring gears 54 and 55, the clutch 61 will be in its inoperative position as shown in FIG. 1. To engage the clutch 61 to effect "high" gear with a one to one ratio between the shaft 24 and sprocket 26, the lever 80 is moved to the "high" gear position shown in FIG. 3 adjacent the stop 86. When thus positioned the nose 87 has engaged the projection 70 moving the same from the clearance 78 into the clearance 77 to rock the lever 67 about it trunnions 68 to move the fingers 69 from the position shown in FIG. 3 to axially slide the clutch 61 to the left on its spline against the tension of the spring 64. This movement will bring the jaws 60 and 63 into positive driving relation to rotate the support 36' which in turn rotates the race 32 through the pins 36 and 37. All planetary gears rotate idly.

When the lever 80 of FIG. 3 is in the "coast" position the nose 87 is disposed to the right of the slot 59 in ring gear 55 as shown in FIG. 9 and to the left of projection 70 and out of effective engagement therewith. Referring to FIG. 10, the lower shift lever 81 is shown in full line in position *a*, the "low" gear position, with nose 87 arresting movement of the ring gear 54 for the reason the nose 87 is disposed in a slot 58. When the lever 81 is in the dotted outline *b* position the nose 87 will be disposed in rotation arresting position in a slot 59 of the ring gear 55. In the *c* position, the "coast" position, the nose 87 is out of the slot 59 and out of engagement with the surface 92 of the projection 70 as shown in full line. When the upper shift lever 80 is moved adjacent the stop 86 of FIG. 3, the lower shift lever 81 will be in the

d, the "high" gear position, of FIG. 10 with the nose 87 having rocked the lever 67 through movement of the surface 92 into the dotted line position.

It will be understood that when the rider of the bicycle is coasting the rider's feet are resting on the pedals and the shaft 24 is either not rotating or is rotating at a lower speed than the sprocket 26. Under such conditions the star gears 43 and 47 will be driven by the sprocket 26 about the axis of the shaft 24, the star gears rolling on the sun gears 40 and 41 without delivering any torque to the shaft 24.

Sealing rings 93 enable the housing 10 to receive a supply of oil to continuously bathe the mechanism disposed therein. By locating the shaft 79 in the frame member 23, it is substantially protected from the elements. Standard or suitable sealing means, not shown, adjacent the slot 94 in the frame member 23 through which the upper shift lever 80 operates, avoid corrosion and contamination of the oil bath.

Those skilled in the art will readily appreciate that the illustrated form of the invention is of a design which will enable the bearing structure to provide extreme usage without failure. The torque imposed on the shaft 24 is always transferred to the sprocket 26 through five or more pins 36 and 37 equally spaced and rigidly mounted in the inner race 32 and support 36'. In low and 2nd gears, the torque of the shaft 24 is divided between five gears. When the clutch 61 is in operative position to provide "high" gear all substantial stress is removed from the planetary gear system. This is also true during "coasting." Also, substantial stability is imparted to the planetary gear system through the association of the pins 36 and 37 with the components 32, 51 and 31' and the sandwiching effect provided thereby with respect to the star and ring gears.

FIGS. 11 through 14 illustrate another embodiment of the invention wherein the transmission includes four speeds, the fourth gear constituting an "overdrive." With reference to FIG. 11, the housing 96 is of a configuration generally similar to that of housing 10 of the previously described embodiment. The housing 96 supports the rotatable shaft 98 upon antifriction bearings, and the general arrangement of the shaft 98, the housing 96, pedals, chain sprocket, and planetary gearing system for the two lower gears is similar to that of the previously described embodiment and components identical or similar to those previously described are indicated by primed reference numerals.

As will be apparent in FIG. 11, the antifriction bearing support of the shaft 98 includes an outer bearing race 100 which may be pressed or otherwise mounted in the housing 96 which includes ball members rotatably supporting the inner ball race 32'. Also, the structure includes at its right end, FIG. 11, an outer ball race cap 102 maintained in position by the snap ring 104 for cooperation with the ball bearing elements 106.

In the four speed embodiment the support 36'' includes an axially extending portion 108 of a cylindrical configuration having splines 110 defined on the exterior surface thereof. An axially slidable sleeve 112 is mounted upon the portion 108 having internally formed splines for cooperating with the splines 110 to key the sleeve with respect to the support 36'' for preventing relative angular rotation, while permitting axial displacement thereto. An annular groove 114 is defined in the sleeve 112 for cooperation with a shift lever as will be later described.

The sleeve 112 is also provided with exterior gear teeth 116 at the end opposite to the groove 114, and the teeth 116 function to selectively engage drive members to drive the sleeve 112 at the selective angular rate desired depending upon the position of the gear shift mechanism.

The inner ball race 118 is keyed to the shaft 98 at 120 and axially maintained thereon by a snap ring 122. As will be apparent from FIG. 11, the inner ball race 118 includes a radially extending portion 124 having annular and radially inwardly disposed gear teeth 126 formed thereon wherein the teeth 126 define a ring gear which will rotate directly with the shaft 98. Also, the inner race 118 is annularly recessed at 128 to form splines in axial alignment with the sleeve teeth 116, and upon the sleeve 112 being shifted to the right with respect to the position shown in FIG. 11, direct driving interconnection between the inner ball race 118 and the sleeve is produced.

A gear guide and bearing element 130 is mounted within the housing 96 and includes an axially extending portion having a hole 132 defined therein for cooperation with the positioning pin 134. The gear guide 130 includes an outer axially extending portion 136 and an inner axially extending portion 138 which are each of a crescent configuration as will be appreciated from FIG. 13, and are each in alignment with the gear teeth 126.

Power transmission from the ring gear teeth 126 to the sleeve 112 is accomplished by an intermediate sun-ring gear 140 having external teeth 142 defined thereon and internal gear teeth 144 formed on the inner diameter. The gear 140 is eccentrically related to the axis of the shaft 98, and maintained in this eccentric relationship by the gear guide portions 136 and 138, which function as bearings for the gear 140. It will be noted from FIG. 13 that the gear guide portions extend more than 180° in a circumferential direction and the gear guides maintain the sun-ring gear teeth 142 in mesh with the inner ball race gear teeth 126 at the upper region of the inner ball race and maintain the inner gear teeth 144 in alignment with the sleeve gear teeth 116, as will be appreciated from FIG. 11. In that the cylindrical segment surfaces of the gear guides 136 and 138 extend through more than half of the periphery of the associated gear teeth, adequate bearing support of the sun-ring gear 140 is achieved.

Shifting of the sleeve 112 is accomplished through a lever 146 pivotally mounted upon the housing 96 by a pivot pin 148. The lever 146 includes a radially inwardly disposed projection 150 received within the groove 114, FIGS. 11 and 12, and the lever also includes a cam follower portion 152 located in the upper portion of the housing 96. A spring 154 of the torsion type is disposed about the pivot 148 and imposes a biasing force upon the lever 146 tending to rotate the lever in a counterclockwise direction, FIG. 11.

Shifting of the transmission between the various speed changing positions is accomplished by a vertically disposed actuating shaft 156 pivotally mounted in the housing 96. The shaft 156 includes a lower head 158 located within the housing upon which the radially extending arm 160 is defined. The arm 160 includes a stop 162 adapted to be selectively received within the

slots 58' or 59' of the planetary gearing systems constituting low gear and second gear in the manner previously described. Additionally, the head 158 includes a cam 164 having a configuration which will be apparent from FIG. 14 including a peripheral surface which is engaged by the shift lever cam follower portion 152. The cam follower 152 is maintained in engagement with the cam periphery by the spring 154 and the radial distance between the cam surface portion 166 and the axis of the shaft 156 is such that when the arm 160 is shifted between the positions for selectively connecting the stop 162 to the first or second planetary gear systems to the sprocket 26', the sleeve 112 will be positioned as shown in FIG. 11, i.e., out of engagement with both teeth 128 and gear teeth 144.

When the head 158 is rotated to the third or direct drive condition the notch 168 defined in the cam periphery will be in alignment with the cam follower 152 permitting the cam follower to radially move toward the shaft 156 and thereby shift the sleeve 112 to the right permitting engagement between the sleeve gear teeth 116 and the inner ball race teeth 128. This relationship establishes a direct drive connection between the shaft 98 and the support 36'', which is directly transmitted to the sprocket 26'.

If the rider desires to shift the transmission into fourth or "overdrive" gear, the shaft 156 is rotated further in the clockwise direction, FIG. 14, which causes the cam follower 152 to ride out of the notch 168 and up upon the cam peripheral surface 166, which causes the lever 146 to rotate in a clockwise direction, FIG. 11, sufficiently to align the sleeve gear teeth 116 with the sun-ring gear 140 establishing a meshing relationship between gear teeth 116 and gear teeth 144. Due to the difference in radii between gear teeth 126 and gear teeth 116 an "overdrive" step-up power transmission is transferred to the support 36'' from the shaft 98, permitting the sprocket 26' to rotate at an angular velocity greater than that of the pedal shaft.

When it is desired to down shift the shaft 156 is rotated in the opposite or counterclockwise direction, FIG. 14, which will permit the desired gearing portion of the transmission to be operatively connected to the sprocket.

It will therefore be appreciated that the embodiment illustrated in FIGS. 11 through 14 permits a concise bicycle transmission to be manufactured having a four speed capability. The components of this embodiment are concisely related such that the transmission may be readily mounted upon the bicycle pedal hanger, and as the adjustment between the various speeds is sequentially accomplished by rotation of the shaft 156, shifting is facilitated in a logical manner.

While the invention is particularly adapted for use in a bicycle, a transmission has been provided which may find use in other fields wherein concentricity between input and output shafts is desired. While only two embodiments of the invention have been illustrated it is appreciated that various changes may be apparent to those skilled in the art without departing from the spirit and scope of the invention.

I claim:

1. In a transmission for a bicycle and the like, a housing, a power input shaft extending through said housing, a power take-off embracing said shaft and disposed at one side of said housing and intermediate the ends of said shaft, bearing structure supporting said power take-off on said housing for rotational movement relative to said shaft, an bearing support upon which said power take-off shaft is carried, a first planetary gearing system located within said housing selectively connecting said shaft and said power take-off, a second planetary gearing system located within said housing selectively connecting said shaft and said power take-off, a support mounted in said housing concentrically related to said shaft and rotatable thereabout, said support being connected to said power take-off, a sleeve member axially slidably mounted upon said support and angularly fixed thereto, gear teeth defined on said sleeve member, key means fixed to said shaft and axially aligned with said sleeve member gear teeth for selective engagement therewith, a ring gear concentrically fixed upon said shaft, speed step-up gear means located within said housing radially aligned with said ring gear and selectively engageable with said sleeve member gear teeth at one axial position of said sleeve member, and sleeve member positioning means mounted within said housing selectively axially positioning said sleeve member upon said support for selective driven relationship with said key means, said step-up gear means, or a neutral position, and gear shifting means controlling the position of said sleeve member positioning means and operation of said first and second planetary gearing systems.

2. In a transmission as in claim 1 wherein said speed-up gear is in the form of a combined sun-ring gear, and gear support means within said housing eccentrically supporting said sun-ring gear with respect to the axis of said shaft.

3. In a transmission as in claim 1 wherein said gear shifting means includes a cam surface, and a cam follower mounted upon said sleeve member positioning means engaging said cam.

4. In a transmission as in claim 1, an inner ball race keyed to said shaft, said key means and said ring gear being defined on said inner ball race.

* * * * *