PATENT SPECIFICATION

(11) 1 581 560

1 581 560

(21) Application No. 9593/77 (22) Filed 18 March 1977
(31) Convention Application No. 668 043
(32) Filed 18 March 1976 in
(33) United States of America (US)
(44) Complete Specification published 17 Dec. 1980
(51) INT CL³ B62M 1/00
(52) Index at acceptance B7E 502 600 601 DP

(19)

(54) DRIVE SYSTEMS FOR RIDER/PROPELLED VEHICLES

(71) I, LAWRENCE GEORGE BROWN, citizen of the United States of America of 3285 Old Highway 395 North, Carson City, State of Nevada, United States of America, do hereby declare the invention, for which I pray that a patent may be granted to me, and the method by which it is to be performed, to be particularly described in and by the following statement:—

The invention relates to drive systems for rider-propelled vehicles such as bicycles, tricycles or the like.

While cycling has been a universal and common mode of transportation for many decades, the drive systems employed at this time as discussed in United States Patent No. 3,889,974, still basically comprise an annular driving sprocket wheel operated by pedal supporting crank arms, an annular driven rear axle sprocket wheel (one speed) or multiple driven varying size rear axle sprocket wheels, and a closed loop chain operably associated with the sprocket wheels. However, the prior art shows that literally thousands of attempts have been made to provide a variety of other types of drive systems for bicycles and the like. At least as early as the Smith United States Patent No. 596,289 of 1897, it has been recognized that the ability of a bike rider to apply input force through rotary crank arms is limited by the rotary position thereof. Thus, the use of various elliptical or oval driving sprocket wheels has been suggested to variously apply the available input force to an annular driven rear axle sprocket wheel as disclosed in United States Patents No. 596,289 of 1897, 885,982 of 1908, 2,693,119 of 1954, 2,827,797 of 1958, 3,259,398 of 1966, and 3,375,022 of 1968.

In addition to the presently commercially available basic single chain drive system, the concept of employing two separate drive systems which are each operable to rotate the one wheel of a bicycle during separate 180° rotation of the crank arms has been known since at least as early as the Crane United States Patent No. 258,559 of 1882. In such systems, each of the crank arms are connected to one of two drive sprocket means by cable or chain type driving means which are effective to transmit force to the wheel by movement from a radial innermost position to a radial outermost position relative to the wheel only during downward movement of an associated lever arm from an upper position to a lower position and are returned from the outermost position to the innermost position by use of one-way clutch means or the like during upward movement of the associated lever arm from the lower position to the upper position. Variations of such drive systems are disclosed in United States Patent Nos. 258,559, 527,396, 598,246, 636,184, 849,342, 3,004,440, 3,375,023, 3,759,543, 3,834,733, 3,888,512 and 3,889,974. Such drive systems have utilized oscillatory type lever arms, reciprocatory type lever arms, and rotary type lever arms. In order to provide a range of variable speeds, various apparatus has been suggested including changing the pedal position relative to the lever arm (United States No. 258,559); changing the location of connection of the drive system on the lever arm (United States No. 527,396, 849,342, 3,375,023, 3,759,543 and 3,834,733), including multiplication of a selected speed (527,396); changing the location of an idler pulley or wheel member engaged with a chain intermediate the lever arm and the drive sprocket means (United States No. 636,184); changing the location of the pivotal axis of an intermediate pivotal link driven by the lever arms (United States No. 3,004,440); changing the fulcrum point of the lever arm (United States No. 3,888,512); and changing the radial location of application of force to a driven shaft intermediate the lever arm and the drive sprocket means (United States No. 3,889,974).

Thus the basic form of the drive systems of present day commercially available bicycles has been known for a long period of time. Essentially, all present day commercially available bicycle drive systems of which I am aware involve a force input means in the form of a driving sprocket wheel attached to a crank shaft driven by crank arms having pedals thereon with the force input sprocket wheel connected by a chain member to force output means in the form of a driven sprocket wheel mounted on and drivably connected to a rear wheel axle. Various drive system devices and apparatus have been proposed for providing varying mechanical advantage between the driving sprocket wheel and the driven sprocket wheel for the purpose of increasing or decreasing the speed of rotation of the rear wheel. Current conventional multiple speed (e.g., 3, 5 and 10 speed) bicycles utilize a drive system known as the derailleur system, which comprises a stack of varying diameter and varying tooth number output sprocket wheels. Such systems have many disadvantages, including cost, ease of maintenance and repair, reliability, limitations on available speed selection and shifting from one speed to another, and lack of adaptability to particular requirements of particular bike riding conditions and individual preferences and abilities of different bike riders.

According to the invention, I provide a drive system, for a rider-propelled vehicle having a driven wheel, operable by a rider by applying force to input means which cause unidirectional rotational movement of the driven wheel and comprising:

variable force transfer means operable by the input means to transfer input force from the input means to the driven wheel so that during the application of force to the input means there is no break in application of forward driving force to the driven wheel;

the variable force transfer means being adapted to control the amount and time of application of input force to the driven wheel during each input force cycle; and

to control the rotary position of the driven wheel relative to the position of the force input means during each force input cycle, wherein said variable force transfer means comprises cam means having a varying contour cam surface.

Embodiments of the present invention will now be described by way of example with reference to the accompanying drawings, in which

Fig. 1 is a schematic side elevational view of a bicycle according to the invention;

Fig. 2 is a schematic plan view to a portion of the bicycle of Fig. 1;

Fig. 3 is a schematic side elevational view of a portion of a conventional bicycle drive system;

Fig. 4 is a schematic side elevational view of a portion of an improved bicycle drive system;

Figs. 5—8 are graphs showing advantages of a bicycle drive system according to the invention as compared with a conventional multiple speed bicycle drive system;

Fig. 9 is a schematic side elevational view of a bicycle drive system according to the invention;

Fig. 10 is an enlarged schematic side elevational view of a portion of a bicycle drive system illustrating the relationship of pedal means, crank arm means, crankshaft means, cam means, and oscillator means according to the invention;

Fig. 11 is a graph showing advantages of a bicycle drive system employing the apparatus of Figs 9 and 10 as compared with a conventional multiple speed bicycle;

Figs. 12 and 13 are side elevational views of alternative forms of design of cam means as illustrated in Figs. 9 and 10;

Fig. 14 is a schematic side elevational view of an alternative form of bicycle drive system embodying some of the inventive concepts;

Fig. 15 is an enlarged cross-sectional view taken along the line 20—20 in Fig. 14;

Fig. 16 is an enlarged schematic perspective view of a portion of the apparatus of Fig. 14, and

Fig. 17 is a partial side elevational view, partly in section and with parts removed, of another alternative form of rear wheel drive means.

Referring now to Figs. 1 and 2, the invention will be described with reference to a conventional bicycle comprising: frame means 10; front and rear wheel means 12, 14; a rider seat means 16; handle bar means 18 operatively connected to the front wheel means 12 for steering the bicycle; and crank arm means 20, 22 having pedal means 24, 26 for transferring force applied through the feet of the rider to the rear wheel means 14.

The drive system comprises: cam means 30, 32 operatively connected to and operable by the crank arm means 20, 22; oscillating arm means 34, 36 connected to and operable by the cam means 30, 32; and speed ratio multiplying rear wheel drive means 38, 40 connected to and operable by the oscillatory arm means 34, 36 and connected to the rear wheel means 14 to cause continuous rotation thereof by a wide range of applied forces in excess of variable non-uniform rider generated forces as applied to the crank arm means 20, 22 through the pedal means 24, 26. In addition, force transfer adjustment means 42, 44 are associated with the oscillatory arm means 34, 36 and operable by the rider through remote control means 46, and force transfer adjustment means 47, 48 are associated with the speed ratio multiplying rear wheel drive means 38, 40 and operable by the rider through remote control means 49 located in juxtaposition to the remote control means 46.

Referring now to Figs. 3—6, the results of harmonic correlation of input force with resistance force in the present system are compared with a conventional ten speed bicycle having a conventional derailleur drive system. Fig. 3 illustrates the input force mechanism of a conventional bicycle drive system comprising a pedal, crank arm, and chain sprocket wheel. Fig. 4 illustrates an input force mechanism of the present invention, comprising a pedal 24, crank arm 22, and oscillating arm means 36. The reference character A represents the likely vertical direction of input force application by an inexperienced or amateur rider. The reference character P represents the most desirable direction of input force application at 90° to the crank arm which is currently popularly believed to be more or less achievable by an experienced professional rider. The relationships between the input forces A and P and the theoretically available output force R, which would represent the pulling force in a drive chain operated by each system, are shown by the formulae of Figs. 3 and 4. Theoretical calculations, which have been substantiated by actual test results, are shown in Figs. 5 and 6.

Curve 110 of Fig. 5 shows that the input force A required to maintain a 200 pound output force R between 0° and 90° pedal-crank arm rotation in a conventional drive system gradually decreases from about 400 pounds at 15° pedal position to about 100 pounds between about 60° and 90° pedal position. By comparison, curve 112 shows that the input force A required to maintain a 200 pound output force R between 0° and 90° pedal-crank arm rotation in a harmonic drive system of the present invention is a substantially uniform relatively low force of about 100 pounds. Thus, between about 0° and 60° pedal-crank arm position, the drive system of the present invention requires substantially less pedal input force A than the conventional drive system to maintain a 200 pound chain output force R.

Curve 114 of Fig. 5 shows that the chain output force R generated by a constant 100 pound input force A between 0° and 90° pedal-crank arm position in a conventional bicycle gradually increases from 0 pounds to 200 pounds. Curve 116 shows that the output force R generated by a constant 100 pound input force A between 0° and 90° pedal-crank arm position in a harmonic system of the present invention is substantially uniform at about 200 pounds. Thus, the drive system of the present invention produces substantially more chain output force R between about 0° and 75° pedal-crank arm position than the conventional system for a constant input force A while also producing a uniform output force rather than a gradually increasing output force.

Curve 118 of Fig. 6 shows that the input force P required to maintain a substantially uniform 200 pound chain output force R between 0° and 90° pedal-crank arm position in a conventional drive system is a substantially uniform force of about 100 pounds. Curve 120 shows that the input force P required to maintain a substantially uniform 200 pound output force R between 0° and 90° pedal-crank are position in a harmonic drive system of the present invention gradually increases from about 0 pounds to about 100 pounds between about 75° and 90° pedal position. Thus, the same chain output force R can be obtained by the present invention with less input force P than by the conventional system.

Curve 122 shows that the output chain force R generated by a substantially constant input force P of 100 pounds between 0° and 90° pedal-crank arm position in a conventional drive system is a substantially constant 200 pounds. Curve 124 shows that the chain output force R generated by a substantially constant input force P of 100 pounds between 0° and 90° pedal-crank arm position gradually decreases from about 700 pounds at 15° pedal-crank arm position to about 200 pounds between 75° and 90° pedal-crank arm position. Thus, between about 0° pedal-crank arm position and about 75° pedal-crank arm position, the drive system of the present invention provides greater chain output force than the conventional system for the same amount of pedal input force.

Another advantage of the drive system is the provision of means for providing an effectively infinitely variable speed range by use of the oscillator means 34, 36, which may be designed to provide a mechanical advantage range from 0 to any maximum within the practical limits of the radial length of the oscillator arm means, in combination with the rear wheel drive means 38, 40, which also may be designed to provide a mechanical advantage range which may vary by multiple integers from a 1:1 ratio (i.e. 0) to increased rations of 2:1, 3:1, etc., or to decreased ratios of 1:2, 1:3, etc. Referring to Fig. 7, a comparison of a drive system of the present invention with a conventional ten speed drive system shows that the present invention can provide not only a much wider range of selectable speeds (i.e., mechanical advantages) than the conventional system, but also a substantially infinitely variable number of selectable speeds as compared with the limited fixed number of selectable speeds in the conventional system. Fig. 7 also shows that the location of the ten fixed speed positions of the conventional drive system are substantially limited by the requirement of an integral number of teeth on the drive sprockets for any selectable gear number. Thus, the location of the ten speed positions tend to be relatively closely spaced in the one extreme or to be variably widely spaced in the other extreme. On the other hand, the drive system of the present invention enables selection of any possible effective gear number as well as unlimited movement from one effective gear number to another effective gear number, within a given speed range, whereas in the conventional drive system the gear numbers must be selected in increasing or decreasing order.

As shown in Fig. 7, ultimate advantageous results of the speed selection improvements are that the bicycle will travel both faster and slower than the conventional ten speed bicycle and the bicycle of the present invention can be ridden at any selectable speed within the speed range, whereas the conventional bicycle can only be ridden at one of the ten speed positions selected in increasing or decreasing order.

Referring now to Fig. 9—15 in a first embodiment of the invention, each of the cam means 30, 32 are shown to comprise a cam plate member 200 fixedly but replaceably attached to the crankshaft member 52 connecting the crank arm means 20, 22 through conventional bearing housing assembly means 56 fixed on the frame means 10. A contoured cam surface 202 extends continuously about the periphery of the cam plate 200 which is mounted in a predetermined location on the crankshaft 52 relative to the crank arm 22. The contour of cam surface 202 may be varied to variously correlate the input force imparted by the rider to the crank arm means to the theoretical resistance R of the drive means 38, 40 or to the actual force and resistance relationships in the drive system. The cam plate member 200 is removably mounted relative to the crank arm means 20 and relative to other drive system components so as to be easily replaceable, whereby cam plate members having variously contoured cam surfaces 202, may be provided in accordance with the characteristics of force application of a particular rider or a particular group of riders, i.e., men or women, adults or children, racer, or recreational riders, etc.

The cam means surface 202 is designed to cause predetermined rotational movement of oscillator means 36 related to the rotational position of the crank arm 22 to correlate input force and actual resistance characteristics of the drive system, with a view toward maximizing the usage of available power input of the bicycle rider while also improving the average power input characteristics of the system.

The illustrative cam contour of Fig. 9 provides harmonic match as hereinbefore described, but may be changed to effect varying results for varying riding characteristics and for varying abilities or capabilities of various riders as hereinbefore described.

Each of the oscillating arm means 34, 36 are shown to comprise a generally L shaped member 204 pivotally mounted relative to the frame means 10 on shaft means which may be in the form of opposite ends of a common shaft member 206 mounted in a common housing 208 fixed to the frame means. An elongated pull arm portion 210 of oscillator member 204 extends radially outwardly from shaft member 206 and is pivotally movable thereabout between opposite extreme rearward and forward positions of oscillation indicated by dotted lines 212, 214. An elongated radially extending slot 216 is provided in arm portion 210 to define a forwardly facing rear surface 218. Suitable gripping means, such as knurling, serration, ribs, threads or the like, are provided along surface 218. The oscillator also has a cam arm portion 220 extending rearwardly at substantially a right angle to arm portion 210. A cam roller 222 is rotatably mounted on the rearward end of arm portion 220 and is maintained in engagement with the peripheral surface of cam 200 by a suitable means such as a spring or a guide slot or the like, to control the position of the oscillator member 204 relative to pivot 206.

A selectively releasable connecting means 224 is slideably adjustably mounted on arm portion 210 for radial inward and outward movement therealong between a first radially innermost position and a second radially outermost position indicated by dotted lines 226, 228. Thus, at the radially innermost position, 226, the connecting means 224 is carried by arm portion 210 along an arcuate path of minimum circumferential length while, at the radially outermost position 228, the connecting means is carried by arm portion 210 along an arcuate path of maximum circumferential length. The connecting means 224 is preferably adjustably connected to arm portion 210 by frictional engagement between surface 229 of member 224 and surface 218 which extends between the innermost and outermost positions so as to enable infinitely variable positioning of the connecting means therebetween. The associated remote control means 46 of Fig. 1 may be in the form of a manually operable actuator device illustrated in Fig. 9 as a control lever 230 which may be mounted on an upper portion of the frame means 10 between the seat means and the handle bar means. A movable connecting member 232, such as a Bowden cable, is suitably connected at one end of the control lever 230 and at the other end to connecting means 224 whereby forward movement of the control lever between maximum rearward and forward positions causes radial outward displacement of the connecting means 224 relative to the arm portion 210 and rearward movement of the control lever causes radial inward displacement of the connecting means 224 relative to arm portion 210. The arrangement is such that the connecting means 224 on each arm portion 210 of each oscillator means 34, 36 is simultaneously uniformly adjusted by the movement of a single control lever 230 which is similarly connected to each of the connecting means 224.

Each of the rear wheel drive means 38, 40 is operatively connected to gripping means 224 by connection means 233, in the form of a wire, rod, cable, etc., and to the rear bicycle axle 234 and drive sprocket wheel 236. The rear wheel drive means 40 of Fig. 9 is in the form of a block and tackle system comprising a continuous closed loop chain member 238 mounted on drive sprocket wheel 236 and sprocket wheels 240, 242, 244, 246. Sprocket wheels 242, 244 are fixedly rotatably mounted on the bicycle frame means 10. Sprocket wheels 240, 246 are slidably guidably mounted in elongated guide means (not shown) attached to the bicycle frame means 10 for reciprocable movement relative thereto. Sprocket wheel 240 is connected to connection means 233 by a pivotally mounted connecting link member 248 so as to be reciprocably linearly movable in response to the movement of the oscillator member. A drive system return means is connected to sprocket wheel 246 by a pivotally mounted connecting member 250 and comprises a cable member 252 fixedly connected at one end to the bicycle frame means 254 and at the other end 256 to the connecting member. The cable member 252 is looped around a pulley member 258 connected to the bicycle frame by a spring means 260. A one-way ratchet means 262 is associated with sprocket wheel 240 to permit movement of the chain 238 relative to sprocket wheel 240 only in the direction of arrow 264 whereby, during movement of sprocket wheel 240 in the direction of arrow 266, the chain is directly moved with the sprocket wheel 240 thereby causing rotation of axle 234 on a 1:1 speed ratio. During rearward movement of sprocket wheel 240 opposite the direction of arrow 266 under the influence of spring means 260, the ratchet 262 releases to permit movement of the chain relative thereto resulting in non-driving movement of chain relative to drive sprocket 236. In order to change the speed ratio of the system from 1:1 to 2:1, a selectively operable one-way ratchet means 268 is associated with sprocket wheel 242. When the ratchet means 268 is engaged with chain 238 on sprocket wheel 242, forward linear movement of sprocket wheel 240 in the direction of arrow 266 causes the chain to move twice as far and results in double rotation of sprocket wheel 236 and rear axle 234 as when ratchet means 268 is disengaged. The ratchet means 268 is also arranged so as to permit relative movement between the chain and the sprocket wheel when return force is applied in the direction of arrow 264 by spring 260. Remote control means 48 comprises a manually operable actuator device such as a control lever 270, which may be mounted on the upper portion of the frame means 10 in juxtaposition to control lever 230. Each drive system 38, 40 has a movable connecting member 272, such as a Bowden cable, connected at one end to control lever 270 and at the other end to ratchet means 268, whereby location of the control lever 270 in a rearward position places the ratchet means 268 in the engaged position and location of the control lever 270 in a forward position corresponding to the forward position of control lever 230 places the ratchet means in the disengaged position.

Thus, the present system enables the selection of a wide range of infinitely variable speeds. In the lowest speed position, control levers 230 and 270 may be in corresponding rearwardmost positions whereat oscillator connecting means 224 are at radially innermost positions on the oscillator arm portions 210 and ratchet means 268 are disengaged to provide a 1:1 speed ratio in chain drive systems 38, 40, or more preferably in opposite positions as hereinafter described.

A substantially infinitely variable low speed range is provided by selective simultaneous adjustment of the oscillator connecting means 224 between the radially innermost positions and the radially outermost positions on the oscillator arm portions 210 by moving the control lever 230 between the rearwardmost and forwardmost positions. Whenever the bicycle rider desires to "shift" the chain drive systems 38, 40 from the low 1:1 speed ratio to the high 2:1 speed ratio, the control lever 270 may be actuated to engage ratchet means 268 with chain means 238. In the high 2:1 speed ratio range, the oscillator connecting means 224 are also fully adjustable between the radially innermost and outermost positions to provide a substantially infinitely variable high speed range. In order to enable smooth, easy, transition between the low speed range and the high speed range, the control levers 230, 270 may be mounted in juxtaposition with suitable correlation of movement. For example, control lever 230 is preferably arranged so that forward movement increases speed and rearward movement decreases speed in each speed range. Control lever 270 is preferably arranged so that, in a forwardmost position corresponding to the forwardmost position of control lever 230, ratchet means 268 are disengaged and the chain drive means 38, 40 are in the 1:1 speed ratio of the low speed range. Thus, when lever 230 has been placed in the high speed low speed range forward position, both levers 230 and 270 may be grasped and moved rearward to simultaneously "shift" the chain drive means 38, 40 to the 2:1 speed ratio high speed range while also locating the oscillator connecting means 224 in the radially innermost "low" speed position in the high speed range, from which the speed again may be gradually infinitely variably increased as the lever 230 is moved forwardly causing the oscillator connecting means 224 to move radially outwardly.

Referring now to Fig. 10, illustrative and presently preferred cam means 30, 32 are shown to each comprise a cam plate member 200 fixedly mounted on crank shaft 52 rotatably mounted in a bearing hub 56 fixedly mounted on frame means portions 280, 282. Crank shaft 52 and cam member 200 are rotatable about central axis 284 in the direction of arrow 286 by the radially extending crank arm means, schematically illustrated by radial line 288, and pivotally mounted pedal means, schematically illustrated at 290, which are rotated in the direction of arrow 292. The crank arm line 288 represents an upper (0°) vertical position and cam member 200 is fully shown in its position at the upper (0°) vertical position of the crank arm and the pedal member 290. The reference designations P1—P16 refer to radial lines representing various rotational positions of the crank arm and pedal member beginning with the 345° rotational position thereof and ending with the lower vertical 180° rotational position thereof. The reference designations C1—C16 refer to the cam member 200 and portions thereof in various rotational positions of the cam member corresponding to the various rotational positions P1—P16 of the crank arm and pedal member.

The associated one of the oscillator means 34, 36 is schematically shown with oscillator shaft 206 pivotally mounted in hub member 208 fixed to a portion of the frame means at 294 to provide a pivotal axis 296, which in the illustrative embodiment, is located on vertical radial line 288 but may be variously otherwise positioned. The oscillator means pull arm portion 210 is represented by and located along a radial line 298, when the crank arm 288 is in the upper (0°) vertical position P4, and follower arm portion 220 is represented by and located along a radial line 300 when the crank arm 116 is in the upper vertical (0°) position P4. In the illustrative embodiment, the central longitudinal axis of the pull arm portion 298 and the follower arm portion 300 are circumferentially spaced approximately 95°. The cam follower means, roller member 222, having a rotational axis at 302, is rotatably mounted on the radially outer end portion of the follower arm portion 300 in continuous engagement with the peripheral cam surface 202 of the cam member 200. The reference designations R1—R16 refer to the roller member 222 and portions thereof in various positions as displaced between a maximum upward position represented by radial line 304 and a maximum downward position represented by radial line 306 along arcs 308, 310, 312 having centers at 296 and representing, respectively, the path of movement of the radially inner peripheral portion, the axis of rotation, and the radially outer peripheral portion of the roller member 222. The reference designations R1—R16 refer to the various positions of the roller member corresponding to the various rotational positions P1—P16 of the crank arm 288 and pedal member 290, as well as to the various rotational positions C1—C16 of the cam member 200. The reference designations O1—O16 refer to the oscillator pull arm portion 298 in various positions, as circumferentially displaced between a maximum forward position represented by radial line 320 and a maximum rearward position represented by radial line 322, which correspond to the various positions P1—P16 of the crank arm 288 and pedal member 290, the various positions C1—C16 of the cam member 200, and the various positions R1—R16 of the roller member 222.

The peripheral surface 202 of cam member 200 comprises an oscillator means driving surface portion 324 extending between lines 326, 328 for causing forward displacement of oscillator pull arm portion 298 from the rearwardmost position O1 at 322 to the forwardmost position O16 at 320. The peripheral surface 202 of cam member 200 further comprises an oscillator means return surface portion 330 extending between lines 326, 328 for guiding rearward displacement of oscillator pull arm portion 298 from the forwardmost position O16 at 320 to the rearwardmost position O1 at 322. Cam surface portions 324, 330 are connected by a cam follower return transition surface portion at and adjacent to line 328 located radially outermost from central axis 284 and a cam follower driving transition surface at and adjacent to line 326 located radially innermost from central axis 284.

The cam follower driving transition surface at 326 is designed and arranged to terminate the rearward return movement and begin the forward movement of the pull arm portion 298 of the oscillator means prior to the time that crank arm 288 and pedal member 290 reach the vertical 0° P4 position. In this manner, forward motion overlap means are provided for causing both oscillator means 34, 36 to be moved forward at the same time during a portion of each revolution of the crank shaft 52. The illustrative arrangement is such that the upward motion of the roller member 222 begins at positions R1 along radial line 306 when cam member 200 is in position C1 with the crank arm 288 and pedal 290 located at position P1 after 345° of rotation of the crank arm 288 from the vertical 0° position P4 and 15° before again reaching the vertical 0° position. At this time, the other crank arm is located 180° opposite the crank arm 288 at the 165° position P15 with the other oscillator pull arm portion being moved forward at the O15 position. As the crank arm 288 rotates upwardly from position P1 at 345° to positions P2 at 353°, P3 at 357°, and P4 at 0°, the roller member 222 is moved upwardly from position R1 to corresponding position R2, R3 and R4 and pull arm portion 298 is moved forwardly from position O1 to corresponding positions O2, O3 and O4. At the same time, the other crank arm rotates downwardly from position P15 at 165° to position P16 at 180° so that, when roller 222 is moving upward at position R4, the other roller is in the uppermost position R16 in engagement with the transition surface at 328. The amount of overlapping forward movement of the oscillator pull arm portions 298 may be varied from cam design to cam design by changing the curvature of transition surface at 326 as necessary or desirable. Fig. 8 illustrates that each oscillator means begins its forward movement before completion of the forward movement of the other oscillator means. Thus, there is continuous application of input force to the rear wheel without input force lag at the 0° and 180° positions of the crank arm means.

The portion 325 of cam surface 202 between the transition surfaces at 326, 328 provides work surface means designed and arranged to position and correlate the pull arm portion 298 relative to the position of crank arm 288 in a manner providing input-output force correlating means for continuously varying the effective radial and circumferential location of application of input force, as applied to the crank arm 288 through pedal 290, to the pull arm portion 298 of the oscillator means through cam means 200 and cam follower means 222 in accordance with various force application and transmittal characteristics, as well as varying resistance and torque characteristics encountered in a bicycle drive system.

In general, one work surface means design consideration is the fact that the effective work range of the crank arm 288 and pedal 290 is from approximately 0° to 180°. In other words, the input force transmitted to the pedal 290 through the foot and leg of the rider can be effectively transmitted to the drive system only in the effective work range during downward movement of the crank arm 288. Furthermore, the maximum effective work position of the crank arm 288 is at the horizontal 90° position P10 where the available input force is fully applied at right angles to the crank arm 288. Another design consideration, which may be taken into account in connection with the present invention, is that the available input force at any given position in the effective work range usually varies between persons and usually varies from leg to leg of the same person. In fact, test results have shown a considerable difference in patterns of available input force between persons who are experienced bike riders and persons who are merely recreational bike riders. Thus, the present invention provides the highly desirable result of enabling cam design for varying capabilities of different riders as well as varying leg thrust capabilities of a particular rider.

The illustrative cam design of Fig. 10 is based upon the general premise that maximum effective force transmittal from the crank arm 288 and pedal 290 to the oscillator means and hence to the rear wheel of the bicycle, can be achieved by matching maximum input force capability of the rider with maximum force transmittal capability of the drive system.

As shown in Fig. 10, as the crank arm 288 and pedal 290 are driven by the available input force of the rider from position P4 at 0° to position P10 at 90°, the cam follower roller 222 is moved further upwardly from position R4 to position R10, which in the illustrative embodiment is generally horizontal, by generally gradually decreasing amounts of circumferential displacement and the pull arm portion 298 is moved further forwardly from position O4 to position O10, which in the illustrative embodiment is substantially vertical. As the crank arm 288 and pedal 290 are driven further downwardly by the available input force of the rider from position P10 at 90° to position P16 at 180°, the cam follower roller 222 is moved further upwardly from position R10 to position R16 by generally increasing amounts of circumferential displacement and the pull arm portion 298 is moved further forwardly from position O10 to position O16.

The results achieved are illustrated by the various circumferential distances travelled by the crank arm 288, the roller 222, and the pull arm portion 298. While the crank arm 288 moves downwardly equal circumferential distances between positions P4—P16, the roller member 222 is moved upwardly gradually decreasing circumferential distances from position R4 to position R10 and gradually increasing circumferential distances from position R10 to position R16. At the same time, pull arm portion 298 initially moves forwardly a relatively large circumferential distance from position O4 to position O5 whereafter the driving arm portion moves forwardly from position O5 to position O15 in relatively small uniform circumferential distance increments and at the termination of the forward movement between positions O15 and O16 again moves a relatively large circumferential distance. As a result the available input forces at positions P5—P15 of the crank arm 288 are transmitted through the cam member 200, the roller member 222, the follower arm portion 300, and the pull arm portion 298 in a manner providing a desired maximum effective output force available on the pull arm portion for transmittal to the rear wheel. Thus, force correlation means are provided to enable the bike rider to provide substantially greater effective output force to the wheel than in the conventional bicycle between P5—P15 positions of the crank arm.

The cam design of Fig. 10 has been tested on a test bike having the aforedescribed transmission system cam design and the results produced thereby have been compared with the results produced by a conventional ten speed bicycle having a conventional chain-mulitple sprocket transmission system. One test comprised static measurement of the vertical force on the pedal of each bike at various rotational positions of the pedal required to balance the same resistance on the rear wheel when each bike transmission system was adjusted to have the same mechanical advantage, i.e., the conventional bike and the test bike were in the same effective gear. The results of static tests for an effective gear No. 33.75 (tooth ratio of 40/32) are indicated hereinafter in Table I and in the graph of Fig. 11, of the drawing.

TABLE I

PP°	P6/30°	P7/45°	P8/60°	P9/75°	P10/90°	P11/105°	P12/120°	P13/135°	P14/150°
PWC #	21.5	13.5	11.3	11.0	9.0	10.0	11.0	13.2	18.5
PWT #	12.8	8.6	8.4	7.6	5.5	4.5	7.2	8.6	14.0
PWC/ PWT %	68	57	34.5	44.7	63.6	122.2	52.8	53.5	37.1
WPC °	30	45	60	75	90	105	120	135	150
WPT °	49.2	61	71.6	81.4	90.3	100.4	110.3	120.6	130.8
WPV °	+19.2	+16	+11.6	+6.4	+0.3	−4.6	−9.7	−14.4	−19.2
9/PWC	.42	.67	.80	.82	1.0	.90	.82	.68	.49
9/PWT	.70	1.05	1.07	1.18	1.64	2.00	1.25	1.05	.64

PP = Pedal position in degrees in the maximum available input force range as measured from the vertical 0° pedal position.

PWC = Pedal weight of conventional ten speed bike in pounds of force applied vertically downwardly on the pedal to balance four pound rear wheel weight applied vertically downwardly at 90° rear wheel position 15 inches rearwardly of rear wheel axis.

PWT = Pedal weight of test bike in pounds of force applied vertically downwardly on the pedal to balance four pound rear wheel weight applied vertically downwardly at 90° rear wheel position 15 inches rearwardly of rear wheel axis.

PWC/PWT = The percentage increase in pounds of force required to balance the same rear wheel weight with a conventional ten speed bike as compared with the test bike.

WPC = Wheel position of conventional ten speed bike relative to pedal position in one to one transmission ratio.

WPT = Wheel position of test bike relative to pedal position in one to one transmission ratio.

WPV = Wheel position variance of rear wheel of test bike relative to rear wheel of conventional ten speed bike.

9/PWC = Conversion of pedal weight data of conventional ten speed bike on 9 pound base at 90° pedal position.

9/PWT = Conversion of pedal weight data of test bike on 9 pound base at 90° pedal position.

Referring to Table I, the test results show that the test bike of the present invention required a substantially lesser input force than the conventional ten speed bike to overcome the resistance of the rear wheel at each measured pedal position. In addition, the cam design of the test bike results in variance of the rear wheel position relative to pedal position so that resistance of the rear wheel is variably transferred to the pedal in accordance with predetermined pedal position design characteristics of the cam.

The cam design is such that at 0° (P4), 90° (P10) and 180° (P16), pedal positions, the wheel position is correspondingly 0°, 90° and 180°. At 15° (P5) and 165° (P15) pedal position, the wheel position is 35.4° and 144.8°, respectively. Thus between 0° (P4) and 15° (P5) pedal position, the relative wheel position is rapidly advanced from the coincident 0° position to a maximum advanced position of approximately 20° and between 165° (P15) and 180° (P16) pedal position, the relative wheel position is again rapidly advanced from a maximum retarded position of approximately 20° to the coincident 180° position. The maximum advanced wheel position attained at the 15° pedal position is maintained substantially constant from the 15° pedal position (P5) to the 30° pedal position (P6) and the maximum retarded wheel position attained at the 150° pedal position (P14) is maintained substantially constant from the 150° pedal position (P14) to the 165° pedal position (P15).

As shown in Table I, between pedal position P6 (30°) and pedal position P10 (90°), the advanced position of the rear wheel of the test bike is progressively decreased relative to the position of the pedal and the corresponding position of the rear wheel of the conventional ten speed bike. The amount of rear wheel advancement progressively decreases from approximately 19.2° at pedal position P6 (30°) to substantially coincident relative positions at pedal position P10 (90°). Thus, the effective resistance of the rear wheel to rotation from 0° to 90° has been redistributed relative to the pedal positions. The effective resistance of the rear wheel to rotation from 0° to approximately 49.2° has been concentrated in the portion of the pedal rotation from 0° to 30° in which range the capability of the bike rider to supply input force on the pedal is minimal and inertial effects may be best utilized to overcome the rear wheel resistance. The effective resistance of the rear wheel to rotation from 49.2° to 90° has been distributed over the portion of the pedal rotation from 30° to 90°, in which range the capability of the bike rider to supply input force on the pedal progressively increases, in a manner such as to progressively reduce the effective resistance of the rear wheel to rotation from pedal position P7 (35°) to pedal position P10 (90°) in a range progressively increasing capability of the bike rider to supply input force.

Between pedal position P10 (90°) and pedal position P16 (180°), the position of the rear wheel of the test bike is progressively retarded relative to the position of the pedal and the corresponding position of the rear wheel of the conventional ten speed bike. The amount of rear wheel retardation progressively increases from approximately 0° at pedal position P10 (90°) to approximately 20° at pedal position P14 (150°), and is maintained substantially constant at approximately 20° between pedal position P14 (150°) and pedal position P15 (165°), and is then rapidly decreased from approximately 20° at pedal position P15 (150°) to a coincident position with the pedal at P16 (180°). Thus, the effective resistance of the rear wheel to rotation from 90° to 180° has been redistributed relative to the pedal positions. The effective resistance of the rear wheel to rotation from approximately 130.8° to 180° (i.e., approximately 50° of rotation) has been concentrated in the portion of the pedal rotation from 150° (P14) to 180° (P16) in which range the capability of the bike rider to supply input force on the pedal is gradually decreasing and inertial effects may be best utilized to overcome the rear wheel resistance. The effective resistance of the rear wheel to rotation from approximately 90° to approximately 130.8° (i.e., approximately 40° of rotation), in which range the capability of the bike rider to supply input force on the pedal progressively decreases, in a manner such as to progressively increase the effective resistance of the rear wheel to rotation from pedal position P10 (90°) to pedal position P14 (150°) in a range of progressively decreasing capability of the bike rider to supply input force.

Referring now to Fig. 11, curve 340 represents the theoretical available output torque at the rear wheel of the conventional ten speed bike, curve 342 represents the calculated available torque at the rear wheel of the conventional ten speed bike based upon the actual test results, and curve 344 represents the calculated available torque at the rear wheel of the test bike based upon the actual test results. The vertical ordinate axis 346 represents available output torque and horizontal abscissa axis 348 represents rotational positions of the rear wheel and crank arm from a vertical 0° upper position to a vertical 180° lower position of the crank arm. The available output torque has been calculated on a relative percentage basis with the maximum available torque of the conventional ten speed bike at the 90° wheel position (and 90° pedal position) represented as a base torque of 100% inch pounds so that the vertical data points of the curves correspond to the calculations 9/PWC and

9/PWT of Table I. Thus, the points CP6, CP7, CP8, CP9, CP10, CP11, CP12, CP13 and CP14 of the conventional bike curve 342 are located on the vertical axis at .42, .67, .80, .82, 1.0 (base), .90, .82, .68 and .49, respectively. The points TP6, TP7, TP8, TP9, TP10, TP11, TP12, TP13 and TP14 of the test bike curve 344 are located on the vertical axis at .70, 1.05, 1.07, 1.18, 1.64, 2.00, 1.25, 1.05 and .64, respectively. The points P6, P7, P8, P9, P10, P11, P12, P13 and P14 of the theoretical conventional bike curve 340 are located on the vertical axis at .50, .71, .86, .96, 1.00, .96, .86, .70 and .50, respectively.

With respect to horizontal axis, the points CP6, CP7, CP8, CP9, CP10, CP11, CP12, CP13 and CP14 of the conventional bike curve 342 and points P6, P7, P8, P9, P10, P11, P12, P13 and P14 of the theoretical conventional bike curve 340 are located at 30°, 45°, 60°, 75°, 90°, 105°, 120°, 135° and 150°, respectively, which represent both pedal positions and wheel positions. At the same pedal positions of the conventional bike, the points TP6, TP7, TP8, TP9, TP10, TP11, TP12, TP13 and TP14 of the test bike curve 344 are located at 49.2°, 61°, 71.6°, 81.4°, 90.3°, 100.4°, 110.3°, 120.6° and 130.8°, respectively, which represent only the wheel position of the test bike.

The areas 346, 348, 350 of the graph between the conventional bike curve 342 and the test bike curve 344 illustrate the change in the theoretical amount of available torque at the rear wheel effected by the present invention. The areas 346, 350, where curve 344 is below curve 342, indicates that the system of the present invention provides a lesser amount of available torque than the conventional bike system from 0° pedal position to approximately 35° pedal position and from approximately 145° pedal position to 180° pedal position. The area 348 where curve 344 is above curve 342, indicates that the system of the present invention provides a substantially greater amount of available torque from approximately 35° pedal position to approximately the 145° pedal position. In addition, the curve 344 indicates a high degree of concentration of available torque between approximately 60° pedal position and 120° pedal position in the maximum range of input force capability of the bike rider.

It is understood that the foregoing test results and calculations do not take into account actual bike riding conditions under which the actual resistance imposed on the drive system by the rear wheel affects the aforedescribed theoretical relationships between the pedals, crank arms, cams and oscillator means. Nonetheless, it is to be further understood that aforedescribed cam design characteristics take into account the actual resistance of the rear wheel and are such as to produce the desired results in the drive system.

While the foregoing illustrative cam design produces particularly beneficial results based upon the concept of matching theoretical maximum input force capability of the bike rider with minimum resistance of the system, for example, the cam design may be changed to match theoretical maximum input force capability of the bike rider with maximum resistance of the system as illustrated by the cam design of Fig. 12. Such a design may be most beneficial for bike racing riders who have the capability of providing high input force at the beginning of the downward pedal movement. As shown in Fig. 12, the design of cam member 200 provides an overlap transition surface between lines 360, 362, a relatively small output torque advantage in the initial portion of the work surface from 0° pedal position beginning at 362 to approximately 15° pedal position beginning at 364; then a relatively large output torque advantage from approximately 15° pedal position at 364 to approximately 100° pedal position at 366; a relatively large output torque disadvantage from approximately 100° pedal position at 366 to approximately 165° pedal position at 368; and a relatively small output torque advantage from approximately 165° pedal position at 368 to 180° pedal position at 370.

For another example, the cam design may be changed to variously match input force capability of the bike rider with resistance of the rear wheel more in accordance with a general theoretical harmonic curve of effective input force, as illustrated in Fig. 13, wherein the cam design provides a relatively large output torque advantage from 0° pedal position at 372 to approximately 15° pedal position at 374; a relatively smaller output torque advantage from approximately 15° pedal position at 374 to approximately 35° pedal position at 376; a relatively small output torque disadvantage from approximately 35° pedal position at 376 to approximately 65° pedal position at 378; a relatively large output torque disadvantage from approximately 65° pedal position at 378 to approximately 130° pedal position at 380; a relatively small output torque disadvantage from approximately 130° pedal position at 380 to approximately 145° pedal position at 382; a relatively small output torque advantage from approximately 145° pedal position at 382 to approximately 165° pedal position at 384; and a relatively large output torque advantage from approximately 165° pedal position at 384 to approximately 180° pedal position at 386.

An important advantage is that the cam designs may be matched to individual input force capabilities of individual bike riders for achieving particular output torque characteristics. For example, general cam designs may be provided for various classes of bike riders, such as racing or recreational bike riders; men and women and boy and girl bike riders; and flat or hilly terrain bike riders. Furthermore, the right and left cam designs for a particular bike system may be varied in accordance with variations in input force capability of the right and left leg of a class of bike riders or of a particular bike rider whose input force capabilities may be measured to determine the most satisfactory cam design for the needs or desires of that particular bike rider. In order to facilitate the provision of varying cam designs for various bike riders, it is contemplated that the cam members be removably mounted on the bike to enable easy and quick change of cam members.

Referring now to Figs. 14—16, the bicycle frame means 10 is illustratively shown to comprise conventional tubular frame portions 400, 402, 404, 406. Portions 400, 402, 404 are fixedly connected to and support a conventional crankshaft-bearing hub 408 in which the crankshaft 52 is rotatably supported. As is conventional, each of the crank arm means 20 is removably fixedly attached to the crankshaft 52 by conventional fastening means 410 with pedal means 24, 26 pivotally mounted on the outer end portion thereof. Each cam means 30, 32 comprises a cam member 200 removably fixedly mounted on one of the outer end portions of the crankshaft member 52 between the associated crank arm member 20 and the end portion of the hub 408 for rotation with the crankshaft member relative to the hub. Each of the oscillator means 34, 36 comprises an oscillator member 204 pivotally removably mounted between the cam member 200 and the frame portion 402 on the outer end portions of a pivot shaft means 412, which may be fixedly mounted in a housing member 414 fixedly attached to the frame portion 402 by suitable fastening means 416, as illustrated in Fig. 16.

Each oscillator member 204 comprises a pull arm portion 210 having a serrated forwardly facing elongated force transfer surface 218 and a laterally outwardly offset roller arm portion 220 rotatably supporting a roller member 222 in laterally outwardly offset relationship thereto for continuous engagement with cam surface 202.

A force transfer connecting means 224 is pivotally mounted on pivot shaft means 420 extending between and connecting and pivotally supporting end portions of pull bar members 422, 424. The connecting means 224 comprises a bar member 425 having tooth portions 426, 428 providing ratchet means releasably grippingly engageable with the serrated surface 218 in various radially inwardly and outwardly adjusted positions. Such release and adjustment may be effected by cable means 430, 432, one of which is attached to one end of bar member 425 at 434 by a length of cable wire 436 extending upwardly from the bottom of surface 218, and the other of which is attached to the other end of the bar member at 438 by a length of cable wire 440 extending downwardly from the top of surface 218. The cable wire 440 may be guidably mounted and confined in a groove 442 at the top of pull arm portion 210 and extends downwardly the rearwardly facing end surface and around the bottom of the oscillator member 204. The arrangement is such that actuation of a control lever attached to cable means 430 to cause movement of wire 436 in the direction of arrow 444 causes bar member 425 to pivot about shaft means 420 to release tooth portion 426 relative to surface 418 whereupon bar member 425 may be pulled downwardly along surface 218 with the inclined outer surface of tooth portion 428 sliding over the serrations until the adjustment force is removed, whereupon the bar member will immediately pivot back into holding engagement with surface 218. The bar member 425 may be similarly released and moved in the opposite direction by force applied in the direction of arrow 446 by wire 440.

Each of the rear drive wheel means 38, 40 are operatively connected to the rear end portions of the pull bar members 424, 426 as by a pivotal connecting pin means 450 rotatably supporting a sprocket wheel 452 mounted between and movable with the bar members. As previously described, a closed loop drive chain member 454 is mounted on sprocket wheel 452 and extends around sprocket wheels 456, 458 fixedly mounted relative to the frame means and the rear axle by suitable bracket means 460 and around movable sprocket wheel 462 connected to return assist spring means 464, attached to the frame portion at 466, by a pivotally mounted connecting bracket 468. If necessary or desirable, chain and/or sprocket wheel guide means 470, 472 may be fixedly mounted on the frame means in any suitable manner. The general operation of the system is as hereinbefore described.

Referring now to Fig. 17, another alternative rear wheel drive means is shown to be mounted in a similar manner between spaced parallel tubular frame portions 630, 632 and 634, 636 and bracket means 638 (only one of which is shown). Each rear wheel drive means comprises an open ended chain member 640 attached at one end to the frame means 10 at 642 and looped around sprocket wheels 644, 646, and rear wheel drive sprocket 648 including a conventional one way clutch mechanism (not shown), with the other end portion 650 of the chain wound on a spring loaded chain reel means 652 to effect a 2:1 speed change ratio between oscillator pull arm 536 and the rear wheel drive sprocket 648. As previously described, such sprocket wheel 644 is connected to the associated pull arm 536 by a connecting rod means 574 pivotally connected to adjustable connecting means 560 at the forward end thereof at 576 and pivotally connected to sprocket wheel 644 at the rearward end thereof at 578.

While the invention has been described hereinbefore by reference to certain illustrative and presently preferred embodiments, and the oscillator means disclosed hereinbefore have been shown to be mounted in a presently preferred generally vertical upwardly extending position for movement between rearwardwmost and forwardmost positions, it is contemplated that the oscillator means may be variously otherwise mounted and positioned. For another example, the cam means disclosed hereinbefore have been variously constructed, arranged, positioned and operably connected between and relative to crank arm means and oscillator means which may be variously otherwise constructed, mounted, arranged, positioned, connected, combined or replaced by alternative embodiments. For another example, various rear wheel drive means have been disclosed hereinbefore which may also be variously otherwise constructed, mounted, arranged, positioned, connected, combined or replaced by alternative embodiments. Furthermore, in connection with the rear wheel drive means, embodiments disclosed herein, it is to be understood that cable and pulley wheel devices are capable of providing the same function as the link chain and link chain sprocket wheels. Also, it is contemplated that the invention is generally applicable to rider-propelled vehicles other than bicycles, such as, for example, tricycles.

WHAT I CLAIM IS:—

1. A drive system, for a rider-propelled vehicle having a driven wheel, operable by a rider by applying force to input means which cause unidirectional rotational movement of the driven wheel and comprising:

variable force transfer means operable by the input means to transfer input force from the input means to the driven wheel so that during the application of force to the input means there is no break in application of forward driving force to the driven wheel;

the variable force transfer means being adapted to control the amount and time of application of input force to the driven wheel during each input force cycle; and

to control the rotary position of the driven wheel relative to the position of the force input means during each force input cycle, wherein said variable force transfer means comprises cam means having a varying contour cam surface.

2. The drive system according to Claim 1, further comprising oscillator means operable by said cam means.

3. The drive system according to Claim 2, comprising a selectively adjustable connector device operatively associated with said oscillator means to provide substantial infinite selective adjustment of the output speed of the driven wheel within a minimum and maximum speed range.

4. The drive system according to Claim 3, comprising a remote control apparatus adapted to selectively operate the selectively adjustable connector device.

5. The drive system according to any of Claims 1 to 4, comprising selectively operable speed ratio multiplication devices in the drive system adapted to selectively change the speed ratio of the driven wheel.

6. The drive system according to Claim 5, comprising remote control apparatus for selectively operating the speed ratio multiplication devices.

7. The drive system according to any of the preceding claims, wherein the input means comprise rotary crank arm members spaced 180° apart and pedal members mounted thereon.

8. The drive system according to any of the preceding claims, comprising a portion of the cam means designed and arranged relative to the input means adapted to cause continuous force transfer to the driven wheel during the transition from force application by a first input device of the input means to the another input device of said input means.

9. The drive system according to any of the preceding claims, comprising a cam portion designed and arranged relative to the input means to cause maximum input force capability to correspond with maximum force transmittal capability and minimum resistance of the drive system.

10. The drive system according to any of Claims 1 to 8, comprising a cam portion designed and arranged relative to the input means to cause maximum input force capability to correspond with minimum force transmittal capability and maximum resistance of the drive system.

11. The drive system according to any of the preceding claims, further comprising reciprocating means adapted to drive the driven wheel.

12. The drive system according to Claim 11, wherein the reciprocating means comprises a chain drive arrangement.

13. A drive system for a rider-propelled vehicle, constructed and arranged substantially as herein described with reference to the accompanying drawings.

LANGNER PARRY,
Chartered Patent Agents,
High Holborn House,
52—54 High Holborn,
London, WC1V 6RR,
Agents for the Applicant.

Printed for Her Majesty's Stationery Office by the Courier Press, Leamington Spa, 1980.
Published by the Patent Office, 25 Southampton Buildings, London, WC2A 1AY, from which copies may be obtained.

1581560 COMPLETE SPECIFICATION

8 SHEETS This drawing is a reproduction of the Original on a reduced scale

Sheet 1

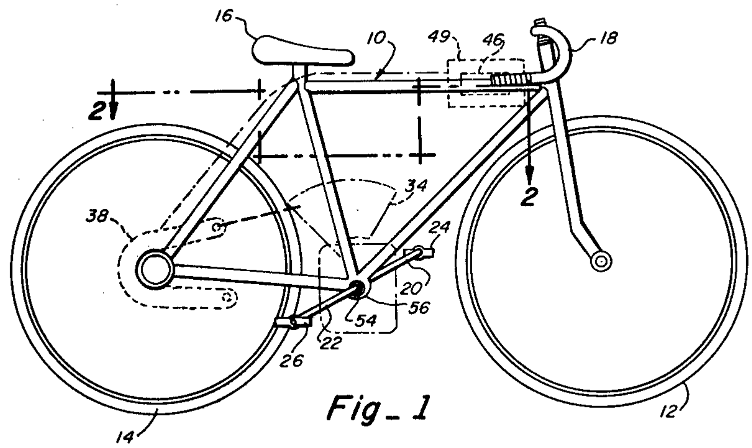

Fig_1

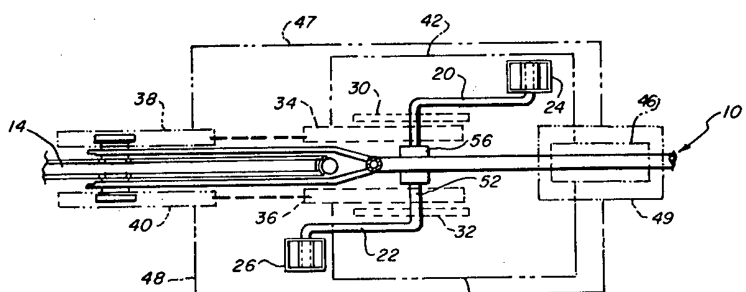

Fig_2

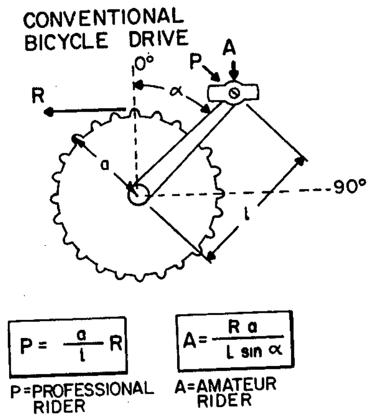

Fig_3

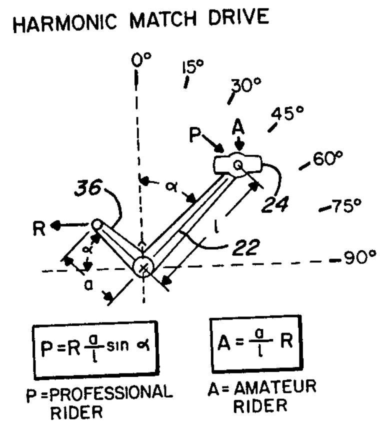

Fig_4

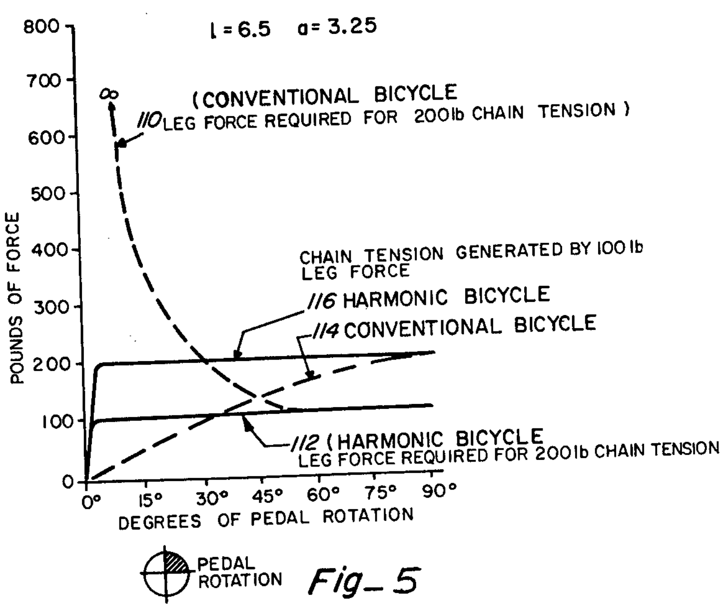

Fig_5

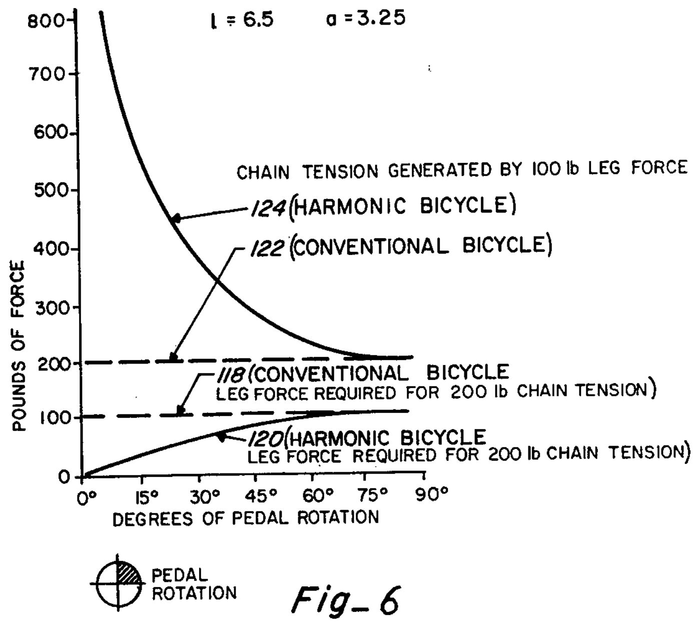

Fig_6

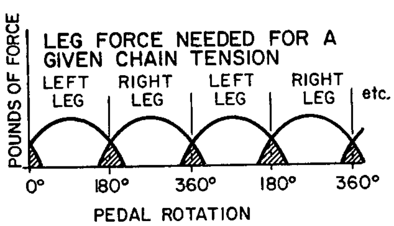

Fig_8

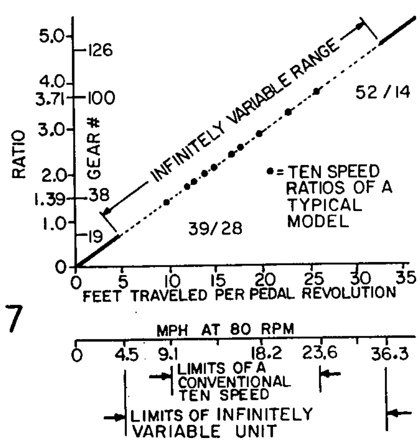

Fig_7

1581560 COMPLETE SPECIFICATION

8 SHEETS

This drawing is a reproduction of the Original on a reduced scale

Sheet 4

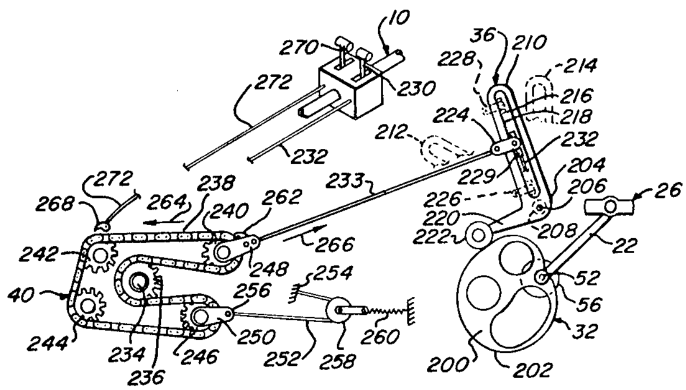

Fig_ 9

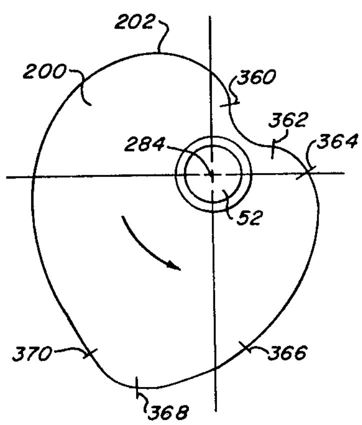

Fig_12

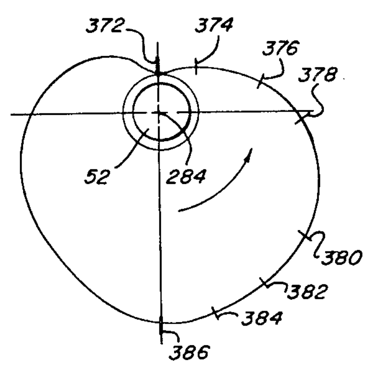

Fig_13

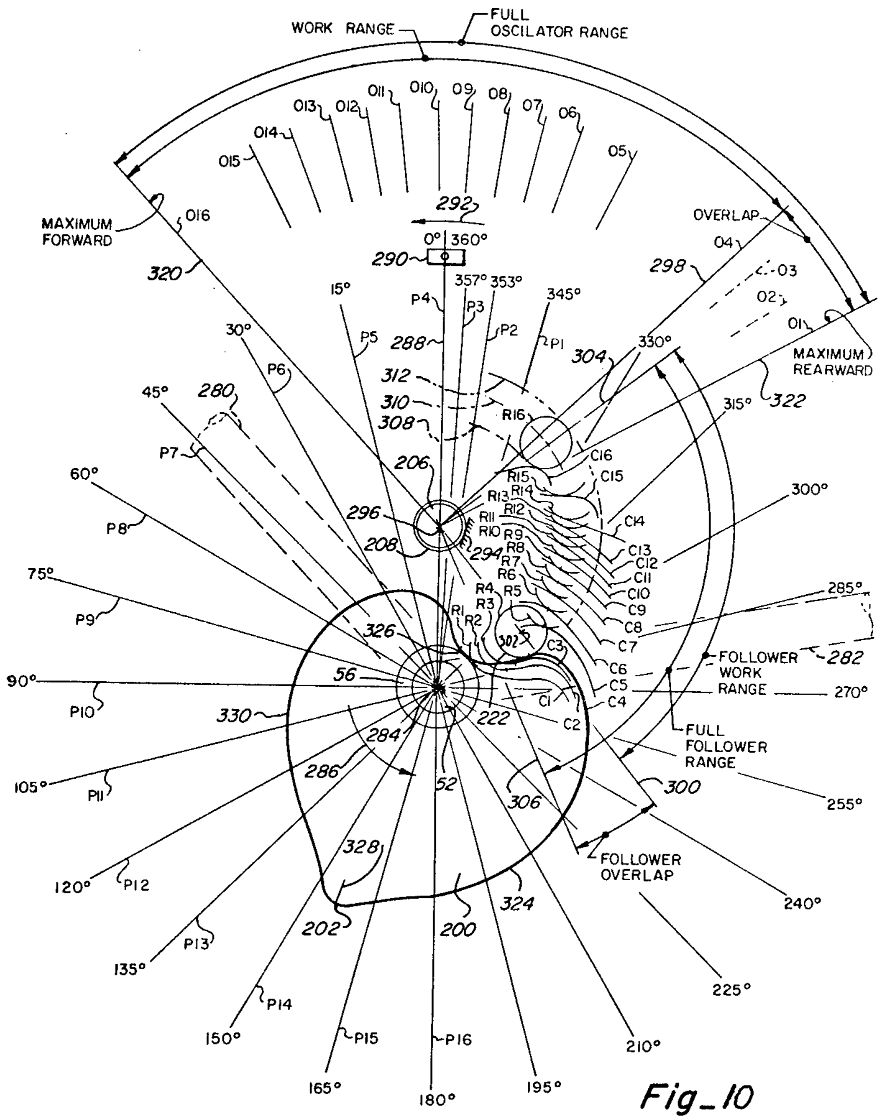
WORK RANGE
FULL OSCILATOR RANGE
MAXIMUM FORWARD
MAXIMUM REARWARD
OVERLAP
FOLLOWER WORK RANGE
FULL FOLLOWER RANGE
FOLLOWER OVERLAP
0° 360°

Fig_10

1581560 COMPLETE SPECIFICATION

8 SHEETS *This drawing is a reproduction of the Original on a reduced scale*

Sheet 6

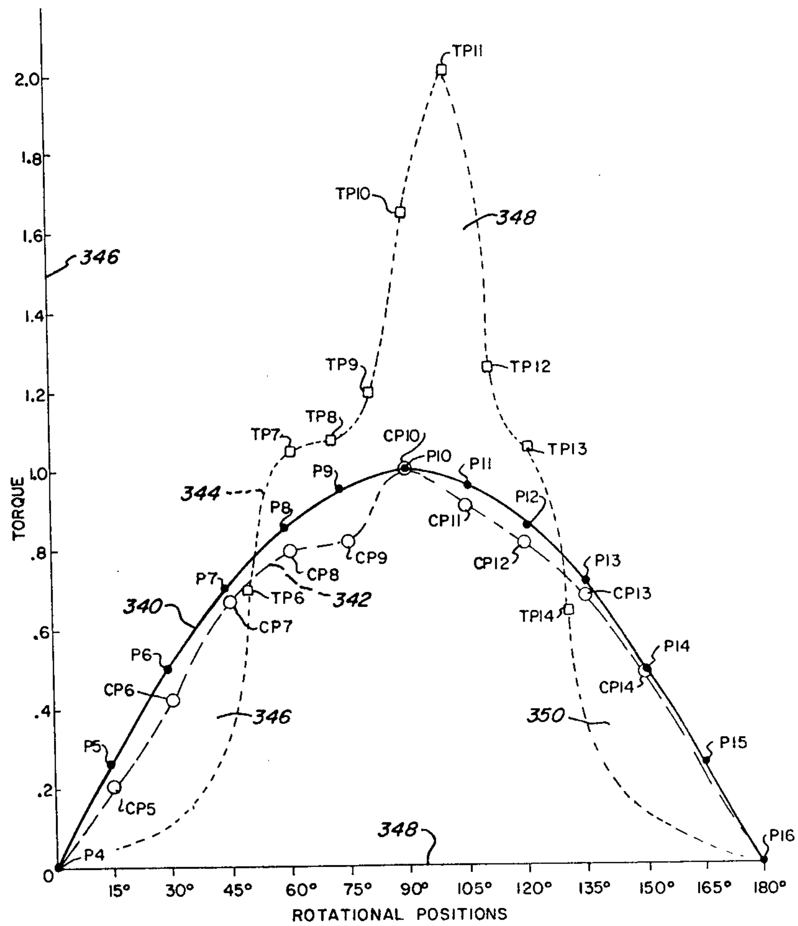

Fig_11

1581560 COMPLETE SPECIFICATION

8 SHEETS This drawing is a reproduction of the Original on a reduced scale

Sheet 7

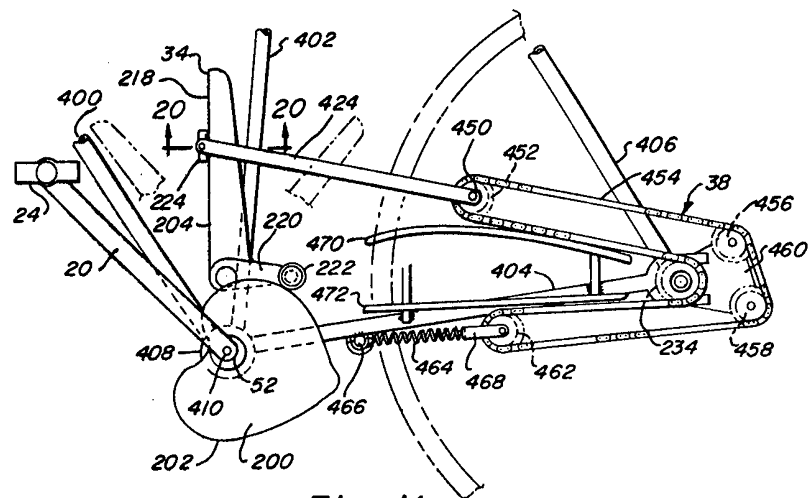

Fig_14

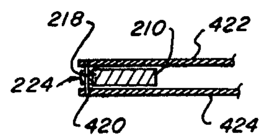

Fig_15

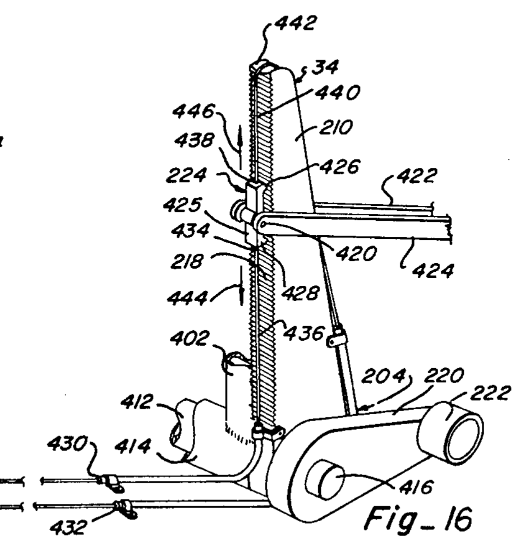

Fig_16

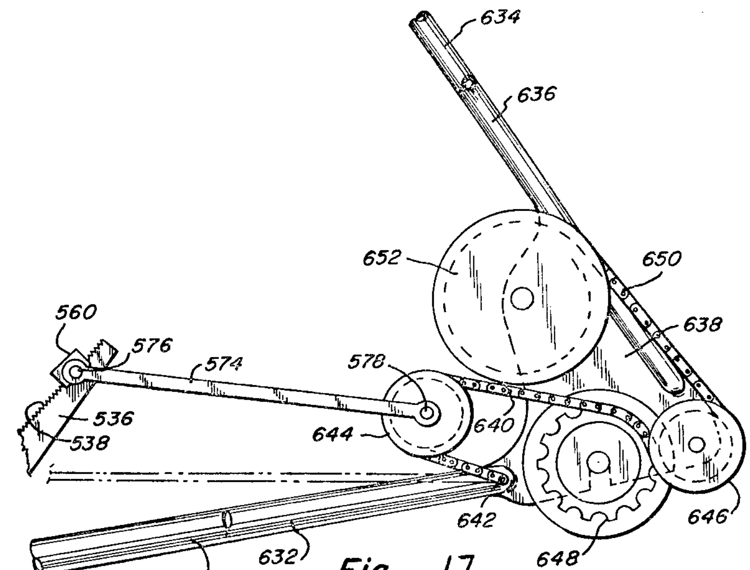
634
636
652
650
638
560
576
574
578
536
538
644
640
646
642
648
630
632

Fig_ 17