

May 4, 1948.

J. D. BELL

2,440,674

ENGINE DRIVE

Filed Jan. 15, 1944

3 Sheets-Sheet 1

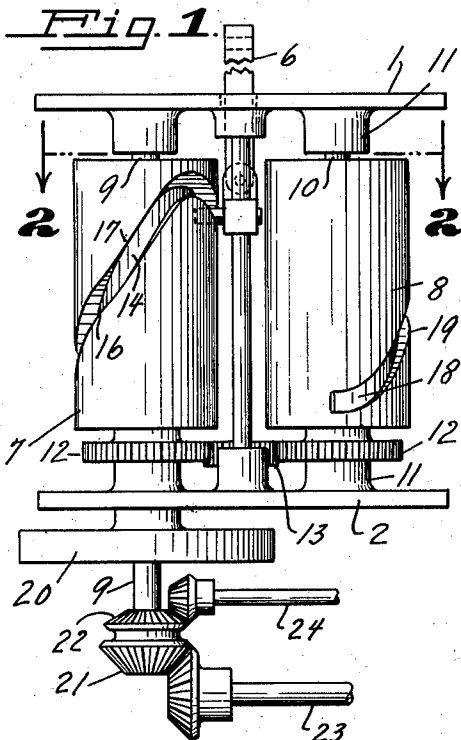
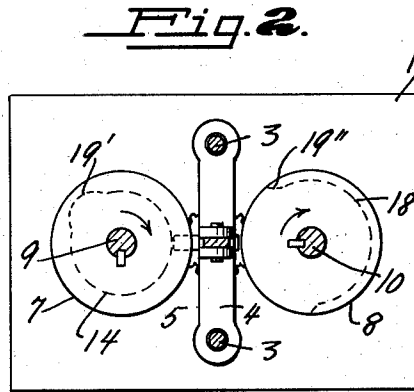
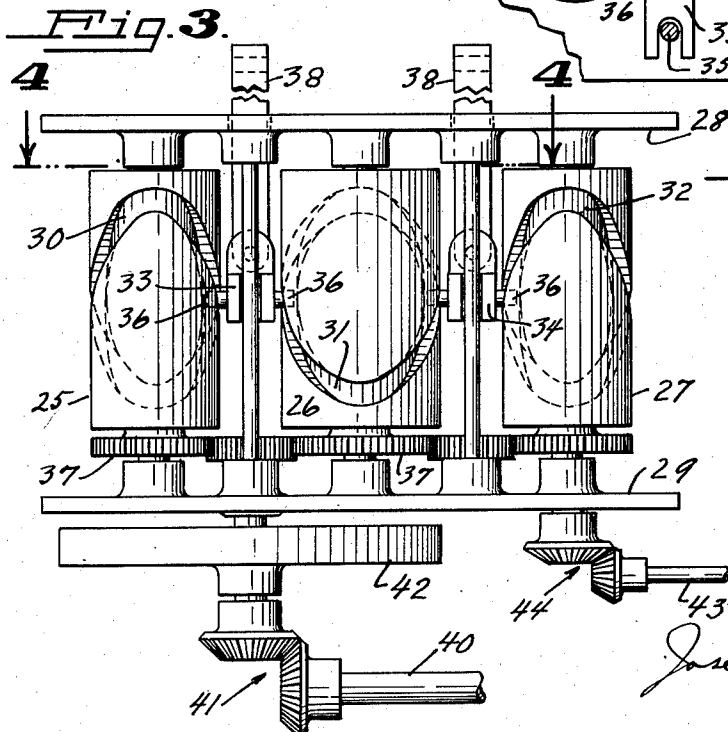
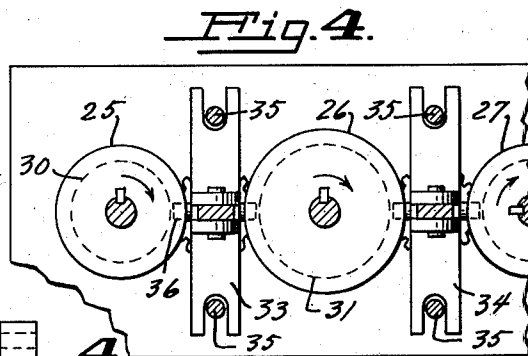
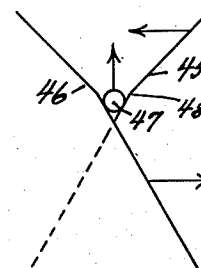


Fig. 5.



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Fig. 6.

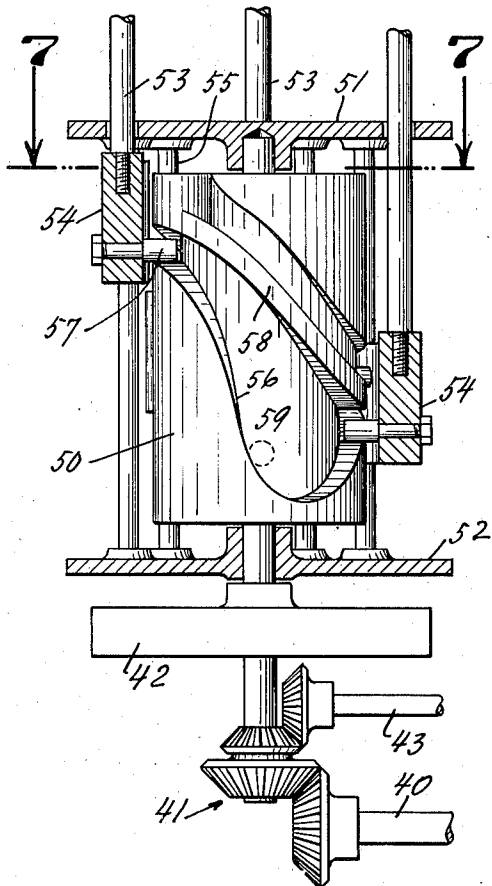


Fig. 8.

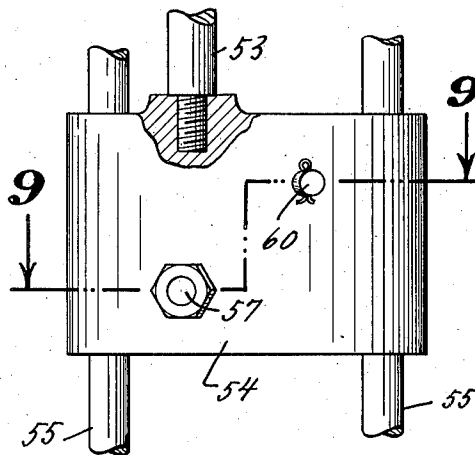


Fig. 9.

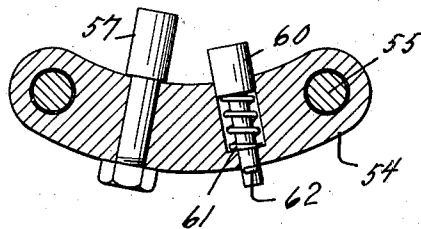


Fig. 10.

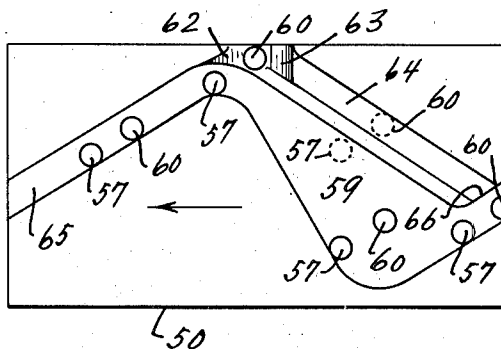
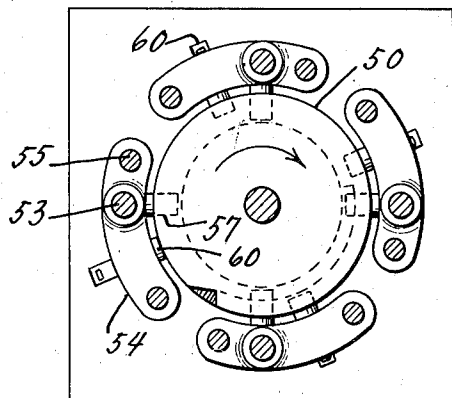


Fig. 7.



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Fig. 11.

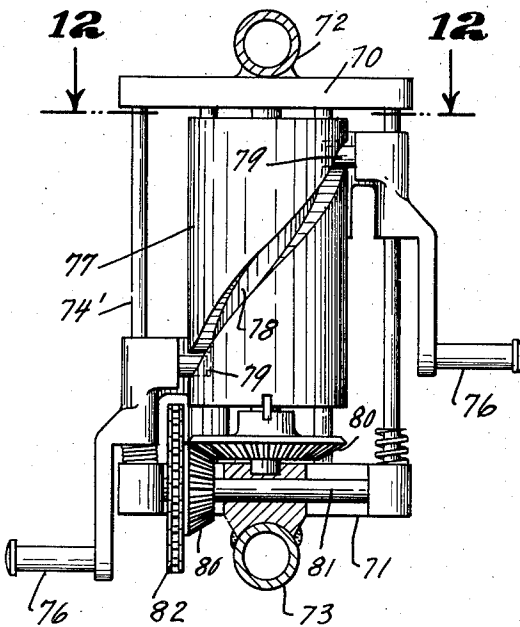


Fig. 13.

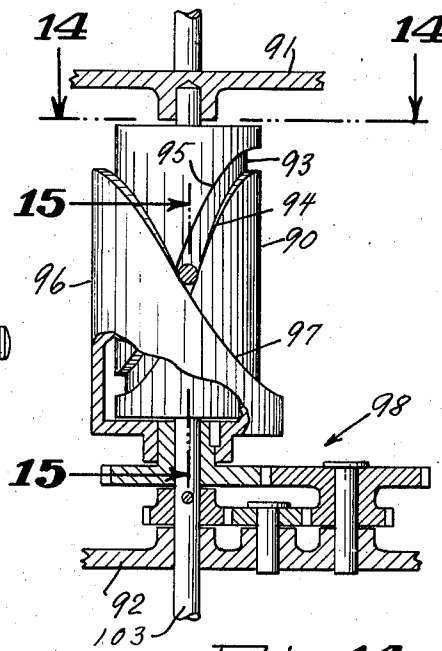


Fig. 12.

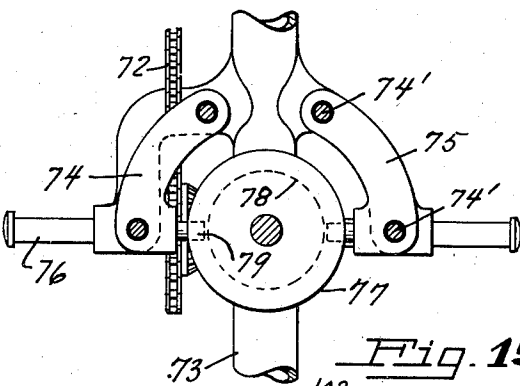


Fig. 14.

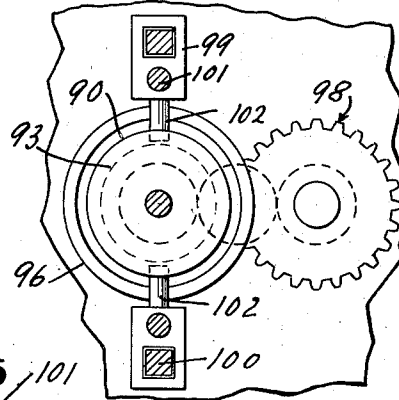
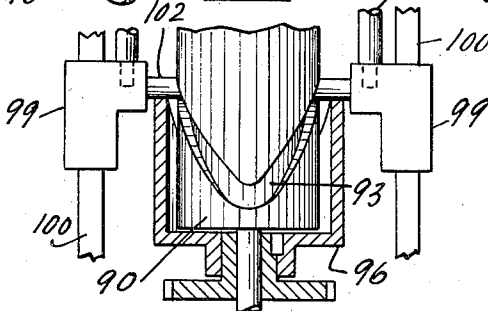


Fig. 15.



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UNITED STATES PATENT OFFICE

2,440,674

ENGINE DRIVE

Joseph D. Bell, San Francisco, Calif.

Application January 15, 1944, Serial No. 518,466

6 Claims. (Cl. 74-58)

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The present invention relates to improvements in engine drives, and its principal object is to provide an engine drive that will give a greater percentage of drive power per pound of pressure exerted by the piston, than does the conventional crank shaft drive or the heretofore proposed cam type drive.

My present invention is a further development of the invention described in my co-pending application, Serial Number 511,156, filed on November 20, 1943, now abandoned, and as far as common subject matter is concerned, the present application should be considered a continuation in part of the former application.

In the co-pending application it was proposed to provide means for turning the reciprocating movement of the piston into rotary motion by means of a pin, driven by the piston, bearing on a helical cam for revolving the latter on the power stroke, while a second helical cam, driven by the first cam, was used to return the pin and the piston on the compression stroke. A still further cam was provided to pull the piston outward during the suction stroke.

These three helical cams were deemed necessary in view of the different work performed by each. For the drive stroke, a helical cam of rather steep grade was desirable to secure the advantage of a favorable wedge action, while on the return stroke a helical cam of light grade was called for to insure favorable lifting action.

For pulling out the piston on the suction stroke again a light grade was desirable, and this cam had to be arranged to bear on the top of the pin instead of on the bottom thereof.

In the present invention it is proposed to use a twin spiral arrangement, with two cams movable in opposite directions, and the push pin of the piston cooperating therewith to simultaneously press on both cam tracks, so as to wedge the two cams in opposite directions.

This arrangement differs from the single cam type in that when the push pin presses on the inclined track of one cam, the deflected force, which in the case of a single cam throws friction on the guide posts for the pin, is now deflected to its companion cam in the form of a driving force against the track. Furthermore, the wedging apart action of the tracks of the two cams at the same time gives a balanced leverage advantage to a degree similar to the advantage one has in steering an automobile with both hands on the steering wheel instead of steering it with one hand.

A further advantage of this arrangement is

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that the pressure of the push pin bearing on the tracks of both cams at the same time is divided for reducing the friction on each track by about fifty per cent.

5 Possibly the greatest advantage is gained on the return stroke of the piston, when the twin cam tracks travelling towards one another wedge underneath the push pin from opposite sides for lifting the same. The advantages of lifting a body by means of two wedges applied from opposite sides is obvious. This feature makes it possible to use a much steeper grade for the inclined track, because friction on the guide post for the pin is entirely eliminated.

10 The introduction of the twin helical cams thus makes it possible to use a steeper grade on the piston return stroke, and possibly a lighter grade on the drive stroke, with the result that the two grades may be made substantially the same which solves one of the principal problems encountered in the helical cam drive.

As compared with the conventional crank shaft drive, applicant's cam drive offers the advantage that with the use of a four inch diameter 25 cam, a full two inch leverage is obtained for the full length of the drive stroke, while the best that can be hoped for in a crank shaft of the same proportion would be an average of about one inch. Besides, the power of the explosion 30 strikes the crank when it has the least leverage, whereas in applicant's structure, the cam has its full leverage when struck by the pressure of the explosion.

Applicant's cam also lends itself to certain 35 refinements that cannot be used in the crank shaft drive. Applicant can, for instance, change the up-grade of the cam to be less steep at the upper end than through its major portion to have greater wedging leverage at the end of the 40 compression stroke. A corresponding construction of the down-grade cam would give greater wedging leverage to the push pin after the greater force of the explosion has been spent. In the crank drive no such compromises can be made.

45 Further objects and advantages of my engine drive will appear as the specification progresses, and the novel and useful features of the invention will be fully set forth in the claims attached hereto.

50 The preferred forms of my invention are illustrated in the accompanying drawings, forming part of this application, in which:

Figure 1 shows a side elevation of one form of my engine drive;

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Figure 2, a horizontal section taken along line 2-2 of Figure 1;

Figure 3, a side elevation of another form of my engine drive;

Figure 4, a section taken along line 4-4 of Figure 3;

Figure 5, a schematic view explaining the twin-cam action;

Figure 6, a side elevation of a further form of my drive;

Figure 7, a horizontal section taken along line 7-7 of Figure 6;

Figure 8, a detail view of a drive pin assembly used in the form of Figure 6;

Figure 9, a section taken along line 9-9 of Figure 8;

Figure 10, a development of the cam groove used in the form of Figure 6, with the surface of the cylinder flattened into a plane.

Figure 11, a side elevation of a form of drive as applied to a bicycle.

Figure 12, a horizontal section taken along line 12-12 of Figure 11;

Figure 13, a side elevation of a further form of my drive, parts being shown in section;

Figure 14, a horizontal section taken along line 14-14 of Figure 13; and

Figure 15, a vertical, fragmentary section taken along line 15-15 of Figure 13.

While I have shown only the preferred forms of my invention, I wish to have it understood that various changes or modifications may be made within the scope of the claims hereto attached without departing from the spirit of the invention.

Referring to the drawings in detail, the form shown in Figures 1 and 2 bears a certain resemblance to Figure 6 of the copending application, previously referred to. The driving mechanism is mounted, in its principal parts, between upper and lower plates 1 and 2 held in spaced relation by any suitable means, such as the rods 3, which latter serve as guide posts for a sliding plunger 4 made in the form of a horizontal bar having a transverse pin 5 slidable in the central portion thereof. The plunger is attached to an engine piston (not shown) by means of a conventional piston rod 6.

On opposite sides of the pins 5 are mounted two cylinders 7 and 8 supported by the shafts 9 and 10 having bearing in the hubs 11 projecting from the plates respectively. The two cylinders are interconnected by suitable gearing 12 including an idler 13, so as to revolve in the same direction and at the same speed.

The cylinder 7 has a helical cam groove 14 cut into the surface thereof, the groove being adapted to receive the pin 5. The groove is endless, ascending on one side of the cylinder and descending on the other side. The lower wall 16 of the groove thus forms an up-grade cam or track on one side and a down-grade cam or track on the other side. The upper wall 17 of the groove forms a cam or track substantially parallel to the lower one.

In operation, as the plunger is forced downward due to the explosion in the combustion chamber, the pin 5, being confined through its mounting to a straight downward path, wedges the cam aside and causes the cylinder to rotate.

Assuming that the cylinder develops sufficient momentum to continue the rotary movement, the bottom part of the cam will reverse the motion of the pin, and the latter will then be made to move upward in a straight path by the up-grade

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portion of the cam wedging underneath the pin. In the upper portion of the cam groove the motion of the pin will again be reversed.

If the next stroke is a suction stroke, the upper wall of the groove, on the down-grade side, will engage the pin and force the latter downwardly, to be lifted again by the up-grade side of the cam as the rotation continues.

It will be noted that the cam selected for the purposes of this invention is rather steep and substantially uniform on opposite sides of the cylinder. This steep grade operates to good advantage on the down-grade side, where the pin wedges the cam sidewise, but is less efficient on the up-grade side, where the cam has to lift the pin.

While being forced upward by the cam, the pin has a tendency to tilt upward and to create considerable friction between the plunger 14 and the rods or guide posts 3. This friction increases with the steepness of the grade and places a limit on the latter. On the other hand, a steep up-grade is desirable to correspond to the steep down-grade on the other side in order to make the opposite sides of the cylinder symmetrical.

In order to eliminate the friction and to render a steep up-grade available, I provide the second cylinder 8 geared to the cylinder 7 to revolve at the same speed and in the same direction. If the cylinders rotate in the same direction, the confronting faces will move in opposite directions.

The second cylinder is formed with a spiral groove 18 extending through half a section of the cylinder and corresponding substantially to the up-grade groove of the first cylinder, so as to present, in its bottom wall 19, a helical track, corresponding substantially to the helical track 16 (on the far side of the cylinder 7, as illustrated).

When the pin 5 reaches the bottom of the groove 14, it is pushed partly into the groove 18 of the other cylinder by a bump 19' in the groove 14, the pin being long enough, however, to leave a part of the same in groove 14.

Thus, the two cams 14 and 18, engaging the ends of the pin from opposite directions, tend to wedge both ends of the pin upwardly, in a shearing action, and thereby eliminate any binding action between the plunger 4 and its guide posts or rods 3.

This construction allows a much steeper grade to be used on the ascending side of the cam. At the end of the up-stroke the pin is pushed back into groove 14, by the tapering off of the groove 18 into the surface of the cylinder, as at 19''.

The shaft 9 of the first cylinder may be used as the power shaft and is provided with a fly wheel 20 and a pair of bevel gears 21 and 22 to serve as take-offs for the main drive shaft 23 and the timing shaft 24.

The form shown in Figures 3 and 4 works substantially on the same principle, except that three cylinders are used in connection with two piston rods and that the endless helical grooves are carried clear around all the cylinders which eliminates the shifting of the pushing pin.

The three cylinders 25, 26 and 27 are revolvably mounted, with their axes in the same plane, between the plates 28 and 29, the central cylinder being preferably somewhat larger than the side cylinders.

Each of the cylinders is formed with an endless helical groove 30, 31 and 32, the grooves in the outer cylinders being symmetrical and the

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groove in the central cylinder being reversed in the sense that in the position shown the bottom section of the groove in the central cylinder faces frontward, while the bottom sections of the grooves in the side cylinders face rearwardly. The axial length of the three grooves is the same, but the grade of the central groove naturally is less steep than that of the two other grooves, since the diameter of the central cylinder is larger.

Two plungers 33 and 34 are provided to slide on posts 35, one between the central cylinder and each side cylinder, and each plunger carries a pin 36 which extends into opposing grooves. The cylinders are interconnected by suitable gearing indicated at 37 to cause the cylinders to rotate at the same speed and in the same direction, with confronting faces moving in opposite directions.

The two plungers are connected to pistons (not shown) by suitable connecting rods 38, and power may be taken off at any one of the gear wheels or interposed idlers. In the drawing the main drive shaft 40 is connected, by bevel gears 41, to one of the idler shafts, which carries the fly-wheel 42, while the timing shaft 43 is driven, through bevel gearing 44, from one of the side cylinders.

In operation, as one of the drive or pusher pins 36 moves down, it wedges opposing walls of the cam grooves in opposite directions, causing the two cylinders involved to rotate in the same direction. Momentum or other drive pistons cause the cylinders to continue this rotary movement, and after the lower center of both grooves is passed, the two ascending sides of the cam grooves or tracks wedge the pin upward, eliminating all side strain or binding along the guide posts. If the next stroke is a suction stroke, it is apparent that the outer walls of the two cam grooves will cooperate in wedging the pin 36 downward.

The second pin 36 works on the same principle and is timed for proper cooperation with the first pin in a conventional manner. It can be readily visualized that any number of piston drives may be added by furnishing further cam cylinders and drive plungers.

The diagram of Figure 5 illustrates the wedging action of two lifting cams 45 and 46 on a pin 47 and particularly shows the wedging from opposite sides for eliminating friction and for reducing wear and resistance on each cam. It further illustrates how the grade of the cams may be changed to apply greater power toward the end of the compression stroke. The bend 48 in each of the cam tracks is here shown as being angular to accentuate the same, but it is apparent that in practice the bend would be smoothed into a curve.

The forms shown in Figures 6-10, inclusive, employ a somewhat different principle. A single cylinder 50, rotatably mounted between plates 51 and 52, is driven by four piston rods 53 arranged symmetrically around the cylinder. Each piston rod terminates in a plunger 54 guided vertically by guide posts or rods 55, which serve also to hold the plates 51-52 in spaced relation.

The piston rods 53 are 90 degrees apart from one another, and the cylinder 50 is formed with a cam track 56 with a steep down-grade extending through a quarter of a circle, and a light up-grade extending through three-quarters of the circle. Each plunger 54 has a pin 57 projecting therefrom to ride on the cam track 56.

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It is assumed, in this form, that the four piston rods receive driving impulses successively. As viewed in Figure 6, the left-hand rod would receive the first driving impulse and would force the plunger 54 downward, causing the pin 57 to wedge the cylinder through a quarter of a revolution. In the meantime the three other piston rods would be lifted by the relatively light cam grade extending through three-quarters of the cylinder's circumference.

Thus, after having been turned through a quarter revolution, the cam track would find the next piston rod 53 ready for the next impulse, and so forth, with the result that the cylinder would receive a rotary impulse each quarter of a revolution, while allowing each piston rod three-quarters of a revolution to return to its initial driving position.

In case the driving force behind one of the piston rods should fail, means are provided for pulling the connecting rod down. This means comprises a second cam 58 arranged above the down grade section of the main cam and having a much easier grade. If the piston 53 fails to drive the plunger down, it is apparent that the revolving cam 58 will pull the pin 57 and the piston rod down and will, at the end of a quarter of a turn, deliver the pin to the main cam on the up-grade side, for the pin to then follow its regular course.

Since the grade of the cam 58 is much less steep than the grade of the cam 56, and thus there is considerable space between the cams, as at 59, it is desirable to positively guide the pin 57 along the lower face of the cam 58 in case the power behind its piston fails. This I accomplish by means of a second pin 60 mounted in each plunger ahead of the main drive pin and at a higher elevation, as shown particularly well in Figure 8. This pin is spring-pressed, as shown at 61 in Figure 9, and normally does not affect the operation of the drive at all. It is shorter than the main pin and its depth of penetration is limited by a stop 62.

The operation may best be explained by reference to Figure 10 which shows the four sets of pins in their respective positions at a given time. Commencing at the bottom of the main cam 56, it will be noted that the bottom is about to approach one set of pins, the main pin 57 riding on the cam and the auxiliary pin 60 being accommodated in the space 59 between the two cams.

The next set of pins (on the right side of the drawing) is shown as climbing the grade of the cam, both pins riding on the cam surface. The third set of pins (on the left side of the drawing), is also shown as climbing the grade, at a higher elevation than the second set.

The fourth set of pins has reached the top of the grade, the main pin 57 riding over the crest of the cam. But the auxiliary pin, while approaching the peak, has struck the upper wall of the cam groove, which is tapered at that point, as at 62, and has climbed the taper so as to rest on the outer wall of the cylinder. The pin, of course, has retracted against the spring 61 for this purpose.

While in this position, the pin 60 may follow one of two courses. If the piston drive is active, and the pin 57 is forced down the steep incline, the pin 60 will follow and will snap down from the outside of the cylinder into the cam groove as soon as the latter widens sufficiently to accommodate the pin. When approaching the bot-

tom of the grade, the two pins will then occupy a position corresponding to that of the pins shown near the bottom of the cam, in full lines.

If, however, the power fails on the drive stroke, the pin 57, at the peak of the cam, will hesitate, and the pin 60 will ride down the incline 63 into the upper cam groove 64, which is of less depth than the main cam groove 65 and will force the main pin to closely follow the bottom face of the cam 58. This course of the two cams is indicated in dotted lines. When reaching the end of the down-grade, the auxiliary pin 60 drops back into the main groove, as at 66, and both pins then follow their regular course.

The main drive shaft 40 and the timing shaft 43 may be connected to the cylinder shaft through bevel gearing 41 in the manner indicated.

Figures 11 and 12 illustrate my invention as applied to a bicycle. The top and bottom plates 70 and 71 are suitably secured to bicycle frame members indicated at 72 and 73. Four rods or guide posts 74 cooperate with the frame members for holding the plates in spaced relation, and also serve as guides for the plungers 74 and 75. Each of these plungers has a pedal 76 projecting therefrom for convenient foot operation.

Centrally between the plungers is mounted the revolvable cylinder 77, with the endless spiral cam groove 78, and the plungers 74 and 75 have pins 79 projecting into the cam groove. The cylinder drives, through bevel gearing 80, a horizontal shaft 81, which again is connected to the rear wheel of the bicycle through a conventional sprocket drive indicated at 82.

Possibly the most promising form of my invention is illustrated in Figures 13 to 15, inclusive. This form works on substantially the same principle as that shown in Figures 3 and 4, but has the advantage of greater simplicity and compactness, insofar as the two cooperating cam members are mounted in concentric relation.

Referring to the figures in detail, the cylinder 90 is revolvably mounted between the two plates 91 and 92, and is formed with an endless helical cam groove 93 furnishing a lower cam track 94 and an upper cam track 95. The cylinder is surrounded by a concentric sleeve 96 cut along its upper edge to present a cam track 97 corresponding substantially to the lower cam track 94 of the cylinder.

The cylinder and the sleeve are interconnected by suitable gearing, indicated at 98 to revolve in opposite directions, with the cam tracks intersecting in the manner shown.

The plungers 99 are mounted on squared guide posts 100 on opposite sides of the cylinder and have suitable piston rod connections 101. Each plunger has a pin 102 extending over the cam track of the sleeve into the cam groove of the cylinder.

Thus the pins, when moving down-grade, under pressure, wedge the two cam tracks apart in a manner similar to that shown in Figure 3, while on the upward stroke the two cam tracks combine in wedging the pin upwards from opposite sides. Power may be taken from the cylinder shaft 103 in the manner previously described.

Since the principal object of the invention is to provide maximum wedging action on the power stroke consistent with adequate wedging action on the compression stroke, the cams have been developed to approximate an ideal condition for accomplishing this object. The cams are formed to give an approximate angle of 25°, with the axis

of the drums and an approximate angle of 65° with the base of the drums.

The wedging action thus produced approximately doubles the power on the drive stroke and halves the power on the compression stroke. This proportion approximately corresponds to the working characteristics of a conventional gas engine of the Diesel or internal combustion type, in which the power consumed by the compression stroke is approximately one-fourth of the power created by the drive stroke.

Thus the shape of the cams creates the ideal condition where a maximum wedging action is allowed for the drive stroke consistent with adequate wedging action on the compression stroke.

It will be noted that in the structure of Figure 13 the pin projects over the two cams from the outside. This allows both cams to bear on the pin on the same side of the piston rod eliminating torque and allowing the two cams to be immediately adjacent to one another so that both cams can have approximately the same grade. While the drawing shows a slight spacing between the drums, this has been done merely for the purpose of making the drawing more easily readable. There is no reason why the drums could not be immediately adjacent to one another so that each would form a bearing for the other.

It is apparent that the cams of Figure 13 can be readily modified in the manner taught in Figure 5 to increase the wedging action toward the end of the compression stroke.

I claim:

1. In an engine drive, a pair of rotary members having endless helical tracks thereon with complementary up and down grades of substantially the same pitch, and a drive pin adapted to rotate the tracks in opposite directions when bearing on the down grades and to be lifted by the tracks when bearing on the up grades, the tracks being relatively steep to produce maximum wedging action on the downgrade consistent with the operativeness of the upgrade portions of the tracks to lift the pin with the tracks arranged in concentric relation and the pin projecting over the tracks from the outside thereof.

2. In an engine drive, a pair of rotary members having endless helical tracks thereon with complementary up and down grades of substantially the same pitch, and a drive pin adapted to rotate the tracks in opposite directions when bearing on the downgrades and to be lifted by the tracks when bearing on the up grades, with the two tracks arranged in concentric relation and the pin projecting over the tracks from the outside thereof.

3. In an engine drive, a pair of rotary members having endless helical tracks thereon with complementary up and down grades of substantially the same pitch, and a drive pin adapted to rotate the tracks in opposite directions when bearing on the down grades and to be lifted by the tracks when bearing on the up grades, the tracks being relatively steep to produce maximum wedging action on the down grade consistent with the operativeness of the up grade portions of the tracks to lift the pin, with the tracks arranged in closely spaced concentric relation and the pin projecting over the tracks from the outside thereof.

4. In an engine drive, a pair of rotary members having endless helical tracks thereon with complementary up and down grades of substantially the same pitch, and a drive pin adapted

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to rotate the tracks in opposite directions when bearing on the down grades and to be lifted by the tracks when bearing on the up grades, the tracks being relatively steep to produce maximum wedging action on the down grade consistent with the operativeness of the up grade portions of the tracks to lift the pin, with the tracks arranged in closely spaced concentric relation and the pin projecting over the tracks from the outside thereof, and the shape of the helical tracks being modified to increase the wedging action toward the end of the lifting action.

5. In an engine drive, a pair of rotary members having endless helical tracks thereon with complementary up and down grades of substantially the same pitch, and a drive pin adapted to rotate the tracks in opposite directions when bearing on the down grades and to be lifted by the tracks when bearing on the up grades, with the two tracks arranged in closely spaced concentric relation and the pin projecting over the tracks from the outside thereof.

6. In an engine drive, a pair of rotary members having endless helical tracks thereon with complementary up and down grades of substantially the same pitch, and a drive pin adapted to rotate the tracks in opposite directions when bearing on the down grades and to be lifted by the

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tracks when bearing on the up grades, with the two tracks arranged in closely spaced concentric relation and the pin projecting over the tracks from the outside thereof, and the shape of the helical tracks being modified to increase the wedging action toward the end of the lifting action.

JOSEPH D. BELL.

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