

Dec. 5, 1967

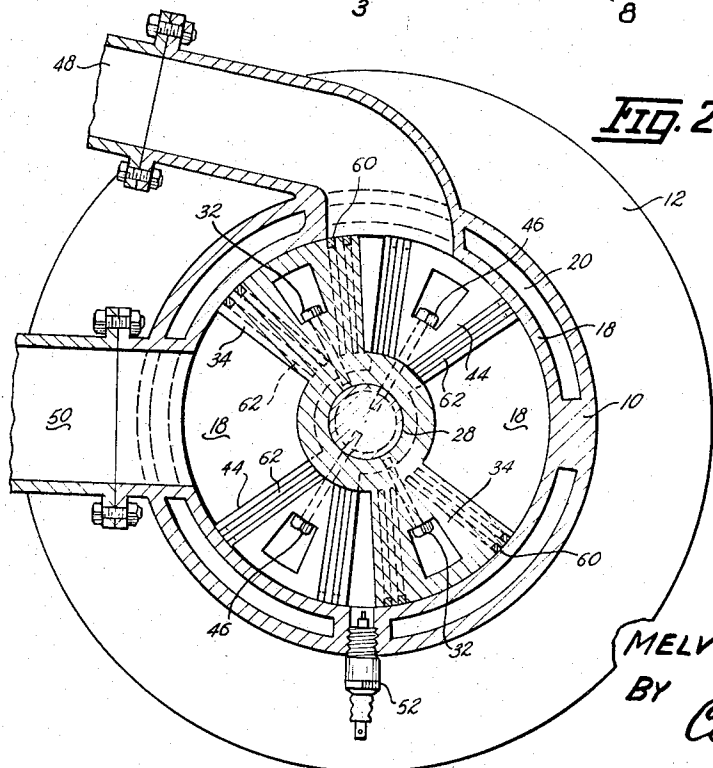
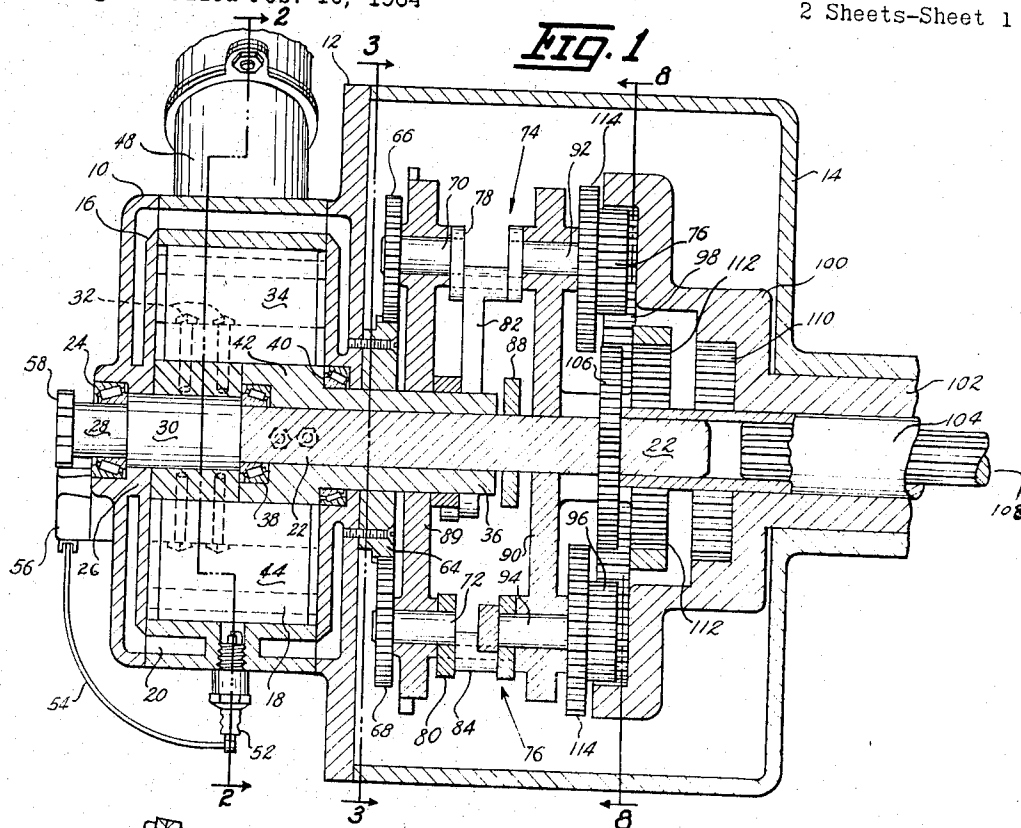
M. T. ROLFSMEYER

3,356,079

ROTARY INTERNAL COMBUSTION ENGINE

Original Filed Feb. 10, 1964

2 Sheets-Sheet 1



INVENTOR.
MELVIN T. ROLFSMEYER
BY
Corey & Corey
ATTORNEYS

Dec. 5, 1967

M. T. ROLFSMEYER

3,356,079

ROTARY INTERNAL COMBUSTION ENGINE

Original Filed Feb. 10, 1964

2 Sheets-Sheet 2

FIG. 3

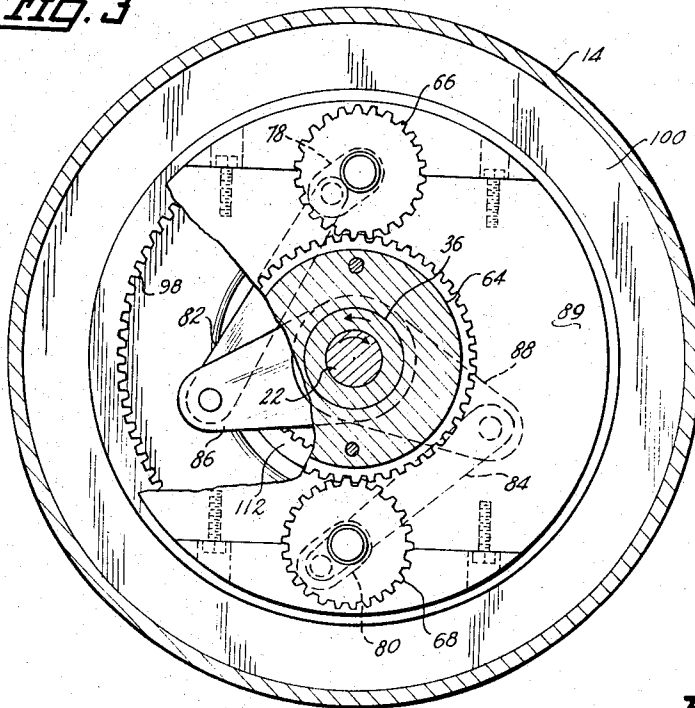


FIG. 4

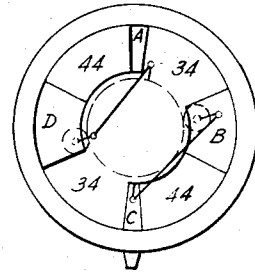


FIG. 5

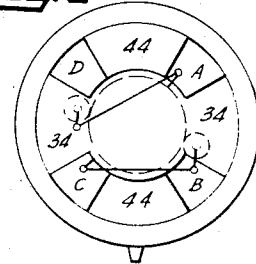


FIG. 6

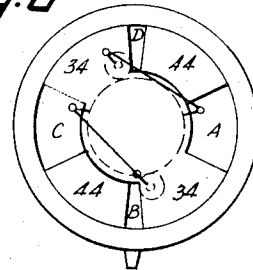


FIG. 7

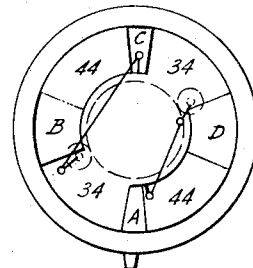
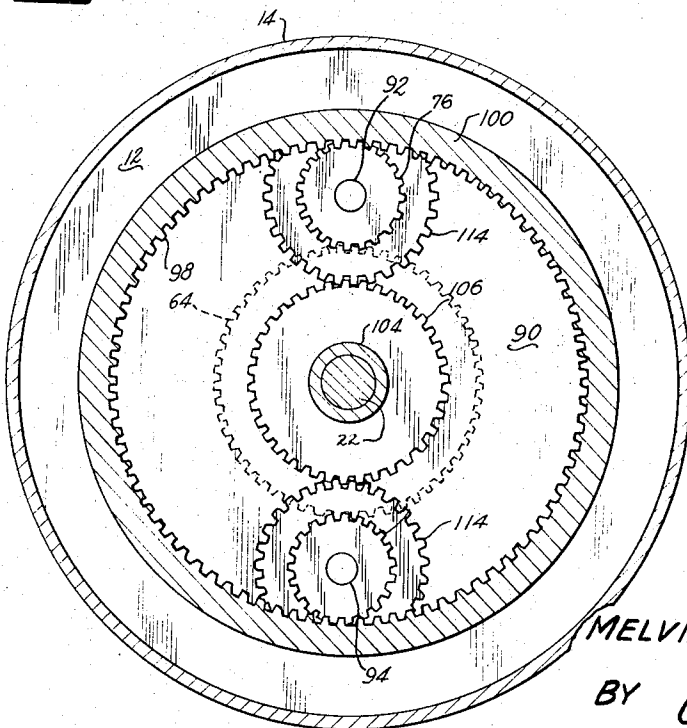


FIG. 8



INVENTOR.
MELVIN T. ROLFSMEYER

BY *Corey & Corey*
ATTORNEYS

1

3,356,079

ROTARY INTERNAL COMBUSTION ENGINE
Melvin T. Rolfsmeyer, Lincoln, Nebr., assignor to Virmel Corporation, Lincoln, Nebr., a corporation of Nebraska
Continuation of application Ser. No. 343,540, Feb. 10, 1964. This application Nov. 29, 1966, Ser. No. 597,797
3 Claims. (Cl. 123—11)

This is a continuation of Ser. No. 343,540, filed Feb. 10, 1964, now abandoned.

This invention relates to an internal combustion engine and more particularly to such an engine in which the pistons rotate within a cylinder while they accelerate and decelerate thus creating relative movement between them.

The basic internal combustion engine, of course, is old, and there are numerous designs of this type of engine known and used commercially today. The multiple piston engine is found in a wide variety of designs including the in-line, V-8, radial, and opposed piston types the operate on either two or four cycles. In these conventional reciprocating piston engines, a certain amount of power is wasted in reversing the direction of the reciprocating pistons. Also, the conventional reciprocating piston engines have a relatively low horsepower-to-weight ratio.

It is therefore a principal object of my invention to provide an improved internal combustion engine which utilizes pistons that rotate within a cylinder.

It is another object of my invention to provide an improved internal combustion engine of an extremely simple and compact design and having a minimum number of parts and capable of being produced at a relatively low cost. An engine constructed according to the principles of my invention also has a relatively high horsepower-to-weight ratio.

It is a further object of my invention to provide an improved internal combustion engine which is substantially free from vibration since the pistons rotate around the drive shaft.

It is a still further object of my invention to provide an improved internal combustion engine of the rotary type which has a greatly increased power output for the same cubic inch displacement as a conventional internal combustion engine of the reciprocating piston type.

It is another object of my invention to provide an internal combustion engine and power transmission mechanism in which the reverse gear is capable of transmitting tremendous power to the output shaft, the resistance to rotation of the shaft helping to turn over the motor.

These and other objects of my invention will be readily apparent from a consideration of the following description taken in connection with the accompanying drawings in which:

FIG. 1 is a longitudinal sectional view of the engine and transmission showing the internal details thereof;

FIG. 2 is a sectional view taken through the cylinder of the engine along the line 2—2 of FIG. 1;

FIG. 3 is a sectional view taken on the line 3—3 of FIG. 1 and illustrating a portion of the crank drive mechanism;

FIGS. 4, 5, 6 and 7 are diagrammatical views of the internal workings of the cylinder illustrating the operation of the engine through the various stages of a cycle; and

FIG. 8 is a sectional view taken on the line 8—8 of FIG. 1 and showing further details of the transmission mechanism.

Referring now to the drawings, a cylindrical-shaped housing 10 has a back plate 12 that provides a partition between the combustion chamber and drive and transmission mechanisms which are contained in a housing 14. Inside and spaced from the cylindrical housing 10 is an interior housing 16 that defines a cylindrical combustion chamber 18. The space 20 between the exterior housing

2

10 and the interior housing 16 is utilized for the circulation of a cooling medium which after circulating would be cooled and returned to space 20 by any suitable means similar to that used in conventional cooling systems and well known to those skilled in the art.

A main drive shaft 22 extends axially through the center of the combustion chamber 18 and partially through the transmission and drive housing 14. Shaft 22 is journaled in an outboard bearing 24 that is seated in a recess at the outside end of the cylindrical housing 10. The recess for the outboard bearing 24 is provided by an annular shoulder 26 formed in the housing 10 adjacent to shaft 22. The shaft 22 is stepped and has a first large diameter portion 28 that abuts the bearing 24 and a second larger portion 30 that abuts the inside of shoulder 26.

Studs 32 serve to secure to the portion 30 of shaft 22 a pair of segmental-shaped pistons 34 located at diametrically opposed positions on the shaft. The pistons 34 lie along a diameter of the cylindrical combustion chamber 18 and are rotatable about an axis coincidental with the axis of the chamber 18.

Telescoped over the inner end of the shaft 22 is a sleeve that provides a second drive shaft 36. The drive shaft 36 is journaled at its outer end on a bearing 38 through which the shaft 22 extends. Thus, bearing 38 provides for relative rotation of shafts 22 and 36 and is axially positioned by abutting the enlarged diameter portion of shaft 22. Shaft 36 also has a stepped diameter, the largest diameter portion 42 being equal to and adjacent the largest portion 30 of shaft 22. The shaft 36 is journaled at its inner end in a bearing 40 that abuts the large diameter portion 42 of the shaft 36 and is seated in the back plate 12 of the housing 10. The large diameter portion 42 of shaft 36 has affixed thereto a second pair of segmental-shaped pistons 44 which also lie along a diametral line of the combustion chamber 18 and which are affixed to the shaft 36 by means of suitable studs 45 (FIG. 2).

From the description thus far, it is evident that there are two pairs of diametrically-opposed pistons in the cylindrical chamber 18. Each pair extends the entire width and depth of the chamber 18 but are connected to separate drive shafts 22 and 36 which are rotatable relative to one another. Thus, the two pairs of pistons 34 and 44 are rotatable relative to each other or "reciprocable" within a limited arcuate distance.

To feed a fuel-air mixture to the combustion chambers formed between the pistons in chamber 18 of the engine, there is provided an inlet manifold 48 that is connected to a carburetor or fuel injection system (not shown) a suitable design of which is known to those skilled in the art. The inlet manifold 48 is connected to the chamber 18 at the top thereof, and an exhaust manifold 50 is connected to the chamber 18 at the side or 90° counter-clockwise (FIG. 2) from the inlet manifold 48. At the bottom of chamber 18, and positioned 180° from the inlet manifold 48 there is an ignition device, such as a spark plug 56 of a design well known to those skilled in the art. The timing of the spark produced by plug 52 is regulated by a timing cam 58 secured to and rotatable with shaft 22 at the end thereof outside of the cylindrical housing 10.

The pistons 34 and 44 each are provided with piston rings 60 fitted in axial grooves along the arcuate outer end of each piston, and also are provided with rings 62 along the sides thereof. These rings provide replaceable seals that separate the individual combustion chambers formed between the pistons.

Referring now to FIGS. 1 and 3, the connecting rod assembly of the drive mechanism will be described. Affixed to the back plate 12 on the side thereof inside the housing 14 is a gear 64 which has an opening in its cen-

ter through which the shafts 36 and 22 extend. A pair of diametrically opposed pinion gears 66 and 68 are meshed with the gear 64. Pinions 66 and 68 are secured to pins 70 and 72, respectively, which are a part of crankshafts 74 and 76, respectively. Connecting rods 78 and 80 are affixed at one end to the main pins of crankshafts 74 and 76, and at their other ends are pivotally connected to levers 82 and 84 which are in turn connected to rods 86 and 88 that are journaled on shafts 22 and 36 respectively. Thus, pinion 66 is operatively connected to shaft 36, while pinion 68 is connected to shaft 22.

The pins 70 and 72 carrying the pinions 66 and 68 are journaled in one side 89 of a flywheel which is therefore driven by rotation of shafts 22 and 36. The shafts 22 and 36 are connected to the other half 90 of the flywheel through the same connecting rods. The pins 92 and 94 at the other ends of the crankshafts 74 and 76 are journaled in the portion 90 of the flywheel to complete the assembly, the portions 89 and 90 rotating together as a unit.

The operation of my novel engine as described so far is as follows, reference being made to FIGS. 4 through 7. With the pairs of pistons 34 and 44 in the relative positions as shown in FIG. 4, there are formed four chambers which I have designated A, B, C and D. The size of these chambers varies continuously as the engine goes through a complete revolution. In FIG. 4, chamber A is opposite the inlet manifold and is just beginning to increase in size. As the pistons reciprocate and thus move apart relative to each other by reason of ignition of a charge of fuel and air in chamber C, chamber A will expand and therefore will draw in through the intake manifold a charge of fuel and air. In FIG. 4, chamber B contains a full charge, and is shown just after it has started to decrease in volume and thus compress the charge. Chamber C is shown in FIG. 4 almost at the peak of compression and is opposite the spark plug 52. Chamber D is just beginning to decrease and exhaust its burned charge and is therefore opposite the exhaust manifold. Because both pairs of pistons 34 and 44 are rotating as well as reciprocating relative to one another, the individual chambers will shift their positions within the chamber 18. FIG. 6 shows the chambers after they have moved approximately 90°. At this time, the charge in chamber C has been ignited driving the pairs of pistons 34 and 44 apart. Chamber A has therefore been expanded to draw in a charge of fuel and air and has also moved to the position in FIG. 6. Thus, chamber A has passed through the intake cycle, chamber B the compression, chamber C the ignition and chamber D the exhaust. Chamber B is now opposite the spark plug 52, and when the charge is ignited, it will immediately expand chamber B while compressing the charge in chamber A. Chambers C and D go through the exhaust and intake cycles, respectively. At the point where the maximum compression of chamber A has taken place the pistons have rotated to the point where chamber A is opposite the spark plug 52, as shown in FIG. 7. At this time the charge is ignited and chamber A is expanded by the force driving apart the two adjacent sides of the pistons 34 and 44. This exhausts the burned charge in chamber B through the exhaust port 50, compresses the charge in chamber D, and draws a new charge into the empty chamber C. Because of the continued rotation of the pistons, chamber A will be fully expanded after it rotates 90° beyond the position of FIG. 7. At this time the compressed charge in chamber D will be ignited expelling the burned gases from chamber A through the exhaust manifold 50. Chambers B and C will go through the intake and compression cycles, respectively. The pistons will continue their rotation, and chamber A, now being empty, will return to the position shown in FIG. 4 at which time the charge in chamber C has been compressed and is ready for ignition, thus completing a complete cycle during which each chamber has passed through the four stages of intake, compression, ignition,

and exhaust. The cycle has required one revolution of the shafts 22 and 36 and in this sense the engine might be considered a two-cycle engine.

Because each pair of pistons 34 and 44 are connected to different drive shafts which are in turn connected by crank shafts to a common flywheel assembly, the relative reciprocating movement of the pistons will drive the flywheels 89 and 90 in the same direction. Turning of the flywheels 89 and 90 causes rotating of the pistons within the cylinder. In effect then, each ignition cycle in each of the chambers serves to move two pistons at the same time, and in effect the four pistons produce the power of eight pistons since relative movement between them drives the flywheels in the same direction. Also, the energy that drives apart the pistons is utilized to compress the charge in one chamber, exhaust the burned charge in another, and at the same time draw a fresh charge into the fourth chamber.

Continuous rotation of the flywheels 89 and 90 will provide the torque to power a vehicle, for example. However, in order to produce various gear ranges for the engine, I have illustrated a simplified transmission. Connected at the end of each crankshaft 74 and 76 is a pinion gear 96. These gears 96 are meshed with an internal ring gear 98 that is a part of and drives the assembly 100 continuously. The assembly 100 includes an extension 102 at the rear of the large ring gear 98. The extension 102 is, in effect, a hollow shaft, and a shaft 104 is received inside extension 102 and has a gear 106 secured to the end thereof. Shaft 104 is hollow at the end containing gear 106 to receive the end of shaft 22. Shaft 22 and shaft 104 are rotatable relative to one another and the shaft 104 is axially movable on a splined shaft 108 relative to both shaft 22 and assembly 100.

The assembly 100 contains a second internal ring gear 110 located to the rear (the right in FIG. 1) of large ring gear 98. The gear 110 is the same diameter as the movable spur gear 106 and its center is on the same axis as the center of gear 106.

A third internal gear 112 of the same diameter as gear 106 is connected to the flywheel 90 on the same axis as gear 106. Thus, when gear 106 is meshed with gear 112, a direct drive providing a "high" gear is produced. When gear 106 is slid into engagement with gear 110, a lower gear is produced since assembly 100 is driven by the smaller pinion gears 96.

In order to provide for a reverse gear, I have affixed to crankshafts 74 and 76 gears 114 which are larger than pinion gears 96 and of a diameter so that they can be engaged with the movable gear 106. When so engaged, gear 106 will be driven in the opposite direction to that produced when it is engaged with either gear 110 or gear 112. This can therefore be utilized to produce a reverse gear. Gear 106 is shown in a neutral position in FIG. 1.

The transmission mechanism shown and described is over simplified but does illustrate how a compact transmission can be associated with my engine and further illustrates an advantage of my engine over those of the prior art.

The torque applied to the shaft 104 when gear 106 is meshed with the reverse gears 114 is in the opposite direction to the rotation of flywheels 89 and 90. Thus, in reverse gear, resistance to rotation of shaft 104 actually assists in driving the pistons thus, producing a tremendous amount of power through this gear.

Having thus described my invention, it will be obvious to those skilled in the art that various revisions and modification can be made therein without departing from the principles thereof. It is my intention, however, that such revisions and modifications as are obvious to those skilled in the art will be included within the scope of the following claims.

I claim:

1. In a rotary internal combustion engine having an electrical ignition system and means for supplying a fuel-

5

air mixture, a cylindrical-shaped combustion chamber and a drive and transmission housing, a main drive shaft extending axially through the combustion chamber and partially through the drive and transmission housing, a second and shorter drive shaft telescoped over a portion of the main drive shaft also extending into the drive chamber, a pair of segmental pistons located in the cylindrical combustion chamber at diametrically opposed positions and secured to the main drive shaft, another set of pistons secured to the second shaft for reciprocatory motion with respect to the first pistons and rotary motion of the pistons as a whole, means for securing reciprocatory and rotary motion of the pistons including a gear secured to the inner wall of the transmission housing adjacent the combustion chamber and concentric with the drive shafts, and a fly wheel of two parts, with one part mounted on the second shaft adjacent the gear and the other part mounted on the main shaft, a pair of crank shafts mounted for rotation transversely in opposite relation through the fly wheel parts, pinion gears on the end of the crank shaft adjacent to and meshing with the first named gear, cranks one on the main drive shaft and another on the second drive shaft and connecting rods from one crank to the crank of the first crank shaft and the other to the crank of the second crank shaft, a driven shaft for furnishing power, and means for furnishing direct drive power from the fly wheel to the driven shaft including an internal gear engaged to the fly wheel and a pinion gear slidably mounted on the driven shaft for engaging and disengaging the fly wheel connected internal gear.

6

2. In a rotary internal combustion engine as defined in claim 1, means for providing a low gear drive including a bell-like driven assembly having a small internal gear with which the slidably mounted gear will mesh and a larger internal gear continuously meshed with a second gear mounted on each of the crank shafts, so that when the said slidable gear is positioned to mesh with said small internal gear in said bell housing, direct drive of the driven shaft is produced at a ratio determined by the relative diameters of the first and second crank shaft gears and the slidable gear.

3. In the engine as defined in claim 1, means for producing reverse drive of the driven shaft including a third gear mounted on each of the crank shafts of such diameter as to mesh with the slidable gear.

References Cited

UNITED STATES PATENTS

20	1,676,211	7/1928	Bullington	-----	123—11
	2,066,952	1/1937	Tornebohm	-----	74—750

FOREIGN PATENTS

25	670,775	9/1963	Canada.
	267,565	9/1927	Great Britain.

RALPH D. BLAKESLEE, *Primary Examiner.*

MARK NEWMAN, *Examiner.*

30 F. T. SADLER, *Assistant Examiner.*