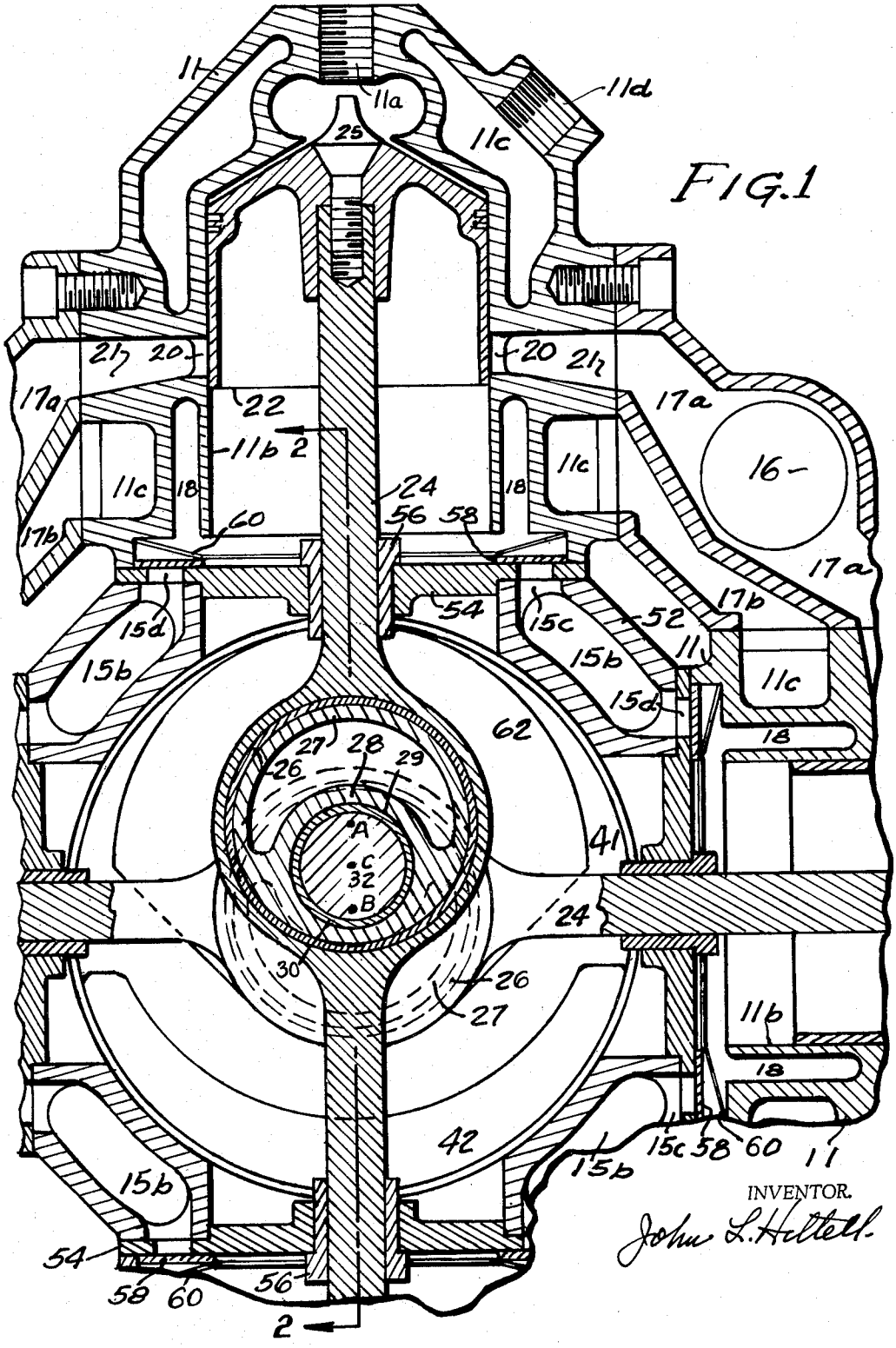
RECIPROCATING PISTON ENGINES

Filed Feb. 25. 1963 3 Sheets-Sheet 1



RECIPROCATING PISTON ENGINES

Filed Feb. 25, 1963 3 Sheets-Sheet 2

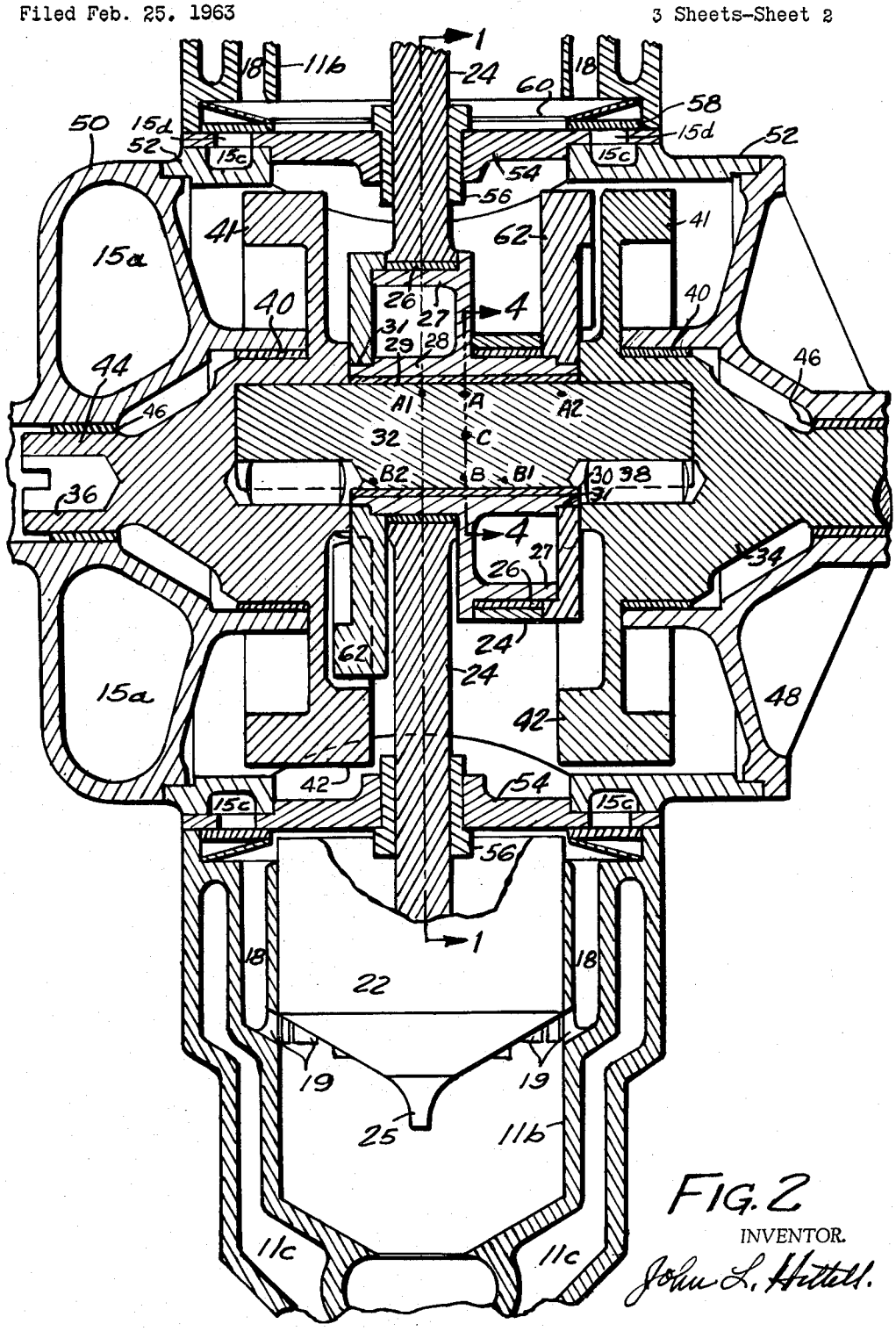


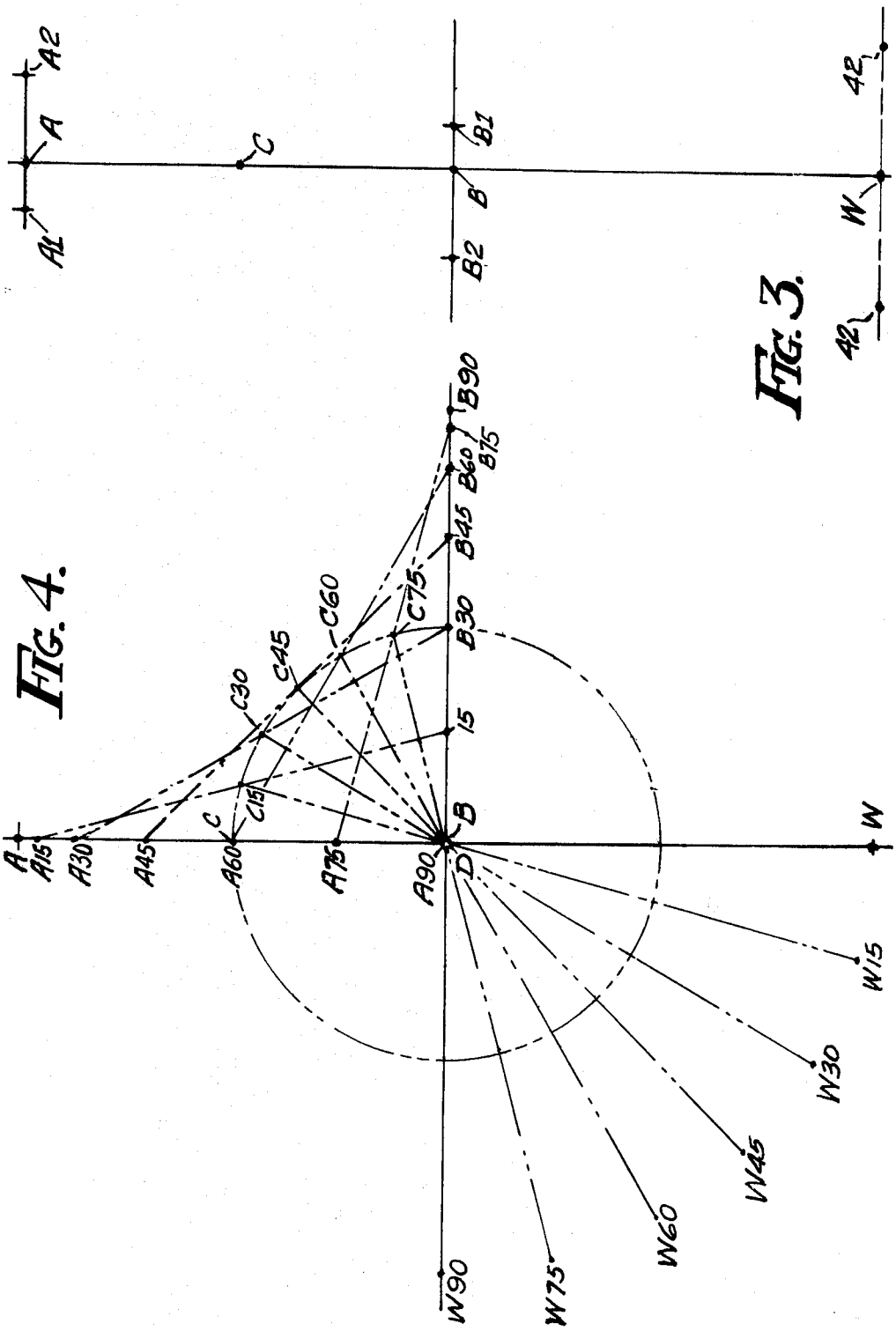
FIG. 2

INVENTOR.

John L. Hittell.

RECIPROCATING PISTON ENGINES

Filed Feb. 25, 1963



United States Patent Office

3,258,992

Patented July 5, 1966

1

3,258,992

RECIPROCATING PISTON ENGINES

John L. Hittell, 9824 Berwick, Livonia, Mich.

Filed Feb. 25, 1963, Ser. No. 260,607
8 Claims. (Cl. 74—603)

This invention concerns reciprocating piston engines, including novel features of power transmitting mechanism.

This application shows and describes certain features being claimed in my copending application Serial Number 498,185 filed September 20, 1965.

In the present state of the art, all reciprocating piston engines vibrate to some degree, becoming much less noticeable with increase in number of cylinders employed. However even with six, eight or more cylinders the vibration is still objectionable and makes it necessary to use vibration absorbing mountings to reduce transmission of vibration to adjoining structures and to people in contact with them, such as the operators or passengers.

In the prior art, certain means have long been known for balancing the strong second harmonic vibration which has been a prominent characteristic of four cylinder vertical engines. Known devices have employed extra shafts carrying counterweights and driven at twice crankshaft speed, involving added cost and ultimate noise and wear in the drive mechanism. They also leave some definite unbalanced higher harmonics of noticeable amplitude, so have not found wide use in spite of the millions of engines needing this type of device which have been produced.

It is recognized that various means have been known in the art whereby pure sinusoidal motions may be generated, but none of these have found very extensive use.

Certain other prior art devices, such as the engine described in the applicant's Patent 2,644,021 now Re. 24,214, have very good dynamic balance with extremely low unbalanced residues of higher harmonics, which under certain conditions can be totally eliminated. However, this type of engine has not been produced in quantities to date, probably due to complication and to wide departure from commercially accepted constructions.

It is also known that certain rotary engines of the general type that is sometimes termed "epicyclic" or epicycloidal and patented as early as April 7, 1903, as well as certain significant recent developments in rotary engines of a very closely related type, can be arranged to have inherent dynamic balance. These prior devices, while theoretically of positive displacement, can only attain this result in practice when the difficult problem of adequate and durable sealing is solved. It is commonly considered among those skilled in the engine arts that it is very much easier to build and maintain durable and efficient sealing when using ordinary cylindrical type pistons and rings.

With this prior art in view, a primary object of this invention is to devise very simple engines using cylindrical reciprocating pistons arranged to inherently permit full dynamic balance at highest speeds, even as a single cylinder engine, and to accomplish this by using a fundamental motion for reciprocation which is free from harmonics and counterbalanced by completely matched means such that all masses are exactly balanced at all times and no harmonics are generated.

A second object is to provide means for constructing engines with these advantages, yet provided in a device which is naturally compact, low in cost and weight and simple to manufacture and service, to thereby attain such light weight by inherent construction rather than by close shaving and high refinement, to permit early development of satisfactory small lightweight engines for small aircraft, for light military equipment for easy long distance air transport, and for general portable use such as in outboard motors, and farm or garden tools.

2

Another important object is to provide simple, durable, quiet and rugged means for driving auxiliary flywheel and counterweight members in continuous and positively maintained synchronism in a reverse direction without the use of gears. An object of the reverse direction auxiliary flywheel is to reduce or even completely cancel torsional reactions on the engine mountings such as are due to internally created cyclic changes of speed of the flywheel masses, as in idling.

Other objects of this invention are such as may be attained by use of the features, combinations and sub-combinations herein disclosed in the various relations to which they may be adapted. While the various means and principles disclosed herein are applicable to engines of one or any other number of cylinders, there are important advantages in using pairs of opposed cylinders, and use of four cylinders at right angles arranged for two cycle operation allows additional advantages such as to afford superb smoothness and light weight as well as simplicity. This arrangement has therefore been selected as the preferred embodiment. The drawings omit some standard and well known accessories and equipment, and show only diagrammatic representations of some other such items. The invention may be fully understood by reference to the drawings in which like characters represent like parts in all views.

In the drawings, FIG. 1 is a partial cross-section stepped to pass through the centers of the four cylinders with the central and main portion of the figure cut on the center of the nearest pair of cylinders along the line **1—1** of FIG. 2, while FIG. 2 shows a section on the line **2—2** of FIG. 1. FIG. 3 is a diagrammatic view showing points A, B, and C arranged as seen in FIG. 2, and projected in line with these points as shown in FIG. 4. In FIG. 4 these points are shown in a section on line **4—4** of FIG. 2, the view being enlarged to clarify the showing of the motion paths of these and other points.

In FIG. 1, one of the cylinders is seen in a vertical position, and in reference herein to cylinders and related parts, words are generally used in the usual sense of vertical cylinders, even though other views show or indicate some other orientation, and the cylinders may be turned in any convenient direction in the use of the engine.

The working fluid may be air or the more usual air-fuel mixture, entering the engine through an inlet passage, which is in free communication with an annular passage **15***a*, also communicating at four places with corner passages **15***b* and with ports **15***c* and apertures **15***d* as may be seen on FIGS. 1 and 2.

The exhaust takes place through four passages **16**, in the four elbows **17**. The passages **16** each join two exhaust passages **17***a*, each elbow providing one passage for each of the cylinders. Each cylinder has a jacket-type inlet passage **18**, seen outside the inner cylinder wall in FIGS. 1 and 2. A plurality of inlet ports **19** communicate directly with the passages **18**, as may be seen in these figures, which also show a plurality of exhaust ports **20** communicating through ports **21** with passages **17***a*. The ports **19** and **20** communicate directly with the cylinder working chambers within the cylinder bores **11***b* in which the pistons **22** operate. Two piston rods **24** are employed. Each rod has two straight end-portions to which the pistons are attached by means of self-locking screws **25**, four pistons being thereby mounted on the two rods **24**. This is the preferred construction, but one piston per rod may be used and might be better in some cases. Between the two end-portions each rod has an eye-portion with a bushing **26** fitted therein. Each of the two bushings **26** has journalled for rotation within it, one of two eccentric portions **27**, of a compound eccentric member **28**. These two eccentric portions are on centers

diametrically opposite with respect to, and equally spaced from, the center of a bore **29** passing through the center of the member **28**. This bore is fitted with a bushing **30** in which is journalled a crankpin **32**.

Two shafts **34** and **36** are alike in most respects; each is shaped to closely fit one end of the crankpin **32** and one of two round-pin keys **38**, and is journalled in one of a pair of aligned bushings **40**. The shafts **34** and **36** are also alike in that they each have integral flywheel portions **41** and counterweight portions **42**, and both are tapered to smaller diameters having bearing portions **44** which are journalled in bushings **46**. They differ in that the shaft **34** is on the power end of the engine and its outside end may be coupled to other equipment in any desired manner, while the shaft **36** has an extending end with an offset cross-keyway and pilot forming a standard drive for auxiliary equipment.

The bushings **40** and **46** are supported in an end cover **48** on the power end, and in a cover and inlet manifold **50** on the opposite end. A crankcase frame **52** has affixed thereto the end cover **48**, the cover and inlet manifold **50** and the four cylinders **11**. Clamped between the cylinders and the crankcase frame are four diaphragms **54**, each of which carries a bushing **56** to guide, seal and support one of the straight end-portions of the piston rods. Four washer-shaped inlet check valves **58** seat on the diaphragms **54**. The series of apertures **15*d*** are covered and sealed by these valves as seen in FIG. 1. The lift of the valve is limited by the depth of the valve space provided in the base of each cylinder for this purpose. The valves are seated by thin springs **60** of the belleville type.

Counterweighted flywheels **62** are mounted with their counterweight portions diametrically across the axis of the crankpin **32**. They are each firmly held in this relation by a pilot having an annular surface. Each has a small pilot annular surface **31** concentric with the crankpin **32**, within a larger annular surface fitting one of the eccentric portions **27** of the eccentric member **28**. Since the center distance between these annular surfaces is the same on each end of the eccentric member, the counterweighted flywheels are made alike and turned to the positions shown before assembly.

Similarly, the cylinders are of identical design and merely rotated one half turn around the cylinder axis in each alternate position. These half turns allow the slightly offset cylinder exhaust ports to meet the elbows **17** in a mid-position, while permitting the cylinders to be offset along the engine axis a short distance allowing the piston rods **24** and bushings **26** to be positioned as shown in FIG. 2. Each elbow has a coolant passage **17*b*** to allow interchange of coolant between cylinders where these passages join the coolant jacket spaces **11*c***. Ports for inlet and outlet of coolant are at **11*d***. The like piston rods **24** appear different in FIG. 2, because they are at different angles, as is shown in FIG. 1.

The screws **25** each have a conical seat under the head, and the apex of the cone is made to intersect at the flat piston seat on the end of the piston rod **24**. This prevents loosening or tightening of the mating surfaces by temperature changes, so retains the initial set-up pressure on the piston even though there may be great differences in expansion or contraction between the screw and the piston, as either may expand or contract independently along the cone angle or the flat seat without any effect on the tightness, since the intersection of the apex of the cone and the flat seating face acts as the origin from which any thermal change of dimension occurs.

Lubricant in the crank space is preferably kept drained into a separate container with the lubricant being drawn therefrom and supplied to the mechanism under pressure by a separate pump in a manner well known in other drained crankcase or so-called dry sump systems. The crank space is isolated from the four corner passages **15*b***, by walls as seen in FIG. 1 and from the under-piston spaces by the diaphragms **54** and the bushings **56**, so most of the lubricant is confined to the crank space, a little working through the bushings **56** for lubrication of the pistons.

To permit complete and perfect balance, pure sinusoidal motion, free from all harmonics, is utilized. The operation of the structure used to attain this motion, and the geometry of the motion, may be most easily explained and understood by applying letters to centerlines and center points of several of the parts. In FIG. 1 these centerlines appear as points, and the point A is on the centerline of one of the bushings **26**, while the point B is on the centerline of the other bushing **26**. Central between theese two points is point C which is on the centerline of the crankpin **32**. The positions of these points are also shown to a much larger scale in FIG. 4, and a series of additional points are also shown as **A15**, **A30**, **A45**, and so on, representing the positions of the point A after the crankshaft has rotated 15, 30, 45, or more degrees clockwise in 15 degree increments from the initial position as also seen in FIGS. 1 and 2. The points B and C also have their various respective positions shown in FIG. 4 at the same series of 15 degree increments of rotation, and another point D is shown at the center of the figure and represents the fixed center of the shaft assembly or crankshaft which, in this embodiment, consists of the shafts **34** and **36** carrying the counterweights **42**, the crankpin **32** and the two round-pin keys **38**, assembled and rotatable as a unit. The path of point C is a circle of a radius equal to the distance from the point D to the point C as seen in FIG. 4, and herein called the distance DC. This distance is made the same as the two equal distances from the point C to the points A and B which have been established and described with the elements **26** to **32**.

The points A, B and C are always on a straight line and the point B is also always on the line through the points D and **B90**, so the equal distances from the points A and B to point C cause the point A to be exactly twice as far from this line as the point C. This holds for all positions, and the distance of point C from the line D–**B90** always equals the distance DC times the cosine of alpha, where alpha is the angle through which the crankshaft has rotated from the initial position indicated. Since the distance of point A from the line D–**B90** is always just twice as great as that of point C, or equal to the distance DA (from D to the initial A position) times the cosine of alpha, the motion of point A is a pure cosine motion with full geometric basic precision, so known to be free of any inherent harmonics of the fundamental motion.

While the point A has been proceeding in the described manner from A to **A90**, the point B has moved from D through **B15** etc. to **B90** along its travel path at 90 degrees to the path of point A, and its distance from D is always equal to the same distance DA times the sine of alpha. Everything is the same as for point A except that for point B the distance is determined by the sine instead of the cosine of alpha. This means that the motions are of identical character, but out of phase by 90 degrees, which is in accord with the well known fact that the sine and cosine functions of an angle are identical in manner of variation, but similarly out of phase. The motion of point B is thus also entirely free from harmonics. It should be noted that the line ACB carrying the three point pattern, and the parts **28** and **62** which are centered on these points, are all forced to rotate in a direction reverse to the rotation of the crankshaft and at a precisely equal reverse angle at all times, by the properties of the geometry existing in FIG. 4, so these parts and the three point pattern can have no rotary acceleration while the crankshaft rotates at constant speed. This freedom from rotary acceleration while the points A and B are guided on their specified lines of travel means that no side forces are applied to the guides to maintain the constant velocity rotation of the masses at the A, B and C points. The guides serve to maintain the geometry as speed and other conditions vary, but side forces at the

guides occur during such changes. Linear acceleration of any masses in or attached to the piston rods applies inertia forces to points A and B, directly through these points and in line with their lines of travel, so also in line with point D. No side forces on the guides and no tendencies for rotary acceleration or deceleration of the rotary masses of the mechanism are created by the sinusoidal linear motions of the masses at A and B.

Constant speed of rotation as above considered is approximately achieved during normal operation. The general purpose of flywheels is to minimize cyclic speed variation which obviously may be caused by fluid pressure forces applied to the pistons of any piston pump or engine. When a gear drive is used to couple a second shaft carrying a counterweight as a mass load, the permissible elastic compensation is very low and the forces necessary to attain the required repeated slight speed changes of the mass load must be transmitted by the drive. A reversal of force at a cyclic rate equal to the number of cylinders times the speed of rotation is a rather high frequency reversal, so forces involved are considerable, even with a small amplitude of speed variation. Gear drives tend to suffer noticeable punishment from these reversals. Any back-lash initially existing allows impacts at the line contacts of the gear meshing. These regeneratively increase the punishment as back-lash is increased by the impacts. Noisy operation and shortened life obviously result.

The type of drive herein disclosed avoids these conditions, partly by being commercially producible with less initial back-lash, but mostly by the fact that good oil films of substantial area exist at all places where back-lash might tend to originate. The large area film absorbs the shock, generally preventing any actual metal to metal impact, but if this should occur under some special condition the area of metal to withstand the impact is much higher, and much of the impact has been absorbed by the oil film. A quiet and durable drive for the reverse rotation inertia members is thereby provided by the disclosed structure, which simultaneously serves to create the motion which is free from harmonics.

It should be noted that by making the eccentric annular surfaces, at **27**, quite large in relation to their offset distance, an arrangement has been created in which the respective diameters of the annular surfaces overlap when the annular surfaces are considered as being superimposed one upon the other. This is made sufficient to allow a straight bore through the eccentric member **28**, permitting a rigid crankpin **32** to pass directly through and be encompassed by both of the eccentric portions **27** at the selected offset, to thereby allow support of the pin **32** from both ends in rigid manner, maintaining parallelism of the crankpin. This structure is thus seen to have additional merit in its ruggedness and rigidity.

In contriving arrangements for complete balancing, the motions and other characteristics of the mechanism lend themselves very neatly to the needs, since for balancing considerations the total of the masses effective at the A, B and C points may be treated as one mass effective at C. It should be noted that the masses at A and B would be made equal since like parts are usually mounted thereon, making equal masses in effect at these centers. The total mass effective at point C is thereby that at A and B plus that directly effective at C, and is easily and simply balanced by a suitable mass effective at the diametrically opposed point W, which may be at a larger radius than C, so needing correspondingly less mass for balance. This reduced mass may then be divided into two parts, both being in line with point W in FIGURE 4, but being embodied as portions of the shafts **34** and **36** positioned in their counterweight portions **42**. In FIGURE 3, the point W is on a vertical line with the points A, C, and B, and lies midway between the centers of mass of the counterweight portions **42**. This vertical line corresponds with the plane of FIG. 4, in which the preceding balance conditions have been established.

With this plane at right angles to the axis, dynamic balance conditions are also satisfied.

To complete the conditions under which full dynamic balance may be attained, the mass effective at point A in FIG. 3 is made up of a mass at point A1, and a smaller mass at A2 acting at a longer moment arm from point A. Any proportions would be fully effective if balanced mass-moments of A1 and A2 about point A are maintained. An equivalent condition obviously applies at point B. In FIG. 4 the points A1 and A2 coincide with the point A and the points B1 and B2 coincide with the point B, so these points follow the same motions as those of points A and B. The masses A2 and B2 may be embodied as masses forming the counterweight portions of the counterweighted flywheels **62** and made effective at points A2 and B2 in addition to the portions used purely for flywheel mass which are balanced about the center of the diameter **31** of these flywheels, thereby being directly effective at point C. The masses A1 and B1 are each equal to that of one of the piston and rod assemblies plus one of the eccentric diameter portions **27** and its adjacent walls.

The preceding balancing explanation has all referred to one quadrant of FIG. 4, but obviously the same type of motions are maintained through the four quadrants by alternate mirror image and identical conditions in other quadrants. The masses B1 and B2 continuously balance the masses at A1 and A2 about the axis through point C, and mass-moments about point C as viewed in FIG. 3 are also balanced longitudinally by the equal mass-moments as explained, so a total mass is constantly maintained effective at point C through all quadrants and intervals therein, and is continuously maintained in dynamic balance by a suitable counterweight mass effective at the diametrically opposed point W.

It should be recognized that the disclosed drive structure has made it possible for the two straight line motions of the piston and rod assemblies, whose lines of action traverse and intersect the crank centerline, to operate without interference with the crankshaft even with all of these parts straddle supported, and further has made it possible for the geometry and construction to generate a positive and low back lash reverse drive without the use of gears, thereby providing a long life and quiet drive for the reverse rotation flywheel and counterweight masses without addition of any drive parts not necessary for other major objects. The ingenuity of this arrangement is further attested by the simple construction sufficing to attain full dynamic balance simultaneously with the other important objects. The reverse rotation flywheel mass may even be proportioned to balance and cancel all torque reactions on the mountings that are due to any speed changes caused by internal forces in the engine, such as piston forces when in idling operation. Such proportioning permits equalling the turbine in smoothness, while notably exceeding its performance in several other respects.

The four cylinder two cycle operation gives as many power pulses per revolution as the usual eight cylinder four cycle engine. Clean forced scavenging accompanied by inherently perfect balance permits significantly higher engine speed and proportionately increased number of power pulses per minute and per mile. Thereby this construction affords smoother and quieter operation than the best engines in common use today, even with very much simplified construction.

The disclosed structure may be adapted to provide full dynamic balance even in a single cylinder engine. This result is secured when three of the pistons and cylinders are removed, leaving one piston rod idle and reciprocable in its guides. This rod would then be proportioned to have the same total weight as the other rod plus the single piston remaining, instead of the same weight as the other rod alone, to preserve equal reciprocating masses. This

leaves all the balance conditions fully amenable to balancing according to the description as has been given.

As will be evident to those skilled in the art, various other rearrangements of the elements of this invention may be made to adapt it to pumps, compressors, fluid motors driven by pressurized fluids and to four cycle engines or to pumps and engines of any number of cylinders, without need of inventive ingenuity, while retaining many of the main advantages. My invention should be recognized as embracing all modifications of the disclosed structure and its method of application which may fall within the scope of the claims.

I claim:

1. In a mechanism for converting reciprocal piston motion to rotary shaft motion and vice versa, a crankpin carried in parallel offset relation to the shaft, eccentric means rotatably journaled on the crankpin and including a first eccentric annular surface rotatably journaling a first piston assembly having a combined mass acting at one point and a second eccentric annular surface axially offset with respect to the first eccentric annular surface rotatably journaling a second piston assembly having a combined mass acting at another point and counterweight means fixedly carried by said eccentric means positioned axially adjacent and diametrically opposite said points having at least one mass acting at at least a third point such that said masses have a combined effective mass acting at the center of gravity of the crankpin during all phases of operation.

2. In a mechanism for converting reciprocal piston motion to rotary shaft motion and vice versa, a crankpin carried in offset relation by the shaft, eccentric means rotatably journaled on the crankpin and including a first eccentric annular surface rotatably journaling a first piston assembly having a combined mass acting at one point a fixed transverse distance from the centerline of the crankpin and a second annular surface offset with respect to the first eccentric annular surface rotatably journaling a second piston assembly having a combined mass acting at another point said fixed transverse distance from the centerline of the crankpin opposite from said one point, two counterweights each non-rotatably carried by said eccentric means, such counterweight positioned axially adjacent and diametrically opposite one of said eccentric annular surfaces and each having a mass acting at separate points whereby said piston assemblies and counterweight masses combine to act at the center of gravity of the crankpin during all phases of operation, and additional counterweight means non-rotatably carried by said shaft having an effective mass acting diametrically opposite to said combined piston assemblies and counterweight mass creating an opposite completely balancing force during all phases of operation.

3. A compound eccentric comprising a primary member having (1) a central bore adapted to be journaled upon a crankpin and the like, (2) an outer annular surface at each end thereof concentrically disposed about the longitudinal axis of the central bore, and (3) two axially spaced annular surfaces each adapted to journal a piston assembly, said two annular surfaces each being eccentrically disposed with respect to the longitudinal axis of said central bore, the centerline axis of each said eccentric annular surface being essentially spaced an equal but opposite distance from the longitudinal axis of the central bore and extending generally parallel to each other, and a pair of counterweights each carried on both (1) one said concentric outer annular surface and (2) one said eccentric annular surface.

4. In an engine and the like a shaft rotatably mounted in a crankcase and carrying an offset crankpin, an eccentric rotatably carried by the crankpin and providing a pair of axially spaced annular surfaces the centerlines of which are equally but oppositely eccentrically disposed with respect to the axis of the crankpin in parallel relation thereto, at least two piston rods slidably supported in guide means which in turn are supported by the crankcase, said piston rods being offset from each other by essentially 90° and each being journaled upon one said eccentric annular surface, two axially spaced counterweights each fixed to said eccentric in diametrically opposed relation to the most adjacent eccentric annular surface and respectively extending axially beyond said most adjacent eccentric annular surface, and counterweight means non-rotatably carried on the shaft to counterbalance the centrifugal force created by the total mass supported by the crankpin during all phases of operation.

5. A compound eccentric member comprising a primary member having a central bore and two ends each having an outer surface concentric with the axis of the central bore, a pair of larger diameter portions on said primary member, each larger diameter portion having a centerline parallel with the axis of said central bore and eccentrically spaced an equal distance on diametrically opposite sides of said central bore axis, a pair of fly-wheel members, one fly-wheel member being non-rotatably carried at each end of said primary member at both (1) the adjacent concentric outer surface and (2) the adjacent eccentric larger diameter portion, each said fly-wheel member having a counterweight portion centered in diametrically opposed relation to the most adjacent eccentric larger diameter portion.

6. In an engine and the like, a shaft rotatably mounted in a frame, a pair of reciprocable members having linear lines of action traversing the axis of said shaft at essentially right angles with respect to both (1) each other and (2) the shaft axis per se, guides accommodating reciprocable movement of said reciprocable members, a crankpin carried by the shaft centered at a fixed distance from the axis of the shaft, an eccentric member rotatably mounted on said crankpin, a pair of eccentric portions comprising part of said eccentric member with said eccentric portions each having a centerline equally but oppositely offset from and parallel to the center of the crankpin, means rotatably journaling said reciprocable members on said respective eccentric portions, a pair of flywheel members affixed to the eccentric member with each fly-wheel member having a counterweight portion positioned diametrically opposite the most adjacent eccentric portion whereby the motion geometry of said structure constrains said counterweight portions to reciprocating lineal motions to attain overall full dynamic balance during engine operation.

7. In an engine, a pair of piston assemblies each (1) rotatably journaled upon a crankpin-supported eccentric rotatably-supported on the crankshaft and (2) slidably disposed within a guide, said piston assemblies each extending essentially transverse to the crankshaft centerline at approximately right angles to each other, one said piston assembly having a mass center of gravity in line with a linearly movable first point and the other said piston assembly having a mass center of gravity in line with a linearly movable second point, said first and second points being equally and oppositely offset from the centerline of the crankpin whereby the combined mass of both piston assemblies acts through the center of gravity of the crankpin.

8. In an engine, a pair of piston assemblies each (1) rotatably journaled upon a crankpin-supported eccentric constituting part of the crankshaft and (2) having a rod slidably disposed in a guide, said piston assemblies each linearly reciprocably extending essentially transverse to the centerline at approximately right angles to each other, one said piston assembly having a mass center of gravity in line with a linearly movable first point and the other said piston assembly having a mass center of gravity in line with a linearly movable second point, said first and second points being equally and oppositely offset from the centerline of the crankpin and thereby having a combined mass centered thereon, and counterweight means carried

by the shaft having an effective mass offset from the centerline of the shaft and acting opposed to said combined mass, the offset distances from the centerline of the shaft to the center of gravity of the combined mass and the center of gravity at the counterweight means mass being selected so that no resultant mass moment is exerted during engine operation.

References Cited by the Examiner

UNITED STATES PATENTS

1,056,746	3/1913	Pitts	123—56
1,090,647	3/1914	Pitts	123—56
1,137,877	5/1915	Manly et al.	74—603 X
1,386,965	8/1921	Simpson	123—74
1,867,981	8/1932	Mudd	74—570 X
1,871,729	8/1932	Newmann	74—598
1,984,454	12/1934	Belyavin	74—598

2,132,595	10/1938	Bancroft	123—56
2,223,100	11/1940	Foster	123—55
2,568,576	9/1951	Witzky et al.	74—603
2,586,621	2/1952	DeLaage	123—74
2,758,897	8/1956	Naab	287—20
2,775,128	12/1956	Young	74—49
2,815,682	12/1957	Kolbe et al.	74—603
2,861,851	12/1958	Young	287—20
2,936,632	5/1960	Palmer	74—49
3,119,462	1/1964	McMahan	123—56

FOREIGN PATENTS

152,704	10/1920	Great Britain.

BROUGHTON G. DURHAM, *Primary Examiner.*

FRED E. ENGELTHALER, MILTON KAUFMAN, *Examiners.*

A. L. SMITH, F. E. BAKER, *Assistant Examiners.*