

[54] ROTARY ENGINE

13,569 5/1929 Australia..... 418/138

[76] Inventor: **Howard R. Chapman**, 20743
Strathern Street, Canoga Park, Calif.
91306

Primary Examiner—Carlton R. Croyle
Assistant Examiner—John J. Vrablik
Attorney—Lowell G. Turner

[22] Filed: **May 24, 1971**

[21] Appl. No.: **146,249**

[52] U.S. Cl..... **418/61**, 418/64, 418/111,
418/138, 418/151

[51] Int. Cl..... **F01c 1/02**, F01c 21/00, F01c 19/00

[58] Field of Search..... 418/61, 111, 138,
418/241, 151, 64; 123/18

[56] **References Cited**

UNITED STATES PATENTS

1,841,841	1/1932	Munn.....	418/61
1,317,056	9/1919	Waagbo.....	418/138
2,724,341	11/1955	Bilas.....	418/111
1,896,256	2/1933	Spencer.....	418/266
1,158,467	11/1915	Evans.....	418/61
1,864,699	6/1932	Varley.....	418/151

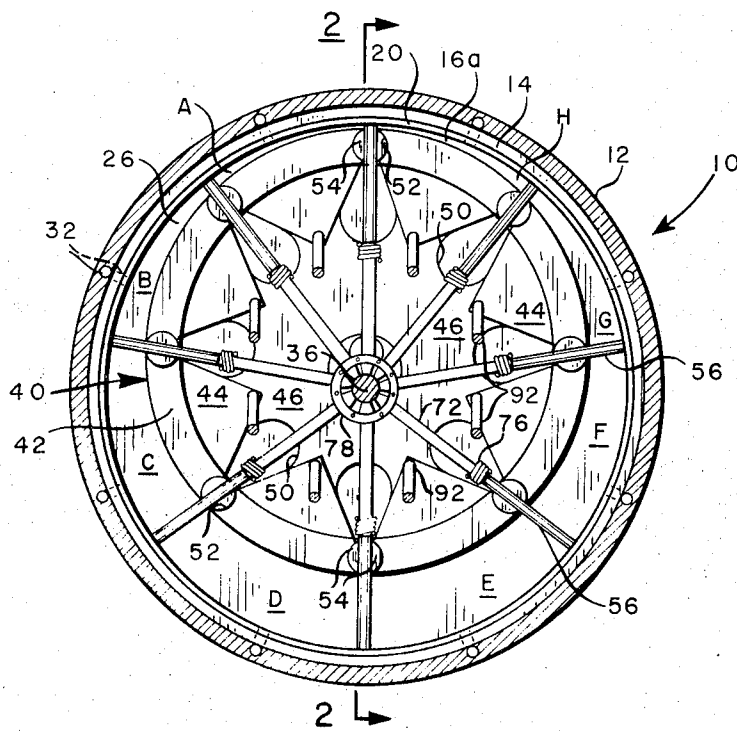
FOREIGN PATENTS OR APPLICATIONS

653,267	11/1937	Germany.....	418/61
---------	---------	--------------	--------

[57] **ABSTRACT**

A rotary engine wherein a stationary cylinder contains an internal rotor mounted on a crank for movement thereupon eccentric to the cylinder and opposed by a counterbalancing flywheel connected to the rotor through an intermediate structure by a plurality of cranks. A plurality of vanes extending radially outward through the rotor engage the cylinder surfaces to provide with the rotor a plurality of sealed gas expansion chambers. Valve means sequentially control the admission of pressurized gases to and the exhaust of expanded gases from the chambers, the gas pressure acting upon the rotor providing the motive power for engine operation. The rotor and the flywheel, as restrained and controlled by the plurality of cranks, move in rotary orbital paths, the vanes reciprocating in a wiping action over the internal surface of the cylinder.

26 Claims, 16 Drawing Figures



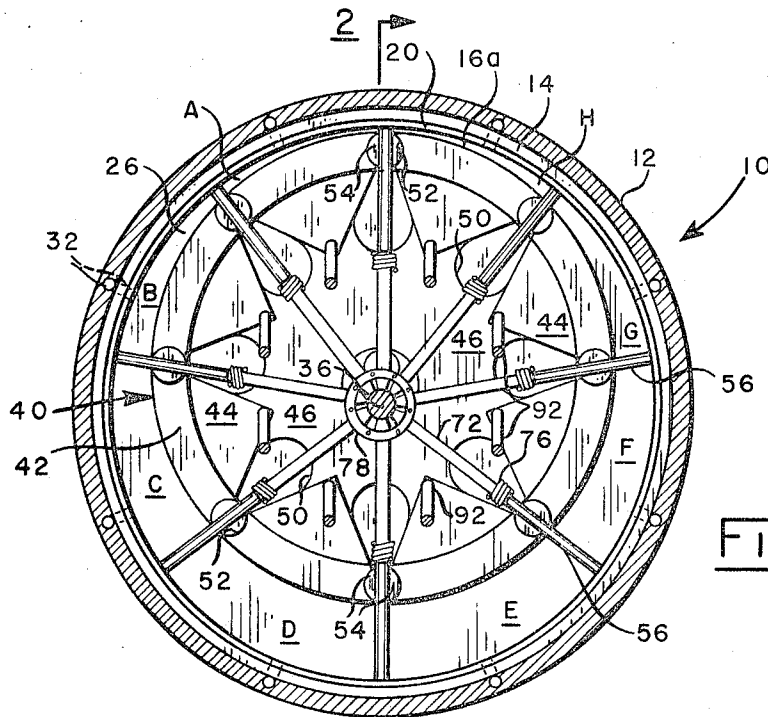


FIG. 1

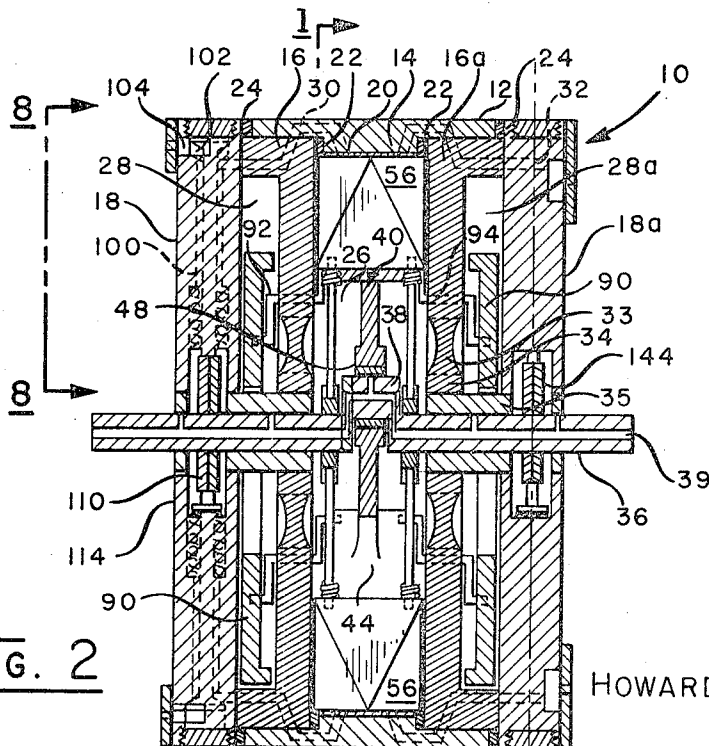


FIG. 2

HOWARD R. CHAPMAN
INVENTOR

Lowell S. Limer
Agent

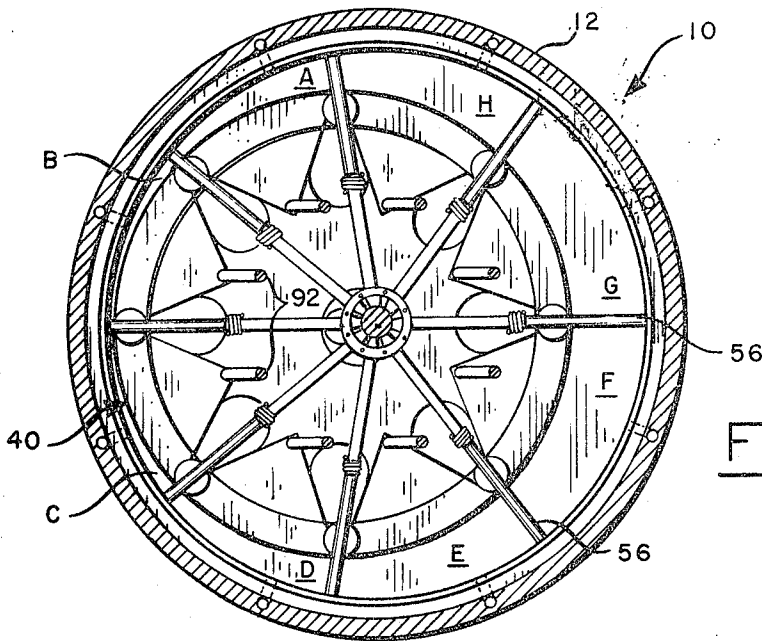


FIG. 3

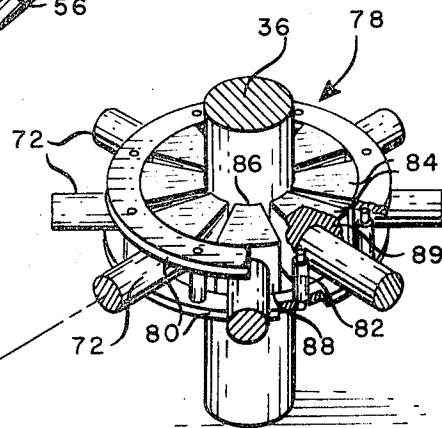


FIG. 5

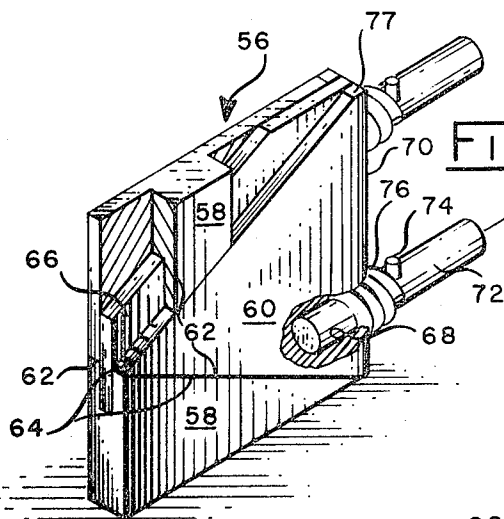
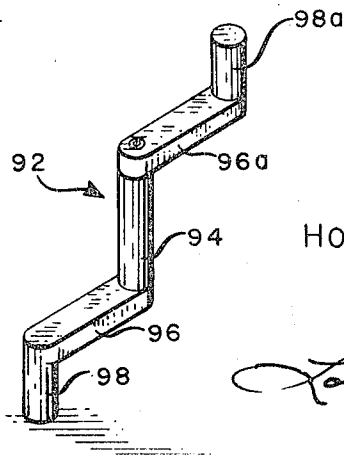


FIG. 4

FIG. 7



HOWARD R. CHAPMAN
INVENTOR

Lowell G. Turner
Agent

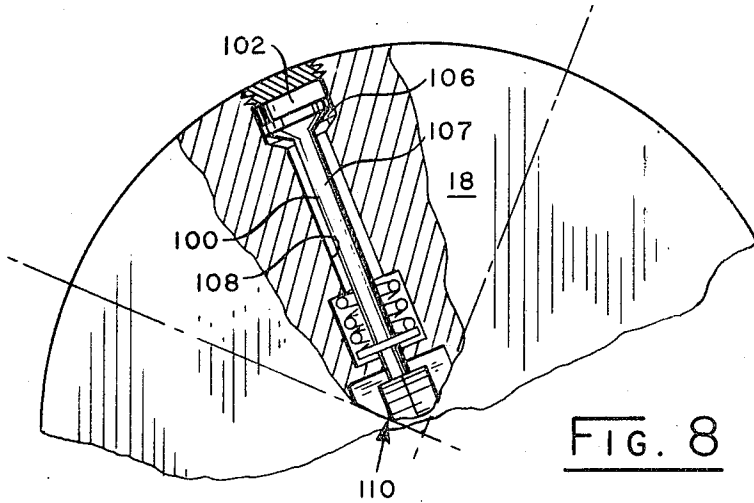


FIG. 8

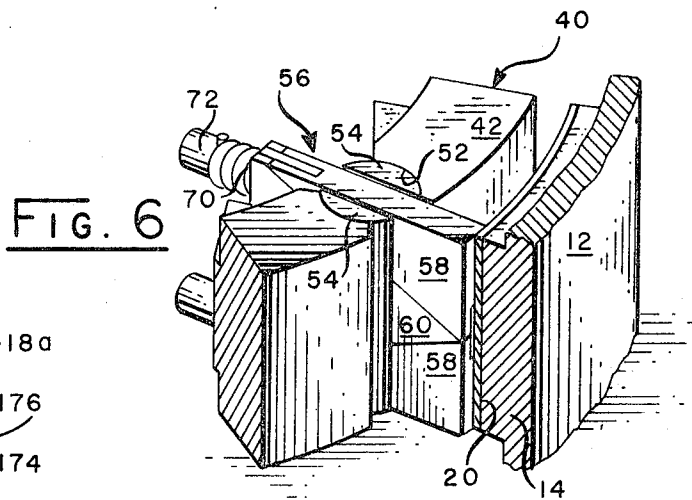


FIG. 6

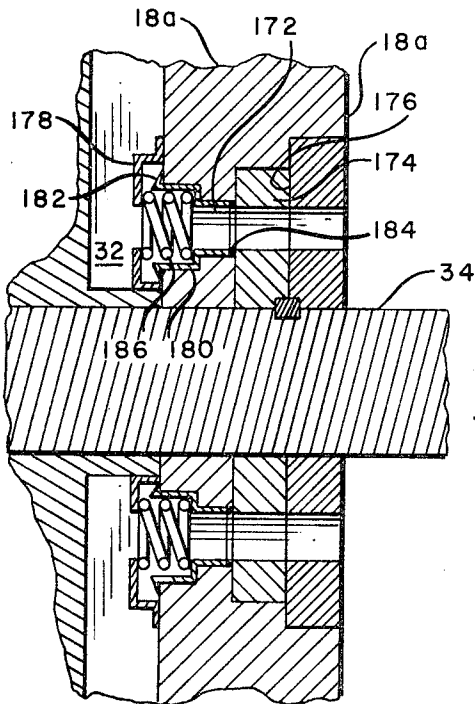


FIG. 14

HOWARD R. CHAPMAN
INVENTOR

Lowell S. Lunn
Agent

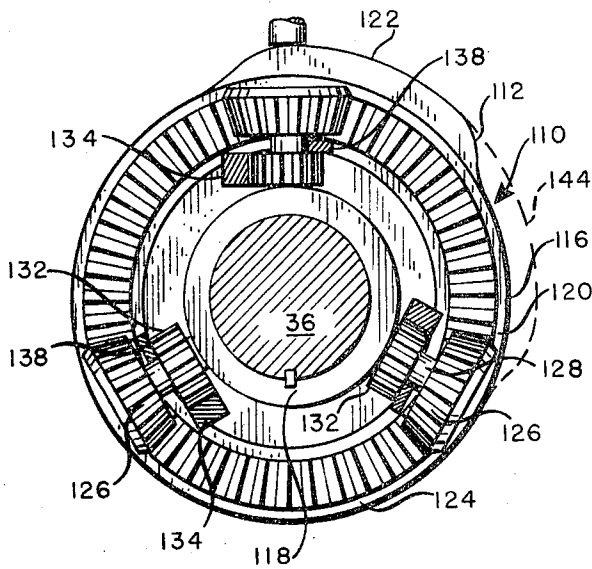


FIG. 10

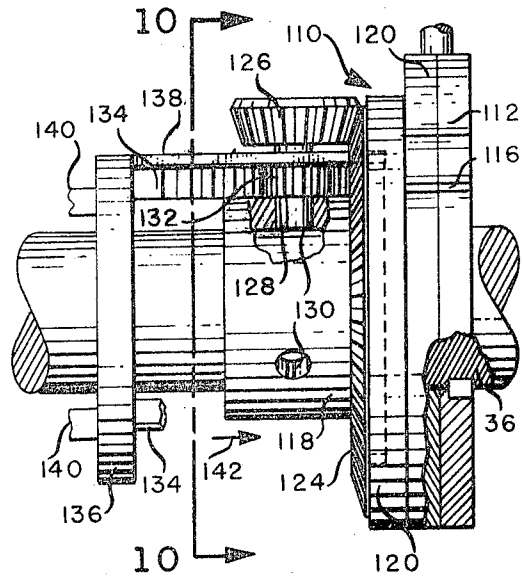


FIG. 9

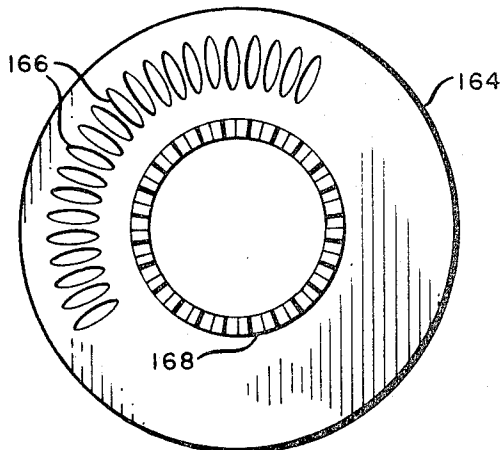


FIG. 13

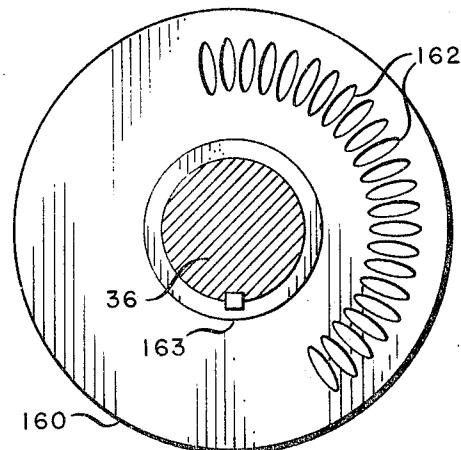


FIG. 12

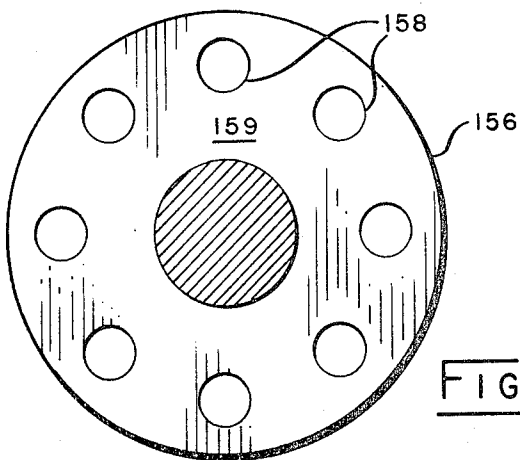


FIG. 11

HOWARD R. CHAPMAN
INVENTOR

Lowell G. Turner
Agent

FIG. 15

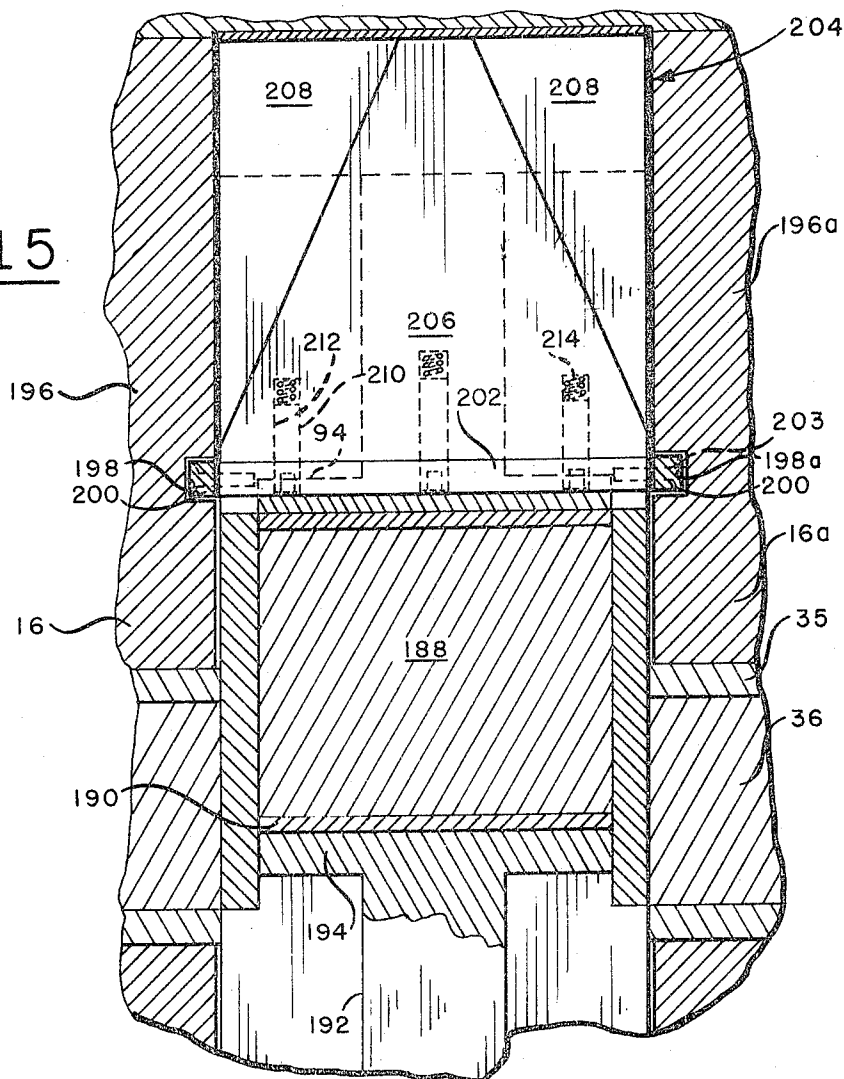
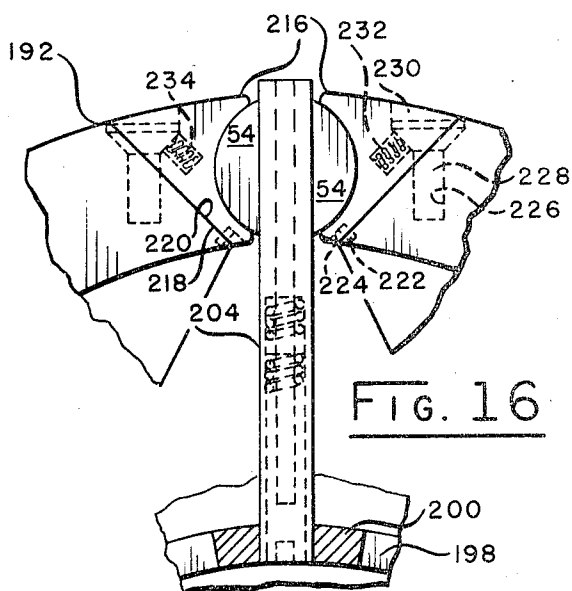


FIG. 16

HOWARD R. CHAPMAN
INVENTOR

Lowell G. Turner
Agent

ROTARY ENGINE

BACKGROUND OF THE INVENTION

Internal combustion engines, particularly those of the reciprocating type have long been a primary source of power. Incidentally, however they have also been a primary contributing factor to the extensive air pollution problems encountered by modern society. Such engines have also suffered from the inevitable effects of mechanical complexity, low power-to-size ratio, and have, in most instances, required the use of relatively expensive fuels for their operation.

Recognizing these basic deficiencies, designers have for many years sought solutions through new approaches in rotary internal combustion engine designs. Progress has been made in several respects. However, until fairly recently no rotary engine has made substantial inroads into the reciprocating engine market; this primarily in view of the inability of such engines to demonstrate technical advances sufficiently significant to capture either the imagination of developers or the financial backing necessary to eventuate in technical and commercial success.

Epitrochoidal configuration engines which are only now approaching acceptance, come closest to this goal by virtue of their reduction in size and complexity, as related to reciprocating engines. Nevertheless, the inability of such advanced engines to reduce the introduction of pollutants into the atmosphere has continued to detract from their general acceptability. The dynamics of the sealing requirements and the severe developmental problems encountered in meeting those requirements have also provided a major deterrent to early acceptance of this engine.

Some rotary engines enclude components having similarity to certain components of my invention. However, in all cases such engines have fallen short of their marks in failing to achieve the breakthroughs requisite to functional acceptability, those novel features incorporated in the presently claimed invention not having been included therein.

While eccentrically mounted rotors with radial vanes have been used in past practice, their 360° rotary actions, together with the resultant necessity for the vanes to continuously wipe vast surface areas, have resulted in low efficiency ratings and short seal life.

An objective of this invention, therefore, is to provide novel means for increasing the efficiency and power output of a rotary engine while decreasing its physical size and mechanical complexity.

Another object to provide a novel rotary power source wherein a minimal movement of mechanical components produces a maximized power output.

A further object is to provide an engine capable of replacing conventional internal combustion engines, while reducing or eliminating pollutants as exhaust products.

Another object is to provide a rotary engine wherein a large rotor area is adapted as a pressurized gas working surface.

Yet another object is to provide a rotary engine mechanism wherein high peripheral speeds are eliminated and lubrication problems are minimized by virtue of encompassing all moving parts within a compact centralized envelope.

SUMMARY OF THE INVENTION

This engine in its simple configuration, includes a fixed, closed ended cylinder, divided into two or more cylindrical compartments by a wall or walls affixed therein. A crank shaft disposed axially through the cylinder includes a crank throw in one of the compartments rotatably supporting a rotor adapted for orbital rotation in another compartment, this eccentric action sequentially bringing each portion of the peripheral rotor surface into proximate spaced relation with a mating portion of the internal cylinder surface.

Attached to and extending radially outward from the crank shaft are a plurality of vanes. These vanes contact and wipe both the internal peripheral surface and the sides of the cylinder. They define with the rotor surface and the cylinder a continuous series of substantially gas-tight compartments in the nature of combustion chambers.

One or more flywheels positioned in compartments adjacent the rotor are connected by a plurality of secondary cranks to the rotor and are adapted to rotate in direct opposition to and to counterbalance the rotor. These cranks also control the nature of the rotary action of the rotor and the flywheel and the articulating wiping strokes of the vanes.

Valves, controlled to function in response to motion of the crank shaft and rotor, sequentially admit pressurized gas to the chambers and exhaust expanded gas from them for powering the engine, substantially one half the peripheral surface of the rotor being pressurized and one half being open to exhaust at all times during operation.

BRIEF DESCRIPTION OF THE DRAWING

FIG. 1 is a sectional plan view of the internal mechanism of the rotary engine of this invention taken essentially along line 1—1 of FIG. 2;

FIG. 2 is a semi-schematic sectional view taken along line 2—2 of FIG. 1;

FIG. 3 is a view similar to FIG. 1 wherein the rotor has moved through a 90° arc;

FIG. 4 is a perspective view of a typical vane cut away to show internal structure;

FIG. 5 is an enlarged perspective view of vane connections to the crank shaft in partial cutaway, to relationship to the FIG. 4 structure being shown;

FIG. 6 is a perspective in cutaway of a vane in its operating relationship to associated structure;

FIG. 7 is a perspective view of a typical secondary crank of this invention;

FIG. 8 is a plan view of the engine taken along line 8—8 of FIG. 2 and cut away to illustrate a typical poppet valve;

FIG. 9 is a side elevational view of the valve cam arrangement partially cut away;

FIG. 10 is a view of the cam taken along line 10—10 of FIG. 9;

FIG. 11 is a plan view of a stator plate for an alternative rotary valve configuration;

FIG. 12 is a plan view of a first metering plate for the rotary valve;

FIG. 13 is a plan view of a second metering plate for the rotary valve;

FIG. 14 is a semi-schematic section view of a rotary exhaust valve;

FIG. 15 is an elevational view shown in semi-schematic section and illustrating an alternative crank throw and vane configuration; and

FIG. 16 is a semi-schematic side view in cutaway section of the vane of FIG. 15.

DETAILED DESCRIPTION OF THE DRAWINGS

The invention is illustrated and described in connection with structural embodiments of a particular character. More specifically, the specification and drawings are directed to preferred configurations, of an engine wherein pressurized gas is utilized as the sole motive power source. As such, it will be apparent to those skilled in the art that certain of the mechanical features described hereinafter may be departed from in certain particulars without departing from either the spirit or the scope of the invention. Moreover, although this engine is described in relation to an engine of the character mentioned, it will additionally become apparent that it is also adaptable for use with certain minimal modifications for application to use as an internal combustion engine, a fluid pump, a fluid motor, or similarly applicable apparatus.

Referring first to FIGS. 1 and 2 of the drawings, the numeral 10 generally refers to the entire engine of the invention. The primary housing means or casing elements of the engine 10 includes a cylinder 12 having a centrally stepped region 14, a pair of spaced walls 16 and 16a abutting the sides of the stepped region 14, and a pair of end walls 18 and 18a, sometimes referred to hereinafter as valve housings. Inasmuch as the inner surfaces of the walls 16 provide wiped cylinder surfaces for engine operation they are sometimes referred to hereinafter as cylinder walls. It is also preferable that the materials from which they and the cylinder 12 are fabricated by a high strength steel or comparable materials. The wiped surfaces should also be ground and honed to a relatively fine surface finish. It is sometimes found preferable to include within the inner or stepped section 14 of the cylinder 12 a sleeve 20 made from a suitably hardened material such as chrome moly steel or a similar metal resistant to high temperatures and pressures and adapted for being continuously wiped by mating metallic elements. The walls 16, 16a, 18, and 18a are normally attached to the cylinder wall 12 by conventional bolting techniques (not shown). Gaskets 22 and 24 are usually provided as pressure seals between the walls 16, 16a and the cylinder 12 and between the walls 18, 18a and the cylinder 12, respectively.

A central compartment 26 and two side compartments 28 and 28a are defined between the structural elements described. The central compartment 26 is usually referred to as the rotor compartment, whereas the two outside compartments 28 and 28a are referred to as the flywheel compartments. Included within the cylinder wall 12 and end wall 18 are a plurality of inlet passages 30, the one shown being illustrative only of the existence of one such passage leading to each expansion chamber described below. Similarly, a plurality of outlet passages 32 are defined within the opposite side of the cylinder wall 12 and the end wall 18a. Each of these passages opens through the cylinder sleeve 20 into the central compartment 26.

The walls 16 and 16a may be of solid construction or, as illustrated, may include webbed sections 33 support-

ing a central boss 34 which encircles a central bearing member 35 contained therein.

Coaxial with the cylinder 12 is a crank means or crankshaft 36 extending through the entire engine housing and outward from either side thereof for appropriate connection to a unit to be powered. Intermediate of the crankshaft ends is an eccentric crank throw portion 38. The crankshaft and throw contain a series of internal passages 39 for supplying oil to the bearings.

Disposed within the rotor compartment 26 is a rotor of circular configuration designated by the numeral 40. The rotor includes an outer rim portion 42 and an internal web portion comprising eight generally pie-shaped segments 44 internally joined with the balance of the web portion 46. Coaxial with the main portion of the rotor 40 is a boss 48 suitably journaled upon the crank throw 38. Thus, rotation of the crankshaft 36 results in the rotor 40 being driven in an eccentric or orbital rotational pattern within and with respect to the rotor compartment 26. It will be noted that the additive radii of the crank arm and the rotor are of lesser total length than the internal radius of the rotor compartment 26. Hence, the outer periphery of the rotor at no time contacts the inner periphery of the cylinder.

Included within the body of the rotor 40 are a series of essentially radially oriented reliefs or cutouts having interior regions 50 of essentially teardrop shape, these regions with cylindrical-shaped holes 52 through the rotor rim 42 to define the complete cutouts. Within each of the cylindrical holes 52 are disposed a pair of pivot seals 54 in centrally spaced relation from one another.

Extending radially outward from the crank shaft are a plurality of vanes generally designated by the numeral 56 (FIGS. 4 and 6). Each vane, in its preferred embodiment, extends radially outward and is positioned between the pivot seal halves 54, the sides of the vane contacting the pivot seal halves in a mutually sealing relationship. The outer extremity of the vanes engage the inner surface of the cylinder sleeve 20 in a firmly sealing and reciprocal wiping engagement. The side surfaces similarly wipe the cylinder walls 16 and 16a.

These vanes 56, which are usually eight in number, include a pair of outer, marginal, or wall-wiping segments 58 and a central, inner or camming segment 60. Each of the outer segments 58 includes an inner bifurcated edge 62. The outer surfaces of the inner segment 60 include edges 64 adapted to act as camming surfaces against the inner edge of the segments 58. Each of the edges 64 includes a tang 66 extending outward between the bifurcations of the outer segment edge 62. This interlocking construction contributes structural integrity to the respective segment portions and facilitates movement of the segments relative to one another for the automatic adjustment purposes to be later described.

Included within the inner edge of each of the segments 60 is a pair of holes 68 within which a pair of shafts or vane support rods 72 are inserted. Compressed and retained upon the rods 72 by pins 74 are compression springs or vane actuators 76 bearing against an edge 70 and urging the segment 60 radially outward. This segment is thereby caused to forcibly engage the cylinder wall 12 and to cam the segment 58 radially and axially outward into engagement with the cylinder walls 12, 16, and 16a in a firm wiping contact. As the vane segments wipe the cylinder walls and vane

edge wear occurs, the camming action of the vane segments tends to maintain an effective seal at all times by continuously and automatically adjusting the positions of the vane segments to correspond to the shape of the walls. This continuous honeing and adjusting process results in an improvement in sealing integrity throughout the operational life of the engine.

Leakage of pressure via the interlocked vane segment edges from the pressure chambers defined between the vanes is prevented by the termination of the interlocking surfaces at a position on the side of the segment 60, as indicated at numeral 77, such that it engages the sealing surface of the chamber wall.

Each two adjacent vanes 56 cooperates with the walls 12, 16 and 16a and the external periphery of the rotor 40 to define a gas expansion chamber. These chambers, eight in number in the presently described embodiment, are identified in FIGS. 1 and 3 as chambers A through H, for purposes of description.

Each of the vanes 56 is free to rotate with the rotor 40 by virtue of the described mounting in the pivot seals 54. This freedom is further facilitated, while simultaneously assuring a radially contacting relationship with respect to the crankshaft 36 by a pair of restrainer assemblies 78. One such assembly is mounted upon each side of the rotor 40 and supports one rod 72 from each vane 56.

Each restrainer assembly 78 (see FIG. 5) includes a pair of restrainer rings 80 retained in spaced relation by a plurality of pins 82, and is in coaxial relationship with the crankshaft 36. Positioned between the restrainer rings 80 and the crankshaft 36 are a plurality of rod adapters 84, each having an arcuate internal surface 86 and an arcuate external surface 88 respectively mating with the surfaces of the crankshaft 36 and the restrainer rings 80. Each rod adapter includes a cavity 89 within which the end of a rod 72 remote from its vane is nested.

Hence, as the respective vanes travel with the rotor 40 and pivot within the pivot halves 54 they also rock independently of one another about the axis of the crankshaft. Throughout this movement they are controlled and prevented from moving radially by the rod restrainer assemblies 78.

The pivot seals 54, together with the rods 72 and the restrainer assemblies 78, are sometimes referred to as vane control means.

Situated within each of the compartments 28 and 28a is a flywheel 90, these flywheels having a combined weight exactly matching that of the rotor assembly. The flywheels 90 may be of any convenient configuration so long as they provide the weight counterbalancing relationship requisite to a smooth engine operation.

The flywheels 90 are connected to the rotor assembly 40 through a plurality of secondary cranks 92. Each crank 92, as best shown in FIGS. 2 and 6, includes a central pin 94 retained through the respective walls 16 and 16a for rotation therein. A pair of opposed arms 96 and 96a have axially oriented pin members 98 and 98a extending respectively into appropriately provided holes in the flywheels 90 and in the pie-shaped segments 44 of the rotor 40 (FIG. 2). The radius of the throw of each of the arms 96 and 96a is identical to the radius of the crankshaft throw 38.

It will be noted, since the secondary cranks 92 are secured through the stationary walls 16 and 16a and additionally engage the rotor and flywheels, that both the

rotor and the flywheels are thereby prevented from moving in a standard 360° rotary pattern around the crank 38. Rather, they are constrained to move in a pattern which may be described as rotary orbital, or which is sometimes referred to as a "wobulation" pattern. During any cycle of operation, each point upon the rotor describes a circular pattern having a radius equal to that of the cranks 38 and 92.

Since the secondary cranks 92 not only control the movement of the rotor, but also accept a portion of the force (principally the centrifugal forces) transmitted by the expanding gases into the rotor, it is necessary that they be fabricated from a material capable of withstanding high tension forces. Cold-rolled steel has been found to be acceptable for this purpose.

The valve housing 18 and 18a each includes a plurality of inlet and outlet valves for controlling the inlet of pressurized gases to the expansion chambers A through H and the exhaust from those chambers of spent gases. Since eight chambers are provided in the usual case, it is necessary that a comparable number of inlet and exhaust valves be provided. It is also usual that all inlet and exhaust valves are of identical configuration. Therefore, only one valve will be described in detail.

A typical valve is identified in FIGS. 2 and 8 by the numeral 100. Being of a conventional poppet configuration, the valve 100 is located within the housing 18, for example, such that its head is positioned within a cavity 102 appropriately provided in the housing and connected to a manifold 104. This cavity includes a nesting surface 106 configured to receive the underside of a head in mating relation. The inlet passage 30 which leads into the expansion chamber is closed by the valve which in a conventional manner prevents the passage of gas into the expansion chamber via the passage 30. The valve 100 has a stem 107 disposed within a drill passage 108 and is conveniently spring-actuated to the normally closed position. This spring actuation also maintains the extremity of the valve stem in forceable sliding engagement with a split cam 110.

The split cam 110, more specifically illustrated in FIGS. 9 and 10, includes a cam lobe 112 and is mounted upon the crankshaft 36 and within a cam housing portion 114 (FIG. 1) of the wall 18. A first cam plate 116 includes an axially extending boss portion 118 and is conventionally keyed to the shaft 36 for rotation therewith. The cam surface 112 of plate 116 is of substantially the configuration shown in FIG. 10, or of such other configurations as may be appropriate in view of the number of valves to be actuated thereby and their relative position in the radial pattern. In the present instance the flat portion of the cam lobe 112 approximates a 45° coverage of the cam surface. Thus, as one valve 100 closes responsive to cam actuation, the next adjacent valve is opened by the ascending surface of the cam. Mounted upon the cam boss 118 is a relatively rotatable and abutting relationship with the first cam plate 116 is a second cam plate 120 having a cam lobe 122 matching the cam lobe 112 and normally positioned such that its outline coincides exactly with that of the first cam plate 116, substantially as illustrated.

Fixed to and extending outward a convenient distance from the face of the second cam plate 120 in a direction opposite the first cam plate 116, is a sun gear 124 providing a portion of a cam actuating means to be described. Engaged with the sun gear 124 are a plural-

ity of bevel gears 126. Each gear 126 is mounted for rotation upon a shaft 128 retained in a hole 130 in the boss 118 and normal to the axis of the shaft 34. Each shaft 128 also carried a spur gear 132, the teeth of which engage those of a rack 134 extending outward in finger-like relation from a force ring 136. This ring is mounted for axially sliding along the shaft 134 and is keyed for rotation therewith. The force ring 136 also includes a guide 138 extending outward therefrom substantially parallel to each of the racks 134. The force ring 136 is connected to the accelerator in the system of the engine of this invention via the attachments 140 in any convenient manner, such accelerator and attachments forming no part of this invention. Opening the accelerator will ultimately cause a force to be exerted against the force ring, causing it to move in the direction of arrow 142. This moves the racks 134, which turn the spur gears 132 and the bevel gears 126. The bevel gears, being engaged with the sun gear 124, cause rotation of the sun gear 124 and its integral cam plate 120, thereby moving the cam lobe 122 radially with respect to its mating cam 118 and extending the length of the cam surface available for valve actuation, generally as indicated by the dotted lines 144 of FIG. 10. This permits the inlet valves to remain open for a more extended portion of the cycle and facilitates the opening of more than one valve at a time, thereby providing for the entrance into the pressure chambers of a larger volume of gas than would otherwise be introduced. This results in the generation of a greater accelerating force by the engine.

The exhaust valves (FIG. 1) function in essentially the same manner as do the inlet valves, except that a single cam plate 144 is provided. This plate is also mounted on the crank shaft 34 to rotate therewith and is of a configuration similar to that illustrated with respect to the cam plate 114, except that a cam lobe (not shown) is provided over essentially 180° of the cam plate periphery, thereby assuring that one half of the exhaust valves will remain open at all times.

An alternative valving arrangement is provided by the structure shown in FIGS. 11-13. The generalized configurations of the inlet and outlet valves, are indicated by the illustrations. Therein the inlet valve components are positioned within a valve housing substantially similar to that described above relative to cam housing 114. These components are positioned coaxially upon the crankshaft 34 such that their ports (to be described) lead into gas inlet passages through the wall 18 to connect with the passage 30 (FIG. 1) entering into the compression chamber.

A stator plate 156 is mounted in the wall 18 such that a series of ports 158 therein exactly match the noted inlet passages leading to the passages 30, thereby providing a stationary member through which pressurized gases may enter those passages. The stator plate 156 incorporates a smooth surface 159 against which a metering plate 160, which is keyed to the rotor shaft 34 for rotation therewith, is adapted to rotate. It is desirable that the contacting or faying surfaces of the plates 156 and 160 be as smooth and frictionless as reasonably possible in order that their relative rotation will not be destructive of the structures of their sealing relationship.

A first metering plate 160 includes a series of elongated ports or holes 162 therethrough. These discretely oriented, but closely spaced ports are described herein

as being elliptical in shape. It is to be understood, however, that they may be rectangular or other shape convenient to the application. These ports generally encompass a predetermined circumferential portion of the plate. This circumferential distance is adapted to encompass in matching relationship four of the inlet ports 158 in the stator plate. The holes 162 are so shaped and spaced that by positioning the metering plate 160 coaxial with and in rotational sliding relationship to the stator plate 156, at least two of the elliptical ports 162 open into each port 158 when those ports are encompassed. This provides an opening for the passage for gases through both plates. The metering plate 160 includes a boss 163 comparable in nature and purpose to the cam plate boss 118 described above.

Positioned upon the boss 163 is juxtaposition against the metering plate 160 is a second metering plate 164. This plate is adapted for rotation with and with respect to the metering plate 160. It also incorporates a plurality of elliptical ports 166 in the same pattern as the elliptical ports 162 on the metering plate 160. the second metering plate 164 also incorporates a gear 168 upon its surface in essentially the same manner and for the same purpose as the cam plate 122 of FIGS. 9 and 10. Not shown in FIGS. 11 through 13, but assumed present are the rack and pinion actuating mechanism described with respect to FIGS. 9 and 10. This mechanism is provided to rotate the metering plate 164 with respect to the metering plate 160, thereby accommodating the matching or mismatching of the elliptical ports. This relative movement of the elliptical ports in the two metering plates results in their alignment or misalignment, providing for the passage of pressurized gas through both ports and additionally through the ports 158 in the stator plate 156, but only to the limited extent that all such ports are aligned. For example, at such time as the plate 164 has approximately six of its elliptical holes aligned with a similar number of holes upon plate 160, a single one of the ports 158 will be aligned with at least two elliptical ports on each metering plate, thereby providing a through-passage for the pressurized gas. Were the plate 164 to be rotated upon the boss 163 by the gear mechanism heretofore described, an additional number of elliptical holes 166 in the plate 164 would be rotated into alignment with those holes 162 in the plate 160, thereby providing additional passageways for the gas through both metering plates and into an additional number of the ports 158. By matching all elliptical ports 166 with all the ports 162 a maximum of four ports 158 in the stator plate 156 can be uncovered, thereby providing a maximum entrance of pressurized gases and a maximum acceleration of the engine. A retraction of the foot pedal, in the acceleration mechanism will obviously result in a reverse rotation of the metering plate 164, thereby again causing the heretofore uncovered ports 158 to be covered, with a resulting deceleration of the engine.

The outlet valve of the rotary exhaust system is illustrated in FIG. 14. The rotor in this arrangement is provided to assure that pressurized gas acting against it will not deflect and deform the plate outward. Additionally, it is necessary that a seal means be provided to obviate the possibility of exhaust gases escaping around the ports when they are intended to be closed. The hereinafter described structure is provided for these purposes.

The stationary engine structure illustrated in enlarged section in FIG. 14 is the valve housing 18a of FIG. 2. Each port 32 leading from a particular pressure chamber also leads to a sealed outlet port 172. Keyed to the shaft 34 for rotation therewith is an exhaust metering plate 174 provided with a series of elongated or elliptical ports sufficient in number to encompass four of the sealed outlet ports 172, i.e., encompassing a circumferential distance upon the plate 174 of approximately 180°.

A plan view of the exhaust metering plate 174 is not illustrated since the elliptical port configuration is otherwise identical to that of the plates 160 and 164. The external marginal surface of the metering plate 174 is positioned against a shoulder 176 of the valve housing 18a such that pressure on the left-hand or pressurized side of the plate tends to force the plate into engagement with that shoulder, thereby providing a gas seal and a resistance to deflection or deformation of the plate. Appropriate anti-friction means is sometimes provided in this faying surface region to obviate friction problems.

Within the sealed ports 172 are positioned sealing means comprising a retainer 178 appropriately fastened to the structure of wall 18a. Within the recessed port 172 is a seal housing 180 having a pressure-deforming seal 182 on its outer periphery and a conventional carbon seal 184 or similar structure in rubbing contact with the face of the plate 174. When desirable, a compression spring 186 may be provided to bear against the interior of the retainer 178 and against an internal shoulder of the insert 180 to maintain the insert in a positively sealing relationship against metering plate 174.

Inasmuch as the engine of this invention has the inherent ability of developing a horsepower rating which is very large in relation to engine size, it is desirable that the bearing upon the main crank throw be as large as possible commensurate with good design principles. This assures that forces applied thereto through the rotor may be accepted without destructive effect upon the bearing. Therefore, it is sometimes preferable that an alternative rotor and vane configuration in the nature of that hereinafter described be provided so as to accommodate a larger bearing area. It is also desirable that means be provided to assure that an effective seal always be maintained between the vanes and the pivot seal through which the vanes travel. The alternative configurations illustrated in FIGS. 15 and 16 provide means for accomplishing these purposes.

Referring, therefore, to FIGS. 15 and 16, the crankshaft 36 is provided with a throw 188 of substantially greater diameter than the throw 38 in the initially described configuration. In the present instance it is also of greater diameter than the balance of the crankshaft 36. It is surrounded by a journal bearing 190. The rotor 192 includes a boss 194 journaled to ride upon and drive the crankshaft 36 through the bearing 190, the large bearing area thereby provided being effective to facilitate the required power transfer. The rotor 192 is otherwise similar in all major respects to the rotor 40 described with respect to FIGS. 1 and 2. A pair of side-walls 196 and 196a, comparable to the heretofore described walls 16 and 16a, respectively include annular grooves 198 and 198a in their inner surfaces. These grooves are coaxially oriented with respect to the crankshaft 36 and are of slightly greater radius than the

maximum radius of the crank throw 188, as related to its distance from the primary axis of the crankshaft 36.

Positioned for slidable movement within the annular grooves 198 and 198a are a plurality of vane control shoes 200. One such shoe is provided within each groove 198 and 198a for each of the vanes utilized in the engine. Fixed to and extending between the two opposed control shoes is a vane support means in the form of a beam 202 affixed by the servos 203 and adapted for movement with the control shoes. This control shoe and support beam combination is also adapted to support and guide the vanes 204 in their translational movement. Thus, the function of the control shoe/support beam combination is similar to that of the vane rods 72 and the restrainer assemblies 78, i.e., to facilitate a radial reciprocation or translation of the vanes in their action within the rotor 192.

The vanes 204 include a plurality of vane segments 206 and 208, these segments being basically comparable to their counterpart vane segments 58 and 60 described above. The central vane segment 206 is provided with lateral support by a series of pins 210 extending into unthreaded holes 212 therein and threadedly retained through the support beam 202 to require simultaneous movement of the vane when the support beam is moved. A compressed spring 214 is positioned in each hole 212 to bear against the ends of the pins 210 and the beam 202, thereby exerting a continuous force radially outward upon the center vane segment 206 to facilitate the action heretofore described with respect to the vane segment 60 of FIGS. 1 and 2.

Each vane 204, as best illustrated in FIG. 16, extends outward through the pivot halves 54 which are identical to those heretofore described. Of additional significance, however, are the seal actuator means which comprise left and right-hand pivot half sealing shoes 216 and their related structure. Each such sealing shoe includes a surface 218 oriented at an acute angle from the radially direction of its adjacent vane 204. The surface 218 mates with and is slideable with respect to a similar mating surface 220 upon the rim of the rotor 192. Inserted into the sealing shoe 216 upon the surface 218 is a retention and guide means, including a pin 222, an outwardly extending portion of which is inserted within and moveable with respect to a slot 224 provided within the rotor mating surface 220. Also providing a portion of the retention and guide means, included within the rim of the rotor 192 and extending generally inward from the surface 220, is a threaded hole 226 within which a screw 228 is engaged. Positioned in matching relationship with the screw hole 226 is a slot 230 within the sealing shoe 216, this slot opening upon the surface 218. The head of the screw 228 extends into the slot 230 such that the sealing shoe 216 is free to slide with respect to the rotor rim surface 220 in a relationship which is controlled and guided by the pin 222 in the slot 224 and by the screwhead 228 in the slot 230. Extending inward from the slot 230 substantially parallel to the surfaces 218 and 220 is a hole 232. A spring 234 is compressed therein and bears against the underside of the screwhead 228 such that a constant force is exerted against the screwhead. This configuration provides for an assist from the pressurized gas. This tends to move the sealing shoes 216 downward, the angular relationship of the surfaces 218 and 220 acting as cams and causing the shoes 216 to move in a wedging action against the pivot half 54. This results in

the production of a continuous and forceable sealing relationship between the shoe 216 and the pivot half 54 and between the pivot half 54 and the vane 204 which it contacts. It will also be reorganized that gas pressure acting upon the surfaces of the sealing shoes within the pressure chambers also acts to further accentuate this cam action and to apply a sealing force.

By virtue of the structures immediately heretofore described the pressure contained within the pressure chambers is effectively maintained so as to efficiently and effectively prevent its loss therefrom. The intent is also to illustrate the basic principles of the invention, recognizing that structural simplifications in the configuration can be achieved within the broad concepts presented.

An operational cycle of this engine is completed in the following manner. Although this cycle is discussed in terms of the configuration of FIGS. 1 - 14, it will be understood that it relates equally to that of FIGS. 15 and 16. Referring to FIG. 1, it will be noted that the rotor 40 is in its uppermost position. Its counterclockwise rotation through a few degrees will result in chamber A being at its minimum volumetric position. Again, at no time does the outer periphery of the rotor 40 contact the inner periphery of the cylinder 12, a space always being defined therebetween. At this time a minimum acute angle of approximately 45° is defined between the vanes 56. Simultaneously, the vanes upon either side of the chamber E on the opposite side of the rotor will be at their maximum acute angle, approximating 55°. Thus, chamber A will be at its minimum volume both from the standpoint of rotor-to-cylinder distance and the distance between the chamber-enclosing vanes, whereas, the chamber E will be at its maximum volume from both standpoints.

With the rotor 40 in this described position, i.e., approximately the 11:00 o'clock position, the inlet and exhaust valves associated with chambers A, F, G and H are closed and the chambers are pressurized, thereby facilitating the output of work by virtue of the expansion of pressurized gases within those chambers.

The exhaust valves of chambers B, C, D, and E, as actuated by the cam 144, are open at this time, expanded gases being forced therefrom by the action of the rotor in its movement, continuously reducing the volume of those chambers. The gas inlet valve 100 located within the channel leading to the port 30, and which opens into chamber A, is caused to open by the rotation of the cam 122 under the valve stem. Pressurized gas from an independent source (not shown) is thereby permitted to enter through the respective ports leading to chamber A, pressurizing that chamber. The pressurized gases reacting against the rotor surfaces in each of the pressurized chambers A, F, G, and H act to move the rotor in a counterclockwise direction, the force applied from each of these chambers having a resultant force component acting through the rotor axis.

Thus, an important objective of the invention, that of increasing engine efficiency, is obtained by providing a structure wherein torque-generating forces are applied over substantially a full 180° of the rotor periphery. This ability to maintain pressurization over a complete half of the rotor surface is a primary factor contributing to the great improvement in efficiency of this engine and a significant increase in torque and indicated horsepower.

Another feature of significance is the motor movement pattern. Since the rotor is mounted upon its crank throw 38 for relative rotational movement thereupon; and since it is nevertheless restrained from 360° rotation about the throw 38 by virtue of the eight secondary cranks 92, the rotor is forced to move in a peculiar pattern. The rotor, in actuality, moves with and is supported by both the primary and the secondary cranks, the secondary cranks providing the means for control of the ultimate movement pattern. As indicated previously, each point upon the rotor during one complete cycle of operation describes a 360° circle independent from every other point. The result is that the rotation of the crank throw 38 and the crankshaft 34 are caused and controlled by this rotor movement, the throw 38 performing a 360° rotation within the rotor 40 and providing the crankshaft with its rotational movement for power output. Hence, the rotational action of the rotor 40 is one, as heretofore-mentioned, of orbital rotation or "wobulation."

The completion of a rotational cycle of the engine after pressurization of chamber A is provided by sequentially closing the exhaust valves in chambers B, C, D, etc., while opening the inlet ports in the same sequential order. The inlet ports are subsequently closed as pressurization is completed. Thus, pressurization is accommodated for each of the chambers in its rotational sequence. As each such chamber is pressurized, its opposite counterpart is exhausted. For example, chamber F is exhausted as chamber B is pressurized. Thus, four chambers participate in the pressure actuation cycle at all times, the other four chambers being exhausted simultaneously, thereby facilitating a maximized efficiency of torque generation and engine operation. Again it will be noted that four chambers encompass one-half the rotor surface area, obtaining for this engine the full measure of its potential insofar as utilization of available surface area for rotor actuation is concerned.

An indication of the relative movement of the system components during operation is provided by the FIG. 3 showing wherein the rotor 40 has moved 90° from the FIG. 1 position, thereby restricting the volumes of chambers B, C, D, and E while increasing the volumes of chambers F, G, H, and A. The relative angular positions of the respective vanes have shifted in the manner described above to accommodate contractions and expansions of the chambers.

Throughout this sequence of operations the continuously shifting weight of the rotor 40 is exactly balanced by the flywheels 90, the secondary cranks 92 providing the connecting and controlling links.

It is also important to recognize that the rotational or cylinder-wiping travel of each of the vanes is restricted by the action of the cranks 92. After the vanes have reached their maximum travel in one direction, as permitted by the restraining action of the cranks 92, they reverse and travel a comparable distance in the opposite direction, until again stopped by the cranks. The action of each of the vanes is thereby controlled in a reciprocal wiping relationship with respect to the internal peripheral surface of the cylinder wall 12, as well as the side walls defined by the interior surfaces of the wall 16 and 16a. This freedom of vane movement is readily facilitated by the actions of the seal halves 54, the angular shape of the rotor segments 44, and the teardrop shape of the cutouts 50, which permit the rotational

movement of the lower sections of the vanes 56 without structural interference.

Thus, during one full revolution or engine cycle a vane traversing the full 360° of rotation in a conventional rotary engine having an 11 inch cylinder diameter would travel approximately 35 inches. However, by restricting its motion to the reciprocal wiping action described relative to the present invention, its travel is limited to 5 inches. This reduction to 1/7 the distance, or by nearly 85 percent, results in a clear and significant advantage in engine wear. This becomes especially apparent during high speed operation.

Maximum torque is generated in that chamber wherein the combination of the maximum gas pressure and a force component closest to an angle normal to crank throw 38 are combined. Hence, in FIG. 1 the representative force percentages, calculated on the basis that driving forces totaling 100 percent of available power (torque output in indicated horsepower) are spread over 180° of the rotor surface. The following chart provides representative data showing the torque percentages produced in each of the chambers E, F, G, and H in the FIG. 1 position, assuming the introduction of 500 in lbs. of gas and varying the length of time the inlet valves are held open, thereby varying the cubic inches of chamber area pressurized.

ENGINE PERFORMANCE

Pressures		Initial volume pressurized, cu. in.	Percent torque contributed				Total torque produced, in. lbs.
Inlet, p.s.i.	Exhaust, p.s.i.		Comp. E	Comp. F	Comp. G	Comp. H	
500	24	1.1	59	26.0	12	3	3,367
500	88	4.0	28.1	46.7	20.3	4.9	7,110
500	143	6.5	20.5	50.0	23.8	5.7	9,785
500	220	10.0	17.5	43.1	31.6	7.8	11,345

The figures of this chart are based on the assumption that the temperature remains constant. However as the gas expands the temperature will drop, as will the pressure. For this reason the exhaust pressures will be somewhat lower than those shown. This chart does, however, indicate in general terms the torque and expansion ratios obtainable.

It is also of interest that with the rotor in the position illustrated in FIG. 1, assuming a cylinder I.D. of 11 inches and rotor dimensions of 9 inches in diameter and 3 inches in width, chamber volumes are approximately as follows:

- H = 1.1 cu. in.
- G = 6.1 cu. in.
- F = 13.6 cu. in.
- E = 22.8 cu. in.

If the gas inlet valve in chamber H is closed at this time an expansion ratio in excess of 20 to 1 is achieved.

The component of force generated within each such chamber passes through the primary axis of the rotor 40, each providing a component of force tending to turn the rotor counterclockwise. The magnitude of the force actually contributed by each of the chambers to the resultant output is a function of the chamber pressure and the angular direction of the force relative to crank arm position. Maximum torque output is achieved when the pressure is at its maximum value and the force through the rotor axis is at 90° from the torque arm. Hence, as chamber volume increases responsive to rotor rotation, chamber pressure decreases. Simultaneously, the force angle relative to the crank arm, being about 70° from normal to the crank arm at the time of maximum chamber pressurization, de-

creases during rotor movement until it is normal to crank arm, then again increasing in the opposite direction to approximately 70° from a direction normal to the crank arm until the chamber exhaust cycle is initiated. An integrating calculation between these extreme positions will yield the precise values of torque output for a rotational cycle.

A typical torque value calculation for a representative engine of this invention is developed by the equation:

$$T = PDWX$$

Then, where T = torque

P = average mean pressure of the four pressurized compartments = 200 psi.

D = rotor diameter = 9 inches

W = rotor width = 3 inches, and

X = crank arm length = 1.25 inches

$$T = 200 \times 9 \times 3 \times 1.25 = 6750 \text{ in. lbs.} = 562.5 \text{ ft. lbs.}$$

Now, when the engine rotates at 200 RPM at the above calculated torque, the indicated horsepower (hp) developed is as follows:

$$\begin{aligned} \text{hp} &= [(2\pi) (\text{RPM}) (T \text{ in. lbs.})] / [33,000 \times 12] \\ &= (6.28 \times 2000 \times 6750) / 33,000 \times 12 \\ &= 42370 / 198 = 208.84 \text{ hp} \end{aligned}$$

When RPM is increased to 3000 the indicated horsepower increases to 320.

The external dimensions of this engine (without accessories) would be approximately as follows:

Diameter of outer cylinder = 12 in.

Length = 12 in.

Its weight would approximate 150 lbs.

From the foregoing it will be recognized that an engine with unique structural characteristics is encompassed and that the features described above and claimed hereinafter additionally provide to this engine a unique capability, that of producing horsepower-to-weight and horsepower-to-size ratios greatly increased over those of prior art engines, of both reciprocating and rotary configurations.

I claim:

1. A rotary drive assembly comprising:

a housing fixed against rotation and having internal cylinder surfaces;

a crankshaft positioned coaxially through said housing for rotation therein and having an offset throw portion;

a rotor positioned upon said offset throw portion for movement relative thereto;

flywheel means in said housing positioned to oppose any force produced by said rotor;

control means restraining movement of said rotor and said flywheel means to a rotary orbital pattern; vanes extending radially outward through said rotor into constant wiping engagement with said internal cylinder surfaces and defining chambers with said

internal cylinder surfaces and said rotor; and valve means for sequentially admitting propulsion gases to said chambers.

2. A rotary engine comprising:

a crankshaft having an offset crank portion intermediate the ends thereof;

a rotor mounted upon said offset crank portion concentric therewith and relatively rotatable thereupon;

stationary housing means concentric with respect to and supporting said crankshaft and enclosing said rotor,

said housing means including wall means separating said housing means into compartments having internal surfaces;

a flywheel positioned within a said compartment and separated from said rotor by said wall means;

crank means rotatably mounted in said wall means and engaging and controlling movement of said rotor and said flywheel in rotary orbital paths within said housing means and in counterbalancing opposition to one another;

a plurality of independently supported vanes extending radially outward through at least a portion of said rotor and into sealing engagement with said chamber internal surfaces and defining with said rotor and said internal surfaces a plurality of gas expansion chambers;

valve means positioned upon said housing means and adapted for sequentially admitting pressurized gases to and exhausting expanded gases from said expansion chambers.

3. The rotary engine of claim 2 wherein said rotor is controlled by said crankshaft and said crank means to oscillate in constant proximately spaced relation from discrete portions of said internal surfaces whereby said gas expansion chambers sequentially expand and contract in volume.

4. The rotary engine of claim 3 wherein said rotor is bearing mounted upon said offset crank portion, said offset

crank portion adapted to be driven in 360° rotation therein.

5. The rotary engine of claim 4 wherein said rotor includes means defining relieved regions therein to accommodate the relative movement of said vanes with respect thereto.

6. The rotary engine of claim 2 wherein

said rotor includes a rim portion having means defining a plurality of relieved regions therein within which said vanes are positioned,

each such relieved region containing bearing means to support and to reciprocate relative to one of said vanes in mutually sealed relation.

7. The rotary engine of claim 6 wherein

said relieved region includes means defining a cylindrical shaped portion, each said bearing means includes a pair of pivot halves

shaped to mate within one of said cylindrical-shaped portions in bearing relation thereto and to a said vane, such vane being positioned between said pivot halves.

8. The rotary engine of claim 7 wherein

said rotor includes a juxtaposition to each said pair of pivot halves seal actuator means bearing against said pivot halves and said rotor rim to improve the seals between said gas expansion chambers and

other portions of said compartments.

9. The rotary engine of claim 8 wherein said actuator means comprises

a sealing shoe defining a portion of said rotor rim and bearing forcibly against each said pivot half, each said sealing shoe and the rotor rim juxtaposed there to including a complementary cam surface inclined at an angle from an adjacently located one of said vanes,

retention and guide means associated with each said sealing shoe and said rotor rim to retain the position of said sealing shoe, and

spring means positioned to cooperate with pressure in said gas expansion chambers to constantly cause said sealing shoe to be cammed into forcible engagement with its mating pivot half and said rotor rim.

10. The rotary engine of claim 2 wherein

said crank means comprises:

a plurality of secondary cranks each having

a central pin mounted for rotation through said wall means;

a pair of opposed arms extending from opposite ends of said central pin;

a secondary pin extending parallel to said central pin from an extremity of each said arm;

said secondary pins respectively mounted for rotation in said rotor and said flywheel.

11. The rotary engine of claim 10 wherein eight of said secondary cranks are provided.

12. The rotary engine of claim 2 wherein

said vanes are oriented in a radial pattern, and wherein

vane control means is provided to restrain said vanes to a rotatably reciprocating movement pattern and to maintain their radial orientation during engine operation.

13. The rotary engine of claim 12 wherein

said vane control means includes a pair of pivot halves in said rotor on opposite sides of and in slidable sealing relation with each said vane; and

vane support means connected to and supporting said vanes and assisting to control their movement.

14. The rotary engine of claim 12 wherein said vane control means comprises:

opposed annular groove means in said wall means concentric with said crankshaft;

a pair of control shoes positioned in mating and bearing relation within said annular grooves adjacent each said vane;

a vane support beam joining each said pair of control shoes and abutting an innermost edge of the adjacent said vane in vane-supporting relation; and

spring means between each said vane support beam and said vane supported thereby, urging said vane radially outward.

15. The rotary engine of claim 12 wherein

said vanes are eight in number equally spaced about the periphery of said rotor, and adapted for independent rotary movement within said rotor to define continuously varying angles relative to one another during rotor rotation.

16. The rotary engine of claim 12 wherein

each said vane includes a plurality of mutually engaged vane segments; and

vane actuating means is provided to urge said vane segments into constant sealing engagement with

said internal surfaces.

17. The rotary engine of claim 16 wherein

said vanes each include a central camming segment of generally triangular shape and having marginal edges sloping outward toward a base of such triangles; 5

two marginal segments engaging in mutual retention the marginal edges of said central actuating segment;

said vane actuating means bearing against said central actuating segment, camming said central segments against said marginal segments and into forcible sealing contact with said internal surfaces. 10

18. The rotary engine of claim 17 wherein springs are provided as said vane actuating means. 15

19. The rotary engine of claim 12 wherein

said vane control means includes, in part, a pair of vane support rods engaging each said vane and extending radially inward therefrom; and

rod restrainer means positioned about said crankshaft, supporting and restraining said vane support opposite said vane engagement rods to limited rotary movement thereof. 20

20. The rotary engine of claim 19 wherein

spring means supported upon said support rods bear against and urge said vanes into forcible engagement with said chamber internal surfaces; and

said rod restrainer means includes a pair of spaced restrainer rings supporting a plurality of rod adapters, each having an end of one said rod engaged therein, said rod adapters having opposed surfaces shaped to mate and move in bearing relation with said crankshaft and said restrainer rings for relative rotational movement with respect thereto. 25

21. The rotary engine of claim 2 wherein

means is provided in each said housing means defining a passage leading to each said gas expansion chamber; 35

said valve means comprises a plurality of radially oriented, normally closed poppet valves in each said wall means, each said poppet valve controlling the admission of gases through one of said passages; and 40

cam means is fixed upon said crankshaft for rotation therewith adjacent each of said walls and contacting said poppet valves for sequentially actuating the same. 45

22. The rotary engine of claim 21 wherein

said cam means includes:

an inlet and an outlet cam; 50

said inlet cam comprising,

a pair of relatively rotatably juxtaposed cam plates having peripheries including matching cam lobes thereupon,

said plates being positioned such that each said poppet valve associated with said gas inlet passages contacts said peripheries of both said cam plates, and 55

cam actuating means in provided to rotate one said cam plate relative to the other; 60

said actuating means including,

gear means upon one of said plates,

a force ring having a gear rack extending therefrom into engagement with said gear means,

said force ring being rotatable with said crank- 65

shaft and adapted for being axially moved by an independent power source, whereby movement of said force ring and gear rack rotates said gear means and said one plate, causing said cam thereupon to rotate relative to said cam on the other said plate and thereby extending the combined length of said cam to extend the opening time of said valves.

23. The rotary engine of claim 22 wherein

said one cam plate includes;

an axially extending boss, and

said gear means includes;

at least one shaft rotatably retained in said boss and extending normal to the axis thereof,

a spur gear and a pinion gear fixed upon each said shaft for mutual rotation therewith,

a sun gear fixed to said one plate and engaging said pinion gear, and

said rack engages said spur gear, for causing rotation of said spur gear, said pinion gear, said sun gear, and said one plate.

24. The rotary engine of claim 23 wherein

at least three of each said rack, spur and pinion gears is provided.

25. The rotary engine of claim 2 wherein

means defining a passage leading to each said gas expansion chamber is provided in said housing means; and

said valve means comprises;

a stator plate concentric with crankshaft and having means defining ports there in leading independently to said passages,

a first metering plate fixed upon said crankshaft for rotation therewith and in slidable surface contact with said stator plate,

means in said first metering plate defining a plurality of discrete elongated ports oriented to substantially encompass a predetermined circumferential portion of said first metering plate and to match in radial position said stator plate ports and to sequentially open a predetermined number thereof as said first metering plate rotates with said crankshaft,

a second metering plate mounted upon said crankshaft for rotation therewith and for controlled relative rotation with respect thereto,

means in said second metering plate defining a plurality of discrete elongated ports oriented to substantially encompass a predetermined circumferential portion of said second metering plate and to match in radial position said stator plate ports and said first metering plate elongated ports,

said elongated ports in said second metering plate being normally oriented to exactly match those in said first metering plate, but rotatable to close selected ones thereof, and

actuating means to cause controlled rotation of said second metering plate relative to said first metering plate.

26. The radial engine of claim 25 wherein

said elongated ports in said first and second metering plates are elliptical in shape.

* * * * *