



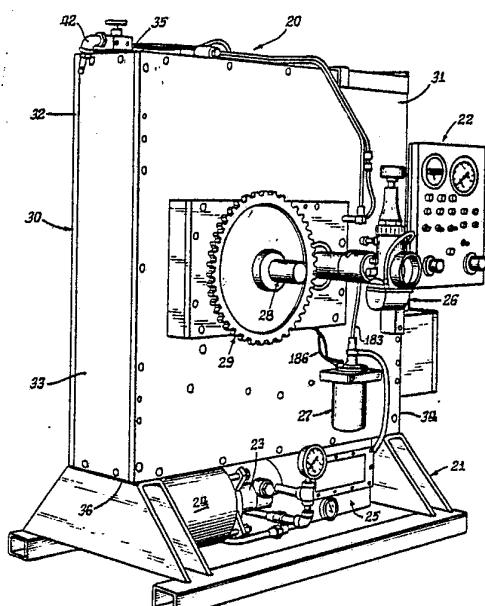
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(54) Title: ROTARY ENGINE

(57) Abstract

A four cycle internal combustion, rotary engine in which a cylindrical rotor having plural arcuately spaced, radially extending, piston carrying cylinders is mounted about a coaxially aligned central combustion chamber and main bearing for rotation therewith about a stationary main bearing support shaft which incorporates a fuel intake and exhaust manifold communicating with the combustion chamber and individual cylinders. A pair of stationary cam plates having registeringly aligned asymmetrical cam tracks are mounted in parallel spaced relation adjacent opposite axial ends of the rotor; the tracks thereof being simultaneously engaged by cooperating pairs of follower rollers slidably coupled to pistons in each of the cylinders in a manner requiring movement of the pistons radially of the rotor in response to rotational movement of the latter about the main bearing shaft. During each rotational cycle of the rotor each piston completes a four cycle operation of intake, compression, combustion and exhaust strokes in which the intake and compression strokes are selectively unequal to the combustion and exhaust strokes in accordance with a selected configuration of the cam tracks whereby to effect maximum fuel efficiency and power output for the engine in accordance with the characteristics of the selected fuel of combustion. A first modified form of the engine is disclosed wherein each cam plate provides a pair of radially spaced cam tracks engageable by individual rollers to positively control movement of the pistons radially of the rotor. A second modified form of the invention is also disclosed in which a plurality of individual rotary engines are arranged in a cluster about a single output shaft selectively coupled thereto via clutch members whereby to enable coupling one or more of the rotary engines to the output shaft in accordance with power requirements.



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ROTARY ENGINE

This invention relates generally to rotary internal combustion engines and more particularly to improvements therein for achieving greater fuel economy and power output.

5 In the familiar reciprocating piston engine used extensively today having pistons coupled to a rotatable crank shaft, the piston stroke is limited by the orbit of the crank shaft. As a consequence, the combustion or power stroke of each piston is only as great as the compression stroke thereof so that when a cylinder's exhaust valve is opened, unspent fuel and gases are allowed to escape to atmosphere resulting in the loss of useable power. Similar impact is encountered in the limitation of the exhaust stroke in such engines 10 to the orbit of the crank shaft. This limits the available time during which the exhaust gases may be expended from the cylinder. In consequence, there is a loss of power due to back pressure of the exhaust gases as each piston moves to the high point of its 15 20 stroke.

Aside from the noted limitations on the power and exhaust strokes of such engines, utilization of an elongated manifold for distributing air/fuel mixtures to the several cylinders thereof leads to 25 inefficiencies of fuel mixing and distribution largely brought about by the necessity of periodically opening and closing the intake valves to the cylinders, thereby

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interrupting the flow of fuel mixture to the cylinders. This leads to uneven delivery and combustion of the fuel mixture to and within the individual cylinders, creating uneven running of the engine.

5 Another area of disadvantage in the conventional four stroke engines resides in the fact that the four cycles of operation for each piston and cylinder, namely intake, compression, combustion and exhaust require two revolutions of the crank shaft for completion.

10 That is to say, for each two revolutions of the crank shaft there is only one combustion or power stroke for each piston. This results in increased RPM's and attendant friction loss and wear and tear on the moving parts for a given horsepower output. Due to

15 the fact that the conventional four cycle internal combustion engine employs a large number of friction producing parts, such as springs which must be compressed, bearings, cams which must be rotated or seals which rub tightly to restrict rotation of the engine, the

20 resulting friction power loss is not available for output horsepower.

While there are other factors and inefficiencies which detract from the overall capability of presently known four cycle internal combustion engines to produce

25 maximum horsepower with minimum fuel consumption, the foregoing indicate the principal areas of disadvantage which the present invention is intended to alleviate.

In particular, the present invention provides an internal combustion four cycle rotary engine,

30 comprising a generally cylindrical rotor having one or more arcuately spaced cylinders each carrying a piston therein and extending radially of the central rotational axis of the rotor, stationary main bearing support shaft means disposed coaxially of said rotor,

35 combustion chamber means mounted concentrically within said rotor and rotatably movable therewith about said

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shaft means, cam means providing asy whole continuous curvilinear cam track means, cam rider assembly means engaged with said cam track means for following the contour thereof, and means coupling a said rider

5 assembly means to the piston in each cylinder of said rotor for effecting movement of each said piston radially of said rotor in response to the following movements of its associated said rider assembly along said track means.

10 Movement of the various pistons in their respective cylinders is unlimited by a fixed crank shaft orbit inasmuch as asy whole cam means are employed for that purpose. Consequently, the piston strokes for the various cycles of intake, compression,

15 combustion and exhaust may be selectively varied and unequal in length and duration by changing the cam configuration. Thus the combustion or power stroke and the exhaust stroke may be considerably greater than the intake and compression strokes. By reason of an

20 increased power or combustion stroke, the engine has the ability to use more of the available combustion gases to produce usable horsepower. This increase in the power stroke is limited only by the overall size of the engine and the available compresses gases of combustion.

25 As a result virtually all of the compressed gases may be used for the purpose of producing usable power outside the engine as opposed to the conventional crankshaft engines which are limited in efficiency by the restrictive length of the crank shaft radius of orbit.

30 As a corollary to the increased power stroke the engine also is fully capable of converting pressurized gases of combustion into usable rotary power due to the fact that the force or pressure of the combustion gases is applied directly against a smooth and ever

35 declining angle of the cam means. Thus the gases are usable to their maximum advantage with regard to the

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leverage applied for rotating the cylinder block. Long camming angles are available for this purpose to produce a constant, smooth, and even leverage action productive of rotary power. As gas pressures increase 5 the camming leverage is at a minimum, gradually increasing as the expanding gas pressure decreases. This opposing action gives the engine an improved characteristic of substantially constant force application in converting rectilinear piston movement into rotary 10 movement of the output shaft.

As the pressures of the combustion gases are expended the exhaust gases must be expelled from the engine and, again, because the engine of this invention utilizes cam actuated piston movement, the camming 15 angle is capable of being decreased in the exhaust mode to provide maximum time for exhaust gasses to escape the confines of the cylinders. This leads to less back pressure against the pistons and a resulting lowered power loss from that factor.

20 In addition to the improved power and exhaust strokes, the cam actuated piston design provides improved engine breathing during the intake cycle principally because the engine rotates about a stationary manifold and main bearing shaft so that there is a near constant 25 or non-interrupted flow of fuel and air into the cylinders of the engine. This improved intake activity provides better distribution of fuel in the air fuel mixture and results in more even and powerful combustion within the cylinders.

30 The preferred design of the engine permits completion of all four cycle modes necessary for complete fuel combustion and exhaust within each 360° of rotor rotation. Thus each cylinder of the rotating cylinder block assembly fires for each complete rotation of 35 the cylinder block rotor. Consequently, the engine is capable of producing increased horsepower at lower

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revolutions per minute, or by changing the appropriate angles of the cam means and the length of piston stroke the engine may conversely produce lower horsepower at higher RPM.

5 In addition to the advantages in the areas of breathing, combustion and power output, the engine also exhibits great advantage in the area of minimizing internal power loss due to friction inasmuch as the engine rotor rotates freely on precision bearings and
10 exhibits very little friction loss by virtue of coupling the cam means with the individual pistons of the engine.

Because the rotor cylinder block of this engine is relatively freely rotatable, it additionally
15 exhibits a marked flywheel effect which permits the engine to produce horsepower higher than the engine's rated capacity for short periods of time. This characteristic is useful, for example, in applications such as an automobile, particularly at
20 start-up, when initially engaging the engine to the drive shaft. This flywheel effect allows the horsepower requirements for an application like an automobile to be reduced so as to size down the engine and provide greater fuel economy.

25 In the accompanying drawings:

FIGURE 1 is a perspective view of an embodiment of the rotary engine;

FIGURE 2 is a partial right hand end elevation of the engine illustrated in Figure 1;

30 FIGURE 3 is an enlarged front elevation with housing cover removed illustrating the interior arrangement of parts for the engine illustrated in Figure 1;

FIGURE 4 is an enlarged foreshortened cross-sectional view with parts in elevation, taken substantially along vantage line 4-4 of Figure 3 and looking in
35 the direction of the arrows thereon;

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FIGURE 5 is a partial front elevation, similar to Figure 3, but at a reduced scale thereover with rotor removed to illustrate the features of associated cam means;

5 FIGURE 6 is a perspective view of a main bearing support and combustion chamber which is mounted coaxially of the rotor;

10 FIGURE 7 is a perspective view of the combined main bearing shaft and intake and exhaust manifold member about which the cylinder block is rotatable;

FIGURE 8 is a perspective view of a slide member associated with the cam rider assemblies illustrated in Figure 3;

15 FIGURE 9 is an enlarged front elevation illustrating the power output shaft and the fuel igniting spark plugs associated with the combustion chamber of Figure 6;

20 FIGURE 10 is an enlarged front elevation similar to Figure 9 showing the relationship of the power takeoff and timing gears;

FIGURE 11 is an enlarged front elevation of the distributor and ignition timing means associated with the drive gear and output shaft;

25 FIGURE 12 is a partial cross-sectional view, similar to Figure 4 of a modified engine;

FIGURE 13 is a partial front elevation of a modified dual track cam means employed in the engine of Figure 12; and

30 FIGURE 14 is an end elevation with portions of the engine casing broken away showing a modified cluster engine employing four individual engines of the type illustrated in Figures 1 through 13.

In Figures 1 through 11 of the drawings features 35 of a working prototype engine 20 are set forth. As shown in Figure 1, engine 20 is shown mounted on a test

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stand 21 equipped with a control panel 22. A sump pump 23, driven by motor 24 is associated with an oil sump well 25 mounted on the underside of the engine. The engine 20 is equipped with a carburetor 26, spark coil 27, output shaft 28 and starting gear 29 adapted for connection with a suitable starting motor (not shown) to effect initial actuation of the engine's pistons for starting purposes.

As shown engine 20 comprises a generally rectangular parallelopiped outer casing 30 having parallel front and back walls 31 and 32, respectively, mounted over parallel side walls 33, 34 and top and bottom walls 35 and 36 to effect an enclosure for the working elements of the engine to be described more fully hereinafter. It is to be understood, of course, that the particular configuration of the casing 30 is relatively immaterial to the present invention other than to illustrate a protective outer covering and support for the working elements of the engine.

As shown in Figure 2 also mounted exteriorly of the casing 30 is an oil filter 37, a 12 volt power supply 38 adjacent the control panel 22, and a network of oil and air lines 39 and 40, respectively, for lubricating the interior working elements enclosed by the casing. A suitable fuel supply line 41 connected to an appropriate fuel tank (not shown) is also provided for supplying combustible fuel, such as gasoline, to the carburetor 26. It will be appreciated that the oil lines 39 communicate between the oil sump 25 via the oil filter 37 and appropriate zones of the engine as will be described in greater detail hereinafter. Similarly, the air lines 40 are coupled to a suitable source of compressed air via fitting means 42 (see Fig. 1) in the particular embodiment illustrated. It is to be understood that while the basic lubricating and cooling system depicted in the enclosed drawings comprises a

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combination of mixed air and oil, the engine is capable of being lubricated by pressurized oil alone or by air in combination with localized oil lubrication.

In brief, in accordance with the cooling and lubricating system illustrated, pressurized air and oil are injected as a misty atmosphere into the enclosed interior of the casing 30 where they are circulated principally by the fan effect of the rotating engine block rotor, throughout the casing. A filter means 44 (see Fig. 3) serves to trap oil mist from the circulating air which exits from the engine casing via openings 45, 45. Condensed oil collects in the bottom of the casing and returns to the oil tank sump 25 via drain means 46 in the casing bottom wall 36. Again such details are of no particular moment to this invention other than to establish the need for appropriate cooling and lubricating systems for the engine's working elements.

Turning now to features of the major working elements of the engine, specific reference is made to Figures 3 and 4 of the drawings. In general there are eight main elements or means involved in engine 20 which are arranged and combined for converting the explosive energy of fuel combustion to usable power available at the output shaft 28. In brief these comprise a main cylinder block rotor means 50 and a bearing and combustion chamber means 52 which is coaxially interlocked with the rotor and provides the main bearing support therefor during its movement about a combined main bearing shaft and intake and exhaust manifold means 54. Each of the cylinders provided in the rotor means 50 houses a rectilinearly movable reciprocating piston means 56 slidably coupled by cam rider assemblies 58 to a pair of parallel stationary cam means 59 and 60.

The organization of the above-listed elements

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is such as to effect rotatable activity or motion of the rotor means and reciprocating activity of the piston means carried therein in accordance with the configuration of the cam means whereby to rotatably drive a primary 5 drive gear means 62 affixed to the outer end of the combustion chamber means 52. As noted such enumerated means 50 through 62 are supported within or on the casing 30 and constitute the major working elements of the engine which will now be described in detail.

10 The rotor means 50 is basically a cylindrical fly wheel cylinder block in which one or more cylindrical bores are provided along arcuately spaced radial axes to form piston cylinders. In the embodiment illustrated herein, four such radially disposed piston cylinders 15 64, 65, 66 and 67 are provided. It should be noted that the permitted number of cylinder bores is determined only by the available space in the rotor and the particular power requirements of the engine. Within such parameters an engine according to this invention 20 may comprise one or more cylinders. Adjacent each cylinder bore and running parallel to and on opposite sides thereof, in the front and back faces of the rotor, are a pair of registeringly aligned parallel spaced cam slide grooves 68, 68 receptive of a pair of cam 25 rider assemblies 58 associated with each piston means 56. Cam slide grooves 68 extend from the outer periphery of the rotor to a central bored opening 69 therein (see Figure 4) which is coaxially receptive of the combustion chamber means 52. The side walls of each 30 groove 68 are undercut to provide overhanging lip portions 70 bordering the lateral sides of such groove whereby to interlock the cam rider assemblies in such 35 grooves.

Within the outer 1/3 diameter of the rotor 35 are elongated slotted openings 71 aligned with the center line of the cam slide grooves 68 and each of the cylinder

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chambers. Such openings 71 extend into the adjacent cylinder chamber for the purposes of affording connection of the piston means 56 therein with associated cam rider assemblies 58, as will appear presently.

5 Between each of the cylinders 64-67, areas of the rotor may be cut away as shown to provide intermediate web spokes 72, or such intermediate areas may be left solid as desired.

It will be appreciated that each of the 10 cylinders 64 through 67 communicates openly with the outer periphery of the rotor 50 and the central bore 69. It also will be understood that in the construction of the rotor 50 the mass thereof is evenly distributed about its central axis to effect static and dynamic 15 balance of the rotor whereby to avoid vibration in operation.

While not shown in the drawings, the central bore 69 of the rotor is provided with the keyway for interlocking connection with the combustion chamber 20 means 52 as will now be described.

With particular reference to Figures 3, 4 and 6 of the drawings, the features of the combustion chamber and main bearing support will be recognized. Combustion chamber means 52 is a multi-purpose part in that it 25 supports the rotor means 50 and rotates with that member on and about the combination stationary main bearing shaft and intake and exhaust manifold means 54 (see Figs. means 3 and 4). As best shown in Figure 6, the combustion chamber means 52 is formed with a cylindrical 30 main body portion 80 provided with an elongated keyway 81 cut into one side thereof parallel to its longitudinal axis. Body portion 80 has an end wall portion 82 partially enclosing one end thereof; the opposite 35 end thereof being partially enclosed by a parallel flanged end wall portion 83 having an annular flange extending radially outwardly of the circumference of

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the main body portion 80 and distinguished by four U-shaped notches or cut out areas 84, 84 located at 90° intervals thereabout. These cut out areas or notches 84 are provided for the purpose of clearing the cam 5 rider assemblies 58 in operation, as will be described more fully hereinafter. The two end wall portions 82 and 83 are distinguished by a large central cylindrical bore 85 extending coaxially through body portion 80 for receiving the stationary main bearing shaft and 10 manifold means 54 along with a main cylindrical sleeve bearing 86 (see Figure 4) which is pressed into bore 85 in assembly.

The cylindrical main body portion 80 of member 52 is further distinguished by four bored openings 87 15 extending radially inwardly from the outer surface thereof to communicate with the central shaft receptive bore 85, as indicated in Figure 6. Each of these bores 87 radially intersects the central axis of the member 52, is aligned centrally of one of the cut out areas 84 20 and has a cylindrical counterbore at its outer end to provide a shallow cup-shaped chamber portion 88 coaxially of the opening 87 and interjoined with the latter via a chamfered frustoconical shoulder area 89. The radially innermost end of each bored opening 87 25 communicates with a transverse oval opening 90 having an elongated axis paralleling the elongated axis of the combustion chamber member 52 and communicating with the shaft 54 via registeringly aligned oval openings 91 formed in the main bearing 86 in assembly. 30 The superposed openings 90, 91 act as valve means in operation as will appear presently. Each of the bored openings 87 and the adjacent bore areas 88, 89 described, comprises an individual combustion chamber communicating with one of the cylinders 64 through 67 with which it is 35 coaxially aligned in assembly. Accurate coaxial alignment of the combustion chambers and the four cylinders

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64-67 is accomplished by mating a key with the keyway means 81 on the exterior surface of the body portion 80 of member 52 and the keyway provided in the central enlarged opening 69 of the rotor.

5 Each combustion chamber is invaded by a spark plug 92 threaded into a bored opening 93 extending inwardly of the outer end 82 of member 52 parallel to the latter's longitudinal axis; such spark plugs having the gapped electrodes thereof openly invading the
10 combustion chambers formed by the bore portions 87-89 (see Figure 4). The outer end 82 of the member 52 is also provided with twelve openings 94, 94 and 95, 95 receptive of appropriate fasteners for joining the main drive gear 62 coaxially to the outer end of the
15 member 52 in assembly; gear 62 being fastened to end wall 82 by eight machine bolts 96, 96 (see Figure 10) and aligned by means of four locating pins 97, 97 projecting outwardly of openings 95. The opposite flanged end wall portion 83 of the combustion chamber
20 means 52 is similarly joined to the rotor means 50 by eight machine bolts (not illustrated) extending through appropriate openings 98 formed in the annular flange portion thereof as shown best in Figure 6.

As previously indicated accurate alignment
25 between the four combustion chambers and the several cylinders is achieved by interlocking key and keyway means formed in member 52 and the central bore of the rotor; the combustion chamber means 52 being press fitted into the central opening 69 of the rotor with
30 the key and keyway means aligned and engaged as noted. In consequence the rotor 50, the combustion chamber means 52 and main bearing 86 are conjointly rotatable about the main shaft and manifold means 54.

The intake and exhaust manifold means 54
35 serves the general purpose of supporting the rotatably movable rotor means 50 and its fixedly associated

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combustion chamber means 52 and main bearing 86 for movement thereabout. It also serves to deliver air/fuel mixtures to the individual combustion chambers and to convey exhaust gases from the cylinders.

5 Particulars of the exhaust manifold means 54 will best be understood from Figure 7 of the drawings taken in conjunction with Figure 4. As there shown, member 54 comprises an elongated cylindrical shaft body 100 having an integral collar portion 101 adjacent one 10 outer end 102 thereof and which is provided with wrench engaging flat surfaces 103 whereby the shaft 54 may be manually rotated for adjustment purposes as will be explained more fully hereinafter. The opposite end 104 of member 54 is provided with threaded openings (not 15 shown) receptive of machine screws whereby the carburetor means 26 may be attached coaxially thereto.

Two large coaxial bores are formed inwardly of the opposite ends of the shaft member 54 to extend partially along the length thereof as indicated at 106 20 and 108 (see Figure 4). Bore 106 is intersected at its inner end by secondary bore 110 disposed at 45° to the axis thereof. The outer end of the slanted bore 110 is intersected by a milled out slot or flat area 113 formed parallel to the longitudinal axis of the member 54. 25 In a similar fashion the central bore 108 is intersected at its inner end by a secondary slanted bore 112 which in turn is intersected by a flattened area 114. The two flattened areas 113 and 114 constitute exhaust and intake 30 valve ports, respectively, which do not intersect in any respect (see Figure 7) and which cooperate with the oval openings 90, 91 associated with each combustion chamber.

The shaft 54 is also provided with internal lubricating passageway means 118 having one or more 35 outlets 119 (see Fig. 4) communicating with the main bearing 86 and having a supply fitting 120 formed at

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collar portion 101 for purposes of lubricating the bearing 86 as the latter moves about the stationary shaft 54; fitting 120 being joined to oil line 39.

As shown best in Figure 4, the interconnecting bores 106 and 110 constitute the exhaust passageway system for the engine while bore 108 and its intersecting bore 112 constitute the main air intake passageway system for supplying fuel and air mix to the combustion chambers 87. From Figure 4, it will be recognized that shaft 54 is coaxially inserted within the main bearing 86 coaxially of the combustion chamber 52 and the rotor 50 with the exhaust and intake ports thereof aligned by rotating shaft 54 so that the slotted valve ports 113 and 114 align with the oval openings 90 and the 15 correspondingly aligned openings 91 in the bearing member 86 in accordance with the intake and exhaust cycles for the several pistons and cylinders of the engine.

In order that the shaft 54 be held in a fixed 20 and non-rotatable condition after adjustment, two shaft clamp numbers 122 and 124 are mounted over and clamped to the outer ends of the shaft member 54; clamp 122 being bolted to the back casing wall member 32 while clamp 124 is similarly fastened to a front cover wall 126 of 25 an auxiliary casing 128, disposed centrally of the engine casing's front wall member 31. With this arrangement, as the rotor 50 moves about the shaft 54 along with the combustion chamber 52, the individual combustion chambers 87, coaxially aligned with their respective 30 cylinders 64 through 67, move sequentially into communication with the ports 113 and 114 at selected points of the rotational cycle for rotor 50. It is to be noted that the combustion chambers and their cylinders are sealed off by the solid outer cylindrical wall of 35 shaft 54 intermediate the ports 113 and 114. Thus the valving system for the engine is provided. It is to be

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noted that with this arrangement the intake port 114 is in communication with one of the four cylinders of the engine at all times so that the flow of fuel and air mixture is uninterrupted by the valving means to 5 provide a steady flow of fuel mixture through the passageways 108, 112. This promotes fuel economy and a smooth running engine. The same holds true with respect to the exhaust passageways 106, 110 to promote uninterrupted outflow of spent exhaust gases.

10 The pistons 56 are best shown in Figure 4 of the drawings as comprising a one-piece construction in which head portion 130 thereof is formed integrally with connecting rod and cross head portions 131 and 132, respectively. The head portion of course, is of cylindrical formation while 15 the crosshead is generally of rectangular configuration having semi-cylindrical ends where the same meet the cylinder walls so as to act as guide means for movement of the pistons in and along the cylinder chambers. The head portion 130 also is provided with three annular rings 134, 20 two of which are compression rings and the third of which is an oil ring to promote lubrication of the cylinder walls. The rings 134, of course, are mounted in appropriate grooves cut for that purpose about the circumference of the cylindrical piston head portion 130. The cross- 25 head portion 132 is provided with a central cylindrical bore extending to the longitudinal axis of the piston for reception of a mating connecting pin 136 by which the pistons are coupled to associated cam rider assemblies 58. As mentioned heretofor, in the illustrated embodiment shown there are four pistons, one disposed in each of the cylinders 64 through 67 for rectilinear reciprocation coaxially of such cylinders and each piston is coupled to a pair of cooperating cam rider assemblies 58 as will now be described in detail.

30 35 As best shown in Figs. 4 and 8 of the drawings, each of the cam rider assemblies 58 comprises an elong-

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ated, substantially rectangular rigid slide member 138 formed with parallel spaced linear rail portions 139 extending along the lateral flanks or margins thereof for sliding reception beneath the lips 70 in the slide 5 grooves 68 provided on opposite sides of each cylinder, as previously described. The cam rider assemblies, as previously noted, are the means by which the rotor develops its rotating power and to this end the connecting pins 136 extend outwardly of opposite ends 10 of the crosshead portion 132 of each piston and through the slotted openings 71 in the cylinder walls for press fitted engagement with cylindrical openings 140 formed adjacent the outer end of each slide member 138 of a cooperating pair thereof (see Figs. 4 and 8). 15 The pins 136 lie in parallel spaced relation to the longitudinal axis of the main bearing shaft 54 and are loosely received in the slotted openings 71. Interconnection of the pins 136 with the slide members 138 rigidly couples such members together so that when the 20 slides move the piston also moves within the confines of its cylinder and vice versa. Engagement of the rail portions 139 with the slide grooves 68 is a loose slide fit to permit easy movement of the slide members along the slide grooves in the rotor.

25 Adjacent the lower end of each slide member 138 is mounted a headed pin member 142 which projects laterally beyond its associated slide member to provide a mounting stud on which a cam follower roller 144 is mounted. Each roller 144 is held in place by a snap ring (not shown) 30 receptive in a snap ring groove 145 formed adjacent the outer end of pin member 142. Each cam follower roller 144 constitutes a bearing assembly of standard construction comprising an outer ring movable about and on rotatable roller bearings held in a hub race member in 35 accordance with known practice.

As shown, in assembly a pair of the bearing

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member rollers 144 are located adjacent opposite sides of the head end 130 of each piston with the associated rollers 144 being coaxially aligned. Thus each piston is effectively supported on rollers 144 by a rigid yoke 5 system comprising a pair of slide members 138 and a cross connecting pin means 136.

Such cam rider assemblies are the means by which the straight line force produced from the combustion of fuel against the piston head in each of the cylinders 10 is converted into desired rotary action of the engine block to rotate the drive gear means 62. This conversion activity is brought about by virtue of the roller bearing 144 being pushed outwardly by the piston after combustion which forces the rollers to ride on declining planes 15 provided by the cam means 59 and 60; the thrust of the follower roller bearings against the cam means causing the free wheeling rotor assembly 50 to reactively rotate and develop desired rotory power.

Features of the cam means 59-60 will best 20 be understood from Figs. 3, 4 and 5 of the drawings. Inasmuch as the cam means 59 and 60 are identical insofar as camming contours and construction are concerned, the two being mere reflections of one another to accommodate their mounting in parallel spaced registry, the description 25 which follows will be concerned primarily with cam means 59 with the understanding that corresponding features of cam means 60 are the same.

With special reference to Figs. 3 and 5, it will be understood that the cam means 59 illustrated 30 therein comprises a generally square shaped heavy metal plate 150 suitably bolted to the back cover wall 32 of the engine casing in coaxial alignment with the shaft means 54 and the rotor means 50. Cam means 60 is similarly fixed to front casing wall 31 in registry 35 with cam 59. In general as best noted from Fig. 5, the plate 150 is cut out centrally to provide an asymmetrical

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peripheral cam track 152 of continuous contour which is engagable by the follower rollers 144 in operation. In the particular embodiment illustrated the track 152 is formed to provide four distinct operations of the 5 pistons, the movements of which are responsive to the engagement and movement of the cam roller means 144 with and along the such cam track.

In brief, track 152 is distinguished by two lobe areas 153, 154 extending inwardly toward the central axis 10 of the engine and cam plate 150, which axis is coincident with the axis of rotation for the rotor means 50. The high point of lobe 153 marks the point of initiating the intake cycle of the engine pistons and may be considered as the zero degree position of rotational 15 movement for the rotor. Reading counterclockwise from this zero degree position to approximately 85 degrees, the camming angle of the track 152 declines or moves away from the central axis of the engine causing each piston to move radially outwardly from the rotor's axis 20 of rotation in accordance with the following activity of the associated roller means 144 along track 152. This intake mode or cycle creates a vacuum atmosphere within the confines of the cylinder as the oval holes or openings 90, 91 of the associated bearing and combustion 25 chamber approach and come into alignment with the intake port 112 provided in the main bearing shaft 54. As the rotor proceeds counterclockwise over the intake port, the piston draws in fuel/air mixture from the carburetor through the intake manifold passageways 108, 30 112, gradually closing the intake port 114 with respect to the involved cylinder as the rotor approaches 85 degrees of counterclockwise rotation.

It is to be noted that there is a supplementary part of cam means 59, labeled 156 in Fig. 5, which is 35 disposed radially inwardly of the intake portion of track 152 to present a secondary cam track 158 in parallel

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spaced relation to the intake portion of cam track 152. A corresponding supplemental plate 160 is provided for cam means 60 in registering opposition to the plate 156 (see Fig. 4). These secondary cam plates are specifically designed to aid intake at engine starting when the 5 RPM of the rotor is insufficient to create the necessary centrifugal forces on the piston and its cam rider assemblies to follow track 152 and draw in a full air fuel charge into the cylinder. Other than the starting 10 mode, the supplemental cam plate portions 156 and 160 do not come in to play and are not normally engaged by the roller means 144 once the engine is in full operation.

The portion of the cam track 152 between the 15 end of the intake cycle to the high point of the second lobe 154, (generally between 85 and 175 degrees of counterclockwise rotation measured from the high point of lobe 153) constitutes the compression cycle for each piston/cylinder assembly.

20 Compression is initiated as the oval openings 90, 91 of the affected cylinder and combustion chamber passes beyond intake passageway 112 and port 114 onto or opposite a solid part of shaft 54 intermediate the intake and exhaust port openings thereof to seal off the cylinder 25 and combustion chamber and prevent the loss of explosive mixture therefrom during compression. The gases are compressed as the associated cam rider assemblies are forced up an inclining cam angle of ever increasing value. As the roller means 144 approaches the lobe 154, the 30 piston is forced in closer to the center line of the engine until it reaches full compression at approximately 175° of rotor rotation.

Following the compression cycle the next step 35 in each piston/cylinder's operation is the combustion mode which occurs from approximately 175° to 237° of counterclockwise rotation. In this mode, the piston in the

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cylinder is under full compression with the intake and exhaust ports in the main bearing shaft sealed so that upon electrically energizing the spark plug associated with the particular combustion chamber involved, the 5 compressed fuel mixture within the sealed cylinder is ignited. When this occurs the piston is forced out and away from the central axis of the engine and its cam rider assemblies are driven against a now sharply declining angle of the cam plates, causing the rotor 10 assembly to be rotatably driven.

Following the combustion cycle each piston/cylinder combination of the rotor assembly passes into the exhaust mode of its operating cycle which occurs from substantially 237° to 360° of counterclockwise 15 rotation. As the rotor assembly moves a cylinder into this exhaust mode, the oval holes 90, 91 in the associated combustion chamber comes into communicating alignment with the exhaust port 113 in the main bearing shaft and the cam rider assemblies are moved along an 20 inclining cam angle forcing the piston gradually inward toward the central axis of the rotor and the peak of lobe 153. This inward radial movement of each piston in response to movement of the cam follower roller assemblies associated therewith, forces the expended 25 fuel and gases out of the exhaust port and manifold passageway 106, 110. The exhaust mode is completed as the rotor approaches 360° of counterclockwise rotation ready to repeat the above described four cycle program.

30 The aforescribed cycles or modes of operation of course occur for each of the four cylinder and piston assemblies of the illustrated embodiment, particular note being made of the fact that each piston completes a full cycle of operation namely intake, compression, 35 combustion and exhaust for each 360° of rotor movement. Importantly it is to be recognized that the piston strokes

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as well as the duration of each operating cycle may be widely varied, if desired, as determined by the selected configuration of the cam tracks.

Such rotational driving of the rotor effects 5 corresponding conjoint rotation of the main drive gear means 62 from which the power output of the engine is taken.

The features of the main drive gear means 62 will best be understood with reference to Figures 4 10 and 9 through 11 of the drawings. The drive gear has several functions. First, it supplies the means for taking power from the engine for delivery to the output shaft means 28. Secondly, it provides means for carrying insulators used to insulate the spark plugs 15 92 and insure minimum loss of electrical power to the spark plugs. In conjunction with this latter function it also acts as a cooperating part of the distributor means for effecting ignition of the spark plugs in proper sequence to the operating cycles of 20 the engine.

For a better understanding of how the foregoing functions are accomplished, initial reference is made to Figure 9 of the drawings which illustrates the combustion chamber means 52 in its assembled position on 25 the intake and exhaust manifold means 54 with end 82 thereof extending through an appropriate opening in the casing wall 31 into the box-like auxiliary casing 128 fixed to the front face of the engine casing. It will be recalled that the combustion chamber means 52 has 30 four bored chambers 93 formed inwardly of the outer end 82 thereof for reception of the four spark plugs 92 in the illustrated embodiment hereof (see Figure 4). The spark plugs of course are threadedly engaged with the walls of the openings 93 as previously described 35 with the gapped electrodes thereof disposed in associated individual combustion chambers for each of

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the four cylinders in the illustrated embodiment.

As is also shown in Figure 9, the output shaft 28 is rotatably supported in and by bearing means 164 mounted in the casing wall 31 and partially supported by 5 the adjacent cam plate 60 (not shown).

Turning now to Figure 10 of the drawings, the detailed aspects of the drive gear 62 will better be understood. It will be noted that gear 62 comprises a spur gear which is fixed to the outer end of the 10 combustion chamber means 52 by bolt and pin means 96 and 97 for rotational movement with the rotor and combustion chamber. Mounted alongside the drive gear 62 and fixed to the output shaft 28 as by key and keyway means 166 is a timing gear 168 which in the particular instance 15 illustrated, is the same size and diameter as the drive gear. The two gears have intermeshing peripheral teeth whereby they rotate at a 1 to 1 ratio relationship. This gearing ratio may be changed, of course, in accordance 20 with desired rotational speed of the output shaft 28 within the skill of the art.

It will be observed that the drive gear 62 contains four large openings 170 which are registeringly aligned with the openings 93 in the combustion chamber means 52 and are designed to receive cylindrical 25 insulators 172 having a central electrode or electrically conductive core member 173 mounted therein (see Figure 4). Such insulator members fit over the outer electrode end of the spark plugs with the conductive core member 173 in contact with the connector electrode end 174 30 thereof, as best shown in Figure 4. This establishes good and positive circuit contact between the spark plugs and the conductor core members 173. Suitable lock bolts 175 engage a projecting semi-circular end shoulder portion 176 of each of the insulators 172 to 35 axially lock the same in their bores 170 and press the same tightly against the central connective electrode

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of the spark plugs. It will be recognized that with this arrangement the drive gear means 62, the spark plugs 92 and the combustion chamber 52 simultaneously move about a common axis in accordance with the movement of the 5 rotor means 50.

In order to insure proper firing of the several spark plugs in accordance with the combustion cycle of operation for each piston of the engine, it is necessary to provide means for timing the ignition 10 of the spark plugs and for transferring electrical energy to the electrodes thereof. To this end it will be noted from Figure 10 in particular, that the shouldered end portions 176 at the outer end of the insulators are semi-arcuate, cutting through a portion of each of 15 the conductive core members 173 thereof with such arcuate cut out areas being aligned at a common radius from the central axis of the shaft 54.

As shown in Figure 11, over the outside cover 126 of the auxiliary housing 128 is mounted the shaft locking collar 124 which is securely bolted to the cover wall 126 as previously described. Locking collar 124 is cut away on one side to provide a straight line shoulder against which rests a distributor insulator and housing 180 having a conductive distributor contact 20 member 182 mounted therewithin. Member 182 is joined by conductor 183 (see Fig. 1) to the spark coil 27 mounted exteriorly of the engine housing or casing 30 and is aligned opposite the path of movement for the conductive core members 173 carried by the drive gear. 25 Additional conductors 186 lead from the coil 27 to a breaker point assembly 187 mounted on a rotatably adjustable timing plate 188 fixed to the front wall plate 126 of the auxiliary housing 128. The breaker points 187 are in operating engagement with a timing cam 190 30 mounted about the output shaft 28 and fixed thereto for coaxial rotation with the output shaft. It will be 35

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noted that the timing cam 190 provides four lobes 191 engagable with the follower 192 of the breaker point assembly 187. Thus for each rotation of the output shaft 28 the breaker points 187 are opened and closed 5 four times corresponding to the combustion cycles of the four piston and cylinder assemblies in the illustrated engine.

The spark coil 27, of course, is connected to the 12 volt power supply 38 so that as the breaker points are opened and closed energy is sequentially 10 delivered to the electrode 182 of the distributor assembly and transferred therefrom to the conductive core members 173 held by the insulator members 172 as members 173 move beneath electrode 182 with the 15 rotatably driven drive gear 62. This distributes electrical energy to the respectively associated spark plugs in positive timed relationship to the rotation of the timing cam 190. Advance or retarding of the spark is achieved by rotating the timing plate 188 which 20 is adjustably held in position by the bolt and slot means 194 as shown in Figure 11. Thus ignition of the combustible fuel mixture within each of the cylinders is effected as desired.

Having described the various elemental portions 25 and parts which go to make up the preferred engine, it is to be noted that because of the unique cam and cam rider assemblies associated with each of the pistons, the latter are permitted a maximum stroke length while keeping the overall size of the engine to a minimum. 30 Due to the fact that the follower rollers of the cam rider assemblies 58 are disposed adjacent the head end of the pistons, the point of force transfer between the follower rollers 144, located adjacent the innermost ends of the slide members 138, and the cam tracks of the two 35 registeringly aligned cam means occurs as close to the center line of the engine as possible. This feature not

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only extends the length of available piston stroke, but permits much steeper angles in the cam track itself during the combustion cycle to produce more available rotating power while greatly reducing the rotational speed required of the rollers 144 as they engage and follow the tracks of the cam means. In addition, due to the provision of the cam slide members the loss of power which would normally be caused by the piston rubbing the sides of the cylinder is essentially eliminated since the pistons are guided rectilinearly by the cam slide members and cross heads so that there are no lateral forces on the pistons. Because of the absence of lateral forces acting on the pistons greater cylinder life and a better sealing action of the piston rings is brought about.

It is of further importance to note that the cam track may be widely varied. This coupled with the fact that all four cycles of piston/cylinder operation, namely, intake, compression, combustion and exhaust are accomplished during each 360° of rotation of the rotor assembly permits wide variation in designing the individual cycles both in duration and length of piston stroke. Such freedom of cycle design permits the engine to produce maximum power with maximum fuel efficiency. As an example, the intake and compression strokes may be only one half the length and duration of the combustion stroke to provide the utmost use of the expanded gases. On the other hand the exhaust stroke can be greatly extended to allow more time for spent gases to be purged from the engine thereby reducing back pressure on the pistons. Because of such available changes in the individual cycles inherent in the cam design, the engine of this invention is capable of efficiently burning and converting to rotating power virtually any rapidly expanding fuel such as gasoline, diesel fuel, alcohol, natural gas, hydrogen, propane,

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butane, etc. Not only is the engine capable of using and burning such fuels, but due to the flexibility of cam design available such burning and combustion can be carried out most efficiently. In addition, 5 because of the capability of extending the combustion cycle, gases of combustion may be more completely consumed thereby reducing exhaust gas pollution.

While the foregoing described embodiment of the engine is efficient and versatile in its 10 operation as noted, a modified version thereof, illustrating the major points of departure over the engine in Figures 1-11, is illustrated in Figures 12 and 13 of the drawings. As shown therein the modified engine 200, comprises a pair of stationary cam means 15 201 and 202 formed integrally with mating outer casing members 203 and 204, respectively. It will be recognized that the principle departure of this structure, over the first described engine 20 is in the provision of double cam tracks in each of the cam 20 means 201 and 202. Roller means 144, as previously described, engage the outer cam track 205 i.e., the cam track disposed radially outermost from the central axis of the engine, while additional roller means 206 engage the radially innermost tracks 207. This provides 25 a cam follower system which responds to positive push and pull action of the engine's pistons and makes for a relatively noise or clatter free operation of the engine, particularly when the pistons reverse their direction from inward to outward movement, relative 30 to the center line of the engine. It also provides for positive control of the intake cycle which is no longer dependent on centrifical forces as in engine 20. Other than the redesign of the outer casing, the cam track means and cam follower assemblies, the modified 35 engine 200 is substantially identical to engine 20 in all other respects.

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Figure 14 shows the features of a second modification of the engines hereinabove described, namely a cluster arrangement of two or more engines 20 or 200 capable of being coupled to a single output shaft 210. Specifically in this modified embodiment an outer casing 212 mounts four individual rotor means 50, for example, arranged in four quadrants about a central output shaft 210. Each of the rotors is coupled to a central drive gear 214 which is keyed to the output shaft 210 over intervening speed clutch assemblies 216. Each clutch assembly basically comprises a pair of gears 218 and 220 of dissimilar diameter mounted on a common clutch shaft 222; gear 218 being engaged with the drive gear 62 of an individual engine rotor and the secondary clutch gear 220 being engaged with the common drive gear 214 coupled to the common output shaft 210. The arrangement of the slip clutch assemblies is such that if gear 220 rotates at a speed greater than that of gear 218 10 the clutch mechanism produces complete disengagement of the two clutch gears. If on the other hand the rotating speed of gear 218 is brought up to match that of gear 220 the clutch engages, locking gears 218 and 220 together and transferring torque to power gear 214 and 15 power output shaft 210. It is to be noted that there are several other known means and methods by which the engagement and disengagement of the drive shaft can be accomplished such as locking pins, standard pressure clutches, etc. within the skill of the art.

Basically, the cluster engine of Figure 14 20 is designed to provide a power plant in which the engine is capable of matching peak horse power requirements as well as meeting offload requirements while operating at a fraction of its potential power output. In brief, 25 the multiple rotary engine of Figure 14 permits two or more engine rotors 50 to be incorporated into one common

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engine while maintaining each rotor selectively independent of its partners. Consequently, it is possible to operate an engine of the character set out in Figure 14 at selected levels of horse power output. For example, 5 if it were to be assumed that each rotor engine were capable of producing 50 horse power, then an engine similar to that shown in Figure 14, containing four rotors, would be capable of producing a total horse-power output of substantially 200 horse power. If, 10 upon occasion, only 100 horse power were required, then only two of the four rotors would be required and the other two could be totally shut down allowing them to be at rest while the other two continued running, unencumbered by the two at rest rotors. In 15 such a circumstance the at rest rotary engines could be maintained, tuned-up or similar operations performed without interruption of the engine's operation on the whole. By way of further example, if the average horse power requirement were only 50 horse power, 20 then only one rotor would need to be activated at any one time and at regular intervals another rotor could then be activated instead of the first rotor, permitting the latter to rest, thus promoting longer engine life and more even wear of moving parts. In effect, a 25 cluster engine of the character indicated in Figure 14 provides the advantage of having an inbuilt back up system, so that in the event of a mechanical failure of any one rotary engine, one or more other engines are available and waiting to be put on the line.

CLAIMS

1. An internal combustion four cycle rotary engine, comprising a generally cylindrical rotor having one or more arcuately spaced cylinders each carrying a piston therein and extending radially of the central 5 rotational axis of the rotor, stationary main bearing support shaft means disposed coaxially of said rotor, combustion chamber means mounted concentricly within said rotor and rotatably movable therewith about said shaft means, cam means providing asymmetrical continuous 10 curvilinear cam track means, cam rider assembly means engaged with said cam track means for following the contour thereof, and means coupling a said rider assembly means to the piston in each cylinder of said rotor for effecting movement of each said piston radially of 15 said rotor in response to the following movements of its associated said rider assembly along said track means.

2. The rotary engine of claim 1, wherein said cam track means is located radially outwardly of the central rotational axis for said rotor and each said 20 rider assembly means effects intake, compression, combustion and exhaust cycle movements of the said piston coupled thereto during a single revolution of said rotor.

3. The rotary engine of claim 1, wherein the configuration of said cam track means is such that the 25 combustion stroke of each piston is a different length than the intake stroke thereof.

4. The rotary engine of claim 1, wherein the configuration of said cam track means is such that the 30 combustion stroke of each piston is a different duration than the intake stroke thereof.

5. The rotary engine of claim 1, wherein the

length of each piston's combustion stroke is selected in accordance with the combustion characteristics of a selected fuel whereby to maximize power output of said combustion stroke.

5 6. The rotary engine of claim 1, wherein the combustion stroke of each piston is of different length and duration than the intake stroke thereof.

10 7. The rotary engine of claim 1, wherein the combustion stroke of each piston is of different duration than the exhaust stroke thereof.

8. The rotary engine of claim 1, wherein the combustion stroke of each piston is of different length than the exhaust stroke thereof.

15 9. The rotary engine of claim 1, wherein the intake and compression strokes of each piston are substantially equal in length and duration, and the combustion and exhaust strokes thereof are unequal and of greater length and duration than said intake and compression strokes.

20 10. The rotary engine of claim 1, wherein the intake and compression strokes of each piston are unequal in length and duration.

25 11. The rotary engine of claim 1, wherein the point of force transfer between said rider assembly means and said track means is adjacent the radially innermost end of the said piston coupled thereto.

30 12. The rotary engine of claim 1, wherein a pair of rider assemblies is associated with each piston and said cam means comprises a pair of parallel spaced registering aligned cam track means disposed adjacent opposite axial ends of said rotor.

35 13. The rotary engine of claim 1, wherein said cam track means is configured to effect intake, compression, combustion and exhaust cycle operation of each piston for a single-revolution of said rotor and

in which such cycles are of unlike duration.

14. The rotary engine of claim 1, wherein said cam track means is configured to effect intake, compression, combustion and exhaust stroke operation 5 of each piston during a single revolution of said rotor and in which such strokes are of unlike length.

15. The rotary engine of claim 1, wherein said combustion chamber means provides an individual combustion chamber communicating with each cylinder of 10 said rotor.

16. The rotary engine of claim 15, including spark plugs mounted in each said combustion chamber for rotational movement with said chamber means about said shaft means.

15 17. The rotary engine of claim 1, wherein said shaft means comprises coaxial intake and exhaust passageway means, and intake and exhaust valve port means communicating with said intake and exhaust passageway means, respectively, and operable to communicate with 20 the combustion chamber of each said cylinder in response to rotation of said rotor and combustion chamber means about said shaft means.

18. The rotary engine of claim 1, wherein each cam rider assembly means comprises a rigid slide member 25 slidably connected to said rotor for movement radially thereof and parallel to the movement axis of an associated piston, means traversing each cylinder for rigidly joining one end of said slide member to the radially outward end of its associated piston, and 30 roller means mounted on the opposite end of said slide member for engagement with said cam track means.

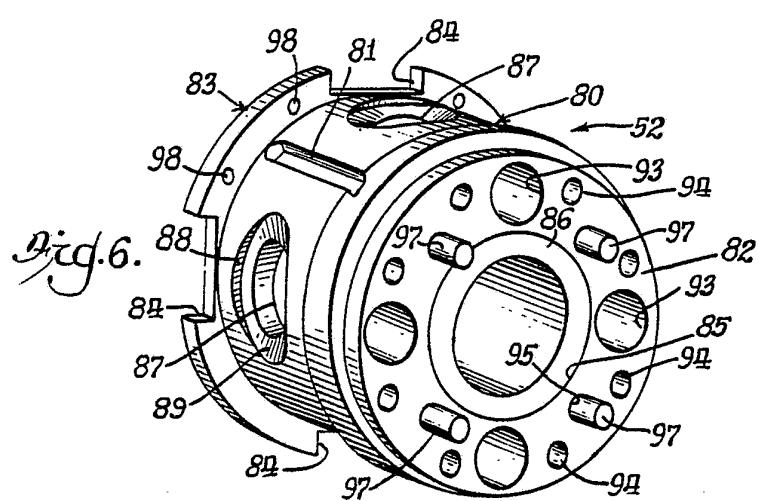
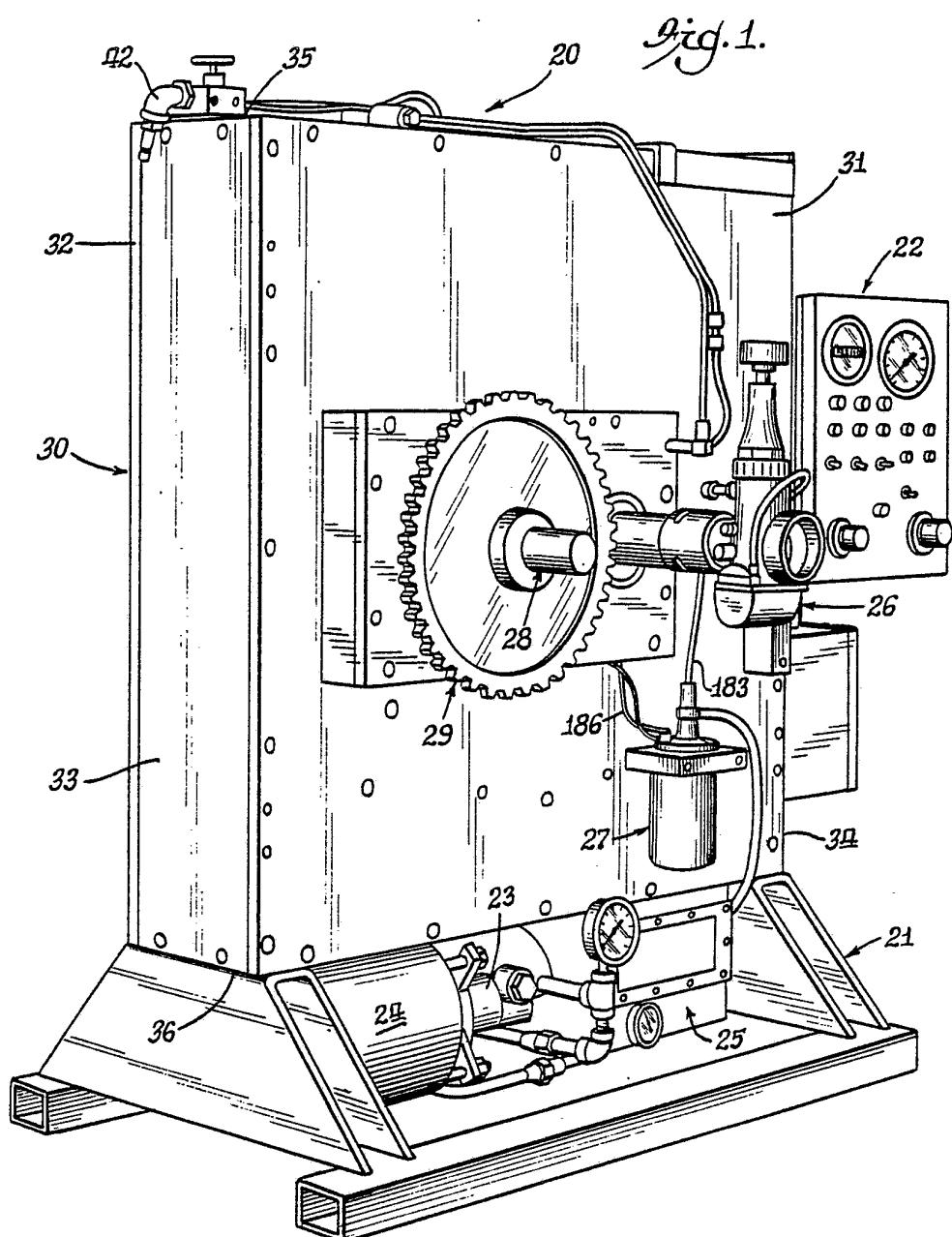
19. The rotary engine of claim 1, in which movement of each piston in each of said cylinders is rectilinearly reciprocal and is regulated in distance 35 and time according to the asymmetrical configuration

of said cam track means for each of the intake, compression, combustion, and exhaust cycles of operation thereof.

20. The rotary engine of claim 1, wherein said
5 cam means comprises a pair of radially spaced cam track
means each of which is engageable by one of a pair of
follower means associated with a said cam rider assembly
means.

21. A rotary engine assembly, comprising a
10 plurality of internal combustion four cycle rotary
engines as defined in claim 1, clustered about a cen-
tral output shaft and having the rotational axes of the
said rotors thereof parallel to said output shaft,
central drive gear means mounted on said output shaft,
15 and speed responsive clutch means coupling each of
said rotary engines to said output shaft whereby the
latter may be powered selectively by any one or more
of said engines.

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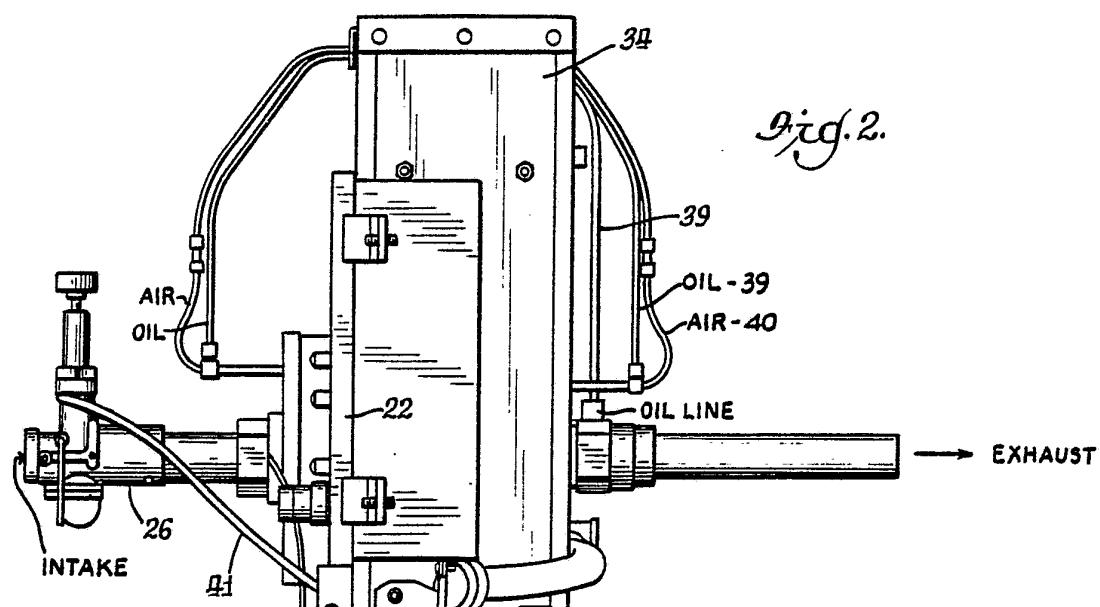


Fig. 2.

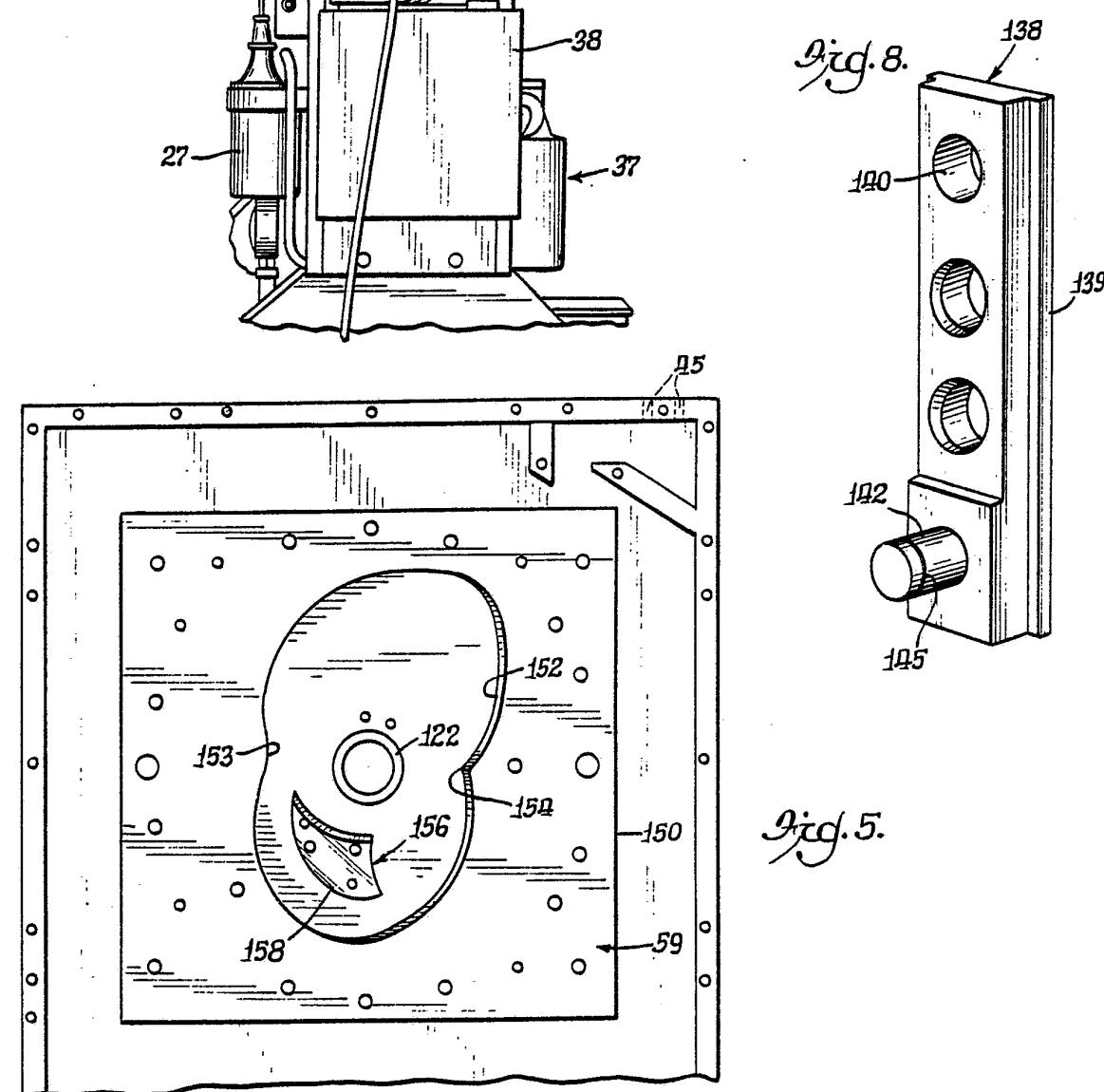


Fig. 5.

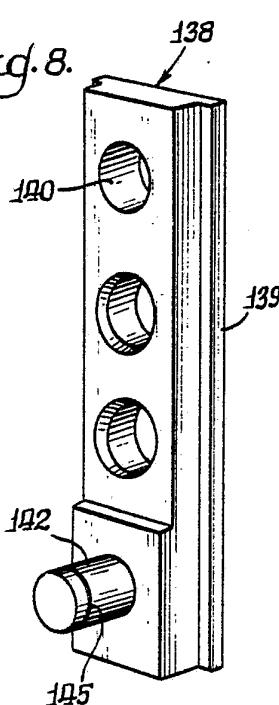


Fig. 8.

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Fig. 3.

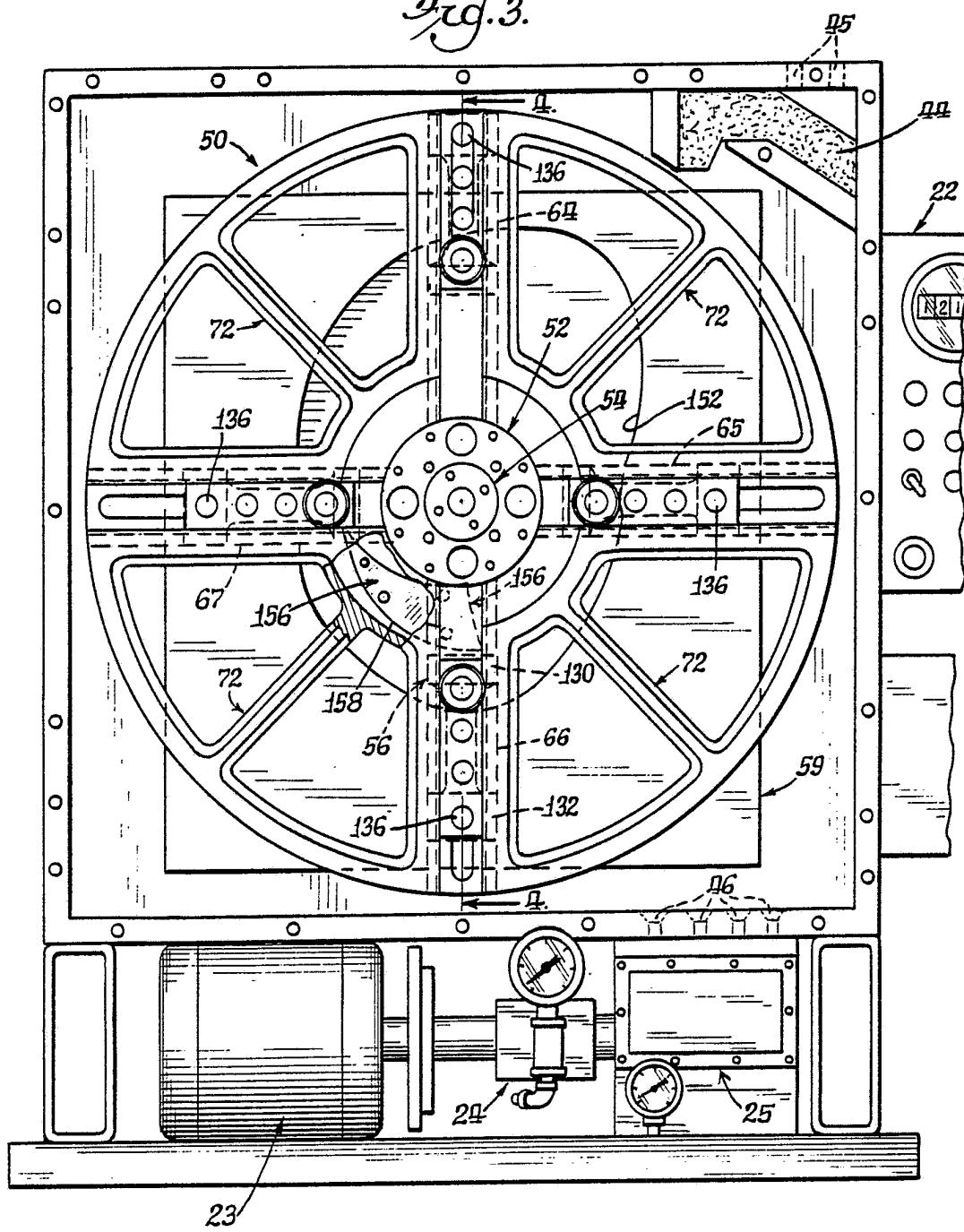
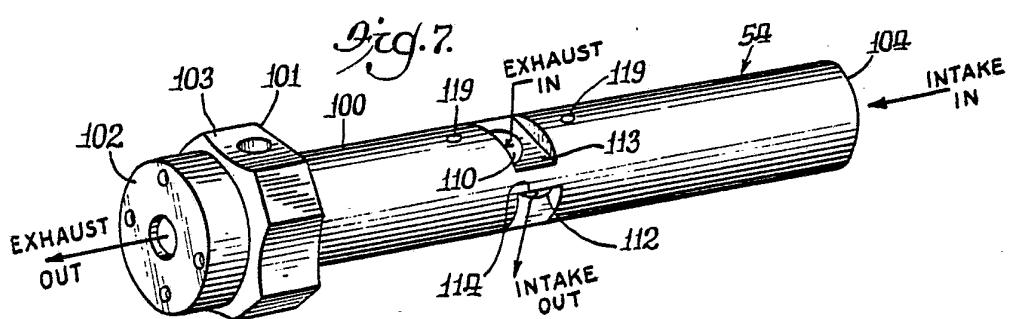
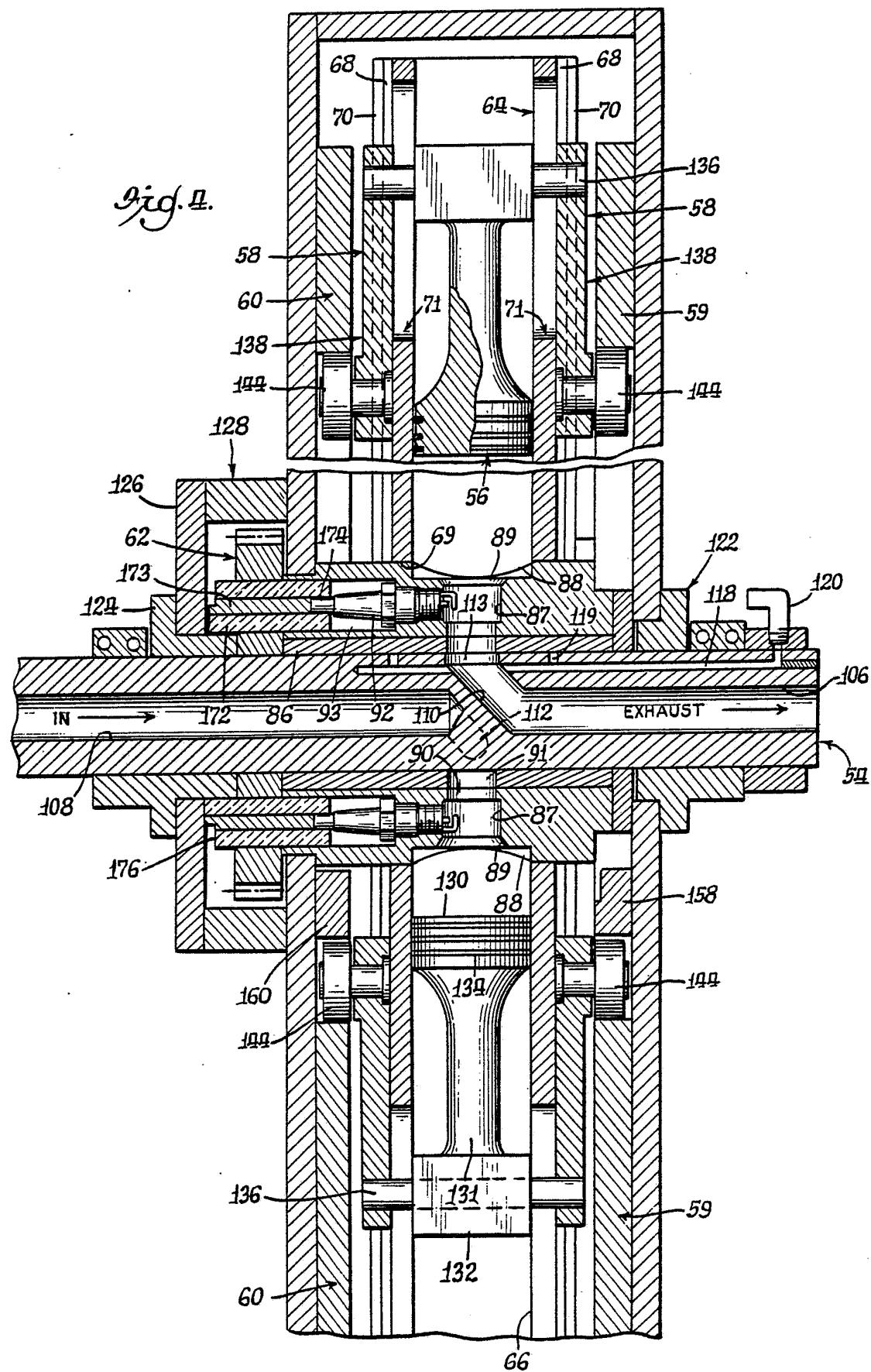


Fig. 7.



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Fig. 11.

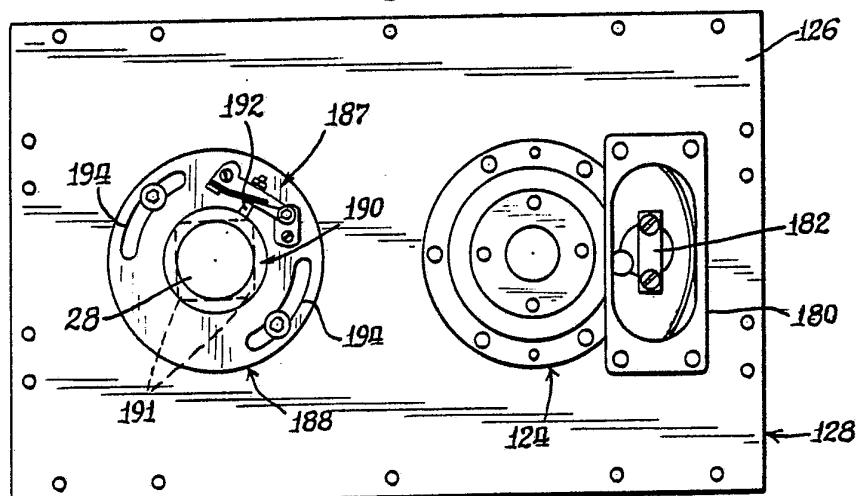


Fig. 10.

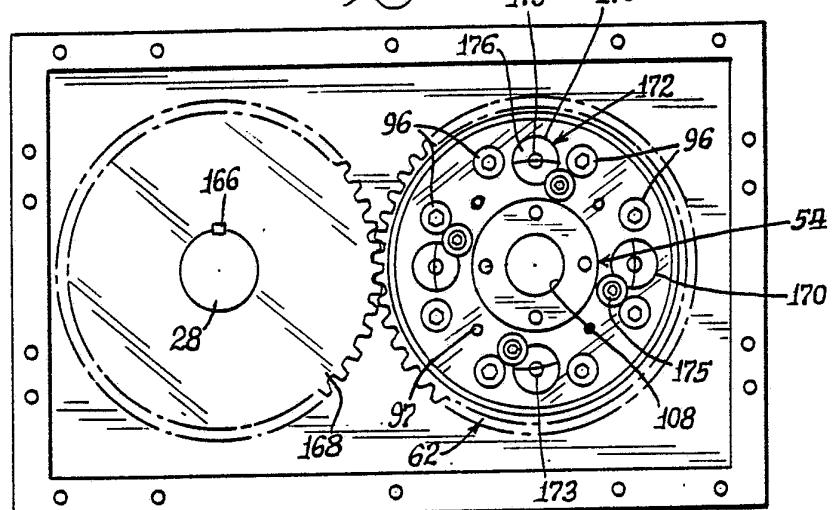
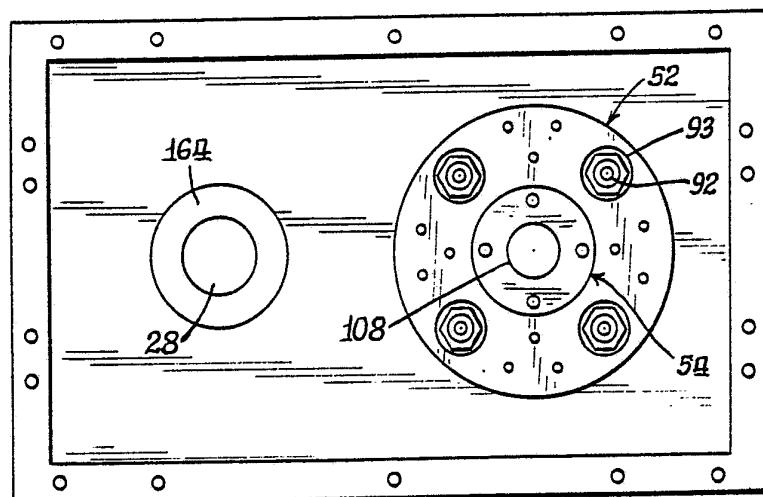
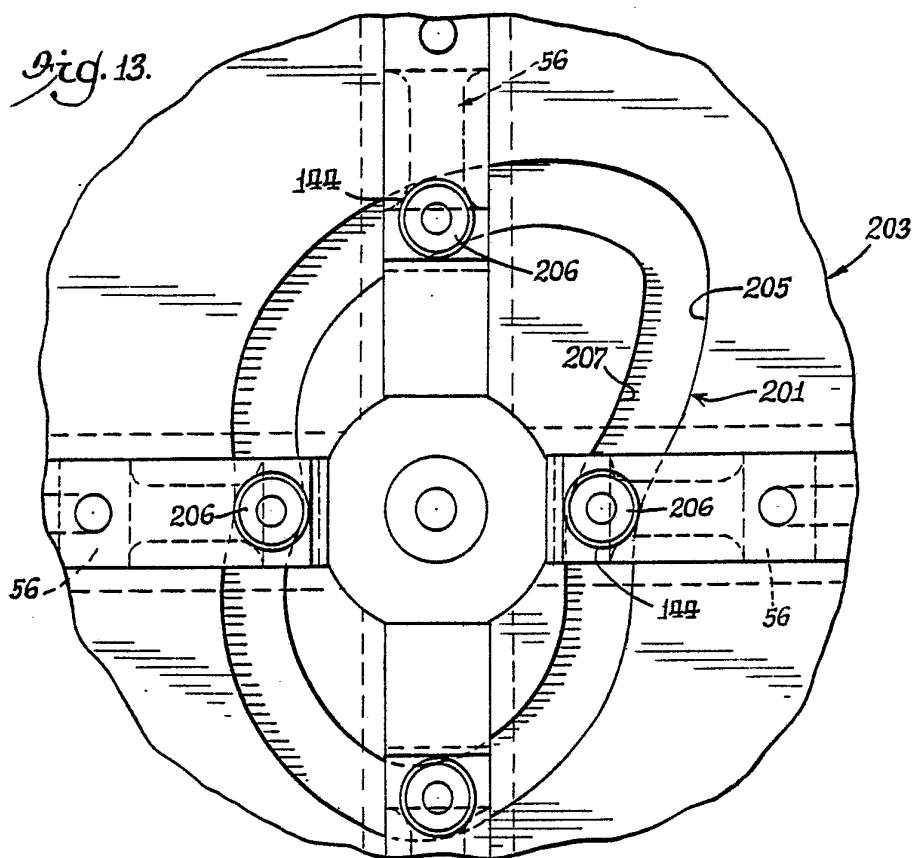
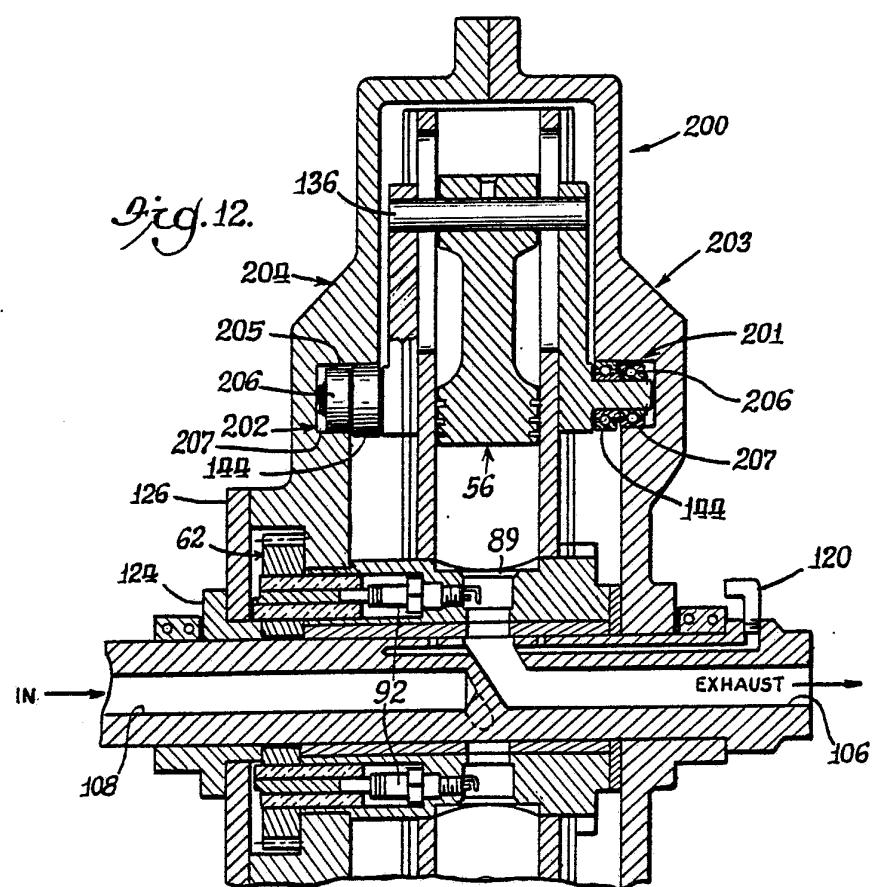


Fig. 9.



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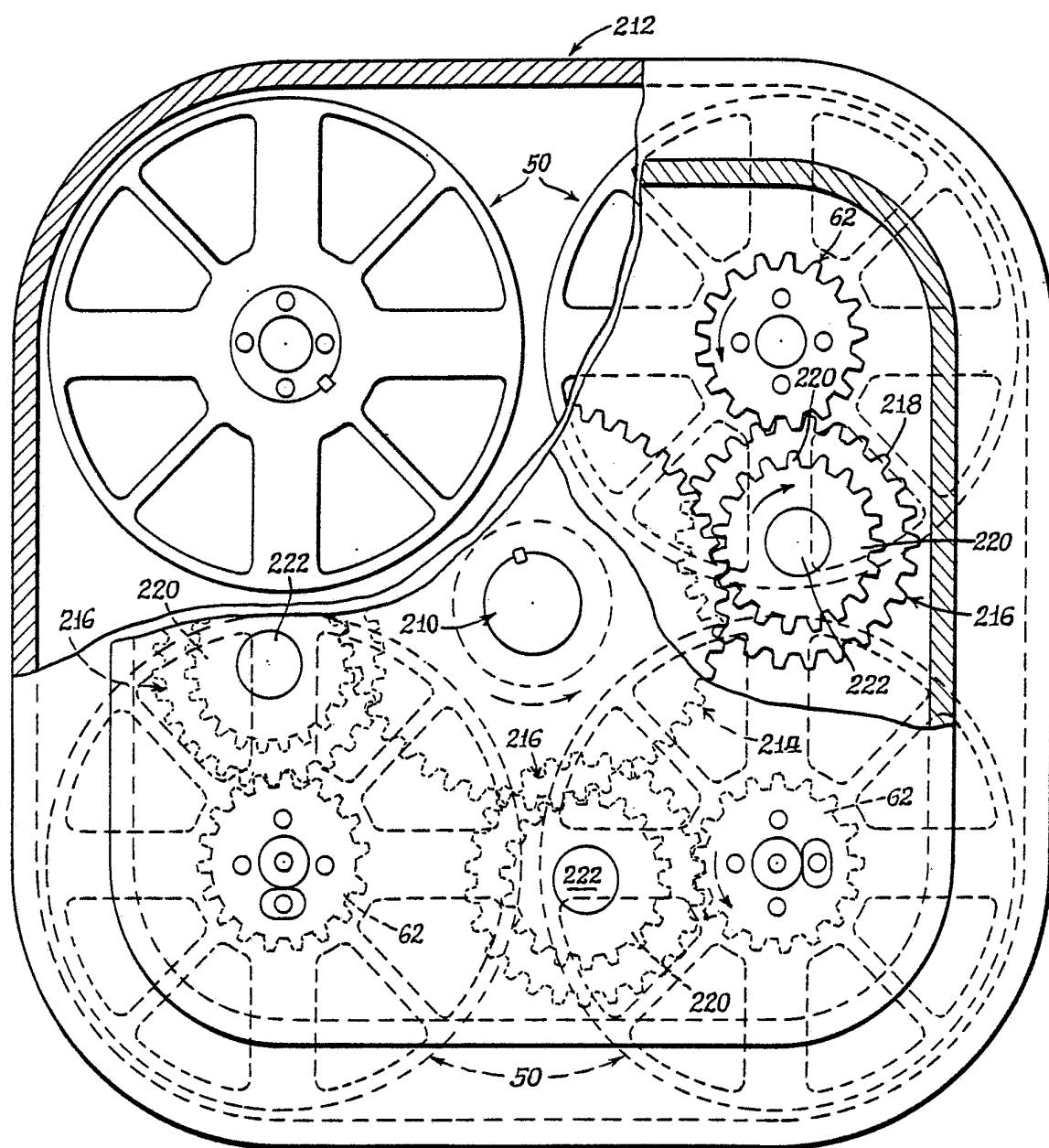


Fig. 14.

INTERNATIONAL SEARCH REPORT

International Application No PCT/US 85/00272

I. CLASSIFICATION OF SUBJECT MATTER (if several classification symbols apply, indicate all) *

According to International Patent Classification (IPC) or to both National Classification and IPC

IPC⁴ : F 02 B 57/10; F 01 B 9/06; F 01 B 13/06 ; F 02 B 41/04

II. FIELDS SEARCHED

Minimum Documentation Searched ?

Classification System	Classification Symbols
IPC ⁴	F 02 B; F 01 B

Documentation Searched other than Minimum Documentation
to the Extent that such Documents are Included in the Fields Searched *

III. DOCUMENTS CONSIDERED TO BE RELEVANT*

Category *	Citation of Document, ¹¹ with indication, where appropriate, of the relevant passages ¹²	Relevant to Claim No. ¹³
X, Y	US, A, 1853563 (HUNTERFORD) 12 April 1932 see figures 1,3; page 1, lines 1-82; 1-21 page 3, lines 2-40 --	
Y	GB, A, 2075594 (CROCKER) 18 November 1981 see figures 2,3; page 2, lines 1-130 --	1-9
A	FR, A, 987175 (ELETTRO) 9 August 1951 see figure 1; page 2, lines 1-36 --	1-9
A	FR, E, 9077 (CONILL) 22 August 1908 see figures 1,2,4; page 1, lines 1-28 -----	1-21

* Special categories of cited documents: ¹⁰

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- "O" document referring to an oral disclosure, use, exhibition or other means
- "P" document published prior to the international filing date but later than the priority date claimed

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step

"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art.

"&" document member of the same patent family

IV. CERTIFICATION

Date of the Actual Completion of the International Search

13th May 1985

Date of Mailing of this International Search Report

10 JUIN 1985

International Searching Authority

EUROPEAN PATENT OFFICE

Signature of Authorized Officer

G.L.M. Kruidenberg

ANNEX TO THE INTERNATIONAL SEARCH REPORT ON

INTERNATIONAL APPLICATION NO.

PCT/US 8500272 (SA 8986)

This Annex lists the patent family members relating to the patent documents cited in the above-mentioned international search report. The members are as contained in the European Patent Office EDP file on 04/06/85

The European Patent Office is in no way liable for these particulars which are merely given for the purpose of information.

Patent document cited in search report	Publication date	Patent family member(s)	Publication date
US-A- 1853563		None	
GB-A- 2075594	18/11/81	FR-A- 2481744 JP-A- 57002423 DE-A- 3117707 US-A- 4381740 CA-A- 1167384	06/11/81 07/01/82 28/01/82 03/05/83 15/05/84
FR-A- 987175		None	
FR-E- 9077		None	