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2C1B 2C2B 2C6 2C7F4
2C7H 2P11A 2P19 2P1A2
2P1A8C 2P2A 2P2D
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4C 4G
F1F 1B
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37D1
F2Q 7
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GB 12
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GB 945185
GB 510679
(58) Field of search
F1B
(71) Applicant
Nizarali Nurmohamed Damji, 36 Mount Road, Hayes, Middlesex, UB3 3LJ
(72) Inventor
Nizarali Nurmohamed Damji
(74) Agents
R. R. Prentice & Co

(54) **Reciprocating piston radial engine**

(57) In an engine having at least one cylinder block which comprises a plurality of radially extending cylinders with a respective piston (130) reciprocally mounted in each of the cylinders, means for interconnecting the pistons in such a manner as to provide a substantially vibration-free engine having no flywheel comprise a crank bearing ring (117), to which connecting rods (119) for the pistons (130) are pivotally connected by pins (118). The crank bearing ring (117) is arranged to drive a crank (106) and is mounted for relative rotation with respect to the crank. Gearing (121, 122, 123, 124) is arranged between the crank (106) and the crank bearing ring (117) and the ratio of the gearing is such that the ring (117) and crank (106) rotate at the same speed but in opposite directions.

ERRATUM

SPECIFICATION No. 2 024 938 A

Page 11, line 19, *for* 25.5°. *read* 25.2°.

THE PATENT OFFICE
19th September, 1980

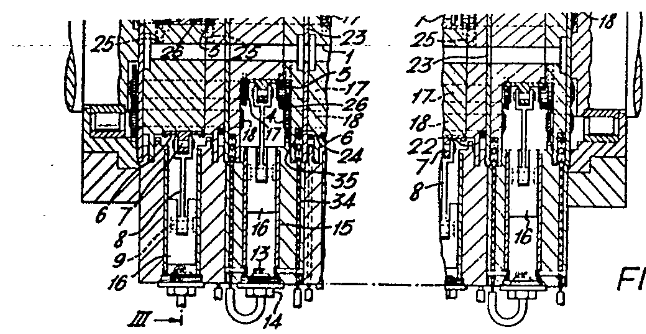
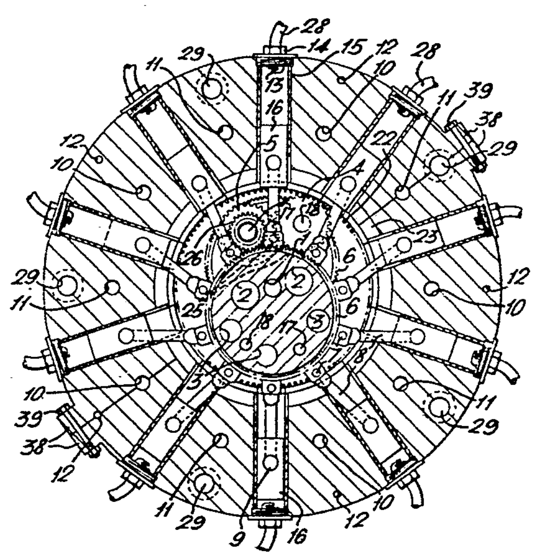


FIG.2.

FIG.3.



The drawings originally filed were informal and the print here reproduced is taken from a later filed formal copy.

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F2T 37A6A1A 37A6D2 37D1 37L
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(56) Documents cited
GB 1231701
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GB 945185
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(54) Reciprocating piston radial engine

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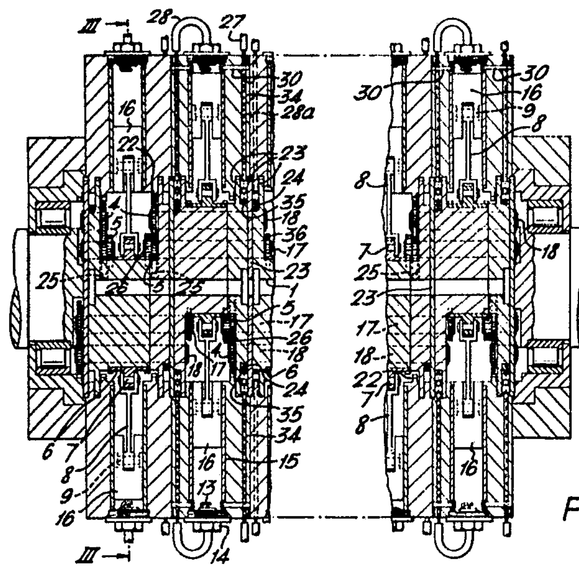
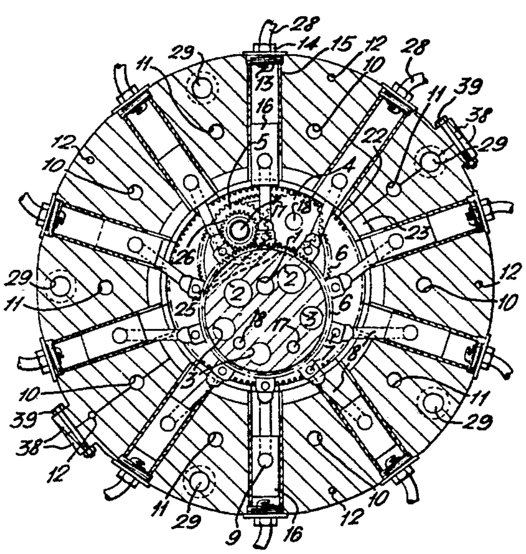


FIG.2.

FIG.3.



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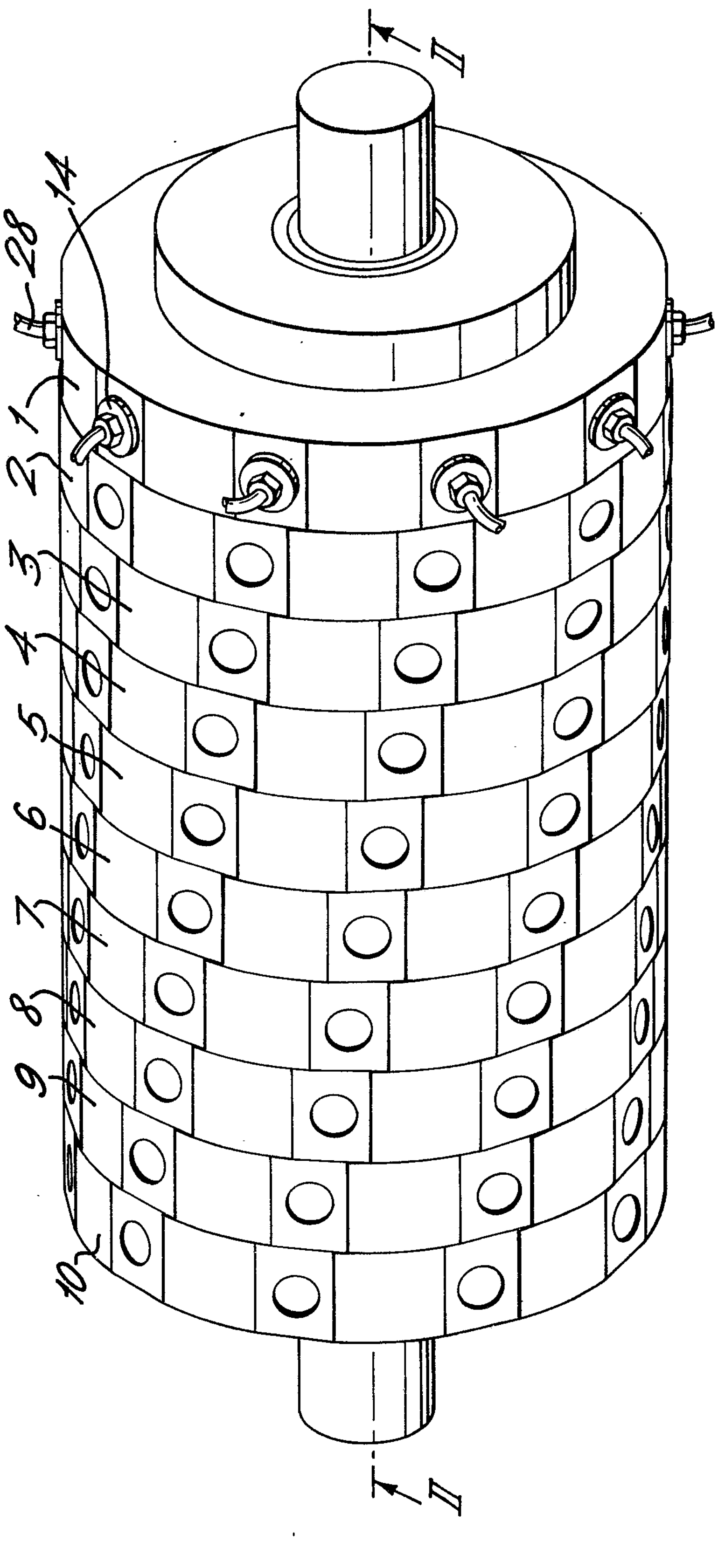

FIG.1.

FIG.2.

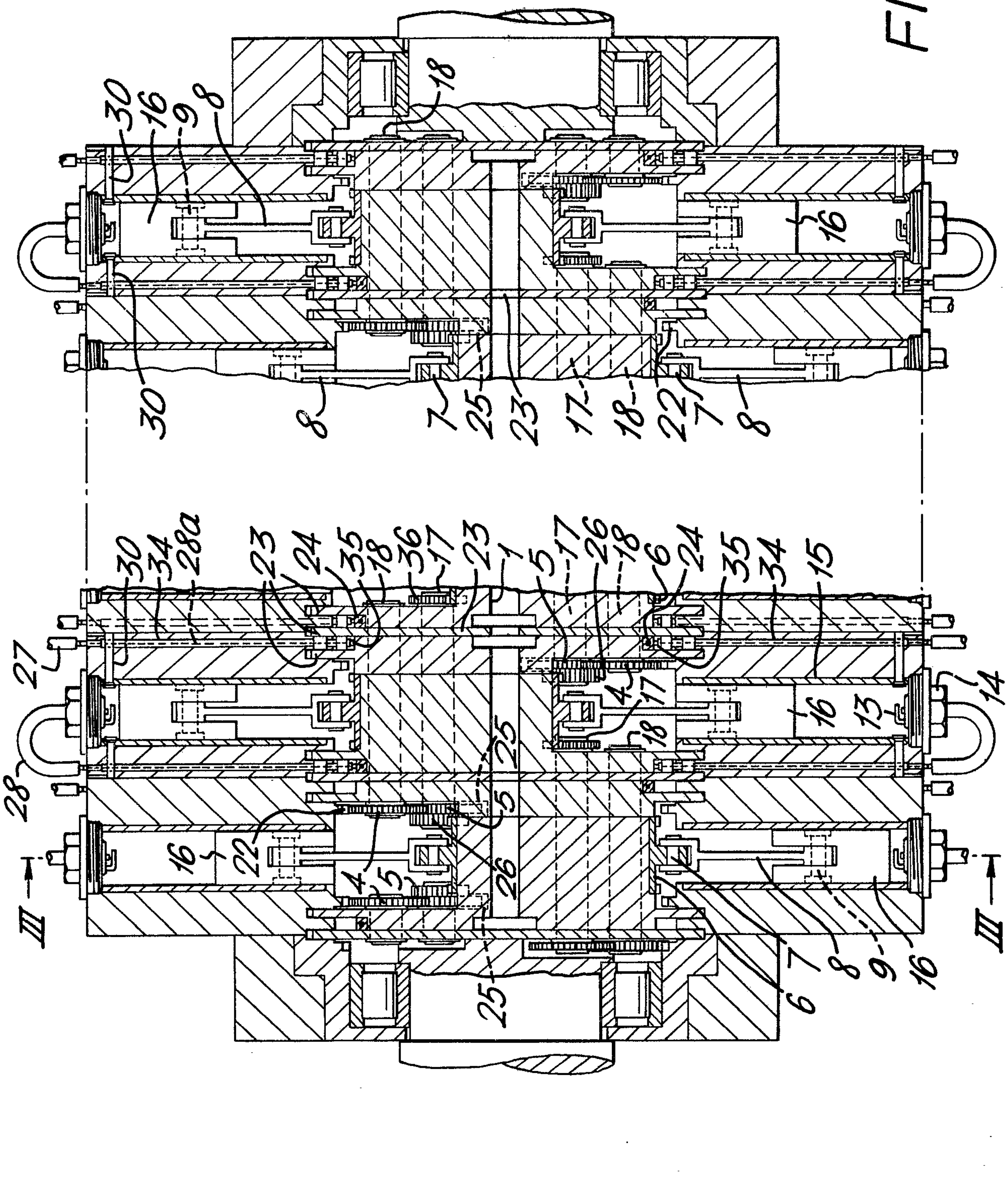

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FIG.3.

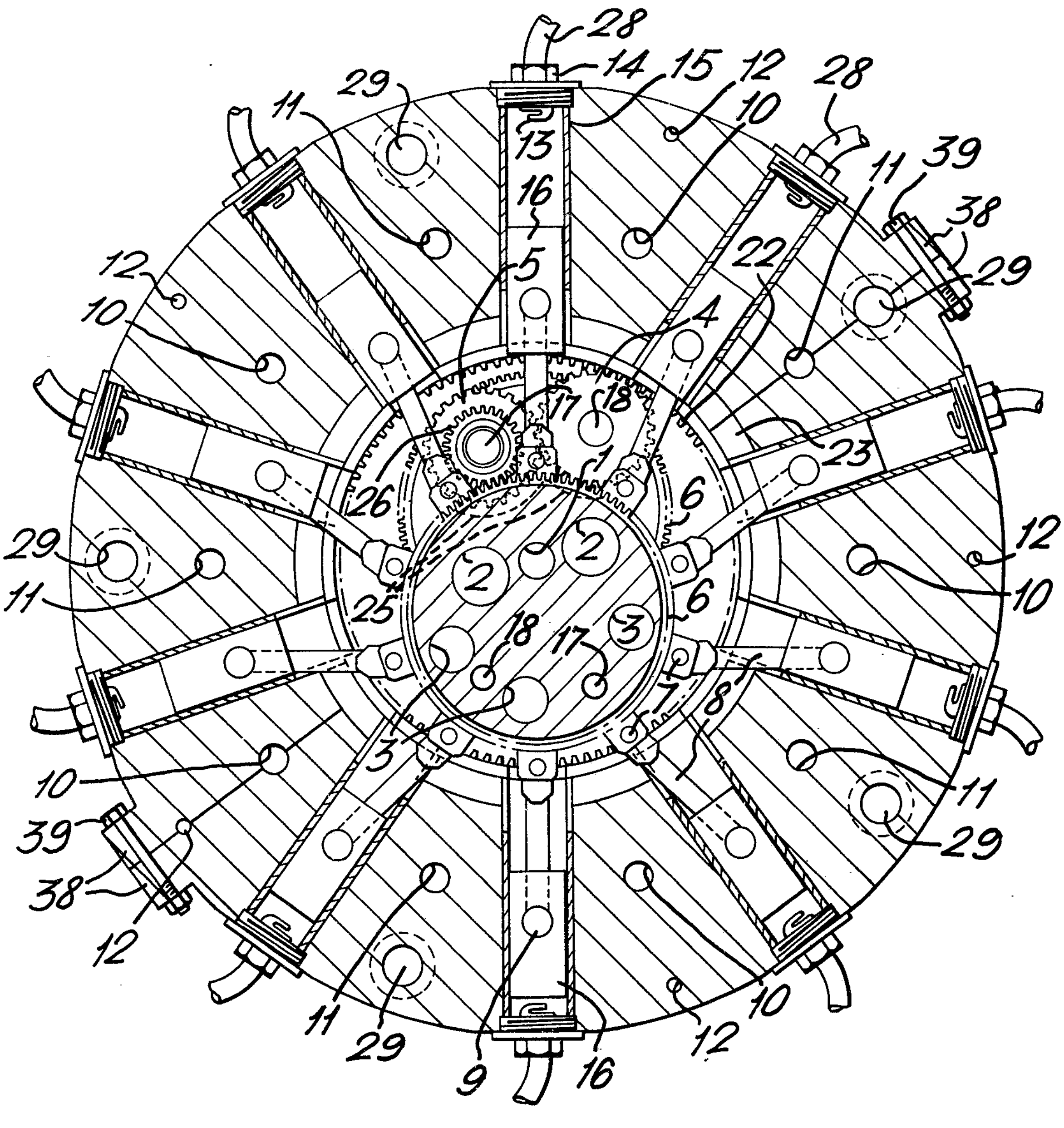
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FIG.4.

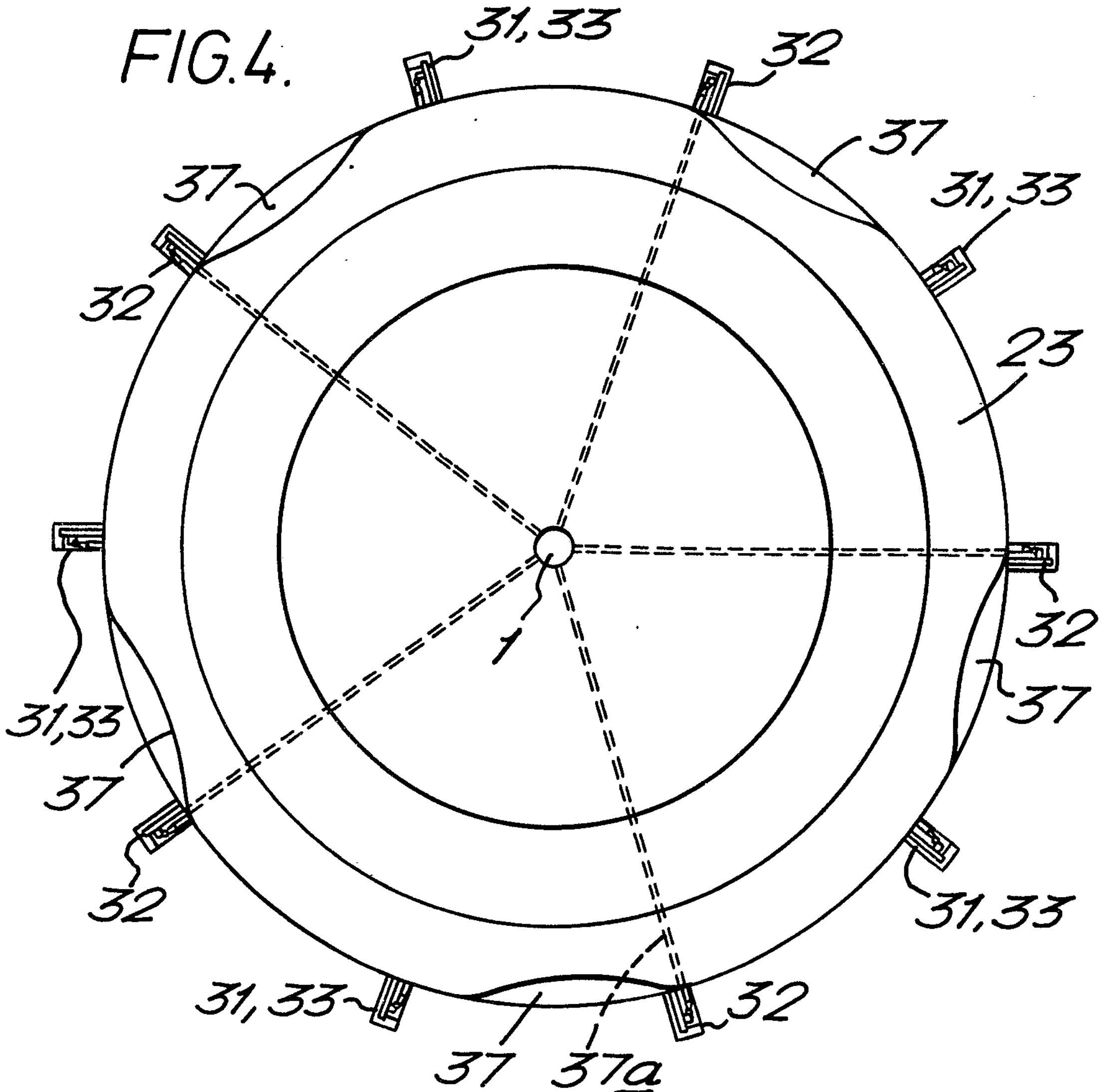

FIG.6.

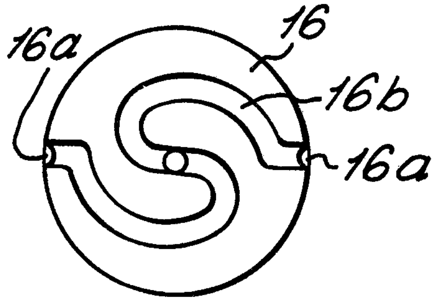

FIG.5.

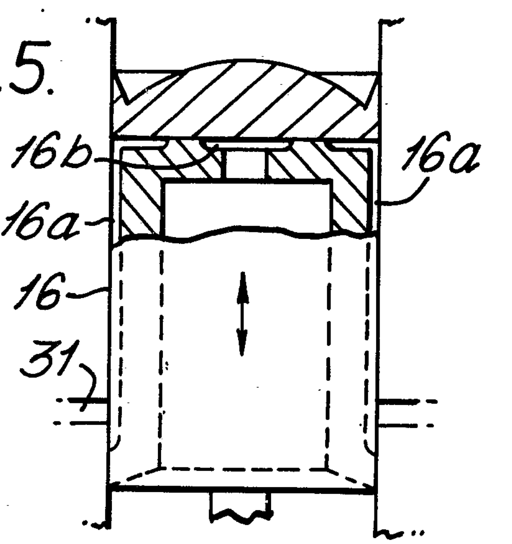

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FIG.7.

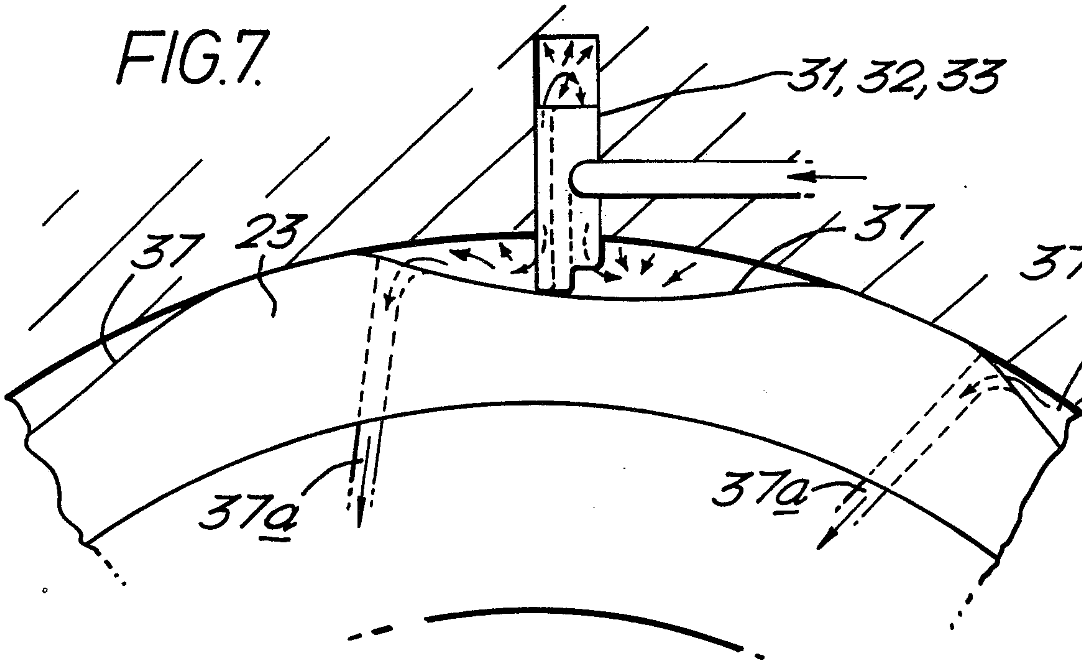
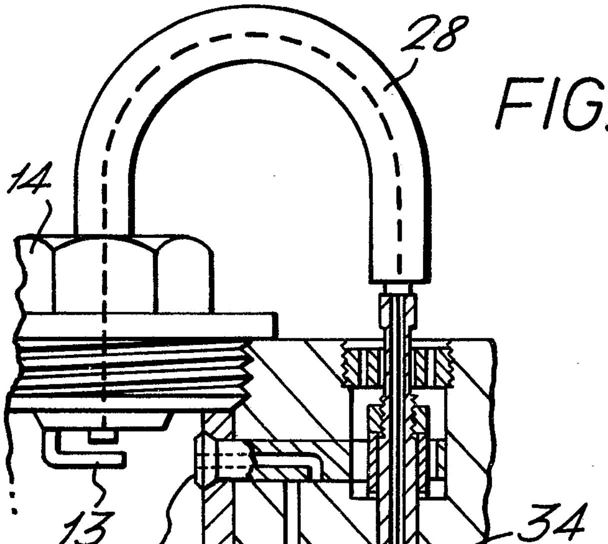


FIG.8.



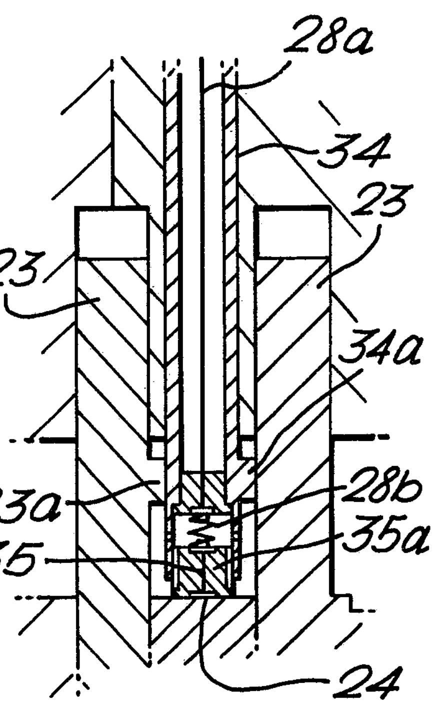


FIG.9.

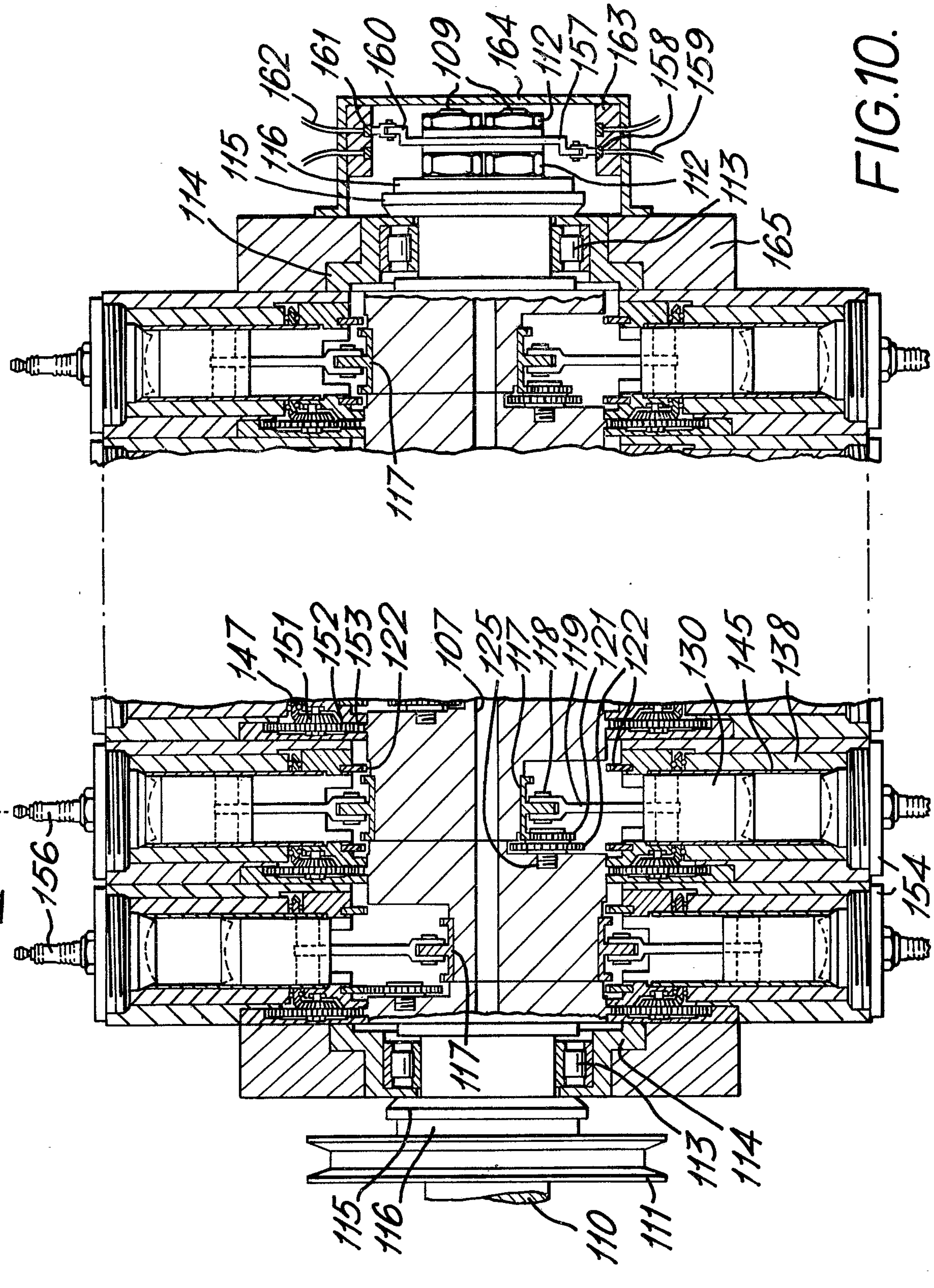

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FIG.10.

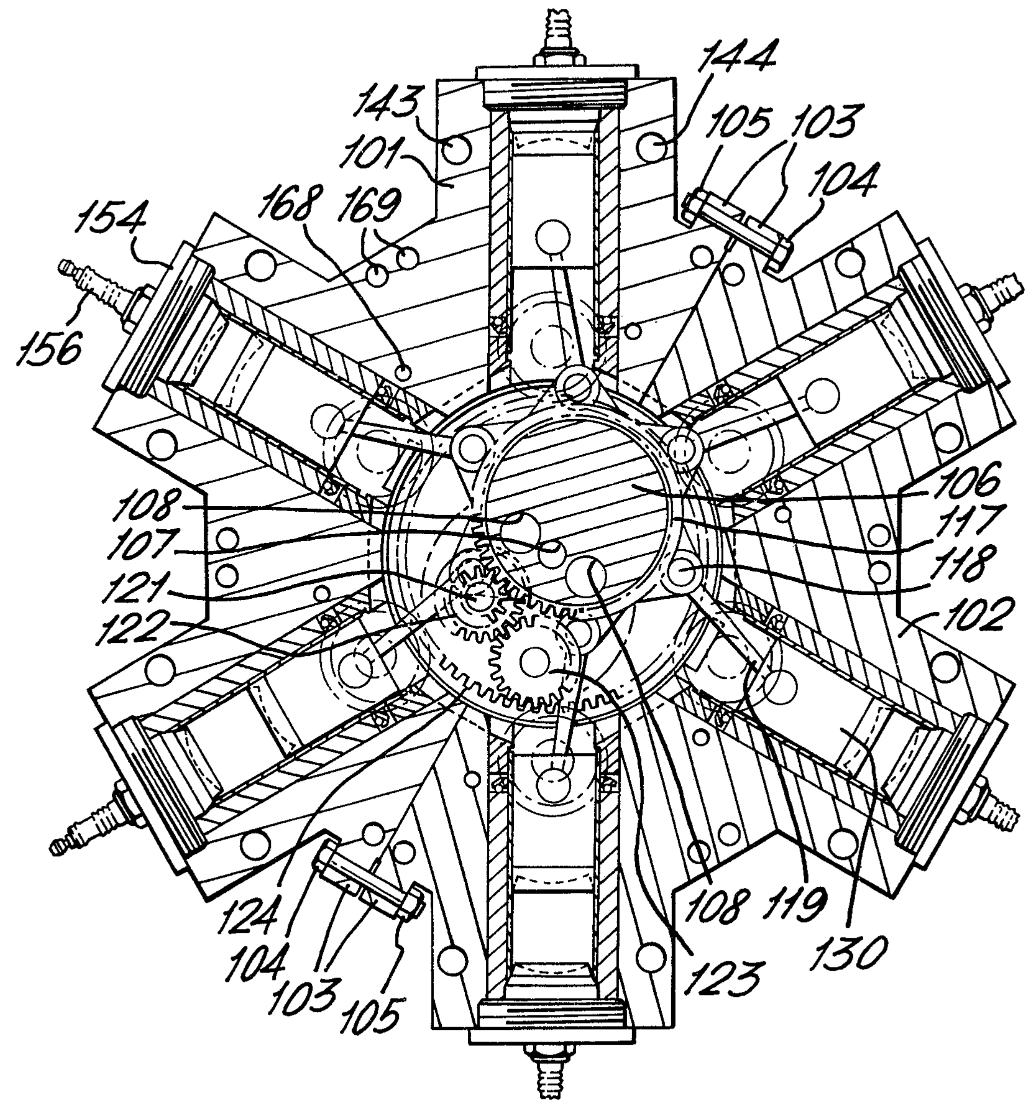
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FIG.11.

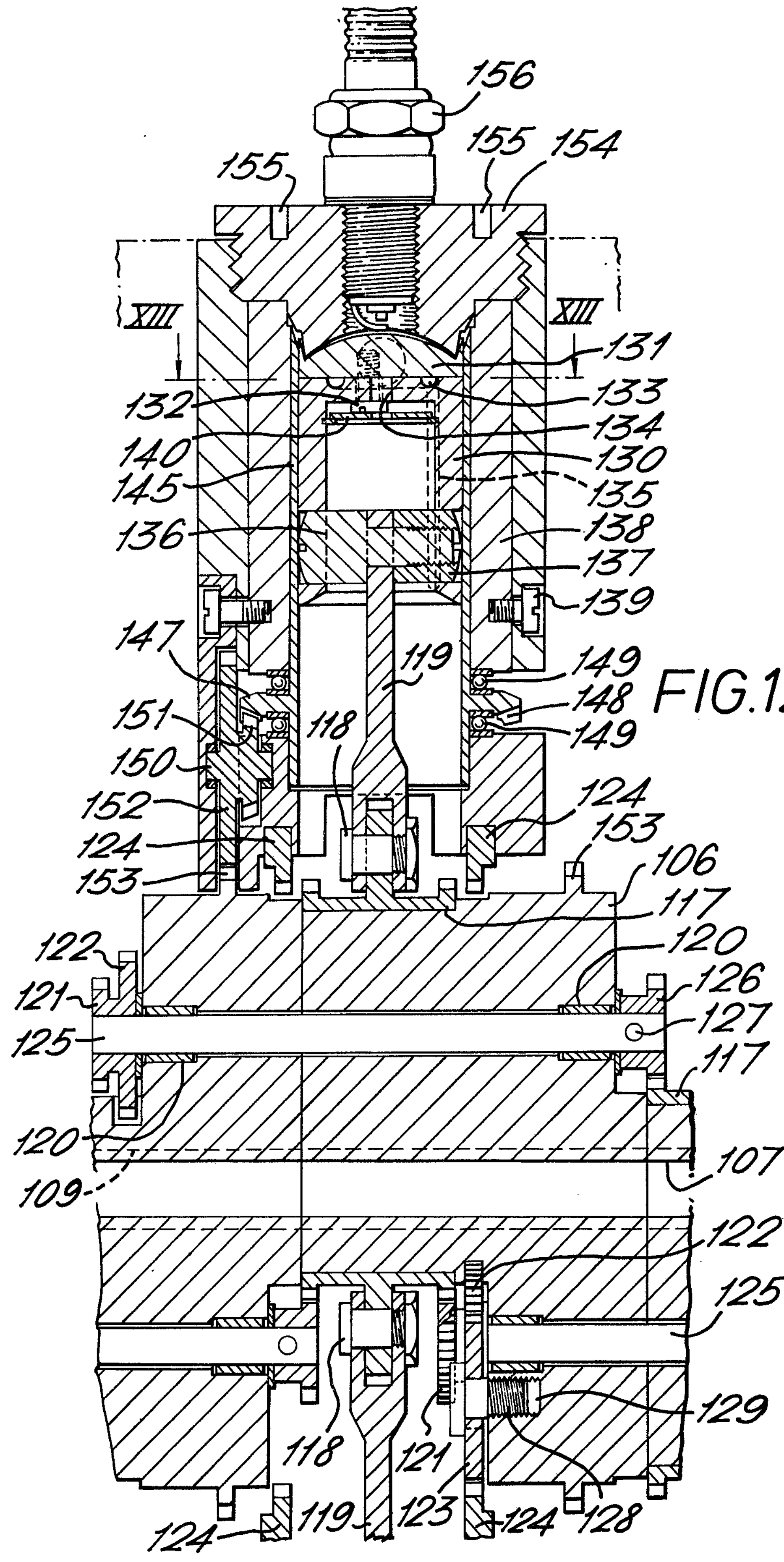
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FIG.12.

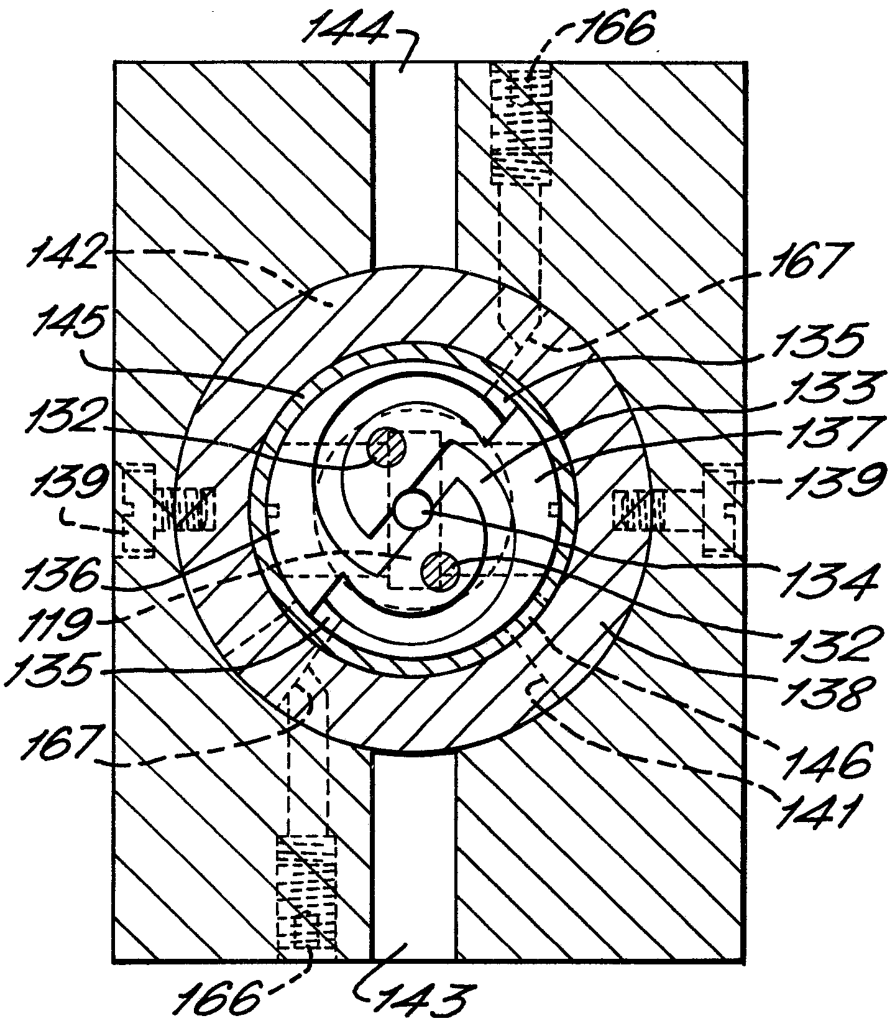
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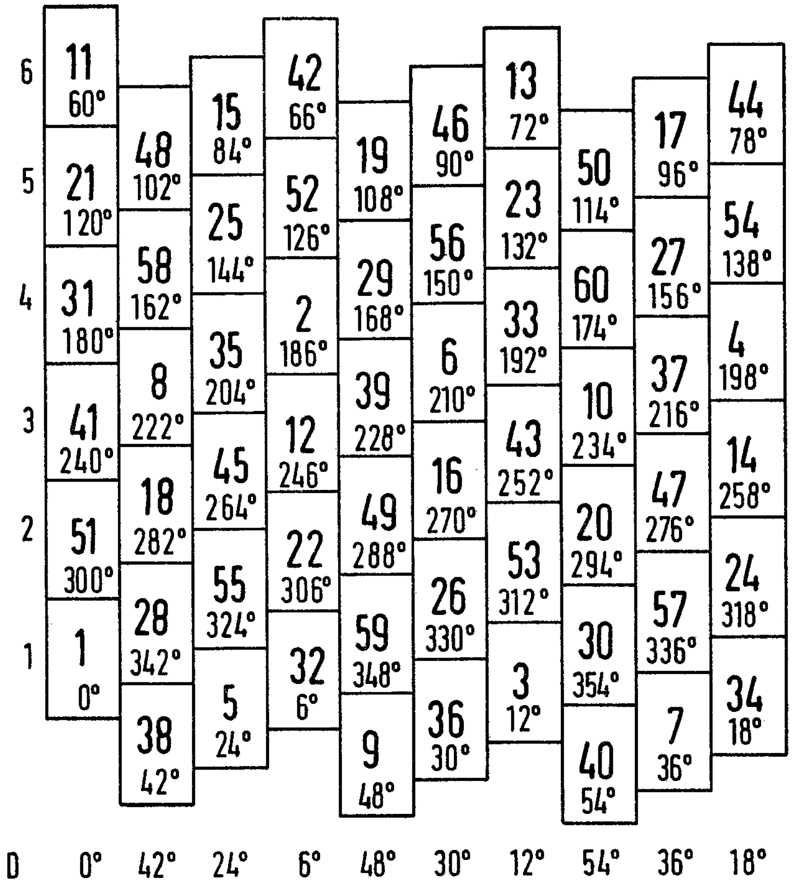
FIG.13.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51 300°
1 0°
48 102°
58 162°
8 222°
18 282°
28 342°
38 42°
15 84°
25 144°
35 204°
45 264°
55 324°
5 24°
42 66°
52 126°
2 186°
12 246°
22 306°
32 6°
19 108°
29 168°
39 228°
49 288°
59 348°
9 48°
46 90°
56 150°
6 210°
16 270°
26 330°
36 30°
13 72°
23 132°
33 192°
43 252°
53 312°
3 12°
50 114°
60 174°
10 234°
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57 336°
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34 18°
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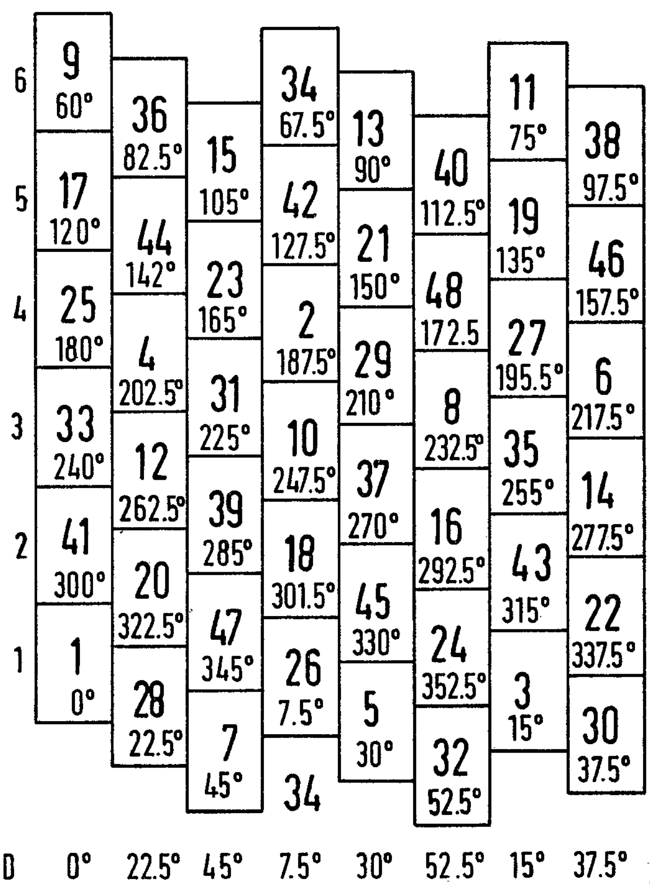
FIG.14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23 165°
31 225°
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47 345°
7 45°
34 67.5°
42 127.5°
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10 247.5°
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13 90°
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32 52.5°
11 75°
19 135°
27 195.5°
35 255°
43 315°
3 15°
38 97.5°
46 157.5°
6 217.5°
14 277.5°
22 337.5°
30 37.5°
D 0° 22.5° 45° 7.5° 30° 52.5° 15° 37.5°

FIG.15

SPECIFICATION

Improvements in engines

This invention relates to engines and is particularly concerned with radial engines.

According to the invention, there is provided an engine which comprises at least one cylinder block having a plurality of radially extending cylinders, a respective piston reciprocally mounted in each of said cylinders and means for interconnecting said pistons in such a manner as to provide a substantially vibration-free engine having no flywheel.

Preferably, each piston is pivotally connected to a respective connecting rod, the connecting rods are pivotally connected to a common crank bearing ring and the crank bearing ring is arranged to drive a crankshaft of the engine. The crank bearing ring is desirably mounted on a crank of the crankshaft for rotation relative to said crank. A plain sliding bearing may be provided between the crank and the crank bearing ring.

According to a preferred embodiment of the invention, the crank bearing ring is provided with gear teeth engageable with a first gear, the first gear and a second gear are mounted on a common shaft rotatably mounted in the crank, the first and second gears being secured together or formed integrally for common rotation, the second gear is engageable with a third gear fixedly mounted on the crank and the third gear is engageable with an internally toothed fixed gear mounted on the engine casing. The ratios of the gears are desirably such that the crank and the crank bearing ring are arranged to rotate at the same speed but in opposite directions.

Each piston may be provided in its upper region adjacent its crown with a channel for the passage of cooling fluid. Each piston may further be provided with a crown which is detachably secured to the top of the piston, the channel being formed in the top of the piston to form, with the underneath surface of the piston crown, a closed passage.

Each cylinder is desirably provided with a rotary sleeve valve, the associated piston being reciprocally mounted in the sleeve valve and said sleeve valve being provided with a port to control induction and exhaust of gases into and out of the cylinder.

A plurality of cylinder blocks may be provided which are secured together in a staggered angular relationship.

The invention will now be further described, by way of example, with reference to the drawings, in which:—

Figure 1 is a diagrammatic perspective view of one embodiment of an engine according to the invention;

Figure 2 is a section taken on the line II—II in Figure 1 in the direction of the arrows;

Figure 3 is a section taken on the line III—III in Figure 2 in the direction of the arrows;

Figure 4 is an elevation of a rotary pump forming part of the engine shown in Figures 1 to 3;

Figure 5 is a longitudinal section through one embodiment of a piston for an engine according to the invention;

Figure 6 shows a detail of the piston shown in Figure 5;

Figure 7 shows part of a rotary pump wheel forming part of an engine according to the invention;

Figure 8 shows, diagrammatically, one embodiment of a valve and tappet assembly for use in an engine according to the invention;

Figure 9 shows a detail of part of an ignition circuit for one of the cylinders of an engine according to the invention;

Figure 10 is a longitudinal section, corresponding to Figure 2, through an engine according to a second embodiment of the invention;

Figure 11 is a section taken on the line XI—XI in Figure 10 in the direction of the arrows;

Figure 12 is a cross-section through part of the engine shown in Figures 10 and 11 of the drawings on an enlarged scale;

Figure 13 is a section taken on the line XIII—XIII in Figure 12.

Figure 14 is a table depicting a scattered cylinder firing sequence according to one example for an engine according to the invention; and

Figure 15 is a table, similar to Figure 14, but showing the scattered cylinder firing sequence for an engine having a different number of cylinder blocks.

Reference will first be made to Figures 2 and 3 of the drawings in which the engine has a crank shaft centre bore 1 which extends in the centre through the entire length of the crank shaft. It carries a fluid which lubricates and cools the engine. The fluid enters this bore through recesses 25 in the cranks of the crank shaft. The cranks are made of three parts and there is a network of channels or further recesses where these parts meet which recesses draw fluid from the engine body and flood the internal cavity of the engine all around the crank shaft. This fluid cools the sides of the combustion chambers and lubricates the moving parts of the crank shaft including the pistons. It is then carried to radiators for cooling through the bore 1.

There are two large bores 2 in the central proximity of the crank shaft through which pass precisely fitting rods for positioning the cranks. The bores and the rods run through the entire length of the crank shaft. High precision is necessary in the bores and rods for positioning the cranks and to maintain the exact accuracy of the crank parts for the formation of the cranks. Slightest play or slackness of the bores to the rods is likely to cause malfunction of the pistons (and hence the engine) through malformation of the cranks.

Further bores 3 are provided in the crank parts through which are fitted bolts to hold adjacent crank parts together. The bolts are locked against loosening in order to avoid possible interference with the moving parts of the crank shaft and hence possible damage to the engine.

A first gear wheel 4 having, for example, 25 teeth is so arranged in the crank that it meshes with a fixed gear 22 which has, for example, 100 teeth. This gives the ratio of first gear wheel to fixed gear of 1:4. The radii of these two gears are also in the same ratio of 1:4. This means that when the crankshaft rotates once, the first gear wheel rotates four times in relation to the fixed gear 22. Again high precision is necessary if distortions such as torsional torque and vibrational disturbances are to be totally eliminated or reduced to negligible amounts. Play and slackness in the gears will give rise to malfunctioning of the pistons. Once accuracy is achieved only wear of these parts will be responsible for malfunctioning of these parts and hence the engine. If these moving parts are perfectly engineered so that they move with great ease without being slack, wear will be slow and negligible particularly when properly lubricated with cooling fluid.

The first gear wheel 4 also meshes with a second double gear wheel 5 which is identical to the gear wheel 4 and has a smaller gear wheel 26 fixed to it. The gear wheel 26 has the same number of teeth as the gear wheels 4 and 5 but it has a smaller radius. Said gear wheel 26 meshes with a toothed bearing ring 6 and has the same ratio with regard to the number of teeth on the ring 6 as the ratio of the gear wheels 4 and 5 to the fixed gear 22, i.e. 1:4. The toothed ring 6 thus has the same number of teeth as the fixed gear 22. Since the gear wheel 5 is identical to the gear wheel 4 and meshes with it, it turns round as many times as the gear wheel 4 upon the turning of the crankshaft. When the crankshaft rotates once, the first gear wheel 4 rotates four times and the second double gear wheel rotates four times so that the small gear wheel 26 also rotates four times. Since the gear wheel 26 and ring 6 are in the ratio 1:4, the ring 6 rotates once for every rotation of the crankshaft. The toothed ring 6 serves as a crank bearing ring as will be hereinafter described.

The crankshaft is provided throughout its length with a series of grooves or recesses 25 which serve to accommodate the span of the moving crank gear wheels 4 and 5, and for easy access of lubricating fluid to the crankshaft centre bore 1. Each gear wheel 5 and associated gear wheel 26 is mounted on one end of a shaft 17 which passes through aligned bores in two adjacent cranks and carries at its other end a further smaller gear wheel 36 which meshes with the crank bearing ring 6 of the adjacent crank assembly. Each gear wheel 4 is mounted on one end of a shaft 18 which also passes through further aligned bores in two adjacent cranks.

Let us imagine that the crankshaft turns clockwise once in relation to the fixed gear 22. Then the first gear wheel 4 rotates anti-clockwise four times in relation to the crankshaft and the second double gear wheel 5 rotates clockwise four times in relation to the crankshaft and the crank bearing ring 6 rotates anti-clockwise once in relation to the crankshaft. What this means is that when the crankshaft rotates clockwise once in relation to the fixed gear 22, the crank bearing ring 6 rotates anti-clockwise once in relation to the crankshaft itself although of course being mounted on the crankshaft it also rotates with said crankshaft. The ring 6 does not therefore turn in any angular direction in relation to the fixed gear 22. It turns in the opposite angular direction to the crankshaft only and at the same rate as the crankshaft. It always points in the same angular direction as the fixed gear 22 just as the earth tilted at $66\frac{1}{2}°$ to the plane of its orbit round the sun, always points in the same direction in relation to the solar system or the pedal of a bicycle always points in the same direction in relation to the frame. The earth, the bicycle pedal and the bearing ring 6 have one thing in common: they all move round a centre and always point in the same direction throughout at any given instance. This sort of arrangement for the bearing ring 6 to point in the same direction is necessary in order to overcome the coupling effect by the piston thrusts round the crank bearing as hereinafter described.

The crank bearing ring 6 may be of the slide bearing type which turns on its bearing opposite in direction to the crankshaft. The locus of a point on this bearing is a circle having a radius equal to the distance between the centre of the crank and the centre of the crankshaft. This locus is the same as that which a piston rod will make at the point of its attachment to the crank bearing ring 6. The diameter of the locus is equal to the length of the piston stroke in the cylinder.

Each piston rod 8 is attached to the crank bearing ring 6 by a respective crank-rod pin 7. In the embodiment of the engine illustrated in Figs. 1 to 3 of the drawings, there are 100 piston and cylinder assemblies arranged in ten banks of ten as illustrated in Fig. 1. Thus, there are 10 crank-rod pins 7 in each crank bearing gear ring 6, said pins being set apart at 36° from the centre of the crank. As has been mentioned before, the centre of each crank-rod pin 7, which is the point at which the rod is attached to the crank, makes a circle as locus with a diameter equal to the travel of the pistons 16 in their respective cylinders which pistons are secured to respective piston rods 8 at the ends remote from the associated crank-rod pins 7 by means of respective gudgeon pins 9.

A plurality of bores 10 are provided in the engine body for conveying cooled fluid from reservoirs (radiators) to the internal parts of the engine for cooling and lubricating. The fluid bores are inclined to the longitudinal axis of the engine and are formed by bores in the adjacent ten cylinder blocks, which blocks have a radial displacement of 3.6° with respect to adjacent blocks. Recesses or channels (not shown) extend from each of the fluid bores 10 on each side of each cylinder block.

A plurality of exhaust bores 11 similar to the bores 10 are also provided in the cylinder blocks for conveying exhaust gases from the combustion chambers via recesses or channels to an exhaust pipe system (not shown).

Further, a plurality of induction or fuel mixture bores 12, similar to the bores 10 and 11, are also provided in the cylinder blocks for carrying a combustible fuel mixture from a mixing unit to the combustion chambers through valves 30 via respective recesses or channels (not shown) provided in one side of each of the cylinder blocks.

Each cylinder is provided in the region of its combustion chamber with a spark plug or other suitable fuel igniter 13. Each spark plug or other igniter 13 is preferably secured in place by screwing it into a respective bush 14 which in turn is screwed into the associated cylinder.

Each bush 14 serves to secure a cylinder sleeve 15 in position in the associated cylinder block, each sleeve being smooth lined inside to allow easy movement of the associated piston 16. High precision is necessary in order to check leaks. Towards the crank end, each sleeve is provided with a pair of slots to allow piston rod swing. A recess is provided all round each of the sleeves on the outer circumference thereof for the purpose of conveying cooling fluid, which fluid also serves to lubricate the pistons.

A preferred embodiment of piston is shown in Figures 5 and 6 in which a channel 16*a* is provided in the skirt at each side and one of said channels is aligned with a recess or channel in one side of the associated cylinder block and the recess in the associated cylinder sleeve 15. The underneath or interior surface of the piston crown is provided with a zig-zag formation channel 16*b* communicating at each end with the channels 16*a* as shown in Figure 6. The effective flow length of the channels 16*a* varies as the piston moves to and fro but does not affect the flow itself. The other channel 16*a* communicates with a channel or recess in the other side of the associated cylinder block from which fluid is sucked by a rotary pump wheel 23 through respective rotary valves 31 (Fig. 4).

The channels or recesses extending from the fluid bores 10 serve as cylinder cooling channels on one side of each cylinder block and each said channel or recess on this side of the cylinder block extends from a fluid bore 10 to one of the associated cylinder bores, meets a recess on the outside of the bush 14, enters the recess on the outer circumference of the cylinder sleeve 15, goes round it in a spiral, and then extends to a rotary valve 33 controlled by another rotary pump wheel 23. Further channels or recesses run on both sides of the associated cylinder blocks from the fluid bores 10 to rotary valves 32 which are controlled by a further rotary pump wheel. In this case, each valve 32 is common to two adjacent cylinder blocks.

The rotary pump wheels are provided on the cranks of the crankshaft, there being three of these wheels to every crank. On the circumference of each rotary pump wheel are five slight and gradual depressions 37 (Fig. 7) and the wheel fits into a respective slot all round it in the associated cylinder block. Five valves 31 set into one of the outer slots engage the circumference of a first wheel 23 and draw cooling fluid from the channels or recesses in said cylinder block into the internal cavity of the engine all around the crankshaft. Five valves 33 are set into the other of the outer slots and engage the circumference of another wheel 23 to draw cooling fluid from the channels or recesses in the adjacent cylinder block into the internal cavity of the engine. Five valves 32 are set into the centre slot and engage the circumference of the centre wheel 23 to draw cooling fluid from the channels or recesses formed between the two adjacent cylinder blocks into the internal cavity of the engine. Each rotary pump wheel 23 is provided with bores or passages 37*a* for conveying the fluid from the depressions 37 into the bore 1.

The gap between two of the wheels 23 accommodates tappet rods 34 and points brushes 35. At the base of the gaps between wheels 23 are power points 24 for distributing voltage supply. Ignition voltages are picked up from these points by points brushes 35 (see Figure 8) in the tappet rod 34 as the crankshaft rotates. Each of the cylinder blocks has a built-in generator (not shown) which is driven by the moving crank gear wheels 4 and 5 and is attached to the gear wheel 5. Currents are produced in the generator by the known moving coil arrangement. Two eddy currents are produced, one going direct to a base point 24 of the crankshaft where it is picked up by the points brushes 35 and then to batteries (not shown) via leads 27 and suitable cut-out devices while the other goes through a series of transistors and capacitors situated within the crank to another base point 24 of the crankshaft where it is picked up by a points brush 35 and delivered through the associated tappet rod 34 to the associated spark plug or other igniter 13 via a high tension lead 28.

The batteries may activate the moving coils and turn the crank gear wheels 36, 26, 5. The gear wheel 5 will then turn the gear wheel 4 which in turn will turn the crankshaft. When the batteries activate the moving coils, eddy currents are induced in the fixed coils which supply the charge to the spark plugs or other igniters 13. Once the engine is running, the current will charge the batteries.

Each cylinder is provided with a pair of tappet valves 30 — one for inlet and the other for exhaust. The valves are situated on opposite sides of the piston bore and are moved by associated tappet rods 34. Each valve has gear teeth at one end which mesh with gear teeth on the associated tappet timing rod. The oscillation of the tappet timing rod moves the tappet valve in and out. As shown in Figure 8, the valve has a tiny bore which opens and closes as the valve is moved in and out. When the valve is moved in towards the cylinder, the valve head is open and the bore is open allowing fuel mixture in or exhaust gases out. When the valve is moved out away from the cylinder, the valve head is closed and so is the bore. Each valve member is mounted for movement in a bore in the cylinder block and a small orifice may be provided at the end of the bore to permit the passage of air therethrough. As the valve moves in, the vacant space in the bore is occupied by air through this orifice and as the valve moves out the air is pushed out through the same orifice.

Each tappet rod 34 is hollow and an insulated wire 28*a* passes up the rod to the associated lead 27 or high tension lead 28 to conduct electricity to the batteries or electric charge to the associated spark plug or other igniter 13 as shown in Figure 8. Highly accurate cam surfaces 23*a* on the sides of the rotary pump wheels 23, as they rotate, press against cam surfaces 34*a* on the tappet rods and oscillate the rods. The oscillations in turn move the valves 30 in and out at the sides of the cylinder bore. The positions of the cam surfaces on the sides of the wheels 23 and the power points 24 have to be precisely located, depending on tests.

As shown in Figure 9, each points brush 35 is embedded in insulating material 35*a* and is connected to the associated wire 28*a* by a spring 28*b* of good electrically conductive material.

Adjacent cylinder blocks are held together by means of bolts which are passed through bores 29 in the cylinder blocks (Figure 3).

Each cylinder block is divided into two parts to facilitate assembly and dismantling of the engine. The fixed gear 22 is not however divided, but takes the form of a ring with the teeth facing internally. The ring fits into a slot in the internal cavity of the cylinder block, and said slot is provided with recesses for accommodating lugs or ears on the fixed gear 22 by means of which said gear is positively located in the cylinder block. Each part of the cylinder block is provided with flanges 38 and the two parts are positively clamped together by means of bolts 39 which are passed through bores in the flanges 38 and clamped in place by means of nuts.

As described above, the fixed gear 22 has 100 teeth which makes it easy to correctly mesh the moving crank wheels since the position of the pistons is dependent on the crank bearing gear ring. Moreover, the teeth of the fixed gear 22 will be at a gap of 3.6° from each other which angle is the factor of radial displacement between adjacent cylinder blocks.

As shown in Figure 1, the engine consists of ten cylinder blocks located adjacent to one another and radially displaced with respect to one another. If we consider the case in which all of the cylinder blocks are radially aligned with one another, then with ten cylinders to each block, ignition could take place at every 36° of revolution of the crankshaft. In such a case, if the first cylinder block and the corresponding crank are set at 0° (360°), it will be found that five of the ten cranks of the crankshaft are alternately at 0° (360°) and the other alternate five cranks are at 180° because the crankshaft is in a plane, the cranks of which are alternating at 180°. The cylinder block and the corresponding cranks are numbered from 1 to 10 and for a given piston cylinder in the first cylinder block set at 0°, the remaining cylinder blocks can be radially displaced independently so that the corresponding piston cylinder in each cylinder block has a desired radial displacement with respect to the first.

Figure 1 shows a linear sequence in which the second cylinder block with its crank at 180° has its cylinders displaced by 3.6° with respect to the corresponding cylinders of the first cylinder block. The third cylinder block has its cylinders displaced by 3.6° with respect to the corresponding cylinders of the second cylinder block or by 7.2° with respect to the corresponding cylinders of the first cylinder block. Similarly, the cylinders of the fourth, fifth, sixth, seventh, eighth, ninth and tenth cylinder blocks are displaced by 10.8°, 14.4°, 18°, 21.6°, 25.2° 28.8° and 32.4° respectively with respect to the cylinders of the first cylinder block. All these displacements have the common factor of 3.6° and there is a displacement of 36° between adjacent cylinders of each cylinder block. By way of example, starting from a given cylinder in the first cylinder block, if the cylinders were to be fired at 180° and the crankshaft were to turn round, the second cylinder to fire would be in the second cylinder block firing at 180° + 3.6° = 183.6°. Thus the firing sequence for the cylinders in each cylinder block would be as follows:—

CYLINDER NO.

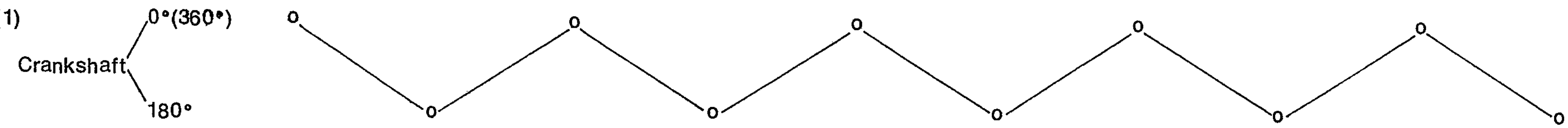
	1	2	3	4	5	6	7	8	9	10
Block 1	0°	36°	72°	108°	144°	180°	216°	252°	288°	324°
Block 2	183.6°	219.6°	255.6°	291.6°	327.6°	3.6°	39.6°	75.6°	111.6°	147.6°
Block 3	7.2°	43.2°	79.2°	115.2°	151.2°	187.2°	223.2°	259.2°	295.2°	331.2°
Block 4	190.8°	226.8°	262.8°	298.8°	334.8°	70.8°	46.8°	82.8°	118.8°	154.8°
Block 5	14.4°	50.4°	86.4°	122.4°	158.4°	194.4°	230.4°	266.4°	302.4°	338.4°
Block 6	198°	234°	270°	306°	342°	18°	54°	90°	126°	162°
Block 7	21.6°	57.6°	93.6°	129.6°	165,6°	201.6°	237.6°	273.6°	309.6°	345.6°
Block 8	205.2°	241.2°	277.2°	313.2°	349.2°	25.2°	61.2°	97.2°	133.2°	169.2°
Block 9	28.8°	64.8°	100.8°	136.8°	172.8°	208.8°	244.8°	280,8°	316.8°	352.8°
Block 10	212.4°	248.4°	284.4°	320.4°	356.4°	32.4°	68.4°	104.4°	140.4°	176.4°

It will be seen from the foregoing table that the firing sequence is linear in that a cylinder in the first cylinder block is fired first at 0° followed by a cylinder in the second cylinder block at 3.6° followed by one in the third cylinder block at 7.2° and so on up to the tenth cylinder block at 32.4°. There is then a jump back to the first cylinder block in which a cylinder is fired at 36°. While such a jump is acceptable in engines in which only a small number of cylinder blocks is employed, it is desirable, where possible, to use a scattered firing sequence in engines having eight or more cylinder blocks.

Such a scattered firing sequence can be achieved if the fourth cylinder block with its crank at 180° has its cylinders displaced radially by 3.6° with respect to the corresponding cylinders of the first cylinder block. The seventh cylinder block with its crank at 0° has its cylinders displaced radially by 7.2°. The tenth cylinder block with its crank at 180° has its cylinders displaced radially by 10.8°. The third cylinder block with its crank at 0° has its cylinders displaced radially by 14.4°. The sixth cylinder block with its crank at 180° has its cylinders displaced radially by 18°. The ninth cylinder block with its crank at 0° has its cylinders displaced radially by 21.6°. The second cylinder block with its crank at 180° has its cylinders displaced radially by 25.2°. The fifth cylinder block with its crank at 0° has its cylinders displaced radially by 28.8°. The eighth cylinder block with its crank at 180° has its cylinders displaced radially by 32.4°. The radial displacement between the first and tenth cylinder blocks thus again ranges from 0° to 32.4° but in a different sequence from that illustrated in Figure 1. There is again a displacement of 36° within each cylinder block. By way of example, starting from a given piston cylinder in the first cylinder block, if the pistons were to be fired at 180° and the crankshaft were to turn round, the second piston to fire would be in the fourth cylinder block firing at 180° + 3.6° = 183.6°.

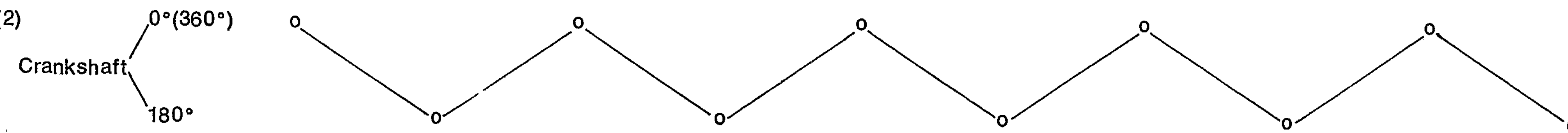
The following Table gives, by way of example, the scattered firing sequence for one complete revolution of the crankshaft of an engine having ten cylinder blocks with ten cylinders in each block:—

(1)



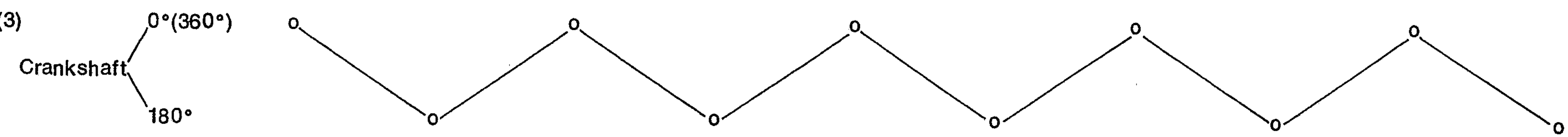
Block No.	1	2	3	4	5	6	7	8	9	10
Displacement °	0°	25.2°	14.4°	36°	28.8°	18°	7.2°	32.4°	21.6°	10.8°
Piston Cylinder No.	1	8	5	2	9	6	3	10	7	4
Firing at °	0°	205.2°	14.4°	183.6°	28.8°	198°	7.2°	212.4°	21.6°	190.8°

(2)



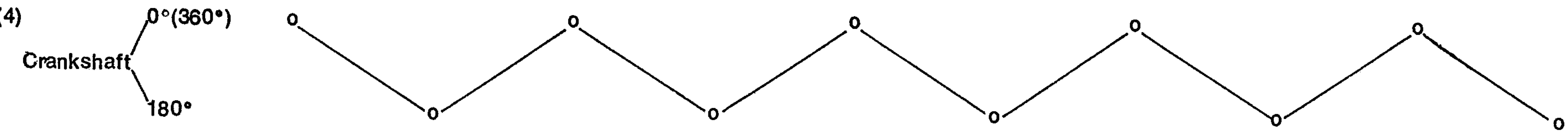
Block No.	1	2	3	4	5	6	7	8	9	10
Displacement °	36°	61.2°	50.4°	39.6°	64.8°	54°	43.2°	68.4°	57.6°	46.8°
Piston Cylinder No.	11	18	15	12	19	16	13	20	17	14
Firing at °	36°	241.2°	50.4°	219.6°	64.8°	234°	43.2°	248.4°	57.6°	226.8°

(3)



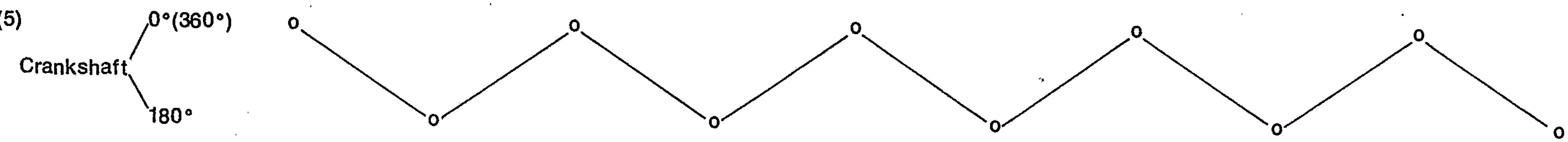
Block No.	1	2	3	4	5	6	7	8	9	10
Displacement °	72°	97.2°	86.4°	75.6°	100.8°	90°	79.2°	104.4°	93.6°	82.8°
Piston Cylinder No.	21	28	25	22	29	26	23	30	27	24
Firing at °	72°	277.2°	86.4°	255.6°	100.8°	270°	79.2°	284.4°	93.6°	262,8°

(4)



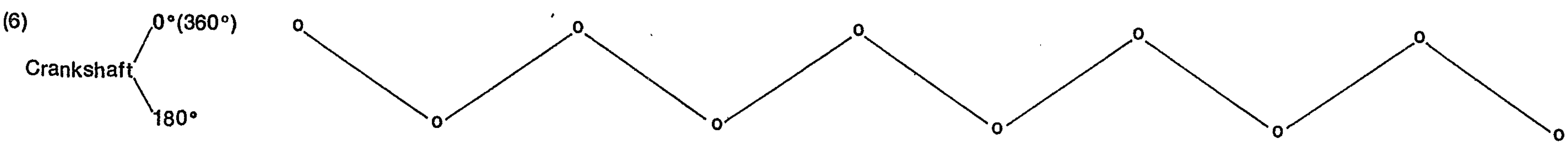
Block No.	1	2	3	4	5	6	7	8	9	10
Displacement °	108°	133.2°	122.4°	111.6°	136.8°	126°	115.2°	140.4°	129.6°	118.8°
Piston Cylinder No.	31	38	35	32	39	36	33	40	37	34
Firing at °	108°	313.2°	122.4°	291.6°	136.8°	306°	115.2°	320.4°	129.6°	298.8°

(5)



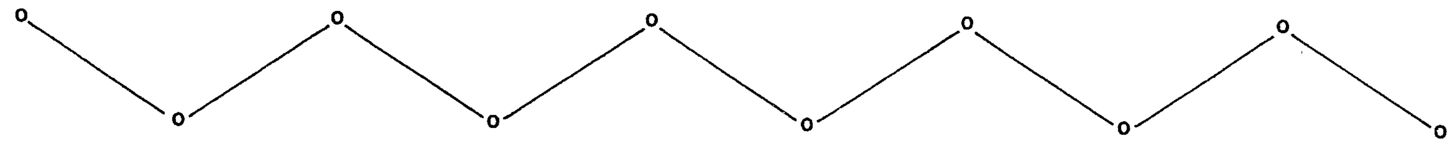
Block No.	1	2	3	4	5	6	7	8	9	10
Displacement °	144°	169.2°	158.4°	147.6°	172.8°	162°	151.2°	176.4°	165.6°	154.8°
Piston Cylinder No.	41	48	45	42	49	46	43	50	47	44
Firing at °	144°	349.2°	158.4°	327.6°	172.8°	342°	151.2°	356.4°	165.6°	334.8°

(6)



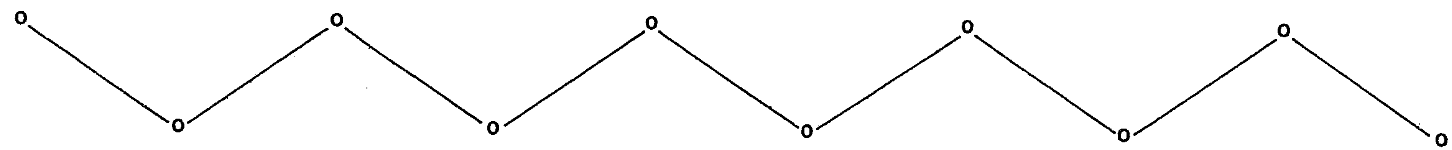
Block No.	1	2	3	4	5	6	7	8	9	10
Displacement °	180°	205.2°	194.4°	183.6°	208.8°	198°	187.2°	212.4°	201.6°	190.8°
Piston Cylinder No.	51	58	55	52	59	56	53	60	57	54
Firing at °	180°	25.2°	194.4°	3.6°	208.8°	18°	187.2°	32.4°	201.6°	10.8°

(7) Crankshaft 0°(360°) 180°



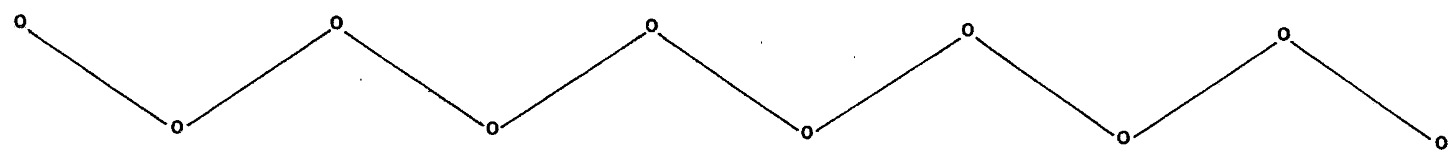
Block No.	1	2	3	4	5	6	7	8	9	10
Displacement °	216°	241.2°	230.4°	219.6°	244.8°	234°	223.2°	248.4°	237.6°	226.8°
Piston Cylinder No.	61	68	65	62	69	66	63	70	67	64
Firing at °	216°	61.2°	230.4°	39.6°	244.8°	54°	223.2°	68.4°	237.6°	46.8°

(8) Crankshaft 0°(360°) 180°



Block No.	1	2	3	4	5	6	7	8	9	10
Displacement °	252°	277.2°	266.4°	255.6°	280.8°	270°	259.2°	284.4°	273.6°	262.8°
Piston Cylinder No.	71	78	75	72	79	76	73	80	77	74
Firing at °	252°	97.2°	266.4°	75.6°	280.8°	90°	259.2°	104.4°	273.6°	82.8°

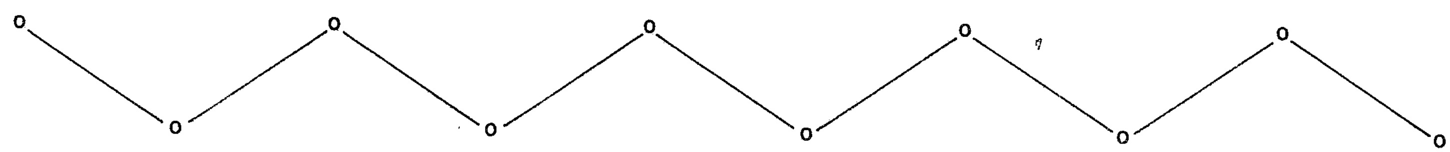
(9) Crankshaft 0°(360°) 180°



Block No.	1	2	3	4	5	6	7	8	9	10
Displacement °	288°	313.2°	302.4°	291.6°	316.8°	306°	295.2°	320.4°	309.6°	298.8°
Piston Cylinder No.	81	88	85	82	89	86	83	90	87	84
Firing at °	288°	133.2°	302.4°	111.6°	316.8°	126°	295.2°	140.4°	309.6°	118.8°

(10)

Crankshaft 0°(360°) 180°



Block No.	1	2	3	4	5	6	7	8	9	10
Displacement °	324°	349.2°	338.4°	327.6°	352.8°	342°	331.2°	356.4°	345.6°	334.8°
Piston Cylinder No.	91	98	95	92	99	96	93	100	97	94
Firing at °	324°	169.2°	338.4°	147.6°	352.8°	162°	331.2°	176.4°	345.6°	154.8°

The linear firing sequence with the cylinders arranged as shown in Figure 1 can be used for any odd or even number of cylinder blocks. However, the preferred, scattered firing sequence can be used only with an engine having an even number of cylinder blocks which is greater than six, i.e. eight or more. The scattered firing sequence can be calculated from the formula $S = D \times G$ in which S is the angle in degrees by which the cylinders of each cylinder block are staggered with respect to the cylinders in adjacent cylinder blocks, D is the firing interval between consecutive cylinder firings and G is the gap in cylinder blocks between consecutive cylinder firings.

In the case of an engine having ten cylinder blocks, a figure of $G = 3$ will give a firing sequence in cylinders in blocks 1, 4, 7, 10, 13, 16, 19, 22, 25, and 28. However, since there are only ten cylinder blocks, the figure 10 or multiples thereof must be subtracted from these results to give the firing sequence 1, 4, 7, 10, 3, 6, 9, 2, 5 and 8 which is the sequence depicted in the above table. G is calculated by selecting either of two prime odd numbers greater than one the sum of which is equal to the number of cylinder blocks and neither of which is a factor of the said cylinder block number. Thus, with ten cylinder blocks $G = 3$ or $G = 7$. In the case of $G = 7$, the firing sequence will be 1, 8, 5, 2, 9, 6, 3, 10, 7 and 4 which is simply the reverse of the sequence with $G = 3$.

The firing interval D is calculated by dividing the total number of cylinders in the engine into 360°. Thus, with an engine having ten cylinder blocks and ten cylinders in each block,

$$D = \frac{360^\circ}{10 \times 10} = 3.6^\circ$$

Therefore $S = D \times G = 3.6^\circ \times 3 = 10.8^\circ$ or $3.6^\circ \times 7 = 25.5^\circ$.

From this the radial spacing between adjacent cylinders in each cylinder block can be calculated by simply adding the figures for S together which gives 36°. Since this figure can also be found by simply dividing 360° by the number of cylinders in each cylinder block ($360^\circ \div 10 = 36^\circ$), this provides a useful check that S has been correctly calculated.

Turning now to the embodiment shown in Figs. 10 to 13 of the drawings, the engine comprises a plurality of cylinder blocks which are bolted together in an angular staggered relationship similar to the arrangement illustrated in Figure 1 of the drawings. However, as can be seen from Figure 11, in this embodiment each cylinder block has only six cylinders as opposed to ten in the previous embodiment so that the cylinders of each block are spaced apart by 60° instead of 36°. If ten cylinder blocks are employed, as in the previous embodiment, the engine will have $6 \times 10 = 60$ cylinders and therefore the cylinders can be arranged to fire at 6° intervals. Therefore, for a linear firing sequence, the second and subsequent cylinder blocks will be angularly staggered by the angle of 6° with respect to the preceding cylinder block and the firing arrangement calculated as in the embodiment illustrated in Figure 1 to ensure that a cylinder fires with every 6° rotation of the crankshaft. However, it is possible to provide more or less than ten cylinder blocks in which case the angular relationship between adjacent cylinder blocks will be varied accordingly. For example, in the case of four cylinder blocks, there will be $4 \times 6 = 24$ cylinders and the cylinders should be so arranged that a cylinder fires at every $360^\circ \div 24 = 15^\circ$ of crankshaft rotation. Thus, the second and subsequent cylinder blocks should be angularly staggered by the angle of 15° with respect to the preceding cylinder block for a linear firing sequence.

As shown more particularly in Figure 11, each cylinder block consists of two half sections 101 and 102 which are provided with flanges 103 which contain bores by means of which the sections can be bolted together by bolts 104 and nuts 105. Each cylinder block contains six cylinder bores extending radially outwards from a central crank cavity. A crank 106 is located in the cavity and is provided with a small central bore 107 and two crank positioning bores 108. The bores 108 in each crank 106 are so arranged that two rods 109 can extend through the bores 108 in all of the cranks 106 when said cranks are correctly aligned so as to maintain the cranks of all of the cylinder blocks in their correct angular relationship with respect to one another that is in a plane alternating at 180° as shown in Figure 10.

Each end of each of the rods 109 is screw-threaded and one end of each rod is engaged in a respective screw-threaded bore provided in a drive shaft 110 on which is mounted a pulley wheel 111. The other end of each rod 109 is secured in place by means of lock nuts 112. The crank shaft thus formed is mounted at each end in bearings 113 provided in respective casings 114 at each end of the engine and sealing rings 115 are provided on the shaft which are urged against the casings 114 by locking rings 116 to prevent the escape of lubricant from the interior of the engine.

Slidably mounted on each crank 106 is a toothed crank bearing ring 117 to which connecting rods 119 of pistons 120 are connected by pivot pins 118. The toothed ring 117 meshes with the smaller gear 121 of a double gear wheel the larger gear 122 of which meshes with a gear wheel 123 which in turn meshes with an internally toothed gear ring 124 which is fixed in the cylinder block. The gear wheels 122 and 123 are identical and the gear wheel 121 has the same number of teeth as the gear wheel 122 but a smaller radius. The ratio of the gear wheel 121 to the toothed ring 117 is the same as the ratio of the gear wheel 123 to the fixed gear 124 so that the ring 117 and the fixed gear 124 have the same number of teeth. The said ratio is preferably 1:4.

As shown more clearly in Fig. 12, each double gear wheel 121, 122 is mounted on one end of a shaft 125 which passes through aligned bores in two adjacent cranks 106 which are secured together by a further gear wheel 126 secured to the other end of the shaft 125 by a pin, grub screw or the like 127. The shaft 125 is rotatably mounted in the bores in the cranks 106 by means of bearings 120. Each crank bearing ring 117 is provided with two rows of teeth, one row of which meshes with a respective gear 121 and the other of which meshes with a gear wheel 126 mounted on the shaft 125 in an adjacent crank. Each gear wheel 123 is mounted on a stub shaft 128 secured in a screw-threaded bore 129 in the associated crank 106 and the fixed gears 124 are located in respective recesses in the associated cylinder blocks.

If the crankshaft rotates once in a clockwise direction, the gear wheel 123 will rotate anti-clockwise for four revolutions and will cause the gear wheel 122, and hence the smaller gear wheel 121 to rotate clockwise for four revolutions and this will cause the toothed bearing ring 117 to rotate anti-clockwise for one revolution. The toothed bearing ring 117 will therefore turn in the opposite direction to the crankshaft but will not turn in any angular direction with respect to the fixed gear 124 in a similar manner to the relationship between the crank bearing ring 6 and the fixed gear 22 in the previous embodiment.

The pistons may take the form of the pistons shown in Figs. 5 and 6 of the drawings. However, in this embodiment of the engine according to the invention, each piston 130 is preferably provided with a separate crown 131 which is secured to the top of the piston 130 by means of screws 132. The screws are secured in place by a spring clip 140 which engages in a groove provided in the inside wall of the piston. A sinusoidal channel 133 is formed in the top of the piston to provide a passage between the top of said piston and the under surface of the crown 131 to permit cooling fluid to pass through the piston for cooling purposes. A bore 34 extends through the piston from the interior substantially to the centre of the channel 133. Grooves 135 extend up the walls of the piston, on the outside surface thereof, to each end of the channel 133 for the supply and outlet of the cooling fluid.

Each piston 130 is connected to its associated connecting rod 119 by a split gudgeon pin comprising a screw-threaded male member 136 and a screw-threaded female member 137 arranged to be located in respective bores in the wall of the piston 130 and to clamp the connecting rod therebetween as shown in Figure 12. A cylinder sleeve 138 is secured in position in each of the cylinder casings of each cylinder block by means of screws 139 and each sleeve 138 is provided with slots 141 and 142 leading respectively to inlet ports 143 and exhaust ports 144 which are connected in turn to inlet manifolds and exhaust manifolds (not shown).

Between each cylinder sleeve 138 and the associated piston 130 is located a rotary sleeve valve 145 for controlling the opening and closing of the inlet and outlet slots 141 and 142 by means of a port 146. The sleeve valve 145 is provided at its lower end with an annular flange 147 which is mounted in bearings 149 and which is provided adjacent its outer edge with a bevel gear 148. The bevel gear 148 meshes with a second bevel gear 151 which is mounted on a shaft 150 for rotation with a larger gear wheel 152. The gear wheel 152 is arranged to be driven by a gear wheel 153 mounted on the crank 106 which is adjacent to the crank on which the associated pistons are supported via the connecting rods 119 and crank bearing ring 117. Thus, rotation of the crankshaft will cause the sleeve valve 145 to rotate at the same speed as the said crankshaft which can be arranged by suitable selection of the ratios of the intermediate gearing 153, 152 and 151, 148.

Each cylinder is closed by a cylinder head 154 provided with a screw-thread for engagement with a co-operating screw-thread provided in the associated cylinder of the cylinder block. The cylinder head can be screwed in place by providing closed bores 155 in the top surface of the cylinder head and engaging a suitable key or other tool (not shown) in the said bores 155. A spark plug 156 is fitted into each of the cylinder heads 154. The spark plugs are connected by high tension leads to a suitable ignition and distributor system (not shown). High-tension voltage for ignition can, however, be derived from a wiping contact arm 157 mounted on the rods 109 (Figure 10) which is engageable with carbon brushes 158 connected by leads 159 to an ignition coil (not shown) for each cylinder block. A second wiping contact arm 160 may also be mounted on the rods 109 engageable with further carbon brushes 161. These latter brushes may be connected by leads 162 to further ignition coils for igniting the plugs in the cylinders of some of the cylinder blocks but are preferably arranged to be connected to a battery for charging purposes. The brushes 158 and 161 are embedded in a ring 163 of insulating material which is located in a housing 164 which is secured to a casing 165 at one end of the engine. The leads 159 and 162 project through bores in the housing 164.

A plurality of bores 166 are provided in the cylinder blocks (Figure 13) for conveying cooling and lubricating fluid to the internal parts of the engine. Each cylinder sleeve 138 is provided with bores 167 which lead from the bores 166 through bores in the sleeve valve 145 to the channels 135 in the associated piston 130 for the purpose of cooling and lubricating the piston. Further bores 168 (Figure 11) in the cylinder blocks serve to convey cooling and lubricating fluid via passages (not shown) to the bevel gears 148 and 151 which act as gear pumps for said fluid. The cooling and lubricating fluid is pumped from the bevel gears into the central chamber of the engine where it serves to lubricate the gears and rings 117 before passing through channels (not shown) in the cranks 106 into the central bore 107 from which it can pass from the engine and be returned to a reservoir or radiator for re-cooling. If desired, rotary pump wheels, similar to the rotary pump wheels 23 in the previous embodiment, may be provided on the cranks instead of or in addition to the bevel gear pumps. The fluid bores 168 in each of the cylinder blocks are inclined with respect to the longitudinal axis of the engine to take into account the angular displacement of the adjacent cylinder blocks so that the bores 168 in each cylinder block are aligned with the corresponding bores 168 in the adjacent cylinder blocks throughout the axial length of the engine.

Adjacent cylinder blocks are secured together by means of bolts which are passed through aligned bores 169 and 170 provided in each of the cylinder blocks. One set of bores 169 serves to secure a cylinder block to the adjacent cylinder block on one side and the other set of bores 170 serves to secure the said cylinder block to the adjacent cylinder block on the other side. The cylinder block at one end of the engine is provided with only one set of bores 169 and the cylinder block at the other end of the engine is provided with only one set of bores 170.

Rotation of each sleeve valve 145 is synchronised with its associated piston 130 so that, when the piston begins its descent from the top dead centre position, the port 146 in the sleeve valve 145 is moved opposite the induction slot 141 in the cylinder sleeve 138 so that a combustible fuel/air mixture is sucked through the inlet port 143 and induction slot 141 into the cylinder. Halfway down the piston stroke, the sleeve valve 145 has been rotated through 90° and the port 146 is moved out of communication with the induction slot 141. The spark plug 156 is then fired to ignite the mixture and drive the piston down to bottom dead centre position. At this position, the sleeve valve has been rotated through a further 90° and, as the piston begins its return stroke to top dead centre, the port 146 is moved opposite the exhaust slot 142 in the cylinder sleeve 138 so that the ignited mixture is exhausted from the cylinder via the exhaust slot 142 and exhaust port 144. The port 146 is in communication with the exhaust slot 142 until the piston reaches top dead centre when the sleeve valve has been rotated through a further 180°. Further piston movement will then bring the port 146 back into communication with the induction slot 141 and the cycle is repeated.

The cylinder blocks of this embodiment may be arranged for a linear firing sequence as above described. However, it is also possible to arrange the cylinder blocks of this embodiment to have a scattered firing sequence using the above-mentioned formula $S = D \times G$. Thus, in this case

$$D = \frac{360^\circ}{10 \times 6} = 6^\circ$$

$$G = 3 \text{ or } 7 \ (3 + 7 = 10)$$

$$S = 6^\circ \times 3 = 18^\circ \text{ or } 6^\circ \times 7 = 42^\circ$$

Radial displacement of cylinders in each block

$$= \frac{360^\circ}{6} = 18^\circ + 42^\circ = 60^\circ$$

Cylinder Block Firing Sequence

= 1, 4, 7, 10, 3, 6, 9, 2, 5, 8 or 1, 8, 5, 2, 9, 6, 3, 10, 7, 4.

A table depicting the scattered firing sequence of the individual cylinders is shown in Figure 14 of the drawings in which each box represents a cylinder and the cylinders are numbered according to the sequence in which they are fired.

Using the above-mentioned formula $S = D \times G$, one can calculate the scattered firing sequence for any engine having any even number of cylinder blocks greater than six and having any number of cylinders in each cylinder block. Thus, for example, for an engine having eight cylinder blocks with six cylinders in each block, the figures for S, D and G can be calculated as follows.

$$D = \frac{360^\circ}{8 \times 6} = 7.5^\circ$$

$$G = 3 \text{ or } 5 \ (5 + 3 = 8)$$

$$S = 7.5^\circ \times 3 = 22.5^\circ \text{ or } 7.5^\circ \times 5 = 37.5^\circ$$

Radial displacement of cylinders in each block

$$= \frac{360^\circ}{6} = 22.5^\circ + 37.5^\circ = 60^\circ.$$

Cylinder Block Firing Sequence

= 1, 4, 7, 2, 5, 8, 3, 6, or 1, 6, 3, 8, 5, 2, 7, 4.

A table depicting the scattered firing sequence of the individual cylinders is shown in Figure 15 of the drawings in which each box again represents a cylinder and the cylinders are again numbered according to the sequence in which they are fired.

The invention is not restricted to the above described embodiments but variations and modifications may be made without departing from the scope thereof.

For example, the fuel mixture need not necessarily be petrol or petroleum products and air but any fluid with potential energy mixed with air in desired variable amounts capable of exploding in combustion chambers of the engine. The engine could also be arranged to run on compressed air or steam.

The lubricating and cooling fluid could be oil, but any fluid or mixtures of fluids capable of lubricating with minimum of viscosity and exchanging heat could be used.

Materials for the engine and parts may be made of light metal or metal alloys such as aluminium or steel.

I consider that the compression of fuel mixture in the cylinder does not ultimately increase the thrust of transmission as the crankshaft has to compress fuel mixture in another cylinder. It merely increases the amount of fuel mixture compared to pressure of the volume in which it explodes and this, as common sense demands, does not increase efficiency but performance at the expense of efficiency. Performance, in my opinion, is wrongly interpreted and misunderstood in place of efficiency. Efficiency of performance is always waste. It is misguided for often it becomes difficult to realise that such efficiency is of negative nature. To think of efficiency of performance is drifting away in the opposite direction to the aim of achieving the efficiency of an ideal machine or engine. On the other hand, to think of efficiency of fuel leads towards getting as near as possible to the ideal machine or engine.

The potential energy of fuel mixture is transformed into heat energy at explosion and mechanical (kinetic) energy at expansion. Compression of fuel mixture releases heat by the compression and since the engine has to be cooled, this is wasted: from potential energy to heat, to mechanical energy, and then reversed again to heat energy. This reversal is an undesirable process and is waste as the engine has to be cooled and the reversal of mechanical energy to heat energy adds an extra burden to the cooling system and reduces fuel efficiency even though misunderstood as increasing performance.

There is no such problem in a fuel system by suction. While in compression, pressure increases and heat is released, in suction pressure decreases and heat is absorbed. Not only is there no extreme heat to be cooled, but suction becomes a self-cooling device and half the problem of cooling the engine is solved, thereby increasing the fuel efficiency. The amount of fuel mixture is not increased as there is no compression. This is in favour of fuel efficiency. Performance may be adversely affected as a result of decrease in the amount of fuel mixture, but there will be an increase in fuel efficiency.

In the case of an engine according to the invention which is arranged to run on steam, the steam will be allowed to expand on the down stroke of the piston after approximately half of the piston stroke thereby considerably increasing the efficiency of the use of steam. Similar considerations apply to the use of compressed air.

There is no need to add weights on the crankshaft for the purpose of balancing the movement of the cranks. The crankshaft of the engine according to the invention assumes the shape of a solid cylinder with cranks as slots in it and acting as balancing weights to each other. The cylindrical shape of the crankshaft automatically self-balances the cranks which are in a plane alternating at 180°. Further, since the firing intervals amount to no more than a few degress of crankshaft rotation and since the combustible fuel mixtures in the cylinders do not have to be compressed, the provision of a flywheel to maintain crankshaft rotation is not necessary.

Moreover, the invention is not restricted to an engine having 100 or 60 cylinders as in the embodiments above described with reference to the drawings but may have more or less depending on the optimum number which would not require the use of a flywheel and which would, desirably, produce the greatest efficiency of fuel consumption. The angle of radial displacement of the cylinder blocks will depend on the number of piston/cylinder assemblies in the engine, for example, with 360 such assemblies, the angle of radial displacement would be 1°, with 1000 assemblies, the angle would be 0.36°, with 50 assemblies the angle would be 7.2° and with 500 assemblies, 0.72° and so on.

CLAIMS

1. An engine which comprises at least one cylinder block having a plurality of radially extending cylinders, a respective piston reciprocally mounted in each of said cylinders and means for interconnecting said pistons in such a manner as to provide a substantially vibration-free engine having no flywheel.

2. An engine according to claim 1, wherein each piston is pivotally connected to a respective connecting rod, wherein the connecting rods are pivotally connected to a common crank bearing ring and wherein the crank bearing ring is arranged to drive a crankshaft of the engine.

3. An engine according to claim 2, wherein the crank bearing ring is mounted on a crank of the crankshaft for rotation relative to said crank.

4. An engine according to claim 3, wherein a plain sliding bearing is provided between the crank and the crank bearing ring.

5. An engine according to claim 3 or claim 4, wherein the crank bearing ring is provided with gear teeth engageable with a first gear, wherein the first gear and a second gear are mounted on a common shaft rotatably mounted in the crank, the first and second gears being secured together or formed integrally for common rotation, wherein the second gear is engageable with a third gear fixedly mounted on the crank and wherein the third gear is engageable with an internally toothed fixed gear mounted in the engine casing.

6. An engine according to claim 5, wherein the ratios of the gears are such that the crank and the crank bearing ring are arranged to rotate at the same speed but in opposite directions.

7. An engine according to claim 5 or claim 6, wherein the first and second gears are mounted on a shaft which is rotatably mounted in the crank.

8. An engine according to any preceding claim, wherein each piston is provided in its upper region adjacent its crown with a channel for the passage of cooling fluid.

9. An engine according to claim 8, wherein the channel has a substantially sinusoidal path.

10. An engine according to claim 8 or claim 9, wherein further channels are provided in the side wall of each piston which communicate with the ends of the first-mentioned channel.

11. An engine according to any one of claims 8 to 10, wherein each piston is provided with a crown which is detachably secured to the top of the piston, the channel being formed in the top of the piston to form, with the underneath surface of the piston crown, a closed passage.

12. An engine according to any preceding claim, wherein each cylinder is provided with a rotary sleeve valve, the associated piston being reciprocally mounted in the sleeve valve and said sleeve valve being provided with a port to control induction and exhaust of gases into and out of the cylinder.

13. An engine according to claim 12, wherein each rotary sleeve valve is provided with a toothed gear arranged to be driven via gearing by the crankshaft of the engine.

14. An engine according to any one of claims 3 to 7 and claim 13, wherein the toothed gear comprises a bevel gear arranged to be driven by a second bevel gear mounted on a shaft for common rotation with a gear wheel arranged to be driven by a gear mounted on the crank.

15. An engine according to claim 14, wherein the ratios of the gearing are such that the sleeve valve is arranged to rotate at the same speed as that of the crank.

16. An engine according to claim 14 or claim 15, wherein the bevel gears are also arranged to serve as gear pumps for the purpose of conveying cooling and lubricating fluid to the internal parts of the engine.

17. An engine according to any one of claims 3 to 7, wherein a rotary pump wheel is mounted on the crank and is arranged to control the operation of valves for controlling the supply of cooling and lubricating fluid to the internal parts of the engine.

18. An engine according to any preceding claim, wherein a plurality of cylinder blocks are provided which are secured together in a staggered angular relationship.

19. An engine according to claims 3 and 18, wherein the cranks of the cylinder blocks are secured together and are provided with aligned bores to accommodate at least one rod extending throughout the axial length of the engine.

20. An engine according to any preceding claim, wherein the or each cylinder block is provided with ten cylinders the longitudinal axes of which are angularly spaced apart by 36°.

21. An engine according to claim 20, wherein ten cylinder blocks are provided.

22. An engine according to any one of claims 1 to 19, wherein the or each cylinder block is provided with six cylinders the longitudinal axes of which are spaced apart by 60°.

23. An engine according to claim 22, wherein ten cylinder blocks are provided.

24. An engine according to claim 22, wherein four cylinder blocks are provided.

25. An engine substantially as described herein with reference to Figures 1 to 9 of the drawings.

26. An engine substantially as described herein with reference to Figures 10 to 15 of the drawings.

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