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INTERNATIONAL APPLICATION PUBLISHED UNDER THE PATENT COOPERATION TREATY (PCT)

(51) International Patent Classification 6 : F01L 11/06

A1

(11) International Publication Number: WO 96/18808

(43) International Publication Date: 20 June 1996 (20.06.96)

(21) International Application Number: PCT/GB95/02952

(22) International Filing Date: 18 December 1995 (18.12.95)

(30) Priority Data:
9425624.5 17 December 1994 (17.12.94) GB

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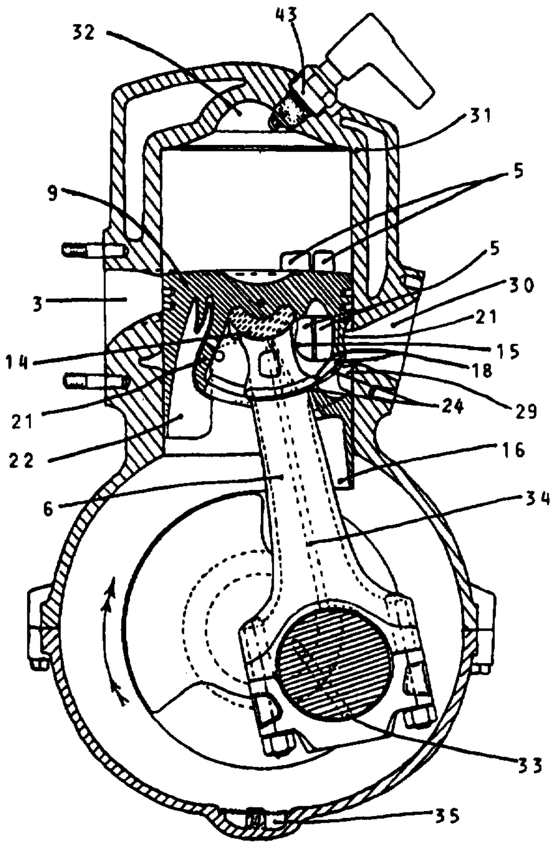
(81) Designated States: AU, BR, CA, CH, CN, CZ, DE, DK, ES, GB, JP, KP, KR, KZ, MX, NZ, PL, PT, RU, SE, SG, US, VN, European patent (AT, BE, CH, DE, DK, ES, FR, GB, IE, IT, NL, PT, SE), OAPI patent (BF, BJ, CF, CG, CI, CM, GA, GN, ML, MR, NE, SN, TD, TG).

Published
With international search report.
With amended claims and statement.

(54) Title: IMPROVEMENTS IN TWO-STROKE ENGINES

(57) Abstract

Valving edges (18) formed on the connecting rod (6) control port (29) from connecting with the piston interior, ensuring that the pressure air supply is not opened to the cylinder via transfer ports (5) until the exhaust is partly completed after piston (9) uncovers exhaust port (3). After (3) is recovered, piston valve (18) allows extended cylinder filling. The exhaust through port (3) may be curtailed by a variable timing rotary valve (42, Fig. 25) acting as a restrictor and as a balancing shaft in conjunction with a contra-rotating balance shaft (40, Fig. 25). Since the crankcase is isolated from the air supply, a turbo-charger or supercharger is required.



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IMPROVEMENTS IN TWO-STROKE ENGINES

Technical Field

Two-stroke engines are now in wide use in three out of four almost distinct categories of usage, with the fourth category due to expand noticeably in the next few years. The categories (each with its own specialised version of the engine) are :-

(i) A "traditional" very simple low cost version — used in Lawn mowers; hand held tools; cycle motors etc.

(ii) An "improved" version with "tuned" ports, disk/reed valves — for higher power marine outboard motors; motor cycles; micro-light aircraft, etc.

(iii) Diesel engines, often requiring one or more "poppet" valves, a low pressure air supply; a high pressure injector and a strongly made heavy construction — for generators; emergency pumps; marine use, etc.

(iv) A "sophisticated" two-stroke using ancillaries (Supercharger; Turbo-charger; Electronic engine management; fuel injectors etc.) developed for "four-stroke" transport use — but with the elimination of: poppet valves and their drive mechanism, also the separate cylinder head requirement.

As announced to-date all the two-stroke versions not using poppet valves or their equivalent mechanism suffer in various amounts from wastage of input air and/or fuel, plus exhaust contamination of input since the piston control of gas flow is not sufficiently effective over a wide range of speeds and loads. Stepped pistons of high reciprocating mass can also benefit from a gudgeon-free; 'gate-valve' design.

Economy with fuel and pollution from exhaust (including "noise pollution") is now becoming important. Traditional designs are often used for short duration activities so the need for economy of fuel tended to be subjugated to cost / simplicity requirements. Future legislation is likely to restrict their use.

The innovative ideas in this application have been crystalised into two distinct embodiment types: the simpler version suitable for adoption into categories (i) and (ii) above, and the sophisticated embodiment for (iii) and (iv) — giving sequential exhaust → overlap → recharge normally without crankcase compression, mainly controlled by three moving parts; Piston, Connecting-rod and Crankshaft. (Controlled rotor adds refinement).

The new piston design such as in the P.C.T. application GB 91/00493 published in the U.K. as patent G B 226 1492 B (August, 1994) can be modified to improve "breathing efficiency" of two-strokes as described below.

Background Art

Categories (i) and (ii). Although a range of variations have been tried for inlet; transfer, exhaust and lubrication methods — the most widely used are similar to the basic version shown in the symbolic drawings on sheet 1, so that versions with a "flat-top" piston can be envisaged as using angled transfer ports which direct the incoming charge to the cylinder such that it is angled to the top of the cylinder rather than the exhaust port. When this is arranged so that the "route-length" of the various passages/ports cause gas resonance then the latest "loop-scavenge" and "tuned" engines, are relatable to these basic diagrams. The symbolic engine in Fig. 1 is at the exhaust + transfer stage with the crank angle "E" indicating the rotation with the exposed exhaust 3 open. In Fig. 2, "I" indicates the smaller crank angle of induction when the port for the long transfer passage 4 is open, shown at a slightly later time in the exhaust + transfer stage. In Fig.3 the operative cycle has moved on beyond "C" where compression started above the piston; while below the piston the corresponding reduction in pressure has caused the reed valve to open (2) allowing the new charge to enter the crankcase as indicated by the three short arrows. Later at Fig.4 the working stroke occurs when the gas temperature (and thus its pressure) has been raised enormously by burning the fuel in the compressed air above the piston, pushing the crank down and around to the position indicated at "W" when the "flywheel" (not shown) has sufficient angular momentum to continue rotation and power output until the following working stroke restores the flywheel's loss of momentum. Repetition at every 2-strokes.

Lubrication either by adding oil to the fuel or by "total loss" to the bearings is not changed in categories (i) and (ii), so is omitted from here: "Dry-lubrication" is well-known.

Engines not requiring use of the crankcase for suction/compression recharge of the cylinder are covered in the following :-

Categories (iii) and (iv). Diesel two-strokes are often provided with an "external" low pressure air supply from either some form of pump or supercharger and often a turbo-charger (this latter powered from otherwise waste energy in the exhaust output), these sources being used sequentially or simultaneously as preferred by the users/designers. The resultant air supply is fed by poppet valve to the cylinder starting near the end of the (port controlled) exhaust sector and continuing to an optimum fill position. With inlet and exhaust at opposite ends of the cylinder scavenging of exhaust fumes is excellent. High compression improves thermal efficiency. A "dry" sump with pressure feed of bearing oil is standard practice. Its main disadvantages concern: high weight; cost/complication of a very high pressure injector — of poppet valves; of multiple piston rings and the fuel oil's slow burn/injection rate which severely restricts engine r.p.m. With scavenging less perfect and reduced ring plus injection restrictions, the proposed innovation eliminates the poppet valve and cams from such engines — and some modern fuel systems can raise r.p.m.

Recent innovations in volatile fuel engines have included: the **Allen** Adiabatic lean burn combustion with approximately 17:1 C.R. (μ +30%) needing controlled water injection to stabilise combustion; **Dr. D. Merritt** high efficiency segregated combustion (allowing high r.p.m. and minimal pollution) and the **Miller** high-boost supercharged variation on the four-stroke **Otto** cycle (this new innovation will permit its equivalent on the two-stroke cycle). Various combined air/fuel injection systems are currently being developed. (e.g."Orbital").

Modern materials and manufacturing methods are being applied to "stepped piston" designs over 60 years old, with considerable success despite high piston mass. See:- **Car Design & Technology,** October 1992, p.28 →30. For different separation configurations, GB A 223 8830 and GB A 224 6394. Prior Art: USA 5,205,245 and GB 901 2349, also **The English Electric Journal,** The English Electric Company Limited, Stafford, England: Volume Thirteen, Number Seven, September 1954, p.295 →302 title "The Napier 'Nomad' Compound Diesel Aero Engine." GB 2 113 800 A; Stepped piston and stepped piston engine.

Disclosure of Invention

The manufacture of a piston in two vertically separated halves allows a more complex shape to be cast than is practical with the conventional design. Restricting the rotation of the crankshaft to one direction allows use of the connecting-rod to piston relationship to be developed into a 'gate' valve simply by suitably changing the shape of the "little end" of the connecting-rod where it is enclosed within the piston. The simplest form allows a traditional two-stroke to transfer the slightly compressed air or fuel mixture from the crankcase either by the slot in the piston lower surface in which the connecting-rod slides and/or by the gate ports in the piston base into intermittent communication with the short transfer ports 5 at the appropriate crank/piston position to cause gas transfer into the cylinder to start appreciably after the exhaust port has been uncovered by the piston and to maintain gas transfer until appreciably after the exhaust port 3 is closed. This ensures good cylinder filling and corresponding high power output (see Figs. 5 to Fig. 7 inclusive.)

The second embodiment is the more sophisticated development (shown in Figs. 8 to Fig.27 inclusive) which keeps the pressurised input air out of communication with the crankcase interior thus allowing conventional low cost automobile replaceable shell plain journal bearings to be used throughout the engine. Thus the conventional sump with lubricating oil out of contact with the rotating crankshaft but pumped under pressure to shell bearings is appropriate. The pressurised air input is obtained most efficiently from an exhaust driven turbo-charger. Since these cannot function until the engine is running some form of air pressure delivery is required. This may be from any suitable conventional source e.g. a **Berk** 3-lobe displacement type; a **Roots** blower; a low cost 'vane' electrically powered unit for starting only. A stepped piston uses the larger diameter lower portion of the dual size piston to act as an air pump feeding into the air passages of the input port 30. The second cylinder of a pair in "V" configuration can be a pumping unit for the first.

Where the pump is continuously driven by the engine the output may be arranged in series or alternatively in parallel with the output from a turbo-charger. (i.e.stepped piston; second cylinder). Where the supercharger drive can be disengaged (especially electrically) the pressurised air may be delivered sequentially with the turbo-charger output. (e.g.boost purposes and/or starting). The very wide choice of ancillary devices available is not shown.

Variations can be made at the design stage of the choice of fuel injection positions; the height of the various ports; the angular position and total number of transfer ports. A restriction on the width of any one cylinder opening is made by the unsupported length of piston ring(s) exposed to the gap. Here, advantage can be taken of variable contact force piston rings which can traverse the port area of the cylinder with reduced wear/vibration; see GB 226 1492 B. The cross-sectional area of the transfer ports may be made uniformly constant with a "streamlined" flaring at the cylinder wall. This may be substituted with a venturi; where fuel injection is into the transfer passage. The transfer passages may be curved to induce rotation of the gasses within the cylinder, and/or angled to direct the air inflow toward selected points within the cylinder (as with conventional two-stroke inputs). For clarity and simplicity the drawings show the transfer passages and ports as open-sided slots. Similarly, many of the sectioned piston elevations omit obstructing portions to eliminate excessive use of dotted lines and to clarify gas flow routes. Where sections are shown for comparison purposes, the various juxtaposition portions may not be in strictly the same plane of section. This will not cause confusion to persons skilled in the art of designing two-stroke engines. As drawn, minimum "gas path" X-section >12% piston area.

The largest angle of connecting-rod swing is minimised by having a high pivot axis at (or even above) the piston crown, and the amount of tip swing clearance required by the gate shutter depends not only on crank/rod/cylinder proportions, but also on the offset in the pivot-hinge line from the cylinder's longitudinal axis. Some timing overlap is desirable between starting the cylinder recharge and ending of the exhaust, in order to facilitate a smooth exchange/substitution of gasses above the piston. Transfer "ends" with air only — thus the "crevice" above and around the piston rings will not contain fuel vapour likely to produce unburnt hydrocarbons as exhaust emissions.

The start, duration and rate of fuel injection will normally be rapidly and repetitively set by the engine electronic management system to eliminate fuel vapour loss via the exhaust port, with transfer ports arranged to give an initial rotary motion within the cylinder by the entering gas — a swirl increased prior to ignition by conservation of angular momentum into the smaller diameter combustion chamber, where "squish" induced turbulence is added from the 'V' internal edges. All are to promote thorough combustion.

Considerable variation is available at the design stage to alter the portions of each crankshaft revolution serving any selected function of the two-stroke cycle and the amount of overlap between adjacent functions; but once decided and assembled, variation is restricted to :- ignition timing, injection variables, fuel selection/"mixture-strength" and charging pressure — except when some additional movable control/restriction(s) is placed in the gas flow route. This advantage may be obtained as follows :-

Fig. 24 introduces two views of a rotating constriction in the exhaust port, in this instance illustrated with a gear drive from the crankshaft providing contra-rotation at crank speed with variable angular positioning of the rotor (Advance - Retard). Starting at Fig. 25 is a group of drawings depicting exhaust rotors which at twice crankshaft r.p.m. are combined with matching balance shafts to provide substantial cancellation of secondary engine vibration. Constriction is provided without contact of the exhaust passage (total sealing is not required) — leakage would be less than 25% of unrestricted flow, and normally between 5% and ½%. To this end the rotator may have a tapered (conical) configuration; allowing close none-contacting adjustment via use of end packing washers, or similar. The 'hot path' rotor would normally be of Titanium; steel, Ti coated/plated ; or "engineering ceramic". Such restrictors may have chain or low cost 'toothed' belt drive allowing some 22° advance to retard of the exhaust constriction with under 2% variation of secondary balance force — such advance/retard normally being controlled by the engine's electronic management unit. Such restrictor allows better control of : gas flow; pollution; power output, and particularly in the "twice r.p.m." case — exhaust noise.

For compression ignition use, selection of high strength/temperature materials; additional piston rings with a change in combustion chamber size and shape, an injector substituted for the spark plug(s), is indicated. For high r.p.m. with multi-fuels and minimal pollution a segregating combustion system is required. The drawings are not intended to necessarily depict the most desirable or efficient engine layout — but to illustrate the various features as clearly as practicable in "2 D." (All performance estimations are subject to test verification.) The non-rotor example in the accompanying drawings used the following :-

Crankshaft — degrees of rotation,

Power stroke	115°		
Exhaust sector	128°	(Exhaust only	50°)
Cylinder recharge	95°	(Recharge only	17°)
Overlap	− 78°		
Compression sector	100°		
Total (one revolution)	360°	(Exhaust rotor examples used different ratios.)	

Using the above (pre and post construction) variables a wide range of user requirements can be satisfied, particularly when flexibly combined with other appropriate innovations.

Brief Description of Drawings

Comparisons are made between the intake area (as a percentage of piston area) in a modern "touring" two valves per cylinder (4-stroke) and the Inlet port/transfer passage area to piston area of this design. After allowance for length of opening; double the number of openings (in two revolutions); speed of port exposure : only a low inlet boost pressure of 0·2 bar (3 psi) at low r.p.m. rising with speed to probably 0·9 bar (13·2 psi) — to compensate for the turbulent intake route and gate valve on the two-stroke — should (with inter-cooling) be required for four-stroke equivalent specific output.

Sheet 1 is diagrammatic/symbolic, Figs. 1 to 4 depicting "traditional" two-stroke cycle. The reed valve allowing one way flow into the crankcase is shown closed at 1 and "pulled" open at 2. The crank angle for inlet at "I" is within and smaller than the exhaust range "E". The transfer passage 4 is long enough to bypass the entire piston height. (Figs.1 and 2).

Fig. 5 shows an engine with less overall height even with the same effective length of connecting-rod and the same crank swing (and piston stroke.) This is via the hot running pivot bearing high inside the piston. The protrusion on the connecting-rod within the piston at 6 is shown just after opening allowing the gas flow shown by three curved arrows from the vicinity of the connecting-rod, through the gate valve, into the short transfer passage 5. The arc of rotation for open exhaust 7, is equal to "E" of Fig. 1. The arc of rotation for transfer of charge from crankcase to cylinder (shown by 8) is much longer and later occurring than the equivalent at "I" of Fig. 2, thus permitting a greater quantity of fresh charge to enter the cylinder for compression — thus giving increased power output.

Sheet 2 illustrates in Fig. 6 the ½ section of a typical category (i) and (ii) piston; with below, a "little-end," and a pictorial view of the latter on the left. 15 indicates one of a pair of transfer exit ports from the piston body which communicate intermittantly with a pair of short transfer ports in the cylinder walls. Fig. 7 (left) a general view from above a typical "flat-top" (no tall deflector as in Figs. 1 to 5) piston, and (right) a schematic diagram of the lower part of the cylinder. The ceramic bearing 14 may have a conic surface (lubricated at 17) which provides the near elliptical motion at the piston surface indicated at 25. Thus the "notch" in 10 can align with a slit at the top of the exhaust port (Fig. 7) when descending, but not do so when rising. This can contribute to a quieter shock front to the exhaust noise, but reduce leakage on the compression stroke. The matching pair of transfer ports 5 are equally spaced each side of the exhaust 27 in the cylinder wall. Often, designers choose to use four transfer passages, with a pair to each piston port 15, in order to present smaller openings for the piston rings to traverse, and to allow a spread of input flows on charging the cylinder. 18 (Fig. 6) is the control edge sliding over edge 24 to define the start of transfer (as in Fig. 5)

Categories (iii) and (iv). Fig. 8 is a graphical outline of the inlet comparison between four-stroke and two-stroke engines described above. Fig. 9 is a section through a typical cylinder showing port arrangement. Fig. 10 is a sectioned piston suitable to the cylinder above. On the left is the corresponding connecting-rod top and little end. Fig. 11 is a pseudo plan section through the above cylinder and the lower part of its piston to allow a comparison of various ports and flow passages. The two pairs of transfer passages 5 are equally spaced on each side of the input 30. Shown in broken line is one possible shape for the very wide range of input manifolds needed for different engine configurations/usage. Three gate-valve flow passages 29 are shown in this instance. The exhaust port 3 is also indicating a possible exhaust manifold using a broken line. The streamlined pillars in the port opening are to safeguard the piston rings. (All drawing detail annotation is listed p.8)

Fig. 12 is a pictorial representation of the piston's two matching halves and below is depicted the little-end nested against a part cylindrical ceramic bearing pad. (The conic-frustum style bearings described in **G B 2261492 B** may be used with variable pressure piston rings — but are not shown here for brevity.)

Fig. 13 is one piston half with two arrows to illustrate the two pathways (duplicated in the other half) conducting cooled pressurised air from the inlet port to the two sets of transfer ports equally spaced on each side of the inlet port 29. These four pathways through the piston are open only once per revolution of the crankshaft when the piston/rod are in the appropriate position. Thus although the piston uncovers the ports 5 twice per revolution, the gate valve is open only on the "up" (prior to compression) stroke. Figs. 14 to 21 inclusive illustrate progressive stages of the new controlled two-stroke cycle.

Fig. 22 shows a compact 90° "V" alternative (for low primary force vibration engines) to the "in-line" three cylinder engine. Broken outlines indicate possible manifolds for use with the inlet 30, and with the exhaust 3, between the cylinder castings. Fig. 23 shows the alternative use of the piston gate valve with the input airflow 'down' through the gate ports – and with the dual sets of transfer ports moved further from 29, and still without allowing the pressurised air to enter the crankcase. Operation is basically the same as in Figs. 14 to 21. Elimination of 'pressure-air' loss through the exhaust port 3 during 'overlap' with cylinder refilling is indicated by use of the exhaust rotary restrictor in Fig. 24. This adds complexity, in exchange for greatly improved gas flow control; improved efficiency; reduced exhaust noise for a wider range of speeds and loads. The A.⟷R. indicates one possible point where the engine management unit can control/vary the relative position of the exhaust restrictor while continuously rotating. The three centre lines through the gear wheel pivots and meshing points indicate a **Watt's** link motion which allows the advance and retard management-control device action-in-a-straight-line over a short range. The exhaust rotor is dynamically balanced and not contacting its casing (hot or cold).

Figs. 25 and 26 are comparable with the cycle Figs. 14 to 21 but adding two contra-rotating, unbalanced opposing shafts running at twice crankshaft r.p.m. arranged such that their mutual resultant un-balance force largely cancels the secondary out of balance vibration due to the piston and part of its connecting rod. In a typical 'in-line' engine this secondary force is approximately ¼ of the primary balance forces. One of the shafts includes the non-contacting exhaust restrictor which need not necessarily have its centre of area/or mass coinciding with the balance weights second harmonic cancelling resultant force. The small arrow '39' shows the instantaneous direction of action of each balance shaft at the crank position drawn. The sequence of action illustrated is from Ignition to just after the start of Exhaust in Fig. 25 and in Fig. 26 from Pre-Transfer where the gate valve is about to open through to a short Compression stroke with a cooled charge which allows the next working stroke to be (as drawn) — 130% greater volume than compression.

Fig. 27, top right is a pictorial view of a typical cylinder barrel casting with; cooling, transfer passages, gas inlet, exhaust and combined cylinder head. Two arrows indicate the location for the exhaust restrictor 42. The bottom drawings are the side elevation and a view on the section 'A A' of a rotor which may not require weights 48, except in the dynamically balanced case of a restrictor running at crankshaft r.p.m.

DRAWING DETAIL — ALL EMBODIMENTS In all the figures the following annotation applies :-

1/ shows a reed valve closed. 2/ reed valve open. 3/ an exhaust port.
4/ "long" transfer port. 5/ "short" transfer port. 6/ High pivot con-rod.
7/"Traditional" exhaust period. 8/ Widened + delayed transfer or inlet period.
9/Asymmetric "half" piston. 10/ Ceramic Insert. 11/ Slot for top ring.
12/ Oil control ring slot. 13/ Location of pressure pad. 14/ Ceramic bearing.
15/ Transfer exit port. 16/ Shield for transfer port. 17/ Oil "Total Loss" type.
18/ Port opening control edge. 19/ "Low-friction" coating. 20/ Little end.
21/ Hinge case. 22/ Shield for exhaust port. 23/ min. angle '18' clears '24'.
24/ Opening edge of slot — defining start of transfer period.
25/ Locus of piston surface when fitted with conic bearings at '20' (shown at '10'posn.)
26/ Transfer openings in the cylinder wall. 27/ Exhaust port in cylinder wall.
28/ Schematic view of part of cylinder showing notch to align with '10'.
29/ Flow passage in piston — inlet port to gate valve (part shown in most Figs.)
30/ Port for (cooled) compressed air (from intercooler).
31/ Integral Head and Cylinder. 32/ Turbulent combustion chamber.
33/ Automotive type lubrication system. 34/ Oil supply passage/route.
35/ "Dry-sump" oil return route to filter/cooler/storage/pump.
36/ one possible position for fuel injection (of several options).
37/ Sealing edge at end of slot for '6'. 38/ 100% balance factor for one cylinder.
39/ Instantaneous direction of action for resultant force in balance shaft.
40/ Balance shaft (2 x crank r.p.m.) 41/ Contra-balance shaft (2 x crank r.p.m.)
42/ Exhaust restrictor. 43/ Appropriate spark-plug.
44/ Core hole. 45/ Hole for spark plug. 46/ Clearance adjustment taper.
47/ Balance adjustment hole (Alternative "internal" balanced restrictor/rotor).
48/ External balance weights (especially for dynamically balanced crank r.p.m.)
49/ Special shape cast surface (interference bonds to adjacent cast surface).
50/ Passage for coolant.

PRESSURE CHARGED TWO-STROKE CYCLE Fig. 14 shows the piston descending, leaving the position of maximum torque; with crankshaft driven in a clockwise direction by the combustion pressures generated in the previous cycle. The connecting rod is at the position of maximum swing. The inlet port 30 has communication via passages 29 to the feed (lower) side of the gate-valve, but the control edges 18 are overlapping the opening edges 24 thus the pressure flow is stopped. The transfer and exhaust ports are closed by the piston walls and combined/reinforced with the piston rings.

Fig.15. Already fairly hot medium pressure gas from combustion has expanded into the now uncovered passages 5, into the piston (above the still closed gate): expanded and cooled appreciably. Air retained here from the previous cycle is further

compressed from the inrush. The exhaust port is closed, but about to open. The open inlet port still has access to the underside only of the closed gate.

Fig. 16. Here the exhaust port is partly open above the slowing piston. The gate valve is closing the inlet supply off from the transfer ports. The high pressure gas is beginning to flow at increasing rate into the exhaust system via cylinder ports 27 into manifold/port 3, simultaneously dropping the pressure above the piston. Compressed air and gas inside the piston and transfer passages 5 are now beginning to exceed the pressure above the piston and starting to flow upward; joining the flow toward the exhaust port.

Fig. 17: The gate valve is about to open with the port control edges 18 just touching the edges 24 (see Fig. 11). From here the fresh supply of air will have access to the inside of the piston for approximately one third of a revolution of the crankshaft. (Cylinder wall transfer cut-off will occur before the inlet-to-gate). The exhaust port is almost fully open: cylinder gas pressure is falling rapidly — even through the turbo-charger (not shown.) Mixed air+gas flow from inside the piston is at maximum; this will assist the acceleration and movement of the recharge air as the gate opens. The small remainder of the previous cycle's compressed air will help to dilute and cleanse the exhaust gasses from the cylinder. A fuel jet orifice is just visible above the piston top at 36. The engine management system will not permit injection to start until the exhaust port is almost closed, as in Fig. 18.

At Fig. 18, the engine is shown with maximum recharge rate from the inlet. Fuel injection commences, with the exhaust about to close. Possibly over 95% of the exhaust fumes has left the cylinder, with the remainder being pressed toward the closing exhaust port by the pressurised flow from passages 5.

Fig. 19; the exhaust port 3 has just closed with the gate valve fully open. Transfer is reducing slightly, with no compression above the piston, but inflow is raising pressure until equalisation occurs above and inside the piston with the delivery pressure now slightly raised from the effect in the moving air's momentum by the constriction in closing port 5.

Fig. 20 shows the piston at the start of compression; with a smaller range of travel/ crank rotation than that available for the power stroke. The transfer ports have just closed. The gate valve is fully open and allows the pressure inside the piston to equalise with the delivery pressure of the supply air. Cooled pressurised air delivered into a cylinder with a low effective compression gives a low temperature before ignition; even though the overall compression is high, there is no "knock" and high thermal efficiency.

Fig. 21 completes the cycle. At or near T.D.C. "spark initiated" combustion occurs in a turbulent combustion chamber 32. The piston control-gate is rapidly closing, with compressed air trapped above and below the gate (but not in contact with the now closed inlet port 30.) The continued swing to the side by the connecting-rod will further compress the trapped air which will absorb heat from the enclosing piston.
This 'compression' may well compensate the pressure drop due to leakage, and help cool the piston. This heated air will later be discharged with the exhaust gasses as described

for Figs. 16 and 17. The exhaust is closed and will remain so for about one third of a crankshaft revolution; giving the "power" stroke. This concludes the description of operation.

The selection of cost effective materials depends on the specific output required and the reliability/longevity required in a given application/environment (eg. marine/tropical). An aluminium alloy cylinder block with chrome (or nickel-silicon carbide coated) sleeved bores allows close working clearance with an aluminium alloy piston skirt,(16, 22.) Titanium (or MMC of Al/Li 8090 + 17% ceramic particulates) is suitable for the crown and "ring-lands" of a piston required to be run very hot from sustained high power requirements.

Best Mode for Carrying Out the Invention

The quest for high specific output has shown that a piston gate-valve based on Fig. 23, using a spherical lower bearing surface within the piston with a corresponding spherical 'low-friction' face on the underside of the "little-end" allows up to five ports 29 with a flow path area of approximately 20% of the piston area. When this little-end under-face is also extended to cover both sides of the "slot" in which the connecting rod 6 moves the pressure air supply is sealed from the crankcase without requiring the little-end to seal with the internal side faces of the piston (as is required Figs. 5; 6; and 10 to 26.) This shape change simultaneously allows the extremities of the little end to be "cut-away" allowing the port 29 to be optionally machined at either side of the piston (thus permitting the R.H. cylinder Fig. 22 to be "rotated" 180°placing port 3 "outside" the "V") and also permits a symmetrical piston which can be repetition cast from only one squeeze-cast die set. The spherical bearing also provides the input air, after passing through the gate valve, to be channelled to the sides of the piston allowing exits 15 and transfer ports 5 to be placed at right-angles to inlet 30 and exhaust 3. The policy of minimising the tooling required for production is followed to its logical limits yet providing a wide choice of engine layouts and capacity at final assembly. (eg. single cylinder; "in-line" three; "V" four; "V" six cylinder etc.)

Cooperating with other patent holders: combining this "spherical version" piston with other related patents and patent applications — G B 226 1492 B (for piston rings); segregated combustion for minimal pollution, based on G B 224 6394; "joining" system for castings — (application G B 950 5581·0) and obtaining convenient modular assemblies for a range of transport and static applications — can provide a very versatile power unit with superior performance and greatly reduced pollution when compared with any previous two-stroke engine. Auxiliary and ancillary devices (turbo-chargers/pumps/management units/ filters etc.)are to user's/vehicle assemblers option. Evolvement of safe transportable hydrogen generators (or compact hydrogen storage) would make this engine pollution free.

Industrial Applicability

With the gradual reduction in world reserves of fossil fuels and an expected increasing demand for vehicles — experienced senior design engineers have already predicted that small lightweight engines — (such as described above), probably using three cylinders "in-line," will be in great demand to meet the anticipated market requirements worldwide. ■

* * * * * * * * * * * * *

Claims

1. I claim a two-stroke internal combustion engine comprising a cylinder with a piston slidably sealed in the cylinder, the cylinder including at least one transfer port and at least one exhaust port, characterised by the transfer port being connected into communication with the inlet air supply via a moving gate-valve formed within the piston, arranged to give substantial control of air flow into the cylinder, said valve being operated and controlled by the relative arc motion occurring between said piston and its constraining connecting rod.

2. I claim a two-stroke internal combustion engine according to claim 1, characterised by having two or more transfer ports positioned within the wall of said cylinder on each side of an inlet port which is so positioned that air flowing from the inlet manifold through said inlet port is constrained to flow through said gate-valve within the piston and out of the piston into the transfer ports during only one portion of each crankshaft revolution thus further controlling air flow into the cylinder.

3. I claim a two-stroke internal combustion engine according to claim 1 in which the inlet port is positioned to feed air into the crankcase below the piston, characterised by said gate valve controlling and restricting air flow out of the crank-case from under the piston in the vicinity of the connecting rod up into the transfer ports communicating with the piston once per crankshaft revolution.

4. I claim a two-stroke internal combustion engine according to claim 3, characterised by having the flow of gas/air controlled by said gate valve delayed in relation to the opening of the exhaust and continuing to flow until after the exhaust port has closed.

5. I claim a two-stroke internal combustion engine according to claim 2, characterised by having said inlet manifold supplied with cooled pressurised air sourced from a supercharger or from an exhaust turbo-charger or from both of these operating in conjunction with each other either sequentially or simultaneously to provide air for the cylinder.

6. I claim a two-stroke internal combustion engine according to claim 2, characterised by having said inlet manifold supplied with pressurised air sourced from the larger diameter of a stepped piston.

7. I claim a two-stroke internal combustion engine according to claim 6, characterised by having the pressurised air originating from the stepped piston being augmented by cooled compressed air originating from an exhaust turbo-charger.

8. I claim a two-stroke internal combustion engine according to claim 2, characterised by having at least two cylinders with at least one of the cylinders and its respective piston is used as a compressor for the air supply to the other cylinder(s).

9. I claim a two-stroke internal combustion engine according to any of the previous claims characterised by having a rotating non-contacting restrictor placed in or near the said cylinder exhaust port to variably curtail exhaust flow thereby augmenting the control of the working fluid obtained by using the said gate-valve alone.

10. I claim a two-stroke internal combustion engine according to claim 9, characterised by said rotating constrictor being dynamically balanced and rotating at crankshaft revolutions per minute as it augments working fluid control.

11. I claim a two-stroke internal combustion engine according to claim 9, characterised by having the said rotating constrictor out of balance but matched with a counter-rotating balance shaft, both said rotating shafts revolving at twice crankshaft revolutions per minute, arranged to reduce by cancellation secondary engine vibration and torque induced vibration while simultaneously augmenting control of the working fluid.

12. I claim a two-stroke internal combustion engine according to any of the previous claims, characterised by having a contoured slot in the edge of the piston top positioned to align with a corresponding slot in the top edge of the exhaust port such as to help mute the audible high transient shock wave at the start of the high pressure discharge of exhaust gas, thus adding a small gas control refinement.

13. I claim a two-stroke internal combustion engine according to any of the previous claims, characterised by having a top connecting-rod bearing with a conic-frusto surface within the said piston arranged to give a small rotational oscillatory motion about the cylinder axis derived from the angular swing of the connecting rod in relation to the piston, thus giving alignment of the slot of claim 12 only during the downward stroke of the piston and thus reducing minor loss of the working fluid.

14. I claim a two-stroke internal combustion engine according to claim 9, characterised in that the rotating constrictor's rotational position relative to the instantaneous position of the crankshaft is under continuously monitored control and adjustment via the actuator directed by the engine management unit such that the rotor's variable curtailment of exhaust flow is suited to all engine speeds, loads, and accelerations.

15. I claim a two-stroke internal combustion engine substantially as shown in Figs. 5 to 27 inclusive and described in the text relating to these Figs.

16. I claim a two-stroke internal combustion engine according to claim 2 or claim 9 or claim 14, characterised by combustion of the fuel/air mixture being achieved solely by compression of the air.

17. I claim any novel feature or novel combination of features hereinbefore described and/or shown in the accompanying drawings.

* * * * * * * * * * * * *

AMENDED CLAIMS

[received by the International Bureau on 16 April 1996 (16.04.96); original claims 1-17 replaced by amended claims 1-17 (2 pages)]

1. I claim an internal combustion reciprocating two-stroke engine containing a piston with internal compartments characterised by being segregated with a partition which also forms the mating surface for the lower bearing of the connecting rod little-end which has no gudgeon pin and is housed inside the uppermost compartment in such shape and configuration that arc sliding of said bearing forms a gate-valve using said partition's one or more ports to permit air from the compartment fed from a pressurised air input to flow at designated crank positions through the open gate-valve into another compartment and from thence into a transfer passage which then conducts the air through the cylinder wall appreciably above the exhaust port's opening level so that the combination of piston plus gate-valve motion subjects the cylinder plus transfer passage interiors to a repetitive fixed sequence of operational modes.

2. I claim an internal combustion reciprocating two-stroke engine according to claim 1, characterised by the sequence of operational modes within the cylinder and transfer passage interiors being distinct as hermetically sealed, moving to exhaust only, followed by exhaust simultaneously with scavenge - evolving to filling - via the transfer passage and then, after exhaust port closure, on to pressure recharge from the inlet manifold via the transfer passage, completing the sequence by returning to hermetically sealed throughout the cycle of compression, combustion and power-stroke.

3. I claim an internal combustion reciprocating two-stroke engine according to claims 1 and 2, with pressurised air from the inlet manifold flowing through the said gate-valve and piston compartments characterised by having one or more transfer passages curved and angled to induce rotation of said pressurised air within the cylinder, such rotation being predominantly about the cylinder's longitudinal axis.

4. I claim an internal combustion reciprocating two-stroke engine according to any previous claim, characterised by allowing the crankcase interior to be in communication with the piston compartment which is fed from a pressurised air input, this input thus being provided from crankcase compression.

5. I claim an internal combustion reciprocating two-stroke engine according to claim 3, characterised by having said inlet manifold supplied with pressurised air sourced from a supercharger or from an exhaust turbo-charger or from both of these operating in conjunction with each other either sequentially or simultaneously to provide pressurised air for the cylinder.

6. I claim an internal combustion reciprocating two-stroke engine according to claim 3, characterised by having said inlet manifold supplied with pressurised air sourced from the larger diameter of a stepped piston.

7. I claim an internal combustion reciprocating two-stroke engine according to claim 6, characterised by having the pressurised air originating from the stepped piston being augmented by pressurised air originating from an exhaust turbo-charger.

8. I claim an internal combustion reciprocating two-stroke engine according to claim 3, characterised by having at least two cylinders with at least one of the cylinders and its respective piston being used as a compressor to supply pressurised air to the other cylinder(s).

9. I claim an internal combustion reciprocating two-stroke engine according to any of the previous claims characterised by having a rotating non-contacting restrictor situated in or near the cylinder exhaust port to curtail exhaust gas flow thereby augmenting the control of the working fluid obtained by using the gate-valve alone.

10. I claim an internal combustion reciprocating two-stroke engine according to claim 9, characterised by said rotating restrictor being dynamically balanced and rotating vibration free at a speed related to crankshaft revolutions per minute as it augments control of the working fluid by adjustable curtailing of the exhaust flow.

11. I claim an internal combustion reciprocating two-stroke engine according to claim 9, with said rotating restrictor being out of balance to reduce or counterbalance crankshaft vibration on appropriate cylinder configurations while augmenting control of the working fluid.

12. I claim an internal combustion reciprocating two-stroke engine according to claim 9, characterised by having the rotating restrictor out of balance but opposed with a counter-rotating balance shaft, both said shafts revolving at twice crankshaft revolutions per minute, so arranged to reduce by mutual interaction secondary engine vibration and torque induced oscillations while simultaneously augmenting control of the working fluid.

13. I claim an internal combustion reciprocating two-stroke engine according to any of the claims 4 to 8, characterised by the pressurised air supplied being passed through an intercooler to assist in cooling the piston while increasing air mass-flow.

14. I claim an internal combustion reciprocating two-stroke engine according to claim 9, characterised in that the restrictor's instantaneous rotational position with respect to the instantaneous position of the crankshaft is under continuously monitored control and adjustment via an actuator directed by the engine management unit such that the restrictor's variable curtailment of exhaust flow is suited to all engine speeds, loads, and acceleration.

15. I claim an internal combustion reciprocating two-stroke engine substantially as shown in Figs. 5 to 27 inclusive and described in the text relating to these figures.

16. I claim an internal combustion reciprocating two-stroke engine according to any of the claims 1 to 14, characterised by combustion of the fuel/air mixture being achieved solely by compression of the air.

17. I claim any novel feature or novel combination of features hereinbefore described and/or shown in the accompanying drawings.

* * * * * * * * * * * *

AMENDED SHEET (ARTICLE 19)

STATEMENT UNDER ARTICLE 19

(To be added at the end of **Background Art** – sheet 3, line 6)

Of historical interest,since Dec.1912, are: DE-A-1576249; FR-A-548197; FR-A-689089; FR-E-16 210 and DE-C-809 117.

It can be seen from the above references that a wide variety of two-stroke mechanisms have been proposed during the evolution of the I. C. engine. Often their failure to conquer the mass commercial market has involved complexity; (extra bell-cranks, springs, multiple seals etc.) more often their being confined to one part of the cycle, eg. crankcase filling.

The proposed innovation is a control engineering systems solution, using the minimum number of rigid moving elements, providing comprehensive working fluid control even on gas passages and ports with cross-sections between 14 to 24 percent of the piston's area, combining efficiency with high specific power output. The above published work has required some claims to be limited in scope to avoid overlap with earlier wide ranging claims, even though they were based on different ideas and contrasting forms or structures.

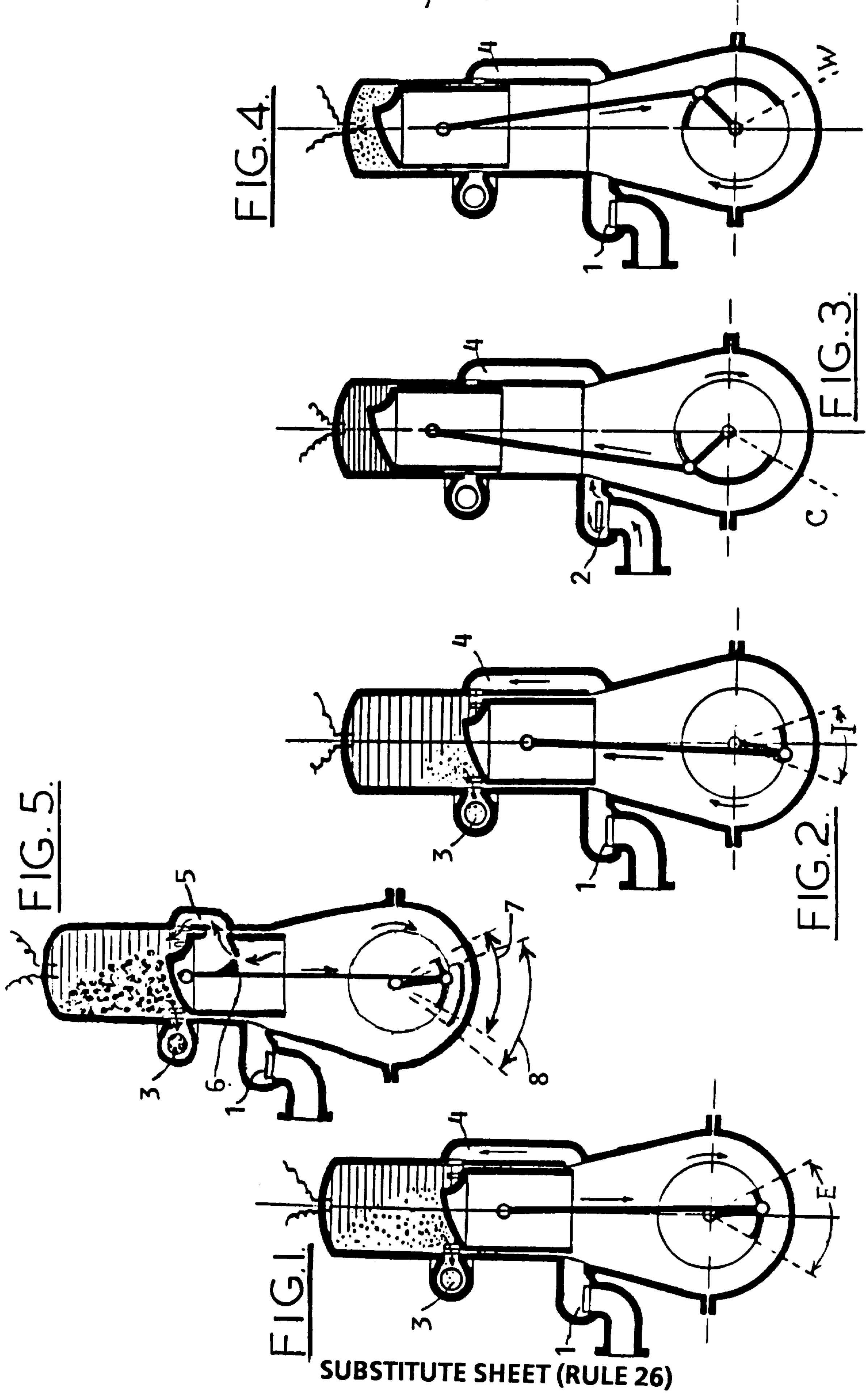

FIG.4. FIG.3. FIG.2. FIG.5. FIG.1.

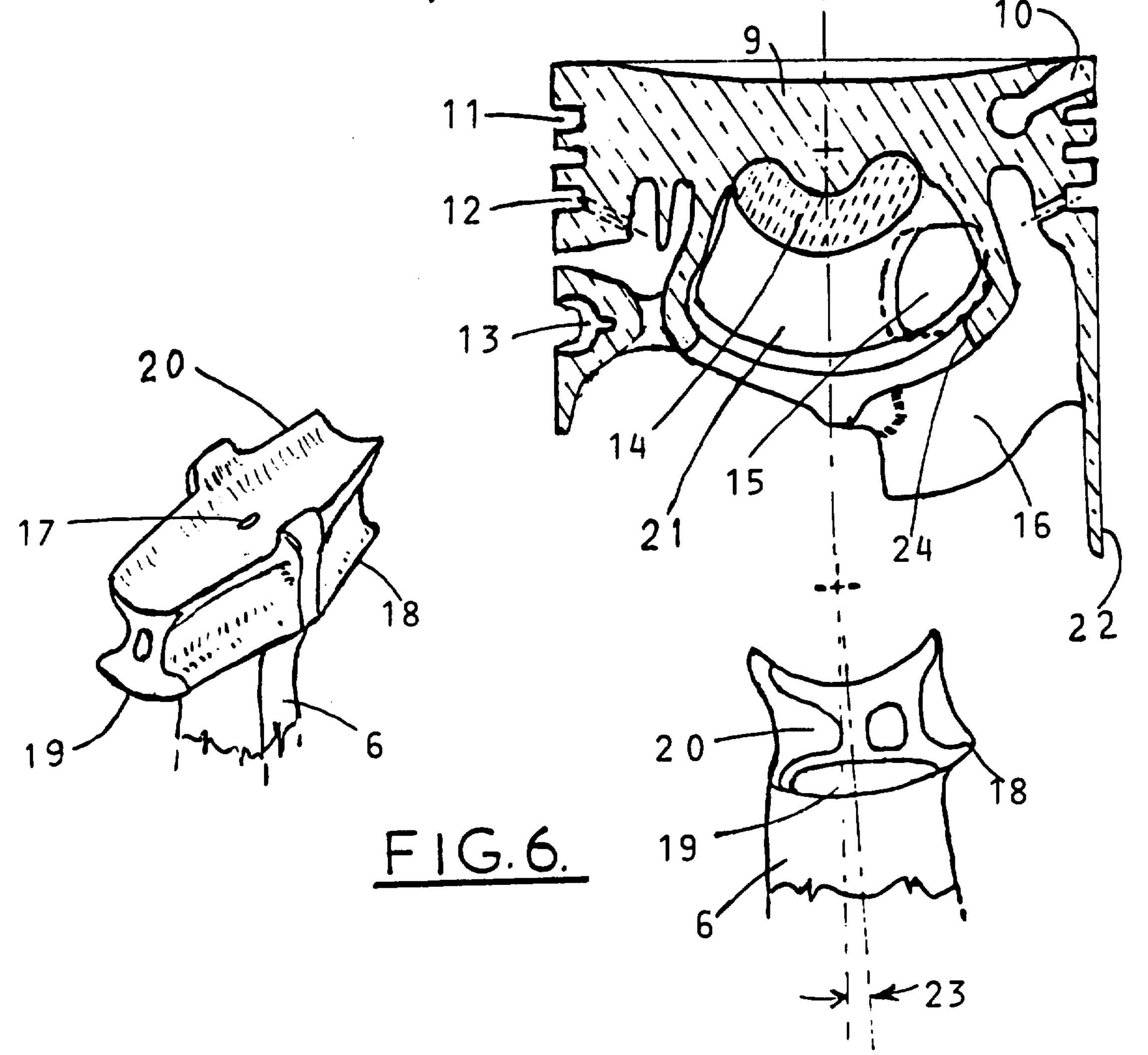
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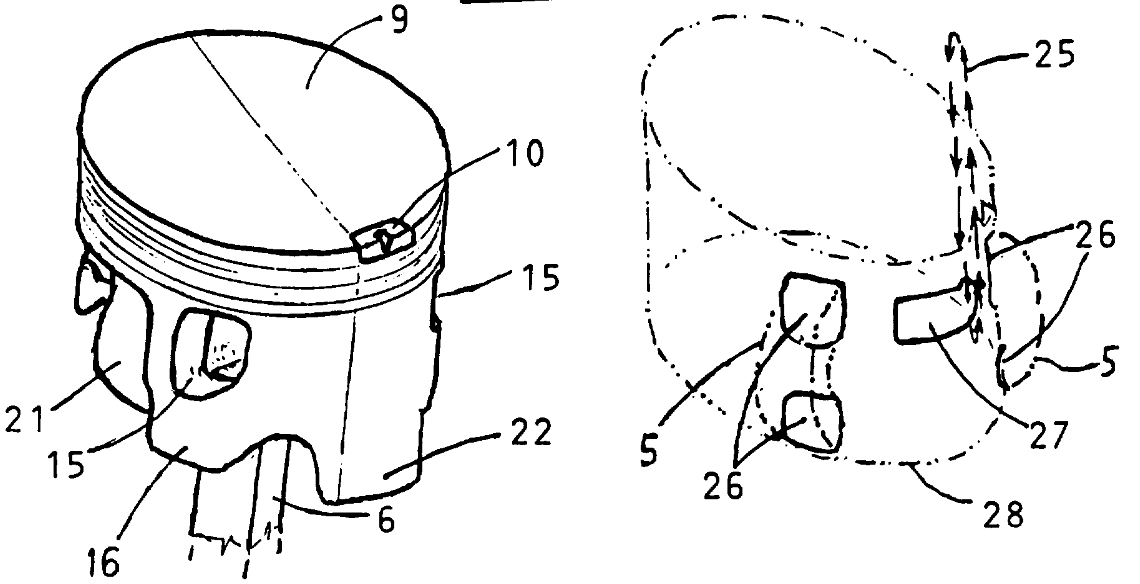
FIG. 6.

FIG. 7.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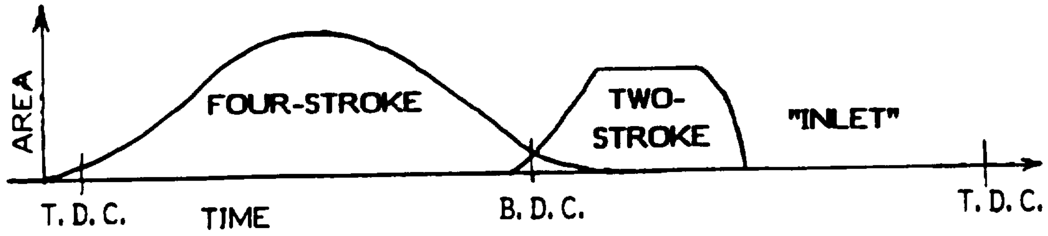


FIG. 8.

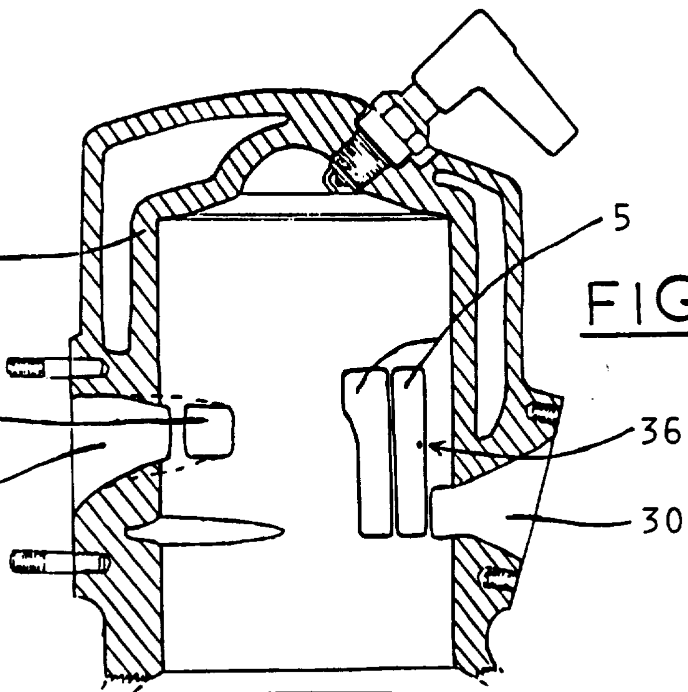


FIG.9.

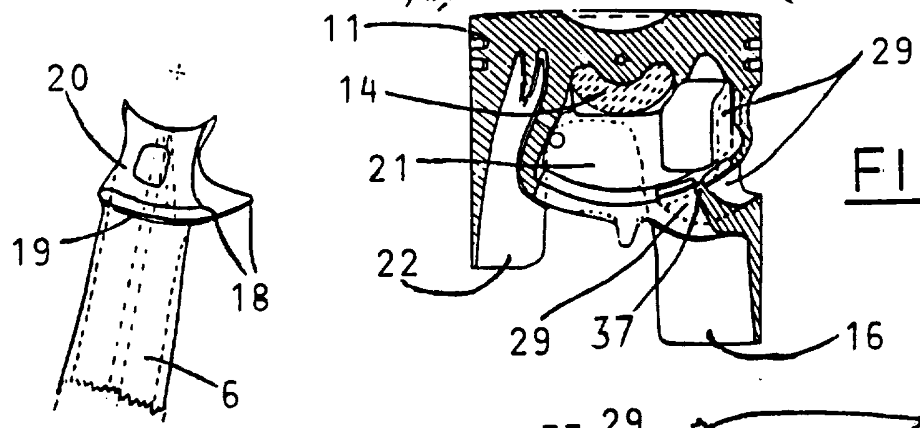


FIG.10.

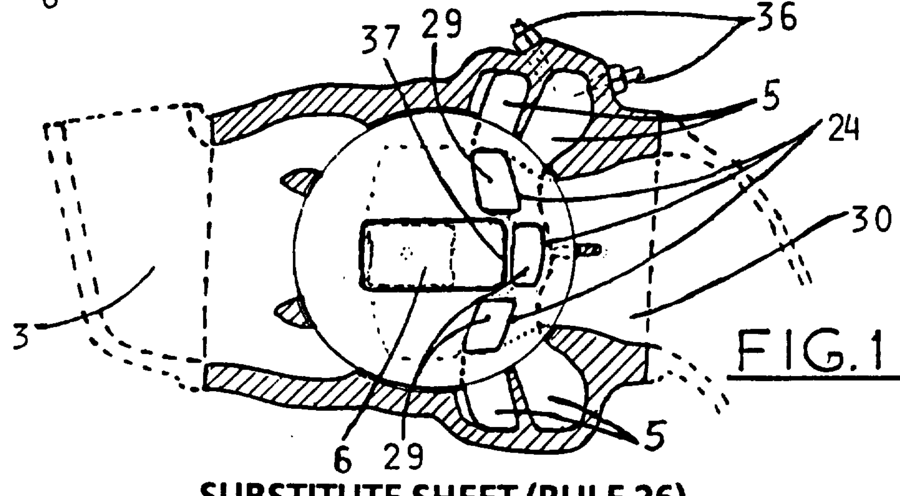


FIG.11

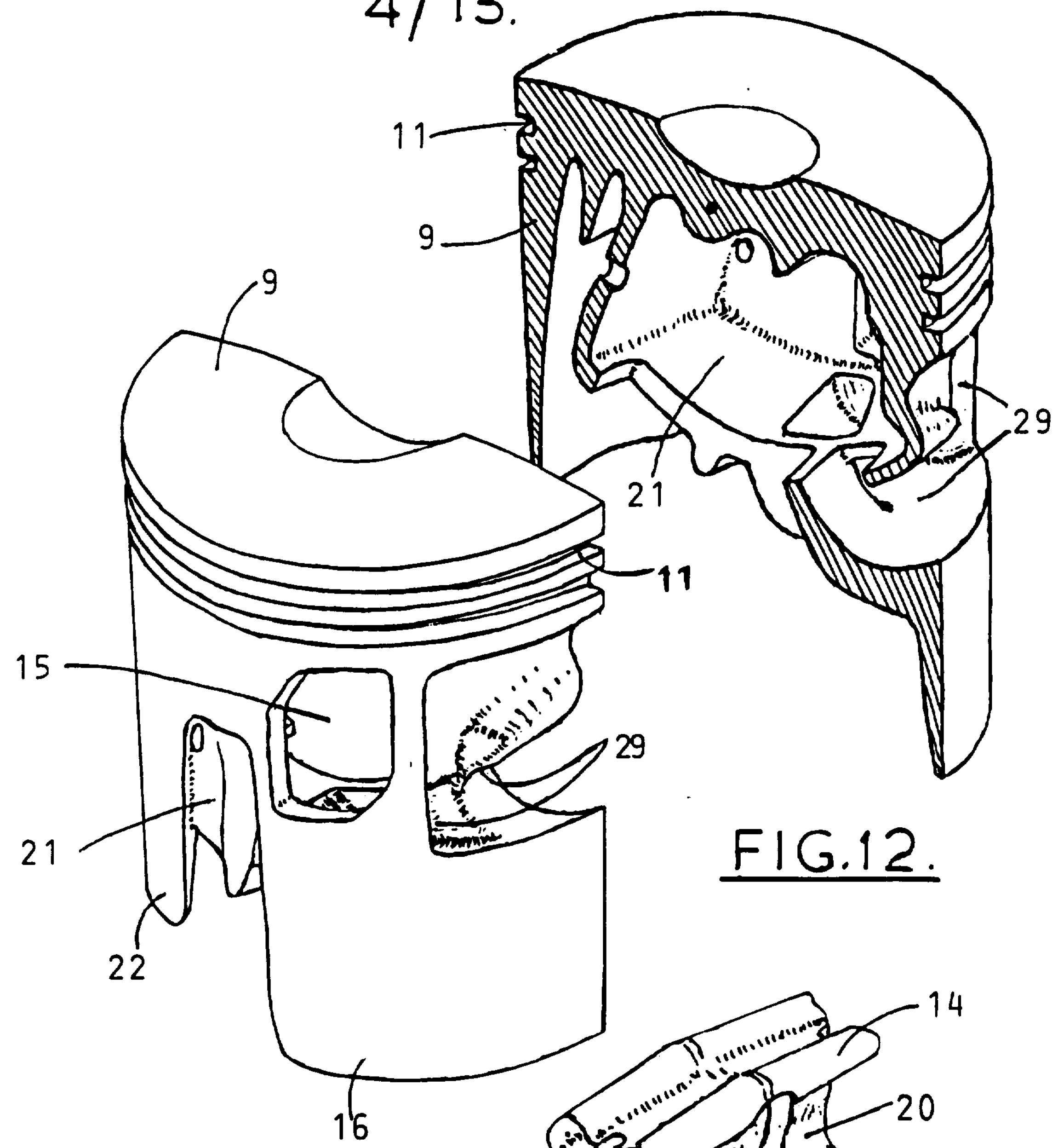
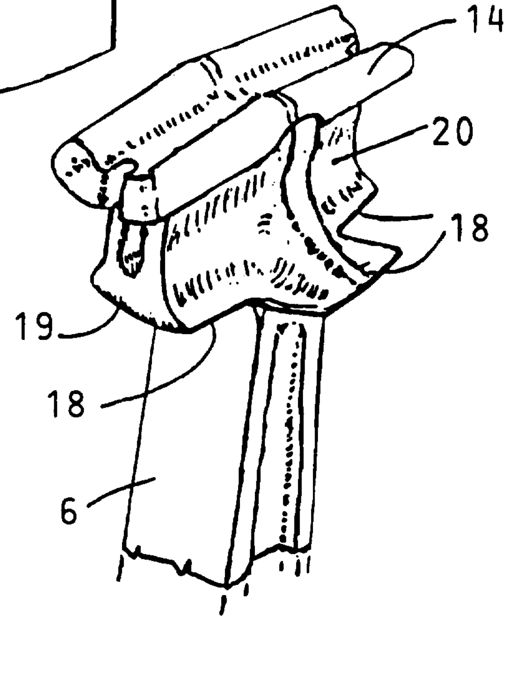


FIG.12.



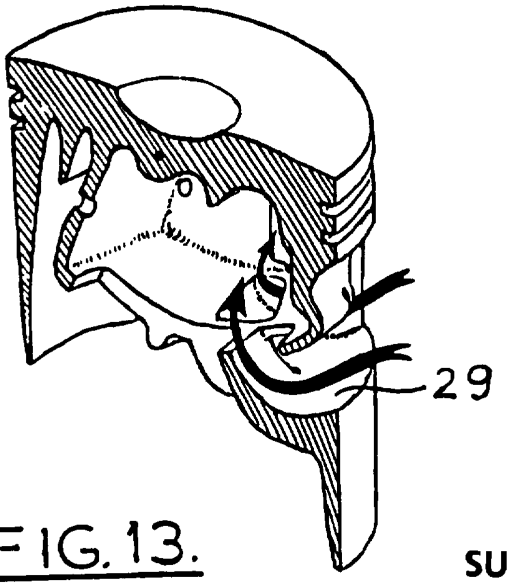


FIG.13.

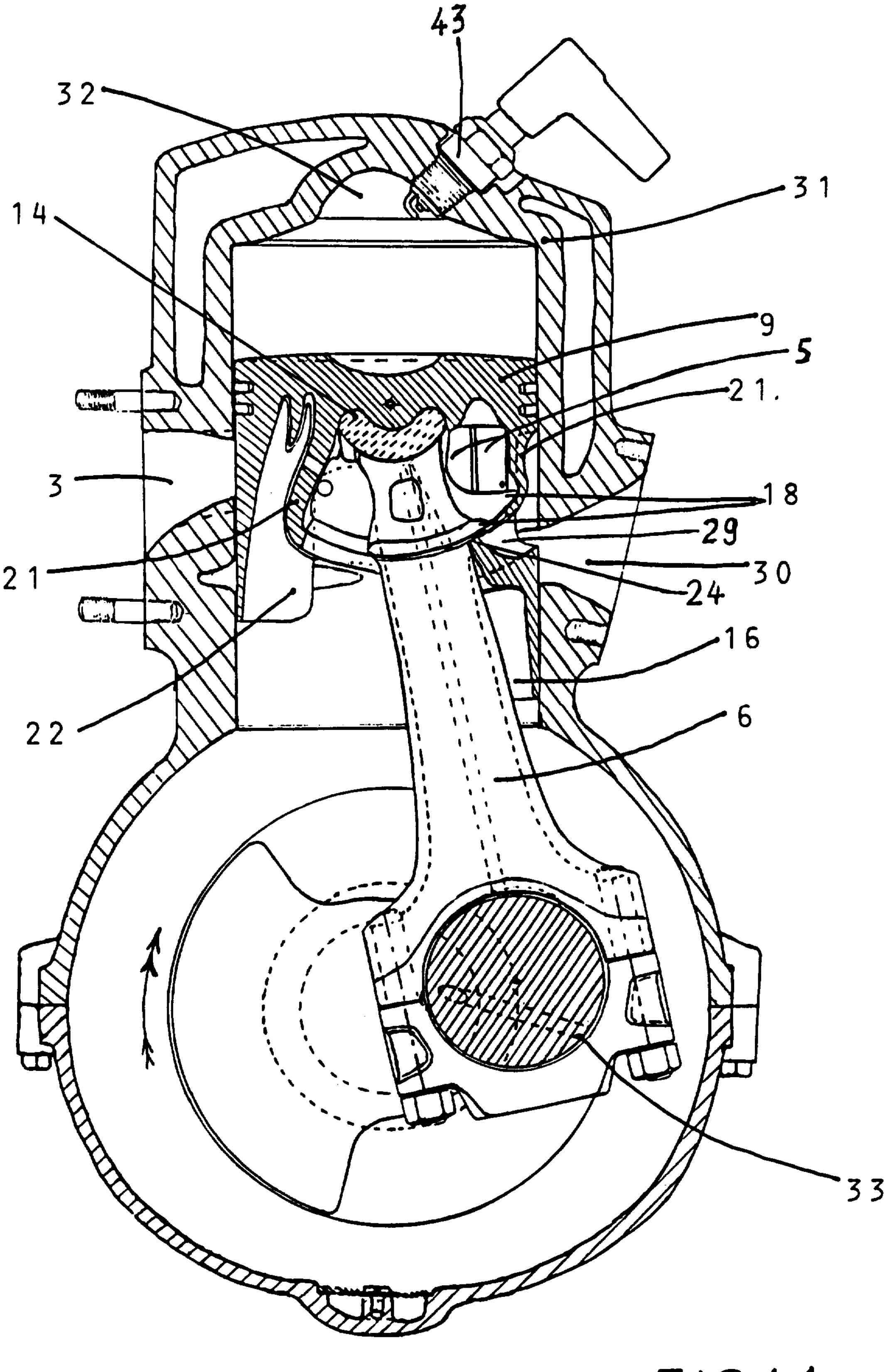


FIG.14.

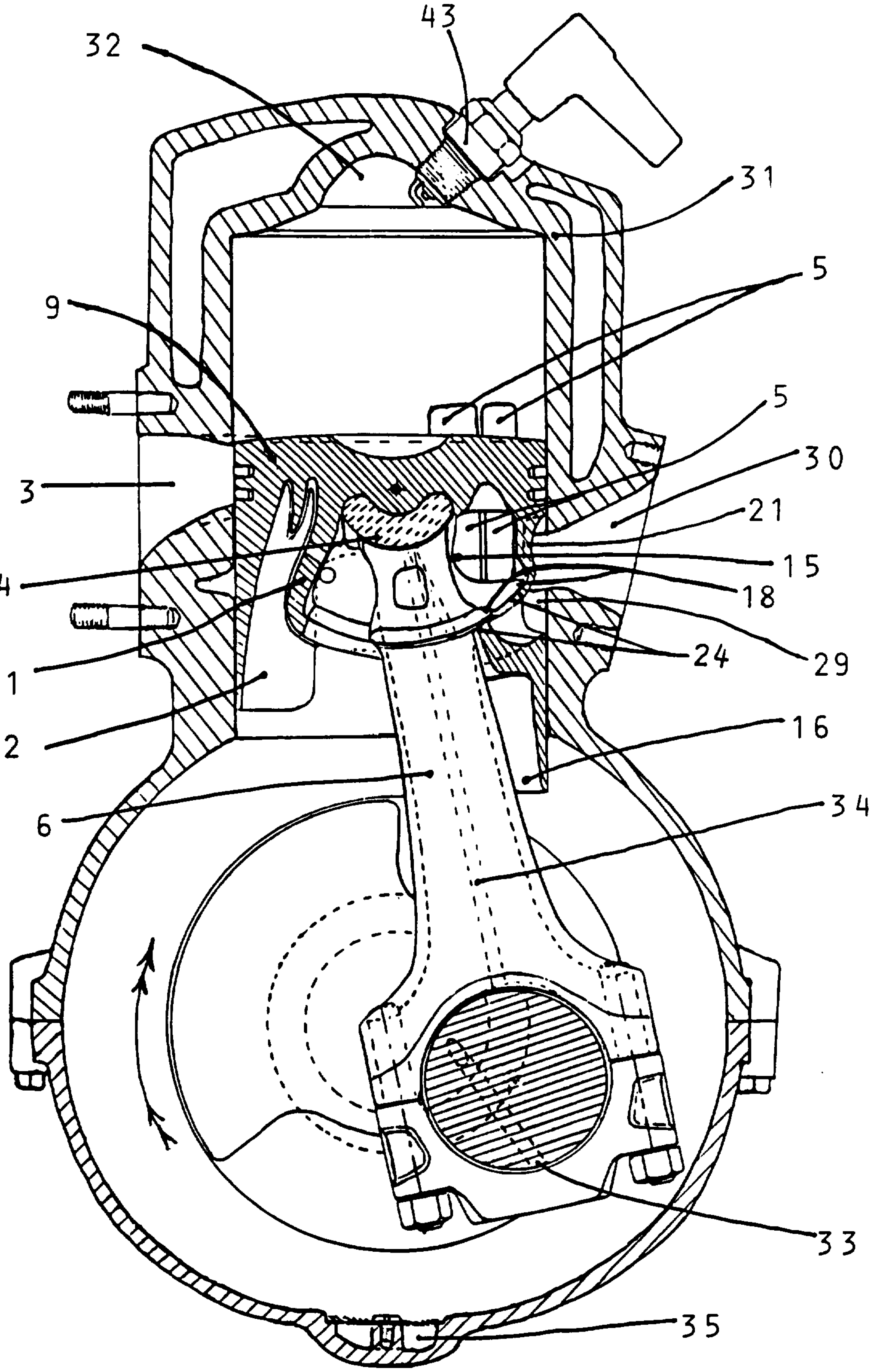


FIG. 15.

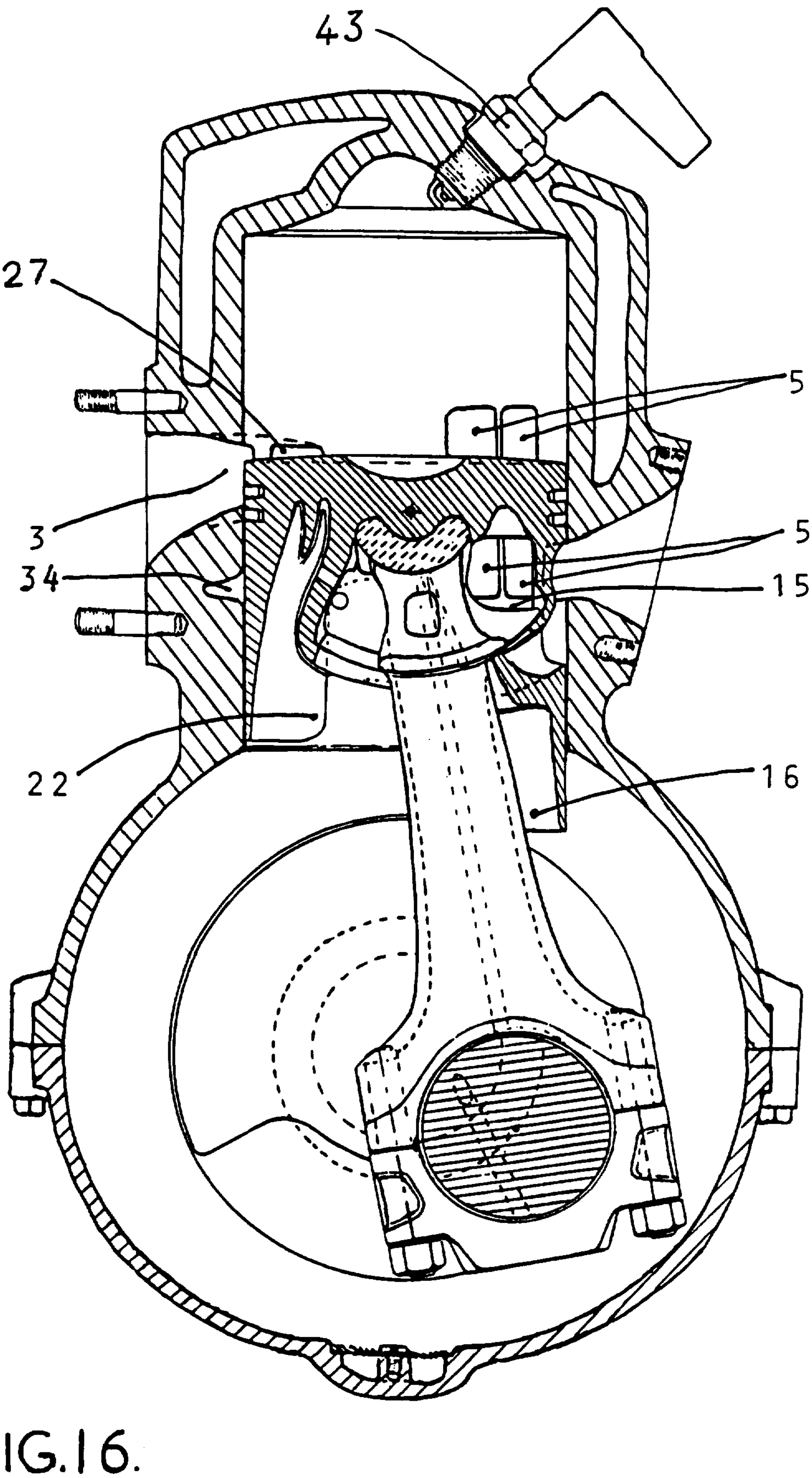


FIG.16.

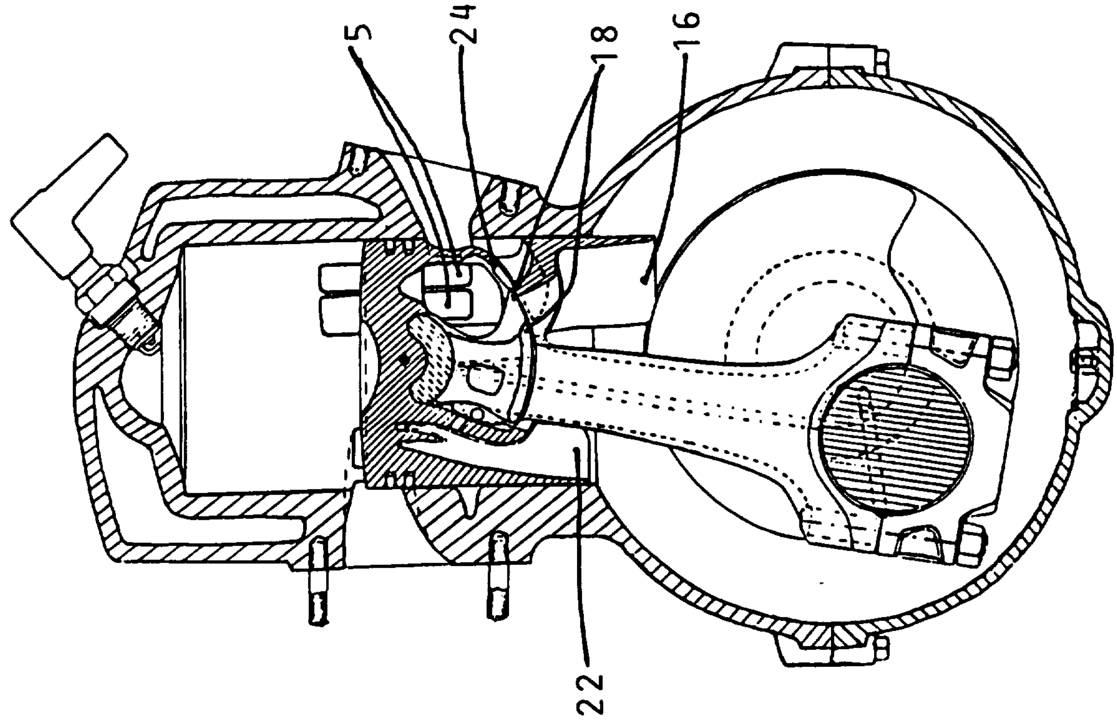


FIG.18.

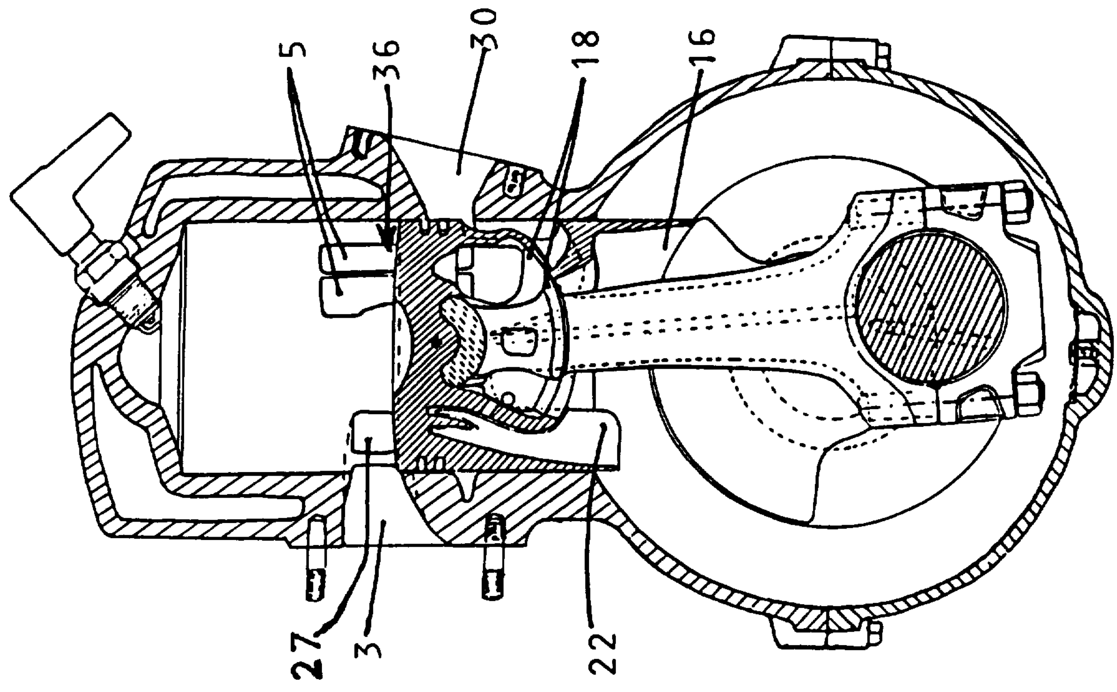


FIG.17.

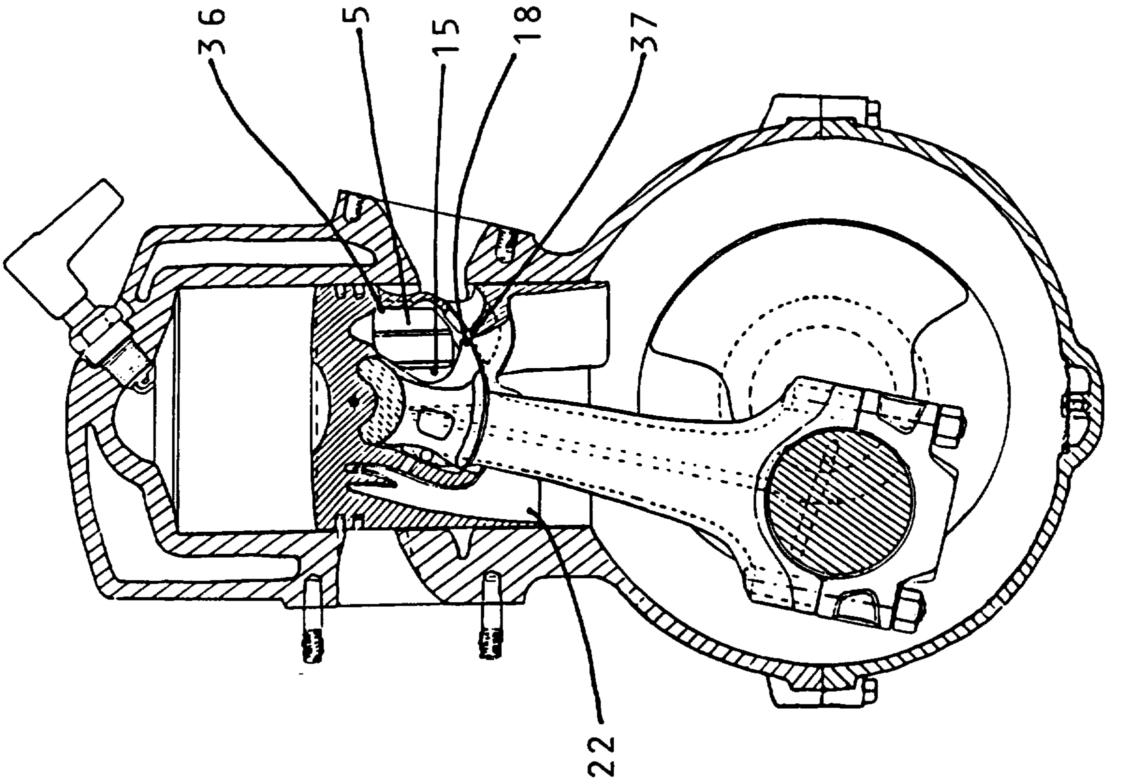


FIG.20.

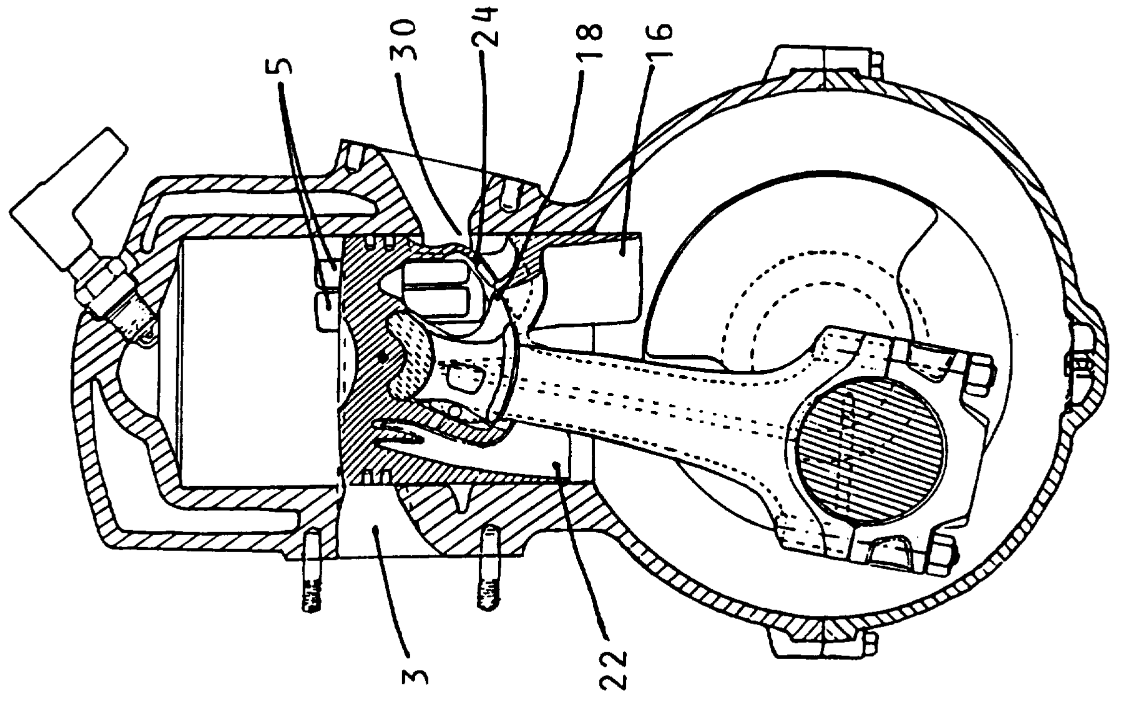


FIG.19.

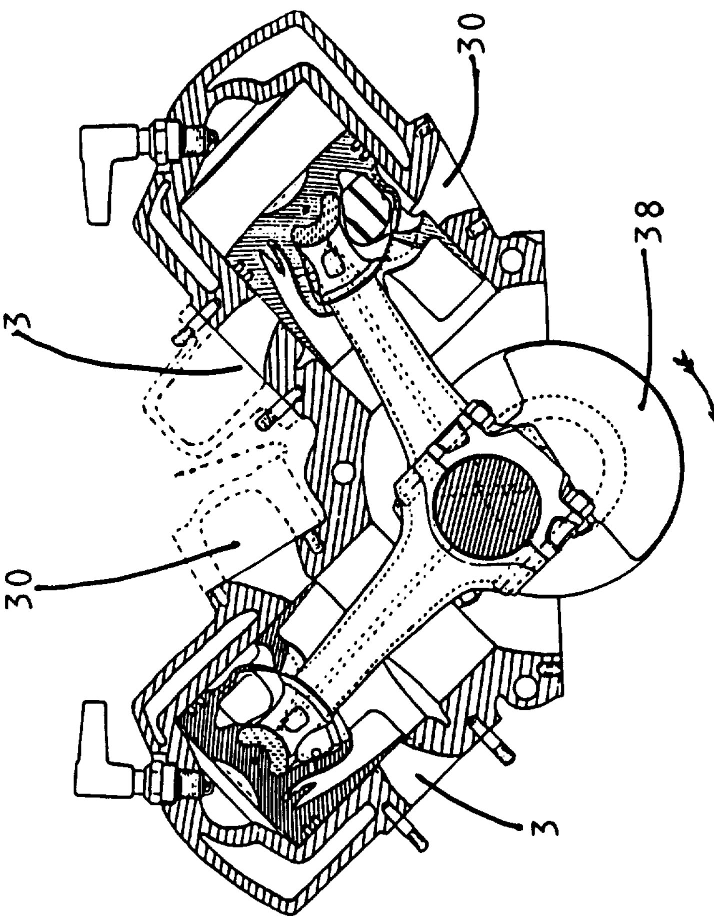


FIG.22.

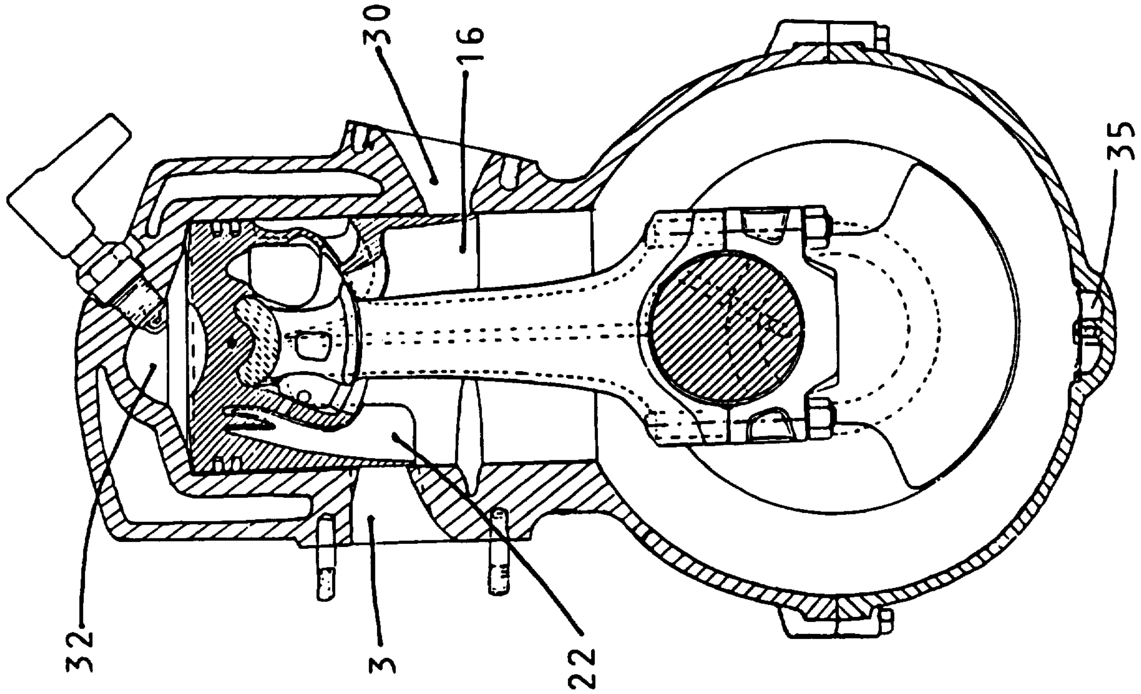


FIG.21.

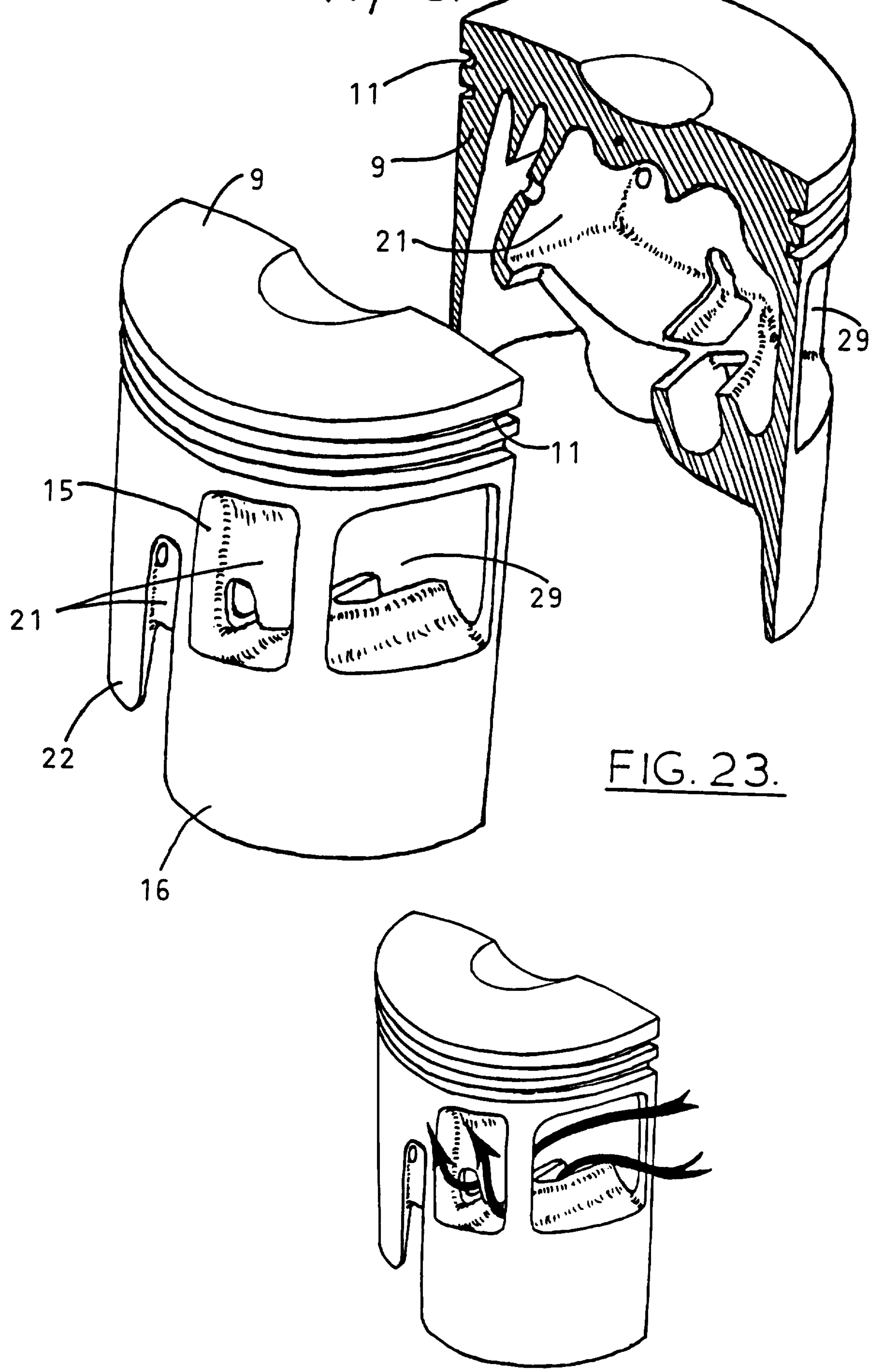


FIG. 23.

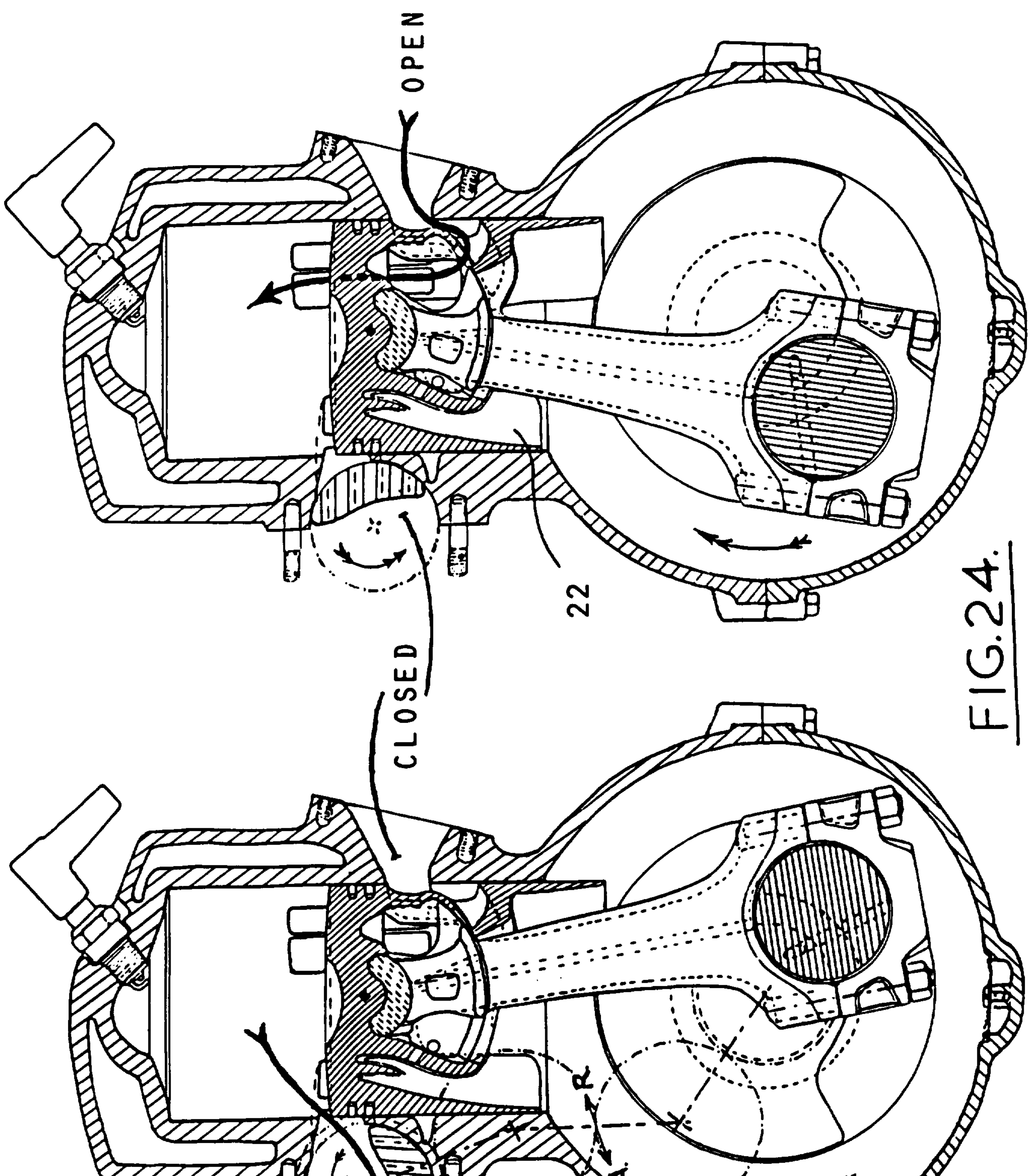
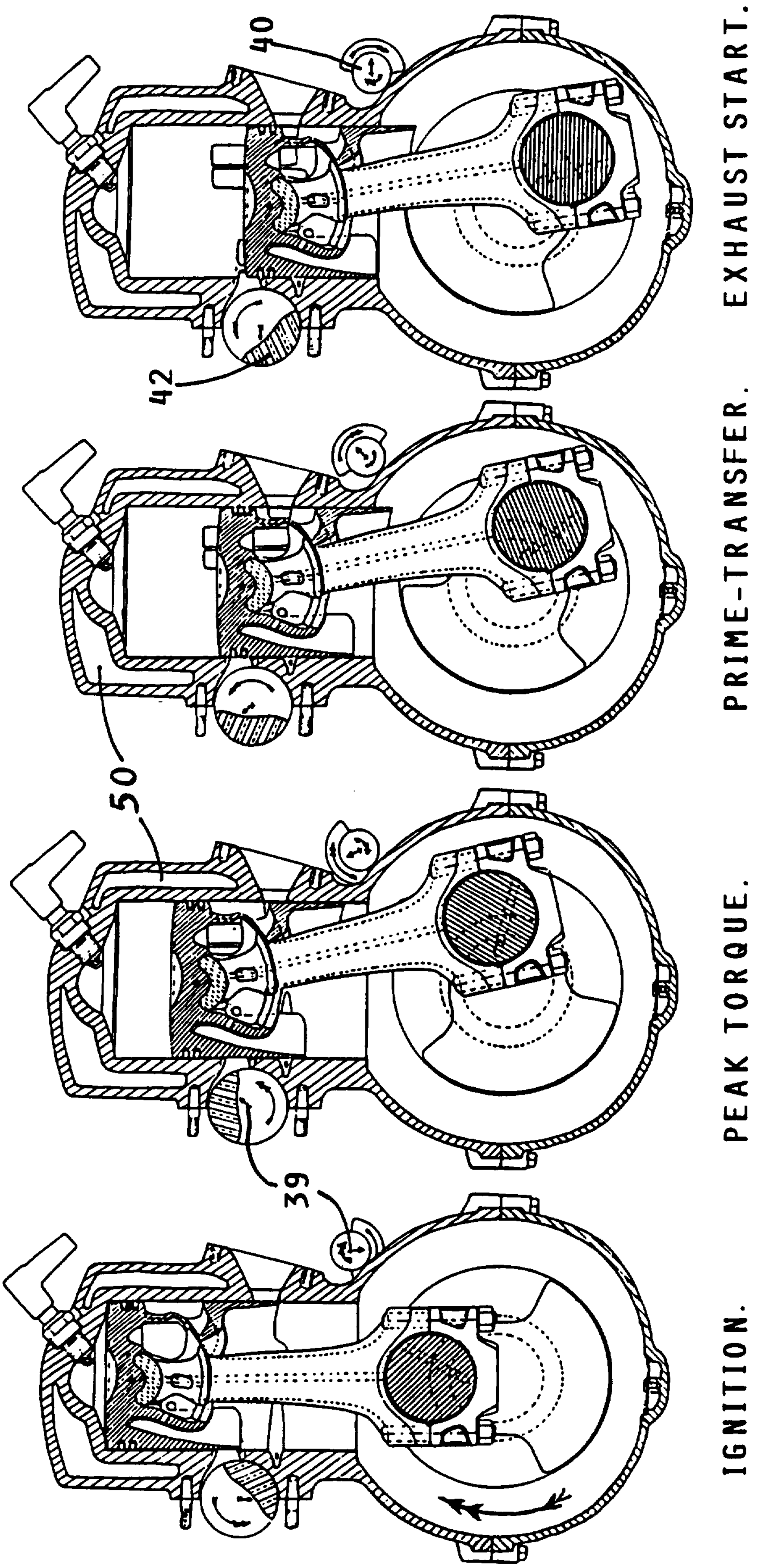


FIG.24.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42
EXHAUST START.
PRIME-TRANSFER.
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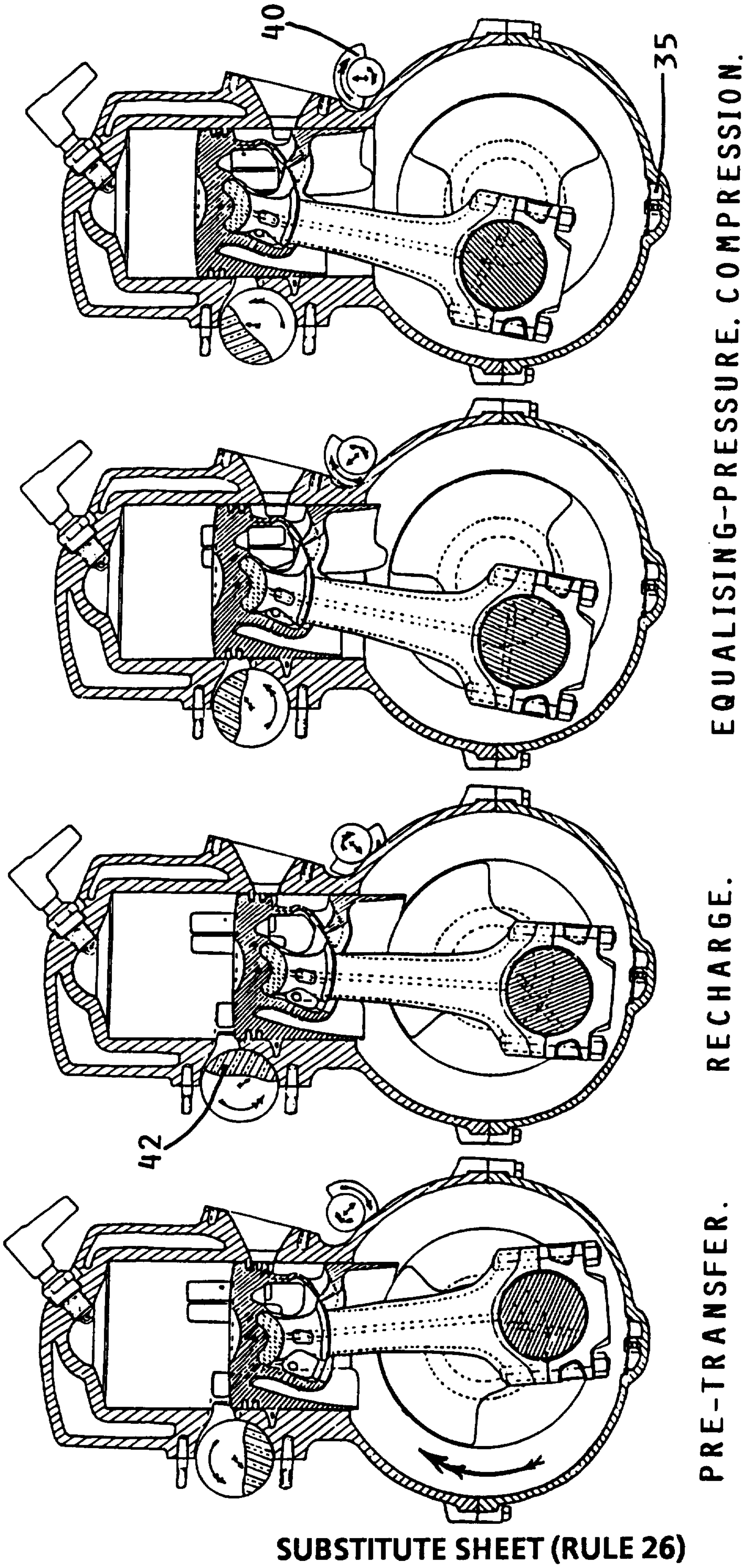
PEAK TORQUE.
39
IGNITION.

FIG. 25.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EQUALISING-PRESSURE. COMPRESSION.
RECHARGE.
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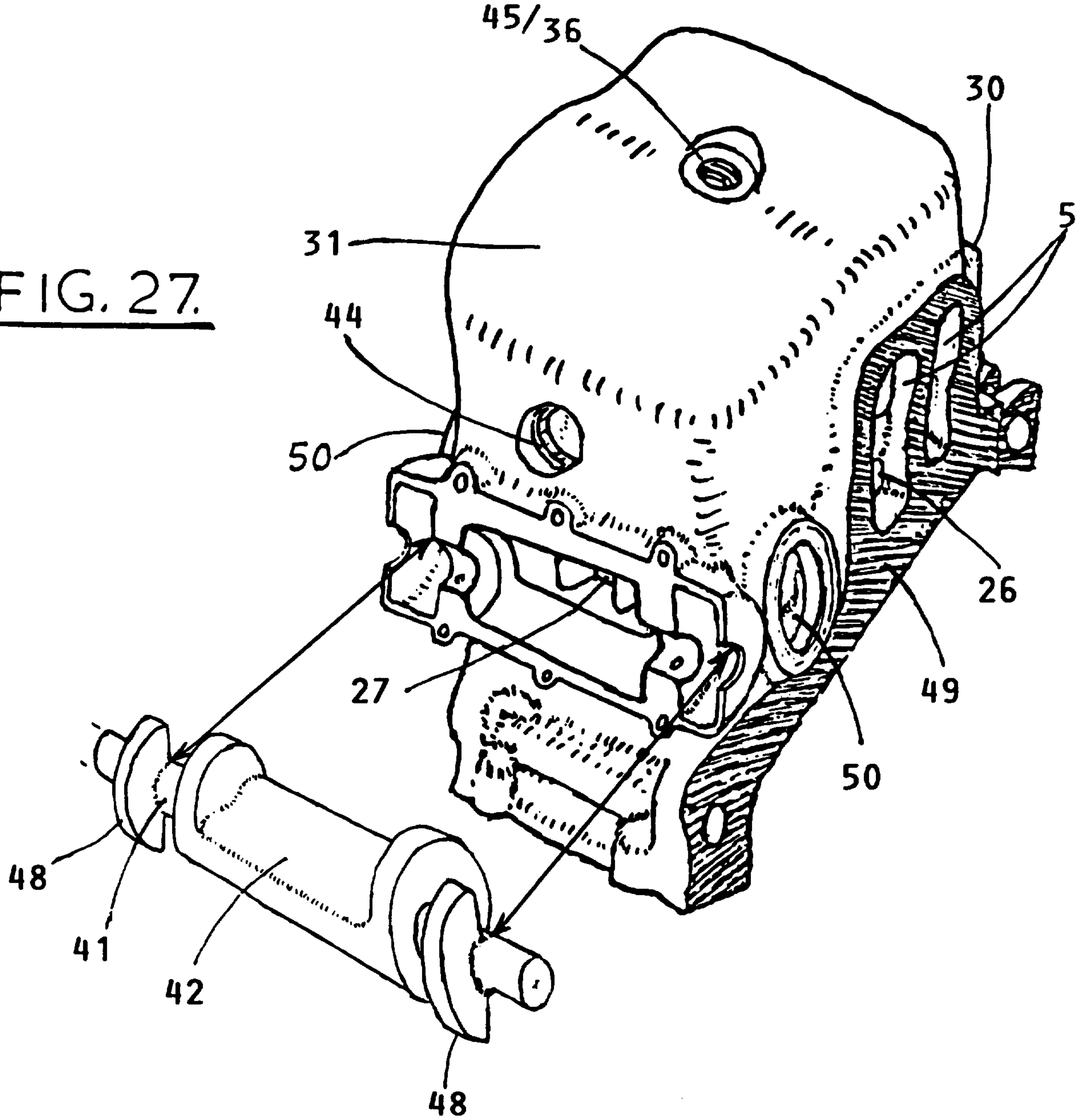
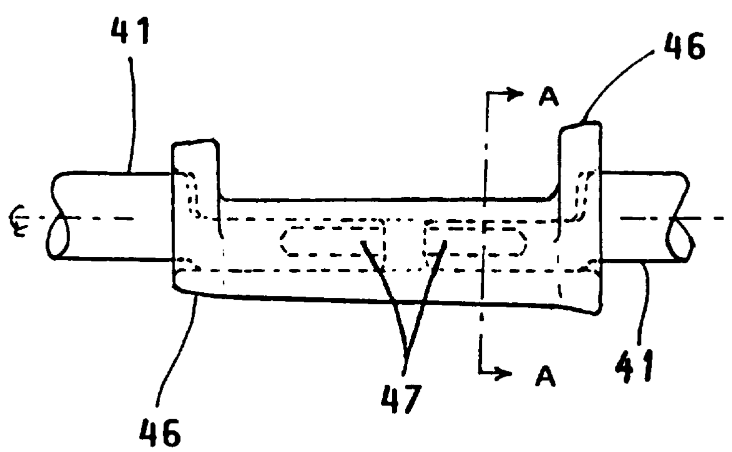
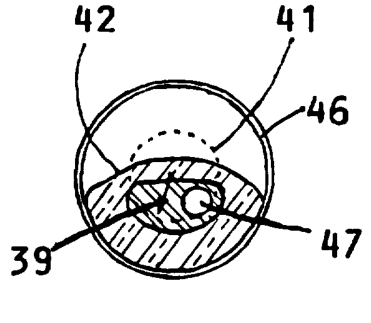
PRE-TRANSFER.

FIG. 26.

FIG. 27.







SECTION A A

INTERNATIONAL SEARCH REPORT

International Application No
PCT/GB 95/02952

A. CLASSIFICATION OF SUBJECT MATTER
IPC 6 F01L11/06

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
IPC 6 F01L F16J

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practical, search terms used)

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category *	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	DE,A,15 76 249 (ANSCHEIDT) 19 March 1970 see claims 1-9; figure 1 ---	1-4
A	FR,A,548 197 (LANDRIN) 9 January 1923 see the whole document ---	5-7
X	FR,A,689 089 (MIJNLIEFF) 26 September 1930 see the whole document ---	1-3
A	FR,E,16 210 (GASS) 10 December 1912 see the whole document ---	1-3
A	DE,C,809 117 (SCHART) 17 May 1951 see the whole document -----	1-3

☐ Further documents are listed in the continuation of box C.

☒ Patent family members are listed in annex.

* Special categories of cited documents :

'A' document defining the general state of the art which is not considered to be of particular relevance

'E' earlier document but published on or after the international filing date

'L' document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)

'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 when the document is taken alone

'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

1

Date of the actual completion of the international search	Date of mailing of the international search report
22 February 1996	29.02.96

Name and mailing address of the ISA	Authorized officer
European Patent Office, P.B. 5818 Patentlaan 2 NL - 2280 HV Rijswijk Tel. (+31-70) 340-2040, Tx. 31 651 epo nl, Fax: (+31-70) 340-3016	Wassenaar, G

INTERNATIONAL SEARCH REPORT

Information on patent family members

International Application No
PCT/GB 95/02952

Patent document cited in search report	Publication date	Patent family member(s)	Publication date
DE-A-1576249	19-03-70	NONE	
FR-A-548197	09-01-23	NONE	
FR-A-689089	26-09-30	NONE	
FR-E-16210		NONE	
DE-C-809117		NONE	