(12) INTERNATIONAL APPLICATION PUBLISHED UNDER THE PATENT COOPERATION TREATY (PCT)

(19) World Intellectual Property Organization
International Bureau

(43) International Publication Date
21 March 2002 (21.03.2002)

PCT

(10) International Publication Number
WO 02/23025 A1

(51) International Patent Classification[7]: **F02B 75/26**, F01B 3/04

(21) International Application Number: PCT/GB01/04077

(22) International Filing Date: 13 September 2001 (13.09.2001)

(25) Filing Language: English

(26) Publication Language: English

(30) Priority Data:
0022669.6 15 September 2000 (15.09.2000) GB
0121762.9 5 September 2001 (05.09.2001) GB

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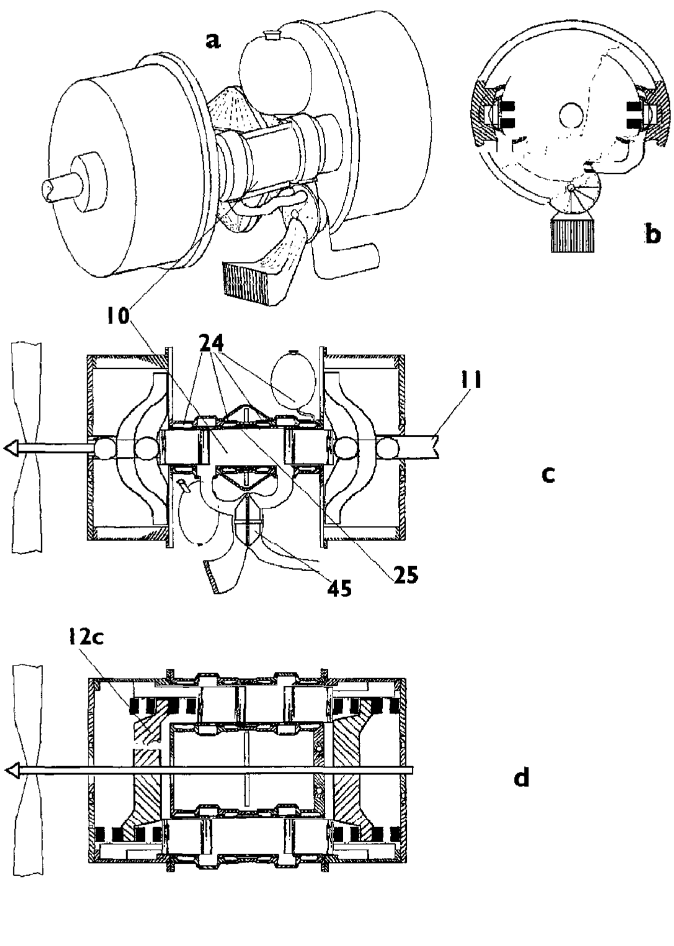
(81) Designated States *(national)*: AE, AG, AL, AM, AT, AU, AZ, BA, BB, BG, BR, BY, BZ, CA, CH, CN, CO, CR, CU, CZ, DE, DK, DM, DZ, EC, EE, ES, FI, GB, GD, GE, GH, GM, HR, HU, ID, IL, IN, IS, JP, KE, KG, KP, KR, KZ, LC, LK, LR, LS, LT, LU, LV, MA, MD, MG, MK, MN, MW, MX, MZ, NO, NZ, PH, PL, PT, RO, RU, SD, SE, SG, SI, SK, SL, TJ, TM, TR, TT, TZ, UA, UG, US, UZ, VN, YU, ZA, ZW.

(84) Designated States *(regional)*: ARIPO patent (GH, GM, KE, LS, MW, MZ, SD, SL, SZ, TZ, UG, ZW), Eurasian

[Continued on next page]

(54) Title: DIESEL INTERNAL COMBUSTION ENGINE



(57) Abstract: An internal combustion engine, preferably diesel, has at least two cylinders (10) and two drive shafts (11a, 11b), each cylinder having two opposed pistons (13) therein, with piston heads (13a, 13b) facing each other and piston rods of the pistons are coupled to the drive shaft or shafts through respective opposed cams (12a, 12b) and roller or slider arrangements. The opposed cams are profiled (23) such as to ensure that the opposed pistons are held at the top dead centre position until combustion is substantially complete. The opposed cams are also profiled such as to ensure that each cylinder as scavenged by more than itas own volume of air and such as to ensure that the exhaust ports close before the inlet ports, thereby ensuring that full supercharger or turbocharger pressure is achieved within the cylinder prior to compression. The pistons and rollers or sliders are fitted with lubrication channels to ensure that friction is reduced.

WO 02/23025 A1

patent (AM, AZ, BY, KG, KZ, MD, RU, TJ, TM), European patent (AT, BE, CH, CY, DE, DK, ES, FI, FR, GB, GR, IE, IT, LU, MC, NL, PT, SE, TR), OAPI patent (BF, BJ, CF, CG, CI, CM, GA, GN, GQ, GW, ML, MR, NE, SN, TD, TG).

Declarations under Rule 4.17:

— *as to applicant's entitlement to apply for and be granted a patent (Rule 4.17(ii)) for the following designations AE, AG, AL, AM, AT, AU, AZ, BA, BB, BG, BR, BY, BZ, CA, CH, CN, CO, CR, CU, CZ, DE, DK, DM, DZ, EC, EE, ES, FI, GB, GD, GE, GH, GM, HR, HU, ID, IL, IN, IS, JP, KE, KG, KP, KR, KZ, LC, LK, LR, LS, LT, LU, LV, MA, MD, MG, MK, MN, MW, MX, MZ, NO, NZ, PH, PL, PT, RO, RU, SD, SE, SG, SI, SK, SL, TJ, TM, TR, TT, TZ, UA, UG, UZ, VN, YU, ZA, ZW, ARIPO patent (GH, GM, KE, LS, MW, MZ, SD, SL, SZ, TZ, UG, ZW), Eurasian patent (AM, AZ, BY, KG, KZ, MD, RU, TJ, TM), European patent (AT, BE, CH, CY, DE, DK, ES, FI, FR, GB, GR, IE, IT, LU, MC, NL, PT, SE, TR), OAPI patent (BF, BJ, CF, CG, CI, CM, GA, GN, GQ, GW, ML, MR, NE, SN, TD, TG)*

— *as to the applicant's entitlement to claim the priority of the earlier application (Rule 4.17(iii)) for the following designations AE, AG, AL, AM, AT, AU, AZ, BA, BB, BG, BR, BY, BZ, CA, CH, CN, CO, CR, CU, CZ, DE, DK, DM, DZ, EC, EE, ES, FI, GB, GD, GE, GH, GM, HR, HU, ID, IL, IN, IS, JP, KE, KG, KP, KR, KZ, LC, LK, LR, LS, LT, LU, LV, MA, MD, MG, MK, MN, MW, MX, MZ, NO, NZ, PH, PL, PT, RO, RU, SD, SE, SG, SI, SK, SL, TJ, TM, TR, TT, TZ, UA, UG, UZ, VN, YU, ZA, ZW, ARIPO patent (GH, GM, KE, LS, MW, MZ, SD, SL, SZ, TZ, UG, ZW), Eurasian patent (AM, AZ, BY, KG, KZ, MD, RU, TJ, TM), European patent (AT, BE, CH, CY, DE, DK, ES, FI, FR, GB, GR, IE, IT, LU, MC, NL, PT, SE, TR), OAPI patent (BF, BJ, CF, CG, CI, CM, GA, GN, GQ, GW, ML, MR, NE, SN, TD, TG)*

— *of inventorship (Rule 4.17(iv)) for US only*

Published:

— *with international search report*

For two-letter codes and other abbreviations, refer to the "Guidance Notes on Codes and Abbreviations" appearing at the beginning of each regular issue of the PCT Gazette.

DIESEL INTERNAL COMBUSTION ENGINE

This invention relates to a diesel internal combustion (I. C.) engine.

Diesel engines have potentially several advantages for aero engines. Magnetos and sparking plugs are such a source of unreliability in petrol aero engines that aviation authorities insist on duplication. Also, the higher specific energy content of diesel fuel means that more payload can be carried. The higher combustion temperature of the diesel engine results in a higher efficiency, hence lower cost and environmental impact. In addition aircraft require high short-term power outputs for takeoff and climb, with sustained economic output at lower powers in the cruise. The ability of the diesel engine to produce a high torque at low speed results in lower noise emissions.

Diesel engines also have similar advantages for seagoing vessels. In particular certain vessels require high reliability and economy in the cruise, coupled with fast response and high short-term power output capability for manoeuvring in port.

The spark-ignition engine has the advantage over the conventional diesel engine in that ignition takes place very rapidly at the point of highest compression. In the conventional diesel engine fuel continues to be burnt well into the power stroke, reducing the thermal efficiency of the engine. There is considerable advantage to be had if a diesel engine can be made to approximate to the combustion cycle of the spark-ignition engine.

Referring firstly to Figure 1, a Jumo diesel engine is diagrammatically illustrated. Manufactured by the German Junkers company during the 1930s, the Jumo diesel engine was a two-stroke aero internal combustion engine with two opposed pistons per cylinder. The pistons 13 came together in the centre of the cylinder 10 at Top Dead Centre (TDC) 1, for fuel injection 25 and combustion, and were furthest apart at Bottom Dead Centre (BDC) 2, for exhaust gas removal and fresh air charging of the cylinders. The pistons drove crankshafts at opposite ends of the engine, 3 and 4 via connecting rods 9. The two crankshafts were linked by a gear-train 5 to rotate a drive shaft 11 coupled to a single propeller 6. The major advantage of the design lay in the fact that the air inlet 7 and exhaust gas exit 8 were effected through ports 19 and 20 respectively in the extremities of the cylinders 10 which were uncovered as the pistons 13 reached the end of the power stroke. Thus conventional poppet inlet and exhaust valves and their associated rocker and tappet assemblies with the associated lubrication, wear and maintenance, were eliminated. Further advantage was gained from the fact that the inlet air was supplied under pressure from a shaft-driven supercharger or exhaust gas-driven turbocharger, thus facilitating scavenging of exhaust gases.

Sections (a) to (f) of Figure 1 show the cylinder and pistons at different parts of the cycle.

The placing of the inlet 19 and exhaust 20 ports was such that as the power stroke ended, the exhaust port 20 opened first, allowing exhaust gases to rush out. (a). As the cylinder pressure fell to near atmospheric pressure, the inlet port 19 opened, admitting air at turbocharger pressure, to scavenge the cylinder 10 of remaining exhaust gases. (b). After the pistons reached bottom-dead-centre (BDC) 2, on the return stroke, the inlet port 19 closed first, then the exhaust port 20, and the air was compressed to high pressure (c, d). As the pistons 13 approached top-dead-centre (TDC) 1, (e), the fuel was injected 25 at high pressure into the space between the pistons 13 and ignited spontaneously, causing the pistons 13 to be driven back again with great force, to drive the crankshafts 3 and 4. (f).

The Jumo design was produced with six cylinders in-line, to produce very smooth running, but it had two disadvantages;

1. The gear trains to the propeller shaft were expensive, caused transmission losses and added weight.

2. In poppet valve diesel engines, the valve opening and closing times can be altered by the shape of the cams which drive them, and the exhaust valve can be closed before the inlet valve, allowing gas at turbocharger pressure to fill the cylinder, before cutting off the inlet, thus enabling a concentrated charge of air, which can sustain a larger fuel charge, and hence higher power output per stroke. In the Jumo engine the inlet port closes first, so the inlet cylinder pressure is close to atmospheric pressure, and power output per stroke is lower.

Figure 2 diagrammatically illustrates another known internal combustion engine, in which the cylinders 10 lie parallel to the drive shaft 11, and impart rotary motion to the shaft by means of an angled cam 12, mounted on it. The angle of slant of the cam 12 is such that the distance between the extremes of the face of the cam 12 as it rotates, is equal to the stroke of each piston 13. The axial face of the cam 12 varies sinusoidally with the shaft angle. The pistons 13 push against the angled face of the cam 12, rotating it. Such engines have been made in petrol- ignition or diesel form, with two or four strokes to the firing cycle.

Engines have been built in the past with pistons at both ends such as is illustrated in Figure 3. The inlet and exhaust gases enter and leave via conventional poppet valves 14 in each cylinder head 15. The design has the advantage of a small frontal area, which is attractive for aero-engine applications. It also has the advantage of the fact that the drive is direct to the shaft 11, without gearing, and that a plain cylindrical shaft is used, without expensive cranks. There are no connecting rods or big and little-end bearings, although there are bearing surfaces 13c required between the pistons 13 and the cam 12. The pistons 13 engage with the cam 12 with ball-ended sockets, or with a shoe, or with roller bearings 16 as illustrated in Figure 4, or with a slipper 35 as illustrated in Figure 11.

In addition the cam 12 does impart side forces to the pistons 13, and does require quite complex machining during manufacture. In practice it may have two or more cycles of axial variation per revolution as illustrated in Figures 5a, 5b and 5c. Figure 5a shows a cross section of the cam 12, Figure 5b shows a profile and Figure 5c shows an isometric view where there are four lobes 17 on the cam 12.

As diagrammatically shown in Figures 6a and 6b several cylinders 10 may also be arranged in a barrel arrangement around the shaft 11, rather like a revolver.

According to the present invention and illustrated diagrammatically herein in Figure 7, there is provided an internal combustion engine comprising at least two cylinders 10 each with two opposed pistons 13a, 13b therein, with piston heads of the pistons 13a, 13b facing each other, the piston rods of the pistons being coupled to a drive shaft 11 of the engine through two opposed cams 12a, 12b respectively.

The two opposed cams 12a, 12b have non-sinusoidal cam profiles. The two opposed cams 12a, 12b also have different profiles to optimise the inlet port 19 and exhaust port 20 opening and closing sequence. In particular by enabling the inlet ports 19 to remain open after the exhaust ports 20 have closed the pressure in the cylinders 10 may achieve turbocharger 45 or supercharger outlet pressure. Furthermore, the two opposed cams 12a and 12b are profiled such as to hold the two opposed pistons 13a and 13b at the top dead centre position 1 until combustion of all the fuel is completed.

The mass of air in a cylinder of an internal combustion engine with opposed pistons becomes fixed when the inlet ports and outlet ports become closed by the pistons as they return from the Bottom Dead Centre (BDC) position. The volume of air at this point is the uncompressed volume. When the pistons reach the Top Dead Centre (TDC) position, the volume of air is at the compressed volume. The ratio of the uncompressed volume to the compressed volume is the Compression Ratio.

It will be appreciated that in an internal combustion engine with the opposed piston arrangement with the drive shaft linked through two opposed cams as described above the pistons are directly coupled to the drive shaft of the engine through the opposed cams and that the rotation of the drive shaft is synchronised to the two opposed cams and that the pistons meet at the Top Dead Centre (TDC) position for every compression stroke of the engine and that the compression ratio of the engine is thereby fixed. It will also be appreciated that because the opposed pistons operate on the opposed cams at a fixed radius 21 (Figure 7) the torque generated by this arrangement will be greater than that from a conventional crankshaft engine design, and that power will be generated at a lower speed.

For a better understanding of the invention and to show how the same may be carried into effect, reference will now be made, by way of example, to Figures 7 to 12 of the accompanying drawings, in which:-

Figure 7 is a diagrammatic illustration of one form of the present diesel internal combustion engine;

Figure 8 is a diagrammatic illustration, similar to Figure 1, showing the working cycle of the present engine;

Figure 9 shows a series of illustrations of a typical two-cylinder engine of the present construction, in which Figure 9a is a perspective view;

Figure 9b is a front view with parts broken away;

Figure 9c is a side view;

Figure 9d is a plan view.

Figure 10 illustrates a cam and roller design for the present engine, with Figure 10a, Figure 10b and Figure 10c showing one form of cam and roller design, whilst Figure 10d shows a modification; and

Figures 11a, 11b and 11c illustrate a modification of a slipper design of cam and piston assembly, with Figure 11c being a diagrammatic section taken along line A-A in Figure 11b.

Figure 12a shows a typical spark-ignition engine combustion cycle with a compression ratio of 10.

Figure 12b shows a typical diesel engine combustion cycle with a compression ratio of 10.

Figure 12c shows a typical diesel engine combustion cycle with a compression ratio of 20.

Figure 12d shows a typical combustion cycle for the present invention with a compression ratio of 20.

The present construction uses the 2-stroke opposed piston concept, but with the cylinders 10 arranged in a barrel around the drive shaft 11, and with the pistons 13 acting on two cams 12 at the front and the rear of the engine respectively as shown in Figure 7. Because the pistons 13 in each cylinder act in opposition, there is no net axial force on the shaft 11, so light thrust bearings 17 can be used, which are rated to carry the axial thrust from the propeller (not shown) or other driven load. Figure 7 is a diagrammatic arrangement, showing a two-cylinder design, with two-lobed cams 12, i.e., each cylinder 10 fires once per revolution. The pistons 13 in each cylinder are 180 degrees out of phase with the other. A simple cam disc 18 on the shaft 11 operates the fuel injector to each cylinder, once per revolution. Induction and exhaust are through respective ports 19, 20 in the cylinder walls, as in the Jumo design, and there are no poppet valves. The only gear drive, shown at one end of the shaft 11, as shown at 22, is for auxiliaries such as high-pressure fuel pump, oil pump and supercharger. Turbo-charging may alternatively be achieved from an exhaust-gas driven turbine 45. The cam 12a is coupled to the pistons 13a which control the inlet ports 19 whilst the cam 12b is coupled to the pistons 13b which control the exhaust ports 20.

Features specific to the present construction include the inlet and exhaust timing.

In the prior Jumo design, the inlet port must open after and close before the exhaust port, as the piston movement bears a fixed relationship to the crankshaft rotation. With the present construction, as illustrated in Figure 8 for example, the cams 12 do not have to have the same sinusoidal profile; indeed non-sinusoidal profiles can be used for each, and the inlet and exhaust cams 12a, 12b, can have different profiles. During the firing stroke, both ports 19, 20 are closed, and it is advantageous for the cams 12 to have the same profile for this part of the stroke, to equalise axial thrust forces, but towards the end of the stroke the exhaust port opens first. (position (a) in Figure 8). As the exhaust gases leave, the cylinder pressure falls rapidly. As it falls below turbocharger pressure, the inlet port opens. (Position (b)). This allows air at turbocharger pressure to sweep into the cylinder and scavenge out remaining exhaust gases. In most two-stroke engines, some exhaust gases remain in the cylinder, diluting the air and reducing output power. In the present construction, the port timing is such that the cylinder 10 is scavenged by more than its own volume of air, before the exhaust port 20 closes, thus removing all exhaust gases. When the exhaust port 20 closes, (c), air continues to enter and reaches turbocharger pressure as the inlet port 19 closes. (d). The air is then compressed by the approaching pistons, until just before Top Dead Centre 1, fuel is injected 25 at very high pressure, and spontaneous combustion takes place, (e). Fuel injection is initiated by a shaft-mounted cam disc 18 operating on the injector 25. Output power is varied by varying the duration of fuel injection. This ability to vary port timing gives an inherent advantage over the Jumo design, with higher power output per cylinder cycle, and higher fuel efficiency.

Additional features relate to shaft bending and engine speed.

The arrangement shown in Figure 7 is not ideal from a balance point of view. The driving forces from the pistons 13 impose bending moments on the shaft 11 which will produce a rotating unbalance exciting force. An alternative arrangement as shown in Figure 9, uses cylinders 10 on opposite sides of the shaft 11, and cams 12 with four lobes, i.e., with two cycles of axial movement per revolution as shown previously in Figure 5. These cylinders 10 fire at the same time, and the axial thrust components will be equal end eliminate bending moments on the shaft 11. In addition this arrangement gives two firing cycles per engine revolution, which means that for a given power output shaft speed will be halved; this is particularly advantageous for a diesel engine used for driving a propeller. Diesel engines are most effective at high cylinder firing rates, whilst for aircraft, the maximum permitted shaft speed is determined when the propeller tip-speed approaches the speed of sound.

In principle a larger number of cycles of movement can be used on the cams to reduce engine speed further. If S is the number of firing strokes per second per cylinder, and L is the number of cam lobes, the shaft speed N is given by

$$N = 120S/L \text{ rpm.}$$

However, only certain combinations of number of cylinders and lobes will give vibration-free operation. Because the pistons are the only reciprocating masses, and these are opposed in pairs, negligible vibration arises from this source. The rotating cams are almost mirror images of each other and so primary balance is achieved.

In a horizontal layout, typical arrangements would be as in Figures 9a-9d. In this case an exhaust-driven turbocharger 45 is shown.

By standardising on a single cylinder and cam size, with the cylinder centrelines on a standard diameter, output can be increased in stages by adding further cylinders whilst minimising casting and machining costs. A water-cooled arrangement is shown, with cooling water in cast ducts in the casing 24 and in direct contact with the cylinder liners 25.

Even with 4-cylinder and 6-cylinder designs, the overall dimensions are very similar to the two-cylinder arrangement shown in Figure 9.

Further features of the present construction relate to the cam and roller or slipper arrangements.

Diesel engines have extremely high cylinder pressures and as a result, high thrust forces and bearing pressures. Hence the cam and roller design has to have fine tolerances to minimise free play. The cam, as well as experiencing the axial force from the piston, experiences sideways forces, and these have a reaction force on the piston. Thus the cylinder walls and piston slider experience lateral forces. The lubrication system must allow for this. Figures 10a to 10d illustrate a cam and roller design provides a combination of rolling and sliding contact. The rollers 26 are carried on a slider 27 which is a precision fit in the slider housing 27a. Furthermore the use of roller bearings 28 within the roller 26 will give best precision and lowest losses. Lubrication channels 29 are provided in the slider from the housing with oil directed onto the bearing surfaces.

From Figure 10 it can be seen that with the parallel roller design there will be a scrubbing action due to the rotation (arrow A in Figure 20a) of the cam at different radiuses on the roller.

Figure 10d shows an improved arrangement in which tapered rollers 26a are used, with a taper angle such that all contact is purely rolling. However this will result in a radial reaction force and lubrication channels 29 are provided in the surface of the slider to ensure low friction forces.

Alternative arrangements to the cam and roller assembly are possible. One design used for barrel engines has ball-ended pistons which slide in an annular slot in the cam and this could be used for the diesel engine. However care would be required in the lubrication system, as access is difficult with this arrangement.

An alternative arrangement is a simple slipper design, with sliding surfaces, as shown in Figures 11a to 11c. A nose 30 of the slipper nearest the piston 13 will experience the full driving force of the piston, and lubrication arrangements are similar to those for the roller design. The oil is fed via a slot 31 in the slider of the piston, and hollow ducts 32 through the slider, to the nose of the slipper, emerging at the leading edge just ahead of the line of contact 33. Preferably, fine grooves 34 in the surface of the cam 12 and the slipper 35 concentrate oil flow at the point of peak bearing pressure. An outermost face of the slipper 36 only carries the force required to compress the air on the compression stroke, and this is much less than the power stroke force, but a suitable lubrication channel 37 is provided. As with the roller design, oil is forced onto the cam 12 during the power stroke, and is directed onto the cylinder liner 10a during the compression stroke, thus ensuring good piston lubrication. An oil scraper ring 38 ensures that excessive oil does not enter the cylinder chamber, whilst an additional oil duct 39 leads oil to the side of the slider to allow for the side-thrust.

This arrangement gives lowest cost and fewer moving parts, but despite using hardened surfaces, wear rates on the slipper and cam may prove high.

Further features of the present construction relate to the compression and combustion strokes.

A typical spark-ignition engine has a maximum compression ratio of about 10, because the fuel-air mixture in the cylinder is likely to ignite prematurely during the compression stroke at higher compression ratios. This is due to the temperature rise which occurs during compression and the premature ignition can lead to engine damage. As the piston approaches Top Dead Centre a spark initiates combustion which leads to rapid fuel burn and a powerful subsequent power stroke. Thus all of the heat energy is added at or just after Top Dead Centre. The resulting high pressure, as shown in Figure 12a is available to drive the piston down for the majority of the power stroke. Figure 12a shows how the cylinder pressure varies with cylinder volume as the piston moves through its complete cycle. The area within the curve 40 represents the energy available per cycle. The area below the curve 41 represents the lost energy, and because of the high area within the curve, this cycle, known as the Otto cycle, has a high efficiency.

A conventional diesel engine, operating at the same compression ratio of 10 would have a lower efficiency. Because fuel injection commences near top dead centre in a diesel engine to prevent spontaneous ignition during the compression stroke, there is a time delay whilst the fuel is injected and atomised. Thus combustion is not entirely completed even when the piston is well down its power stroke, and peak combustion pressure is not available as in the spark-ignition engine. In practice the pressure in a conventional diesel engine stays almost constant during the power stroke, as shown in Figure 12b. The energy available per cycle 42 is reduced, and hence efficiency is lower than in a comparable spark-ignition engine. In addition because combustion is still taking place well into the power stroke, partially-burnt fuel is usually emitted with the exhaust gases.

In practice diesel engines will not operate reliably at compression ratios as low as 10, because of irregular spontaneous ignition. Instead a lower limit of about 14 is used, and more usually, compression ratios in the range 20 to 25 are used. These higher compression ratios result in a higher power output 43 and efficiency as shown in Figure 12c because of the higher pressures achieved, and typical diesel engines are considerably more efficient than spark-ignition engines. However, because of the higher combustion pressures achieved, conventional diesel engines experience much higher crankshaft forces and have to be made much stronger and hence heavier, than spark-ignition engines.

In the present construction, the opposed cams 12a and 12b are profiled 23 such that the heads of the opposed pistons 13a and 13b are held together at the top dead centre position 1 for the duration of the fuel injection period, and for some time longer until combustion is complete. The pistons 13a and 13b are then released ,and the full combustion pressure is available for the power stroke. Thus the present invention combines the high efficiency of the Otto cycle with the high efficiency of the high-compression ratio diesel engine to produce an engine of even higher power output 44 and efficiency as shown in Figure 12d. In addition, because combustion is substantially completed before the power stroke commences, emissions of partially burnt fuel are much reduced.

In the present construction the opposed cams 12a and 12b have profiles for the power stroke which are mirror images of each other, and therefore the forces generated on the cams 12a and 12b are equal. The tangential forces are in the same direction and impel the drive shaft 11 around to produce the motive power for the propeller 6 or other load. The axial forces on the cams 12a and 12b however are equal and opposite and balance each other out along the drive shaft 11. Thus a heavy engine casing is not required and a diesel engine may be constructed which is considerably lighter than a conventional diesel engine of the same power output and the lack of unbalance forces during the power stroke results in an engine with lower noise emissions.

It will be appreciated that the combination of the two opposed pistons per cylinder, two crankshaft, two-stroke diesel engine with the opposed cam design having cam profiles such as to replicate the Otto combustion cycle enables a diesel engine to be built with higher efficiency and power output, lower weight, lower emissions and reduced noise by generating full power at a lower speed.

It will be further appreciated that the use of cam profiles such that the exhaust port closes before the inlet port thereby ensuring that full supercharger or turbocharger pressure is attained in the cylinder prior to compression, results in a higher power output and thermal efficiency of the engine and that the use of multiple cam lobes optimises the number of firing strokes and engine revolutions, particularly for aero-engine applications, and improves engine balance.

It will be further appreciated that the use of taper-roller bearings minimises wear between cams and rollers whilst the option of using a simple slipper design with tailored lubrication channels in the sliders reduces the number of moving parts and reduces friction arising from axial, radial and side-thrust forces.

Finally, it will be appreciated the present construction has considerably fewer moving parts than a conventional petrol engine, previous cam engines or the Jumo diesel engine. This makes for easier construction, improved reliability, lower weight and lower maintenance costs.

CLAIMS

1. An internal combustion engine comprising at least two cylinders each with two opposed pistons therein, with piston heads of the pistons facing each other, piston rods of the pistons being coupled to a drive shaft of the engine through two opposed cams, one cam controlling at least one inlet port and the other cam controlling at least one exhaust port, each cam having a non-sinusoidal profile and the two opposing cams having different profiles such that at the completion of the expansion stroke each cylinder is scavenged by more than its own volume of air before its associated exhaust port closes and such that the exhaust port of each cylinder closes before the inlet port thereby enabling full supercharger or turbocharger pressure to be attained in each cylinder and such that at the completion of the compression stroke the pistons are held together at the Top Dead Centre position until fuel injection and combustion are substantially completed thereby enabling a high thermal efficiency to be achieved.

2. An engine according to claim 1 wherein each piston has an associated slider arrangement by which the associated cam is coupled to its piston.

3. An engine according to any one of the preceding claims wherein the bearing surfaces between each of the pistons and the cylinder walls and of the two opposed cams and its associated sliders are cooled and lubricated by channels in each piston and slider which direct oil onto the bearing faces of the cylinders, cams, pistons and sliders.

4. An engine according to any one of the preceding claims wherein the cylinders are arranged in a barrel formation around the drive shaft

5. An engine according to any one of the preceding claims, and having four cylinders and associated pistons and cams.

6. An engine according to any one of the preceding claims, and having six cylinders and associated pistons and cams.

7. An engine according to any one of the preceding claims and being a two-stroke engine.

8. An engine according to any one of the preceding claims and being a diesel engine.

9. An internal combustion engine, substantially as hereinbefore described, with reference to Figures 7 to 11 of the accompanying drawings.

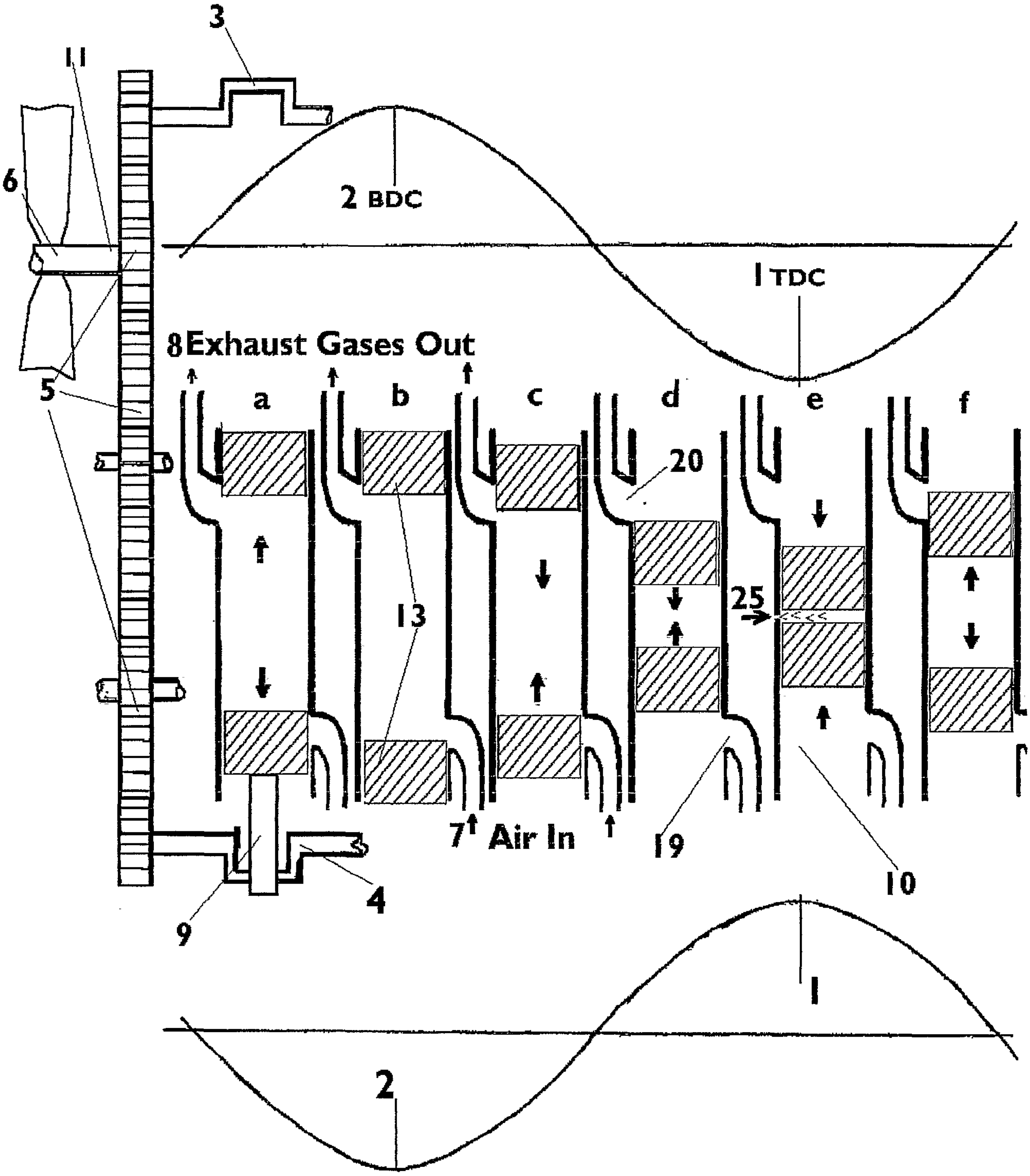

3
11
6
2 BDC
1 TDC
8 Exhaust Gases Out
5
a
b
c
d
e
f
20
13
25
7 Air In
19
10
9
4
1
2

Figure 1

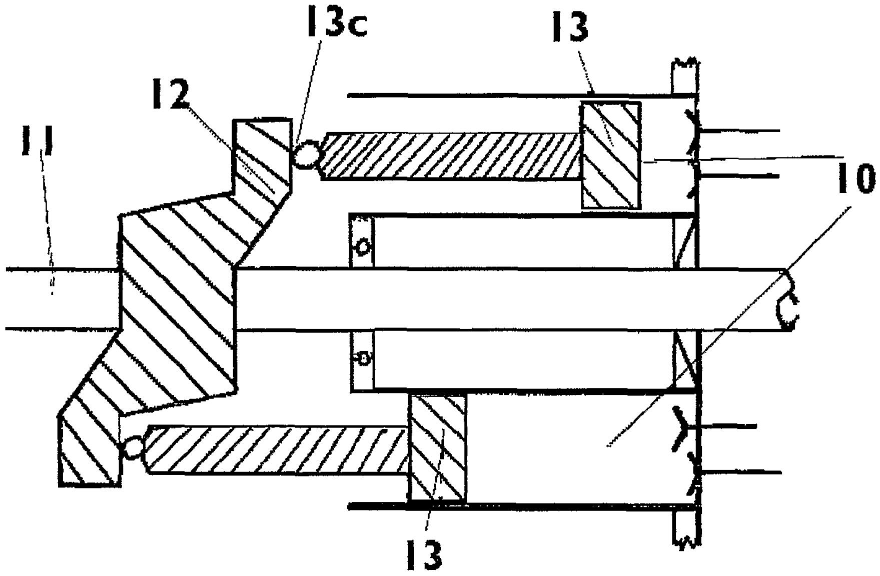


Figure 2

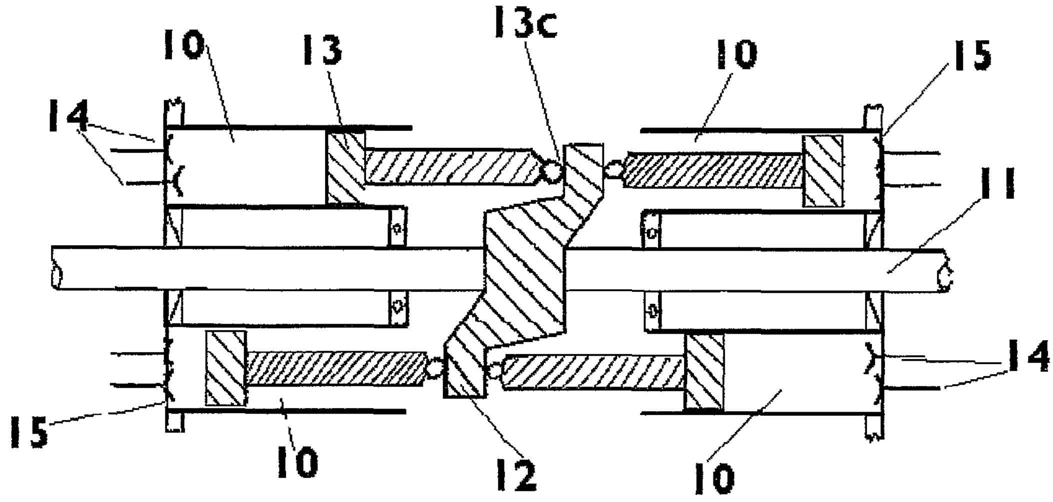


Figure 3

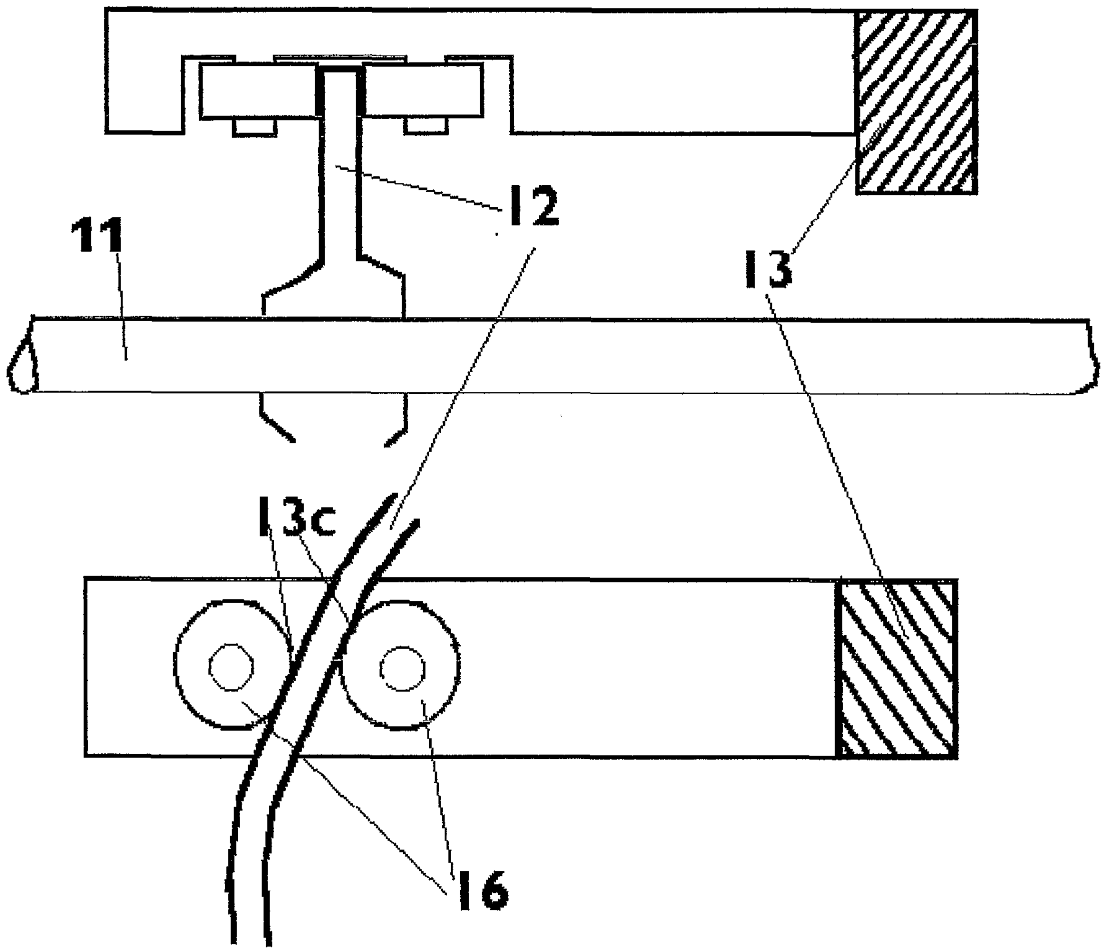


Figure 4

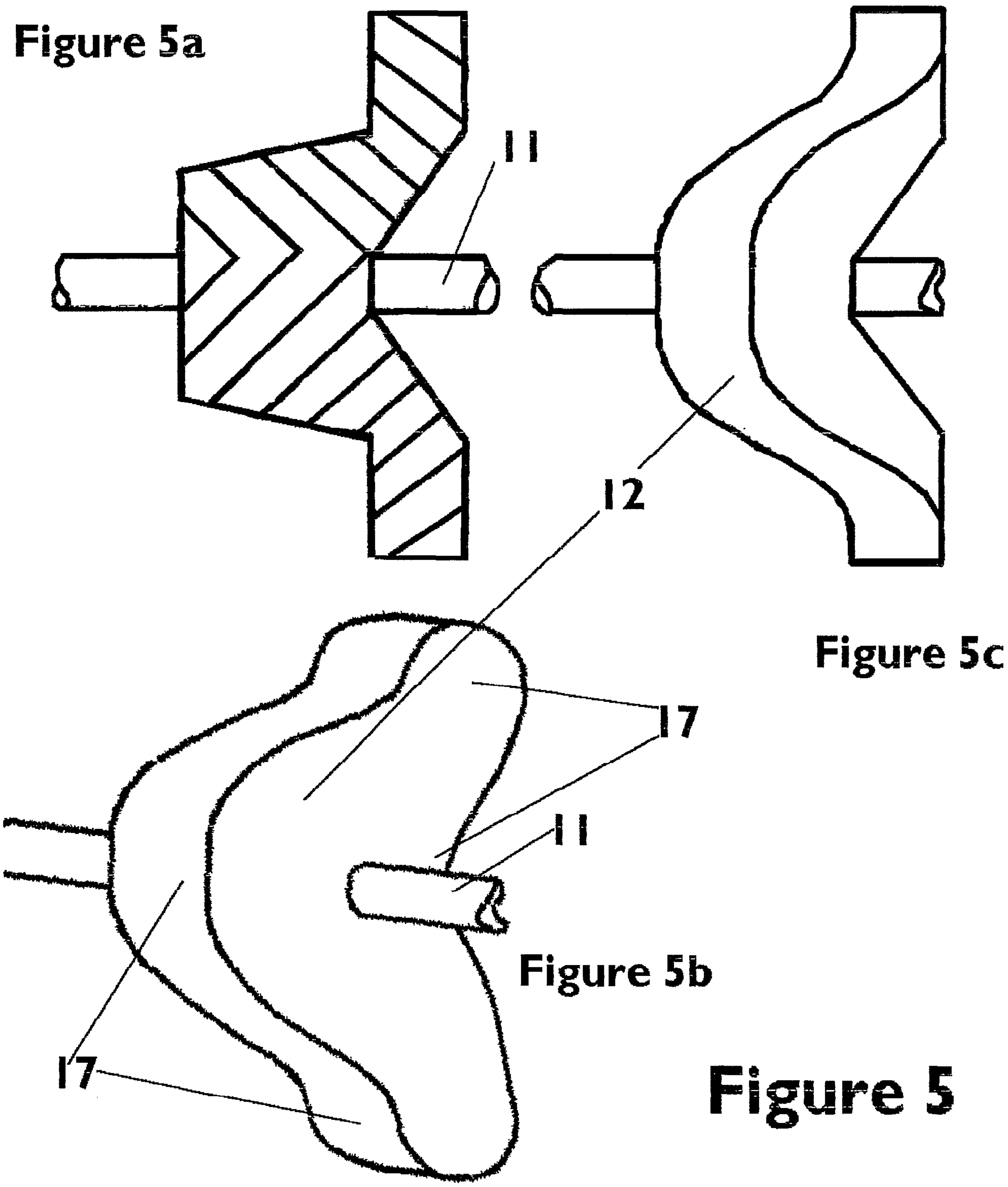


Figure 5

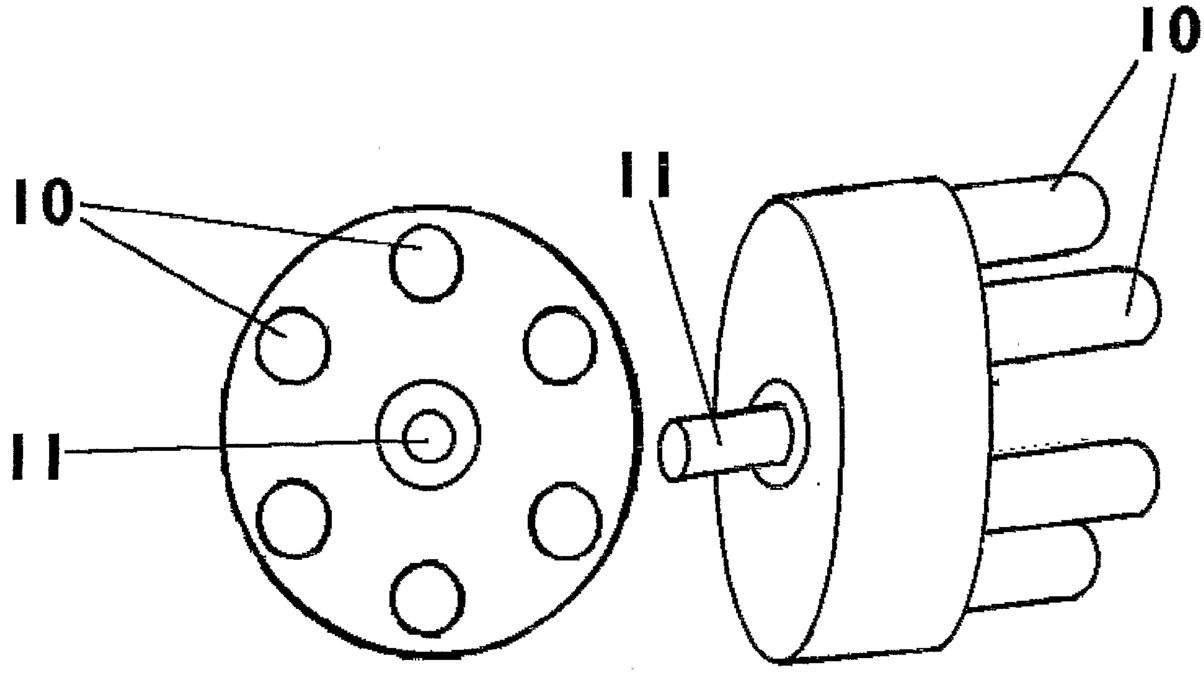
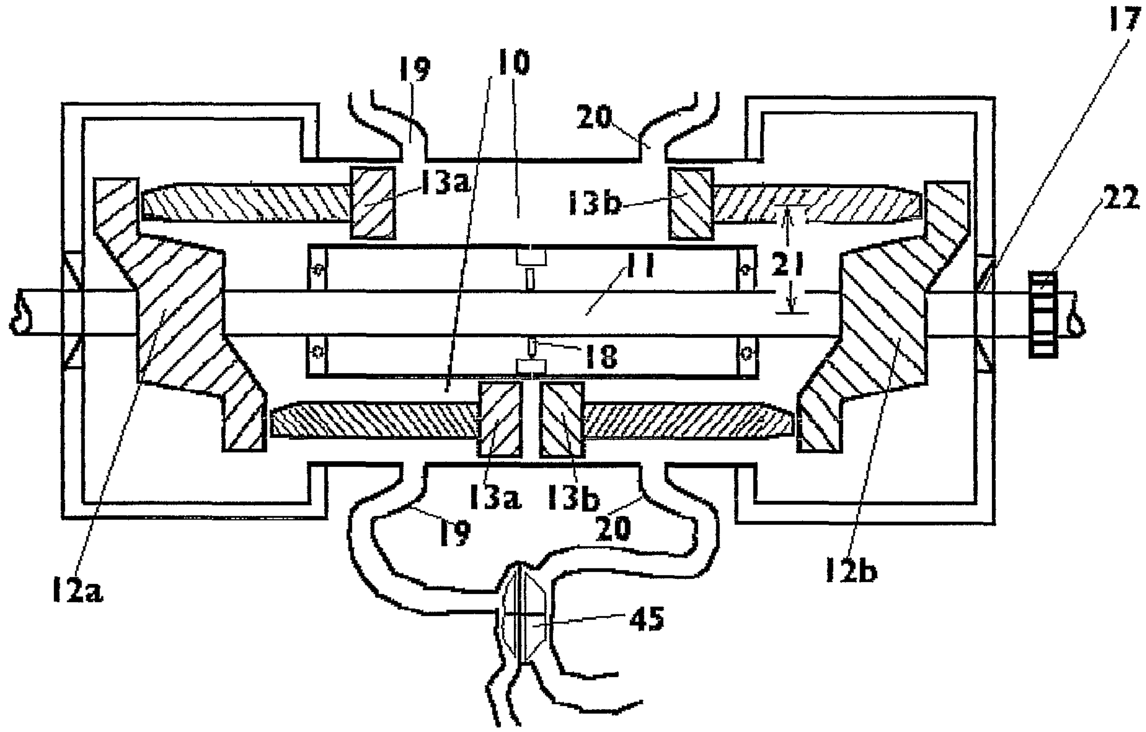


Figure 6a Figure 6b

Figure 6



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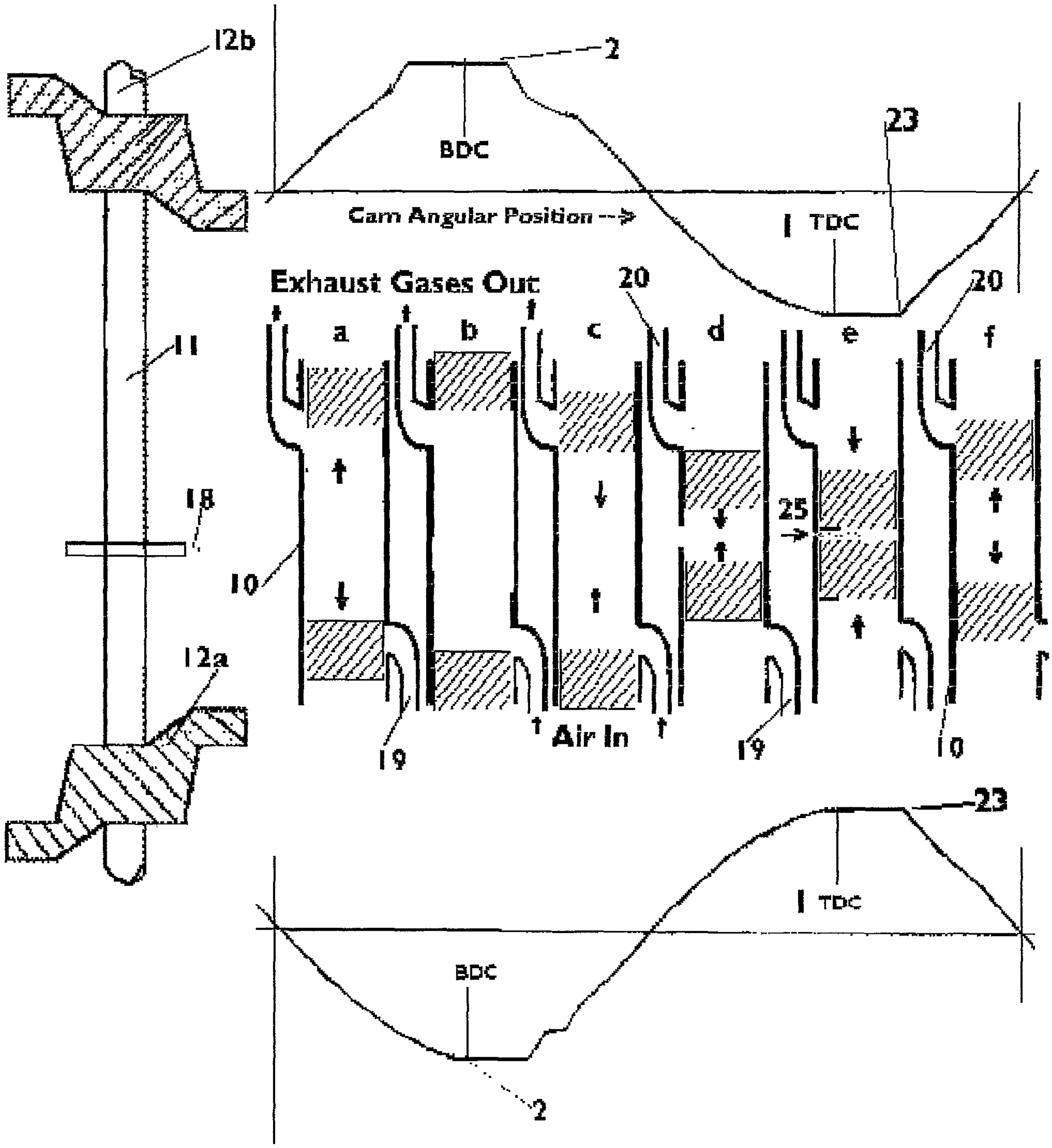

12b
2
BDC
23
Cam Angular Position -->
TDC
Exhaust Gases Out
20
a
b
c
d
e
f
11
18
25
10
12a
19
Air In
23
TDC
BDC
2

Figure 8

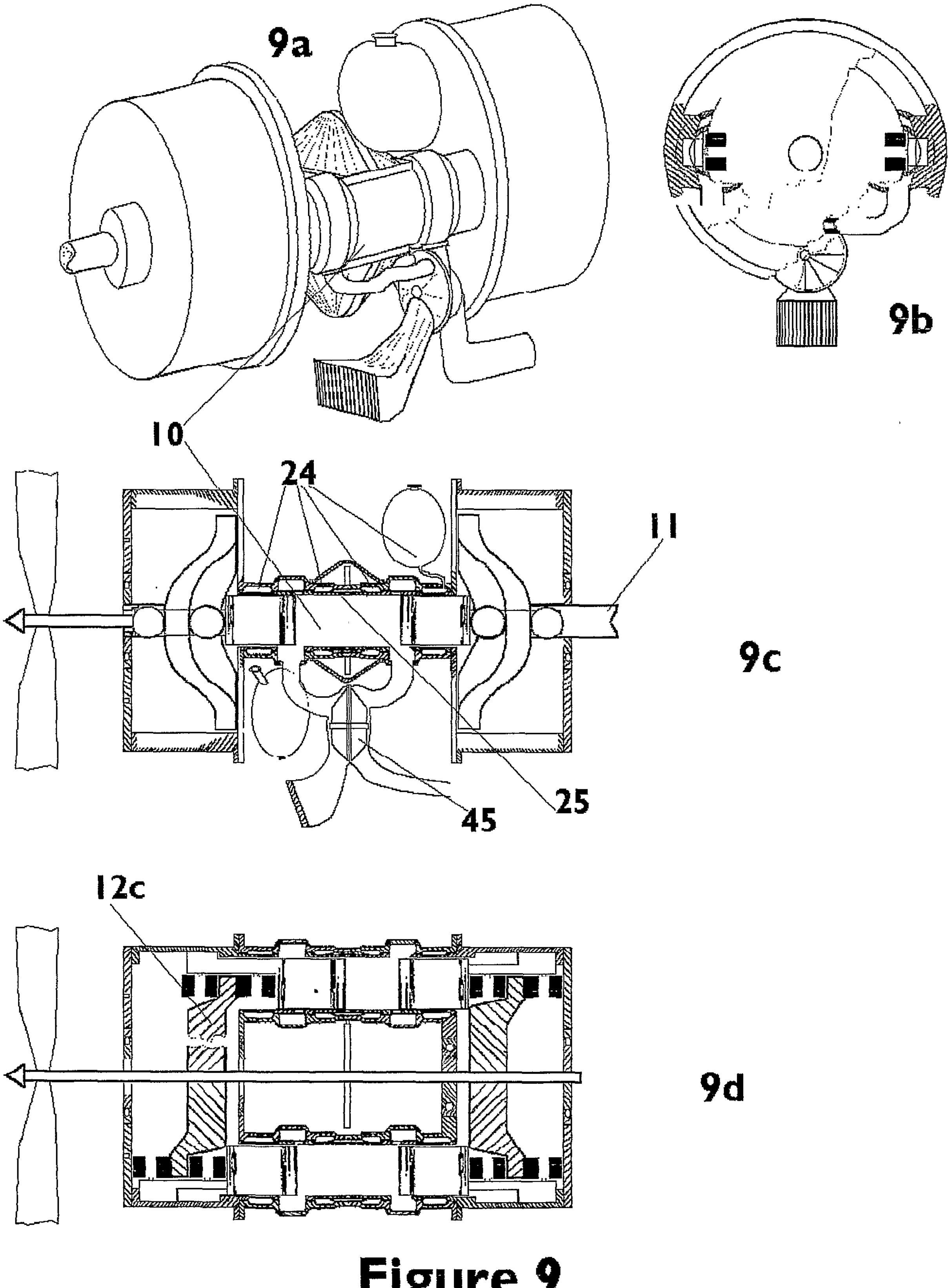
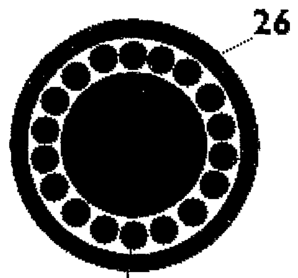


Figure 9

Figure 10b



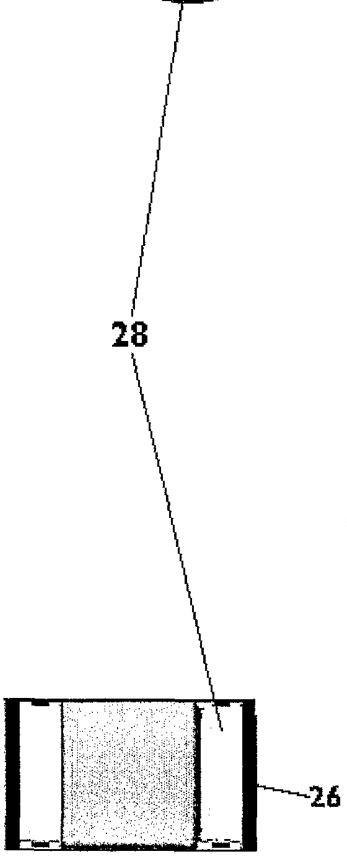


Figure 10c

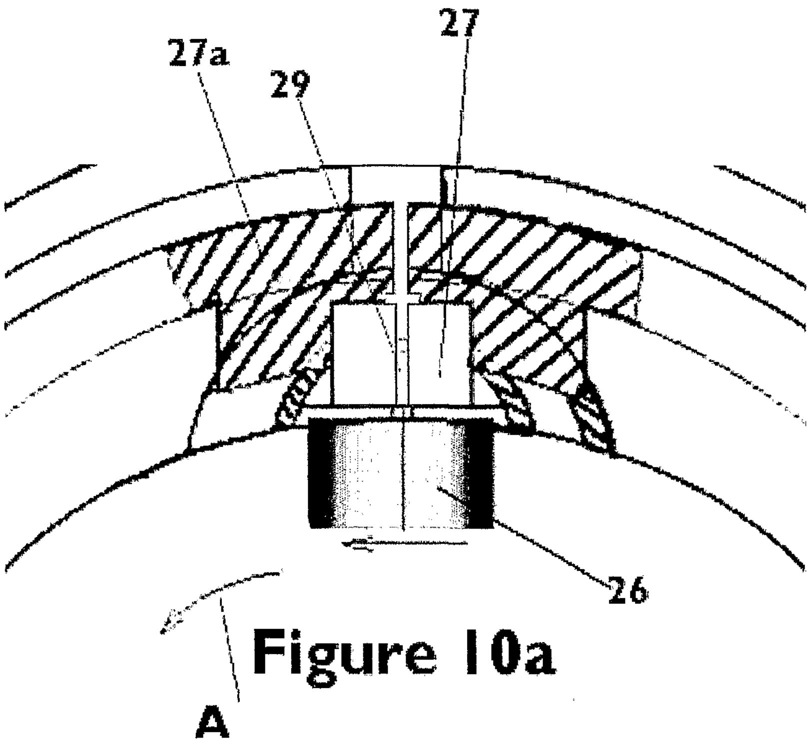


Figure 10a

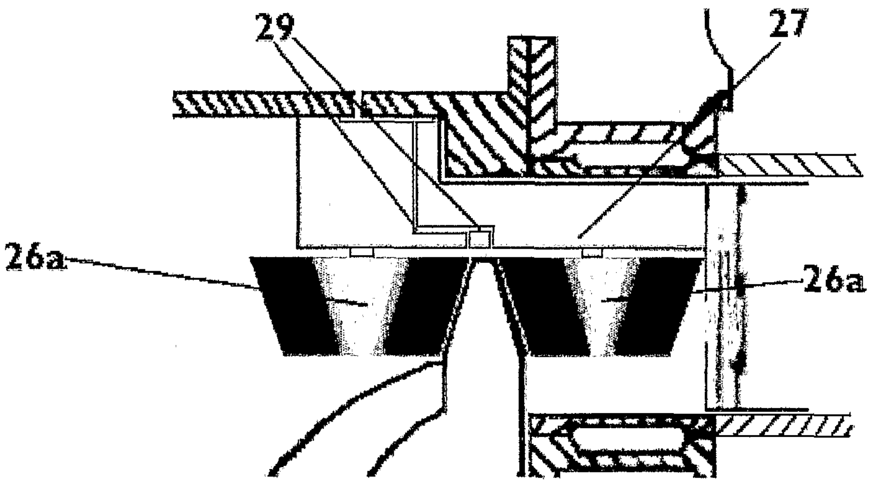
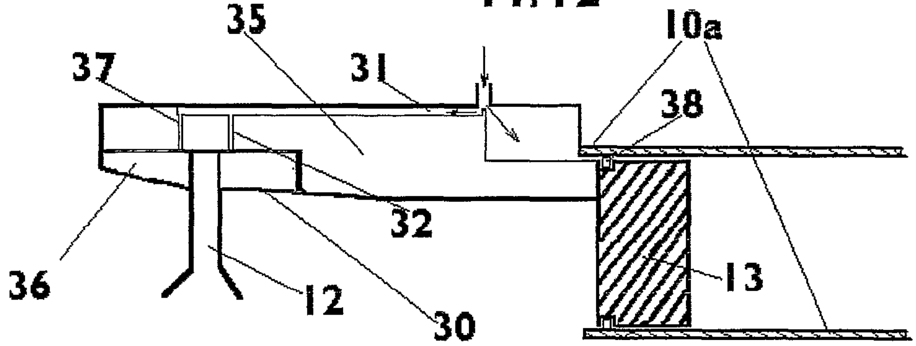
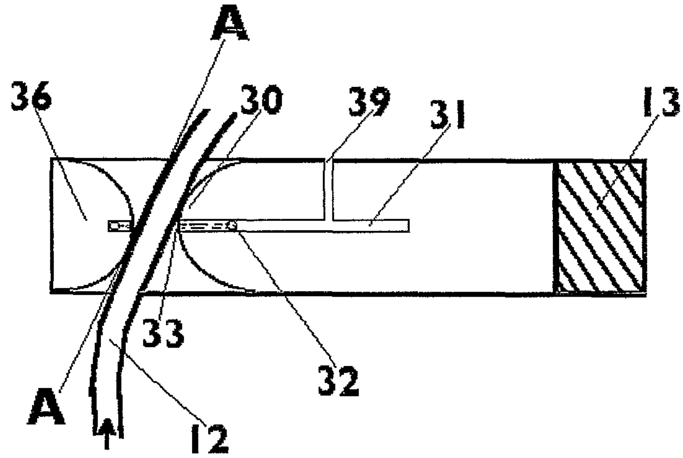


Figure 10d

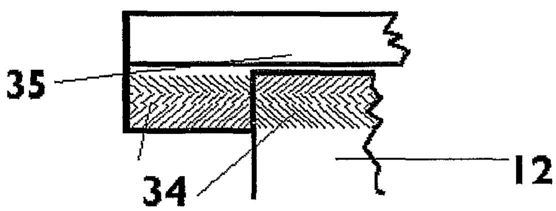
Figure10



11a

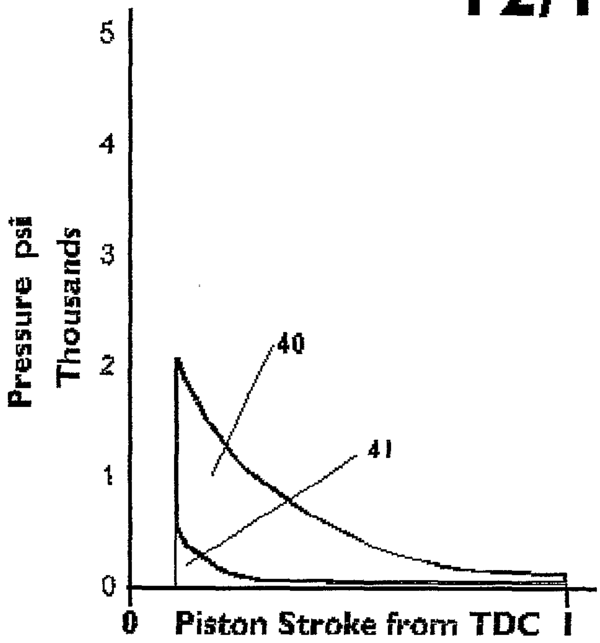


11b

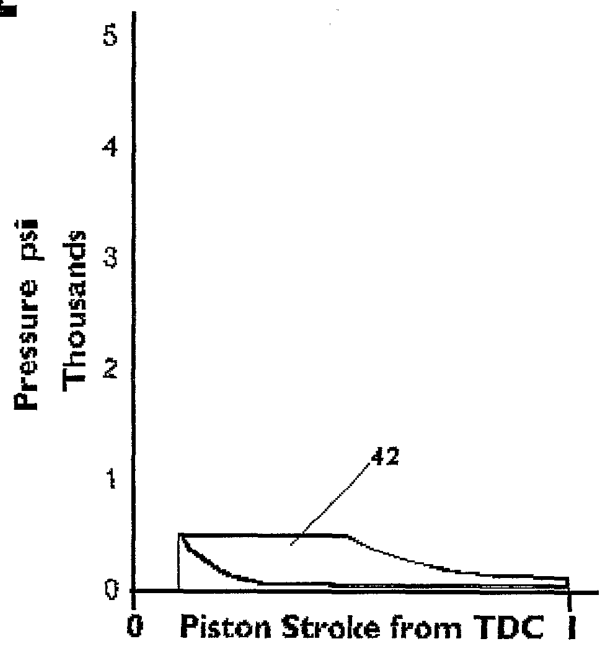


11c

Figure 11



12a



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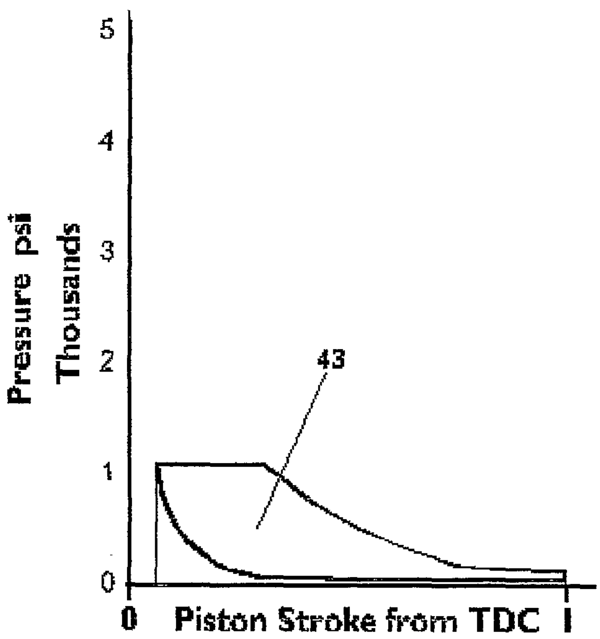

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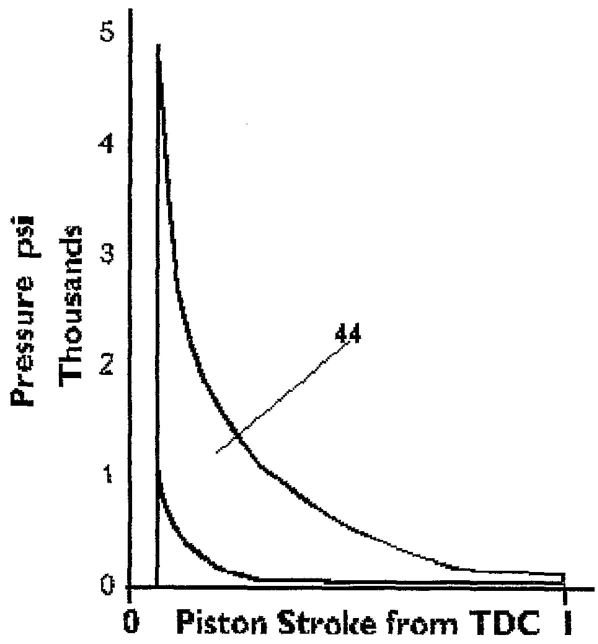

12d

Figure 12

INTERNATIONAL SEARCH REPORT

Inter | Application No
PCT/GB 01/04077

A. CLASSIFICATION OF SUBJECT MATTER
IPC 7 F02B75/26 F01B3/04

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 7 F02B F01B

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)

EPO-Internal

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category °	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	US 5 323 738 A (MORSE JONATHAN E) 28 June 1994 (1994-06-28) abstract figures 1-4 column 3, line 38 -column 5, line 50 ---	1-9
X	US 4 565 165 A (PAPANICOLAOU JOHN P S) 21 January 1986 (1986-01-21) abstract figures 1-5 column 3, line 39 -column 4, line 65 ---	1-9
X	US 4 996 953 A (BUCK ERIK S) 5 March 1991 (1991-03-05) abstract figures 1,5,7 column 9, line 67 -column 10, line 68 ---	1-9
	-/--	

[X] Further documents are listed in the continuation of box C.

[X] Patent family members are listed in annex.

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"P" document published prior to the international filing date but later than the priority date claimed

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"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

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"&" *document member of the same patent family*

Date of the actual completion of the international search	Date of mailing of the international search report
24 October 2001	31/10/2001

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

Intern Application No
PCT/GB 01/04077

C.(Continuation) DOCUMENTS CONSIDERED TO BE RELEVANT

Category °	Citation of document, with indication,where appropriate, of the relevant passages	Relevant to claim No.
X	US 5 799 629 A (LOWI JR ALVIN) 1 September 1998 (1998-09-01) abstract figures 1,2,5 claims 1-9 column 17, line 6 -column 18, line 9 -----	1,2,4-9

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Inter l Application No
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