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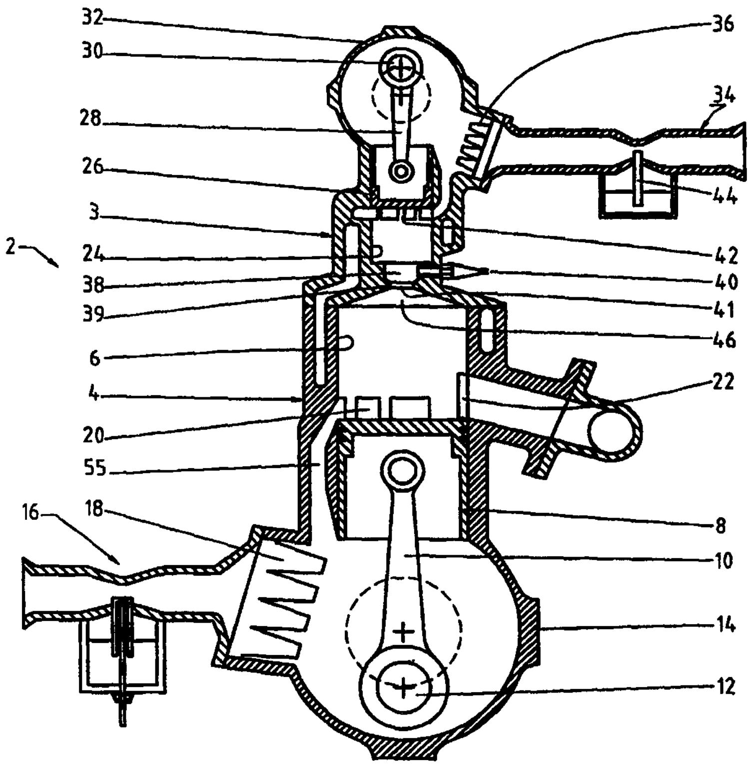
(57) Abstract

A method of operating an internal combustion engine (2) including the steps of: introducing a first charge of air or air fuel mixture into a main cylinder (6) in which a main piston (8) is mounted for reciprocating movement; introducing a second charge of air fuel mixture into an auxiliary cylinder (24) in which an auxiliary piston (26) is mounted for reciprocating movement; causing the auxiliary piston to compress the second charge; igniting the compressed second charge; and transferring burning fuel and/or combustion products produced from the second charge to the main cylinder whereby any fuel in the first charge is burnt.



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INTERNAL COMBUSTION ENGINES

Background of the Invention

This invention relates to internal combustion engines.

More particularly, this invention is concerned with the combustion efficiency of internal combustion engines, particularly two and four-stroke combustion engines.

It is well known that two and four-stroke internal combustion engines do not operate very efficiently at low loads. This is because it is usually necessary to supply an overly rich air fuel mixture to the combustion chamber in order for reliable combustion to take place. The need to supply an overly rich air fuel mixture is necessary because the compression is lower at lower loads. This results in lower combustion efficiency because some of the fuel is not properly burnt and accordingly the efficiency of the engine is much lower than that which should be theoretically obtainable. Unburnt or partially unburnt fuel also causes problems with pollutants in exhaust emissions from the engine.

Some attempts have been made to address this problem. In particular, various proposals have been made to stratify or concentrate an incoming air fuel mix so as to have a comparatively small zone where there is a correct mixture of air fuel or a rich mixture of air fuel and to cause ignition in that zone. The main charge of air fuel is ignited by flame front from the burning fuel in the zone even though the main air fuel mix in the main charge is lean. For various reasons, however, none of these proposals has been very successful.

Summary of the Invention

An object of the present invention is to provide a novel internal combustion engine configuration in which an auxiliary cylinder and piston are provided into which a correct (or substantially correct) air fuel mixture can be supplied or an over rich mixture can be supplied.

Ignition takes place in the auxiliary cylinder and the combustion products are arranged to be transferred to the main cylinder and piston. Because hot combustion products are transferred into the main piston and cylinder, reliable ignition can take place in the main cylinder even though the air fuel mixture therein is lean or substantially leaner than that which is normally required for reliable combustion.

According to the present invention there is provided a method of operating an internal combustion engine including the steps of:

introducing a first charge of air or air fuel mixture into a main cylinder in which a main piston is mounted for reciprocating movement;

introducing a second charge of air fuel mixture into an auxiliary cylinder in which an auxiliary piston is mounted for reciprocating movement;

causing the auxiliary piston to compress the second charge;

igniting the compressed second charge; and

transferring burning fuel and/or combustion products produced from the second charge to the main cylinder whereby any fuel in the first charge is burnt.

Preferably, the method includes the step of supplying a generally constant air fuel mixture to the auxiliary cylinder and the method includes the step of varying the amount of fuel in the charge supply to the main cylinder in order to vary the power output from the internal combustion engine.

The invention also provides an auxiliary cylinder and piston device which can be retrofitted to an internal combustion engine.

A method of controlling the operation of an internal combustion engine comprising providing a main and auxiliary piston which are arranged for carrying out operating cycles in auxiliary and main cylinders respectively, admitting a combustible mixture into the auxiliary cylinder and controlling the operating strokes of the auxiliary and main pistons so that said combustible mixture is constrained to move into a combustion chamber located between said cylinders and igniting the combustible mixture in the combustion chamber.

Preferably the method includes the further step of admitting a full charge of air to the main cylinder during each induction stroke of the main piston.

Preferably further, the method includes the step of admitting a substantially fixed quantity of fuel into the auxiliary cylinder during each operating stroke thereof but a variable quantity of fuel is admitted to the main cylinder. During idling, zero fuel may be admitted to the main cylinder.

On starting the engine, because there is full compression it is unlikely that a choke will be required.

The invention also provides an internal combustion engine including:

at least one main cylinder with a main piston mounted for reciprocating movement therein;

at least one auxiliary cylinder with an auxiliary piston mounted for reciprocating movement therein;

fuel supply means for supplying first and second charges to said main and auxiliary cylinders respectively;

ignition means for igniting fuel in the second charge; and

transfer means for transferring burning fuel and/or combustion products from combustion of the second charge to said main cylinder to thereby cause burning of any fuel in said first charge.

Preferably, the combustion engine includes timing means which is operable to control the timing of the auxiliary piston relative to the main piston so that the combustion products from the ignition of the fuel in the auxiliary cylinder are transferred to the main cylinder at or near top dead centre thereof.

Preferably the auxiliary cylinder has a substantially lower swept volume than the swept volume of the main cylinder. Preferably further, the ratio of swept volumes is in the range 0.05 to 0.5:1. Preferably the swept volume of the auxiliary cylinder is about 1/9 or 1/10 the swept volume of the main cylinder.

More particularly, the ratio of swept volumes does depend on the types of combustion cycles which the auxiliary and main cylinders operate on. For instance, where the auxiliary and main pistons operate both on two-stroke or both on four-stroke cycles, the ratio of the swept volumes is about 1/9 or 1/10, as indicated above. The same ratio applies in the case where the auxiliary piston operates under a two-stroke cycle with crankcase compression and the main piston operates under a four-stroke cycle. In the case, however, where the auxiliary piston operates under a two-stroke cycle without crankcase compression and the main piston operates under a four-stroke cycle, the ratio of the swept volumes is about 1/20 because in this case the relative timing of the pistons would be such that some of the charge admitted to the main cylinder would be compressed into the auxiliary cylinder prior to ignition.

For an engine which mostly runs at fixed loads such as an aircraft engine, the ratio might be as high as 1:1.

Preferably further, the second fuel supply means is arranged to supply an air fuel mixture to the auxiliary cylinder at a substantially constant charge per cycle, that is to say a rate which is not dependent on the required power output of the internal combustion engine. The first fuel supply means is varied for varying the amount of fuel in order to control the power output of the internal combustion engine but the volume per charge of air supplied to the main cylinder remains unchanged at an optimum amount for full compression. In this arrangement, the auxiliary cylinder always operates with a correct air fuel mixture supplied thereto or with a slightly rich air fuel mixture so that reliable combustion always takes place in the auxiliary cylinder. The main cylinder, however, does not need to have a correct or rich air fuel mixture in order for combustion to take place. Accordingly lean mixtures can be supplied to the main cylinder and because ignition is dependent upon the hot combustion products from the auxiliary cylinder, it is not necessary that a correct mixture be present in the main cylinder. Accordingly, at low loads very lean mixtures can be admitted to the main cylinder and reliable combustion can take place. At idle, it is possible that no fuel is admitted to the main cylinder.

The main cylinder may operate on a two-stroke or four-stroke cycle. With suitable timing arrangements, the auxiliary cylinder can operate on two-stroke and four-stroke cycles as well.

In one preferred arrangement, the main cylinder operates on a four-stroke cycle and the auxiliary cylinder operates on a two-stroke cycle. The timing is such that the main engine operates at twice the rpm of the auxiliary engine. It is particularly preferred in this arrangement to arrange that the timing of the main and auxiliary pistons is selected so that crankcase compression can be avoided in the auxiliary engine. More particularly, it is preferred that the auxiliary piston is retarded by an angle of say 80° to 100° and preferably 90° relative to the timing of the main piston. In this way the inlet ports to the main cylinder can open substantially simultaneously with the inlet port for the auxiliary cylinder. During the induction stroke of the main piston, air fuel is drawn into both cylinders.

Preferably further, the first fuel supply means does not include a throttle valve. Preferably further, the second fuel supply means does not include a throttle valve. Accordingly, a substantially full charge of air is admitted to the respective cylinders during each operating cycle. Thus high compression is maintained throughout the operating cycles leading to increased efficiency.

Preferably, the auxiliary cylinder operates on a two-stroke cycle and the main cylinder also operates on a two-stroke cycle.

In some arrangements, the auxiliary cylinder has its own crankshaft which is timed relative to the crankshaft of the main piston. In alternative arrangements, the auxiliary piston may operate as a slave piston which is movable in accordance with pressures developed in the main cylinder.

In other embodiments, it is possible for the auxiliary piston to be moved by means of bearings or cam surfaces provided on the crankshaft of the main piston.

Preferably, the engine of the invention utilises fuel injectors because it is much easier to control the quantity of fuel and timing of admission of fuel to the various cylinders compared to carburettors.

Preferably further, intake air for the main cylinder and the auxiliary cylinder are derived from the same source, that is to say from the same inlet manifold, air cleaner or the like. In this way if there is a variation in inlet air pressure or volume it will have the same proportional effect on both the main and auxiliary cylinders. The inlet manifold may include a valve or other arrangement to control the relative volumes of air delivered to the main and auxiliary cylinders in each operating stroke. The valve can be fixed when the engine is tuned or adjustable according to operating conditions.

The invention also provides a novel head which can be fitted to the block of a normal four-stroke engine of conventional design. The head includes the auxiliary cylinder and piston and enables the modified engine to operate in the same way as described above.

The invention also provides a pilot engine device which can be fitted in the threaded spark plug hole of an internal combustion engine. The pilot engine device includes the auxiliary cylinder and piston and enables the composite engine to function analogously to the engines described above.

The invention will now be further described with reference to the accompanying drawings.

Brief Description of the Drawings

FIGURE 1 is a schematic sectional view of an internal combustion engine constructed in accordance with the invention, the auxiliary and main cylinders operating on two-stroke cycles;

FIGURE 2 is a schematic diagram showing piston displacements for an engine of the invention which has two-stroke cycles for both the auxiliary and main pistons;

FIGURE 3 shows a modified form of the internal combustion engine shown in Figure 1;

FIGURE 4 shows a further modified form of the internal combustion engine of the invention;

FIGURE 5 shows a further modified form of the internal combustion engine of the invention;

FIGURE 6 shows a modified form of internal combustion engine of the invention;

FIGURE 7 shows a variation of the internal combustion engine shown in Figure 6;

FIGURE 8 shows a modified internal combustion engine which utilises a fuel injector in the auxiliary cylinder;

FIGURES 9 and 10 show a modified form of engine of the invention in which the auxiliary piston does not have its own crankshaft;

FIGURES 11 and 12 show a modified geometry of engine in which the auxiliary and main pistons share a common head;

FIGURE 13 is a schematic axial view through a V-configuration engine;

FIGURE 14 is a schematic plan view showing the location of the cylinders and crankshafts;

FIGURES 15 and 16 show a modified engine configuration in which the auxiliary and main cylinders have a common head and inlet valves are provided for the main inlet ports;

FIGURES 17, 18 and 19 show a further modified engine of the invention;

FIGURES 20, 21 and 22 show a modification of the invention in which the auxiliary piston operates on a two-stroke cycle and the main piston operates on a four-stroke cycle;

FIGURE 23 is a schematic diagram showing piston displacements where the auxiliary piston has a two-stroke cycle and the main piston has a four-stroke cycle;

FIGURES 24A to 24P are diagrams illustrating the timing for engines of the invention which require crankcase compression for the auxiliary piston and cylinder;

FIGURE 25 is a schematic diagram useful in determining valve timing and ignition timing where the auxiliary piston operates on a two-stroke cycle and the main piston operates on a four-stroke cycle;

FIGURE 26 is a schematic axial view of a V-configuration engine;

FIGURE 27 is a schematic plan view showing the location of the cylinders and crankshafts;

FIGURES 28, 29 and 30 illustrate a modification of the engine shown in Figure 1 but including cams for operating the valves on the main cylinder;

FIGURE 31 shows a variation of the engine shown in Figures 28, 29 and 30;

FIGURE 32 is a schematic diagram showing piston displacements where the auxiliary piston has a two-stroke cycle without crankcase compression and the main piston has a four-stroke cycle;

FIGURES 33A to 33P are diagrams illustrating the timing of the engines of the invention in cases where no crankcase compression is required for the auxiliary cylinder;

FIGURE 34 shows a schematic side view of an engine in which the auxiliary pistons are operated from the same crankshaft as the main pistons;

FIGURE 35 is a fragmentary view showing the relative locations of some of the cylinders of the engine shown in Figure 34;

FIGURE 36 is a schematic diagram showing piston displacements where both the auxiliary and main pistons have four-stroke cycles; and

FIGURE 37 is a schematic cross-section through a modified form of the invention.

Description of the Preferred Embodiments

The principles of the invention are applicable to many different internal combustion configurations. The principles of the invention will be described with reference to various embodiments and the same reference numerals will be used to denote the same or corresponding parts throughout the various embodiments.

Figure 1 diagrammatically illustrates an internal combustion engine 2 constructed in accordance with the invention. The engine comprises a block 4 having a main cylinder 6 in which a main piston 8 is mounted for reciprocating movement. A head 3 is connected to the block 4 in the usual way. The main piston 8 is connected by means of a conrod 10 to a main crankshaft 12. The main piston and cylinder operate on a two-stroke cycle with air fuel being admitted into a crankcase 14 from a "carburettor" 16. The carburettor 16 usually would not require a throttle valve because a full charge of air is always admitted to the crankcase 14. Downward movement of the piston 8 is arranged to produce crankcase compression in the usual way for the air or air fuel mixture admitted to the crankcase 14 from the carburettor 16. Air fuel is admitted into the crankcase via one way reed valves 18. During the compression stroke, compressed air fuel mixture is admitted into the cylinder 6 through inlet ports 20. The engine 2 includes an exhaust port 22 in the main cylinder 6. It will also be observed that an inlet passage 55 from the crankcase 14 to the inlet ports 20 is inclined upwardly so as to direct admitted charges of air fuel away from the exhaust port 22. This will tend to prevent fuel being admitted to the main cylinder 6 being lost through the exhaust port 22.

The head 3 includes an auxiliary cylinder 24 in which an auxiliary piston 26 is mounted for reciprocating movement. The auxiliary piston 26 is connected to a conrod 28 coupled to an auxiliary crankshaft 30 which rotates in an auxiliary crankcase 32. The auxiliary piston 26 is arranged to operate on a two-stroke cycle and air fuel is admitted to the crankcase from an auxiliary carburettor 34 via reed valves 36. The auxiliary cylinder 24 opens to a combustion chamber 38. A spark plug 40 is arranged to have its electrodes in the combustion chamber 38.

Figure 1 shows a single main piston 8 for clarity of illustration but the invention is applicable to multi-cylinder engines and each main cylinder would have associated therewith an auxiliary cylinder.

The arrangement is such that when the auxiliary piston 26 executes its crankcase compression stroke, air fuel from the crankcase 32 is admitted through inlet ports 42 into the auxiliary cylinder 24.

The main crankshaft 12 is coupled to the auxiliary crankshaft 30 such that they rotate in unison. This can be achieved by means of a timing chain (not shown in Figure 1) or the like. The movement of the piston 26 is opposed to that of the main piston 8, i.e. the piston 26 moves towards and away from the piston 8 at their top dead centres and bottom dead centres respectively. The configuration is such that the auxiliary piston 26 and the main piston 8 are generally simultaneously at top dead centres and bottom dead centres.

The auxiliary carburettor 34 is arranged to supply a nearly correct mixture or slightly rich mixture to the auxiliary crankcase 32. Accordingly, the fuel delivery tube 44 in the carburettor can continuously open, that is to say it is not controlled by means of a needle valve during normal operation of the engine 2, even though the power output from the engine is altered.

During a compression stroke of the piston 26 (and also of the piston 8), the fixed air fuel charge in the auxiliary cylinder 24 will be compressed and transferred by the piston 26 into the combustion chamber 38. The chamber 38 may be shaped for optimum performance. It may include a constricting shoulder 39 to help retain the rich mixture therein. In addition, the mixture admitted to the chamber 38 may be swirling so as to help ignition and burning after ignition. At or near peak compression, the spark plug 40 is operated so as to ignite the charge in the chamber 38. Combustion products expand and are then discharged through a transfer duct 41 into a combustion chamber 46 of the main cylinder 6 so as to cause reliable combustion of the air fuel charge which has been compressed therein by the main piston 8. As shown in Figure 1, the combustion chamber 46 may be located in the head 3. Because the combustion in the combustion chamber 46 is initiated by means of the combustion products from the combustion chamber 38, the air fuel charge in the combustion chamber 46 will reliably ignite substantially independently of the leanness of the charge. Accordingly the carburettor 16 can be arranged to admit very lean mixtures or air only into the crankcase 14 and yet reliable combustion can take place.

During the expansion stroke, the main piston 8 will move downwardly so as to produce useful torque on the main crankshaft 12 until the exhaust port 22 is opened. Both cylinders 6 and 24 are exhausted through the exhaust port 22 which has the significant advantage that it avoids the need for separate exhaust for the auxiliary cylinder.

Figure 2 diagrammatically illustrates displacements of the auxiliary and main pistons during the working cycle. The upper line 54 indicates displacements of the auxiliary piston 26 and extends from top dead centre TDC indicated at point 50 to bottom dead centre BDC indicated at point 52. The lower line 56 indicates displacements of the main piston 8 and extends from top dead centre TDC indicated at point 51 to bottom dead centre BDC indicated at point 53. The first line segment 56a indicates the part of displacement of the main piston in which the exhaust ports 22 are closed. Once the line 57 is reached the inlet ports 20 are open, as indicated by line segment 56c. The inlet ports 20 will remain open until the line 59 is reached and thereafter the exhaust ports 22 will remain closed as indicated by line segment 56d. In the line segment 56a, the main piston 8 will be undergoing an expansion stroke and in the line segment 56d the main piston 8 will be undergoing a compression stroke. As is apparent from Figure 1, the exhaust port 22 will open before the inlet ports 20 open, as indicated by the line segment 56b. The exhaust port 22 will remain open during the compression stroke for longer than the inlet ports 20 remain open. In Figure 1 which is a two-stroke/two-stroke engine, the TDC points 50 and 51 are in phase as are the BDC points 52 and 53. Embodiments are discussed below in which the main piston 8 executes a two-stroke combustion cycle including valves on the inlet ports and/or the outlet ports. In these arrangements the timing of opening of the inlet and outlet ports are similar as those illustrated in Figure 2, but some cases there may be retard or advance in the relative timing.

It will be appreciated by those skilled in the art that the internal combustion engine described above has the substantial advantage that reliable combustion of the main air fuel charge admitted to the main cylinder 6 can be efficiently burned more or less independently of the leanness of the mixture. Accordingly, it is possible to control the required output power by controlling the amount of fuel supplied by the main carburettor 16, the charge of air per cycle remaining at the full level throughout.

Under idle conditions, it would be advantageous to arrange for the auxiliary carburettor 34 to supply the only fuel bearing charge to the engine, the main carburettor 16 being arranged not to supply any fuel during that time, although air is advantageously admitted to the crankcase 14. The efficiency of the engine is enhanced because a full charge of air can be admitted to the main cylinder 6 even under idle conditions. This enables the compression ratio to remain at a relatively high level.

Although it is not shown in Figure 1, it is preferred that air to the carburettors 16 and 34 would be coupled to the same inlet manifold, air filter and the like (not shown in Figure 1). In this way any changes in the air flow rates would affect both of the carburettors 16 and 34 in a proportionate way and therefore the volumetric ratio of air charges available to the auxiliary and main cylinders would remain substantially the same.

In a preferred embodiment of the invention, the swept volume of the auxiliary cylinder to the swept volume of the main cylinder is in the range from 0.05 to 0.5. Normally the swept volume of the auxiliary cylinder will be about 0.1 of the swept volume of the main cylinder. It will also be appreciated that the arrangement shown in Figure 1 can be applied to multi-cylinder engines where a number of main pistons would be coupled to the main crankshaft 12 and a number of auxiliary pistons would be coupled to the auxiliary crankshaft 30.

It will also be appreciated that the carburettors 16 and 34 can be replaced by fuel injectors. Fuel injectors have the advantage that with suitable electronic controllers which are well known in the art, the quantity of fuel and timing of the admission of the fuel can be varied more easily so as to tune the engine for optimum performance.

Many modifications of the engine shown in Figure 1 are possible and some are discussed below in relation to Figures 3 to 37. These engines operate in a generally similar manner to one another, at least insofar as fuel combustion is concerned. All of these arrangements include a main cylinder 6 and piston 8 and an auxiliary cylinder 24 and piston 26, combustion being initiated in the combustion chamber 38 of the auxiliary cylinder and the combustion products transferred to the main combustion chamber 46. If a fuel bearing charge has been admitted to the main cylinder 6, reliable combustion will take place because of the transfer of hot or burning combustion products. Also, the combustion fuel in the chamber 38 will cause a significant increase in pressure in the chambers 38 and 46 which also will facilitate burning in the main combustion chamber 46. In this way the main cylinder is able to operate at effectively very lean air fuel ratios.

Figure 3 diagrammatically illustrates an arrangement which is similar to Figure 1 except that the location of the main carburettor 16 has been changed. In this arrangement, during the induction stroke of the main piston 8, a charge of air is introduced into the crankcase 14 through the reed valves 18. Air is compressed by crankcase compression in the crankcase 14 and is transferred through a transfer duct 60 to the inlet ports 20. The main carburettor 16 is located in the transfer duct 60 so as to admit the main charge of fuel into the compressed air mixture prior to admission into the main cylinder 6. The inlet ports 20 can be shaped so as to direct the air fuel charge admitted to the main cylinder 6 towards the combustion chamber 46. In the illustrated arrangement, a small proportion of the air compressed in the crankcase 14, say 20%, is passed into the passage 55 which opens into the inlet ports 20. This provides a layer or stratum of air adjacent to the crown of the main piston 8 whereas the fuel rich air admitted from the transfer duct 60 is directed upwardly towards the main combustion chamber 46. This arrangement helps to avoid fuel loss through the exhaust port 22.

Figure 4 diagrammatically illustrates a further modification of the invention. In this arrangement, the main and auxiliary carburettors 16 and 34 are omitted and a fuel injector 62 is provided so as to inject a charge of fuel into an inlet duct 64 which communicates with the inlet ports 20. As in the arrangement of Figure 3, part of the air compressed in the crankcase 14 passes from the inlet duct 64 through the inlet ports 20 so as to form a layer above the crown of the piston 8. This again prevents fuel loss through the exhaust port 22. The injector is arranged to direct the main charge of fuel towards the inlet ports 20 and a small proportion of the fuel through a transverse duct 66 for admission to the auxiliary cylinder 24 through the auxiliary inlet ports 42. It will be appreciated that crankcase compression in the crankcase 32 is not required. Crankcase compression of air occurs in the crankcase 14 for both cylinders 6 and 24. The engine otherwise functions analogously to that shown in Figure 1. Multi-jet injectors could be used and arranged to give a rich mixture for the ports 42.

Figure 5 diagrammatically illustrates a further modification of the invention. This arrangement is somewhat similar to that shown in Figure 4 except that the inlet ports 20 to the main cylinder 6 are eliminated. In this arrangement all of the compressed air from the crankcase 14 passes through the duct 64 into the inlet ports 42 of the small cylinder 24. This occurs when the small piston 26 is approaching bottom dead centre and the inlet ports 42 are opened. Air is then free to pass through the transfer duct 41 into the main cylinder 6. The fuel injector 62 is arranged to inject the appropriate amount of fuel for the main cylinder 6 at this stage. The timing of the injector 62 is preferably such that initially air only is admitted to the cylinder so that it will purge exhaust gases from the previous stroke out of the exhaust port 22. This avoids fuel being lost through the exhaust port 22. The injector 62 will then deliver the appropriate amount of fuel to the air so that the fuel rich mixture will be located in the upper part of the cylinder 6 or in the combustion chamber 46. Near the end of the induction stroke, the injector 62 may be again operated so as to deliver a second charge of fuel which is destined to reside in the auxiliary cylinder 24 for compression by the auxiliary piston 26 during its compression stroke. Alternatively, a separate injector 63 shown in broken lines may be provided for supplying the fuel for the auxiliary cylinder 24 near the end of the induction stroke.

It will be appreciated that, just prior to compression strokes of the pistons, there normally will be three distinct zones in the cylinders. The first zone is essentially air only adjacent to the main piston 8. The second zone is an air fuel mix in the main combustion chamber 46. This mix can be relatively lean. The third zone is a correct air fuel mixture in the auxiliary cylinder 24.

The arrangement of Figure 5 has a further advantage in that in the induction and exhaust parts of the combustion cycle air, air fuel and exhaust gases flow in the same direction through the cylinders. This generally improves flow of gases through the engine.

Figure 6 shows a further modified engine of the invention. This configuration is somewhat similar to that shown in Figure 3 except that an auxiliary transfer duct 68 is provided from the main crankcase 14 in order to supply compressed air to the auxiliary cylinder 24. The auxiliary carburettor 34 is located in the transfer duct 68 so as to pass a substantially fixed air fuel mixture through the auxiliary inlet ports 42 into the auxiliary cylinder 24. In this arrangement, crankcase compression need not take place in the auxiliary crankcase 32. The operation of the engine is generally similar to that described in relation to Figure 3. Even though the auxiliary piston 26 undergoes a two-stroke combustion cycle, in the arrangement of Figure 6 it does not require crankcase compression. This enables the auxiliary crankshaft 30, auxiliary piston 26 and its rings and so on to be lubricated by conventional techniques which are normally available only in four-stroke engines. In other words, lubricating oil can be located in the crankcase 32 in a manner which is analogous to that utilised in most conventional four-stroke engines.

Figure 7 diagrammatically illustrates a further modified engine of the invention. In this arrangement, air is admitted to the main crankcase 14 through reed valves 18 where it is compressed by crankcase compression. Compressed air from the crankcase 14 passes through reed valves 70 to a compression chamber 72. Compressed air from the chamber 72 is then delivered to the main and auxiliary carburettors 16 and 34. Again in this arrangement, crankcase compression is not required in the auxiliary crankcase 32. The provision of the chamber 72 ensures that any variations in the flow or pressure of air within the chamber 72 affects the carburettors 16 and 34 in the same or a proportionate way. This ensures that the correct ratios of air and air fuel are supplied to the main and auxiliary cylinders. In the arrangement shown in Figure 7, the outlets to the compression chamber 72 include butterfly valves 77 and 79 which can be used for adjustment of the relative flows of air to the inlet ports 20 and 42. Normally the butterfly valves 77 and 79 would be fixed and would only be adjusted on tuning of the engine.

Figure 8 diagrammatically illustrates a further modified engine of the invention. This arrangement is similar to Figure 4 except that the fuel injector 62 is arranged to inject fuel directly into the auxiliary cylinder 24 and directed towards the combustion chamber 38. In this arrangement it is more or less inherent that there will always be a relatively rich mixture in the combustion chamber 38. At high engine loads, part of the charge of fuel will pass through the chamber 38 and transfer duct 41 into the main cylinder 6 but this should not affect the combustion efficiency because, as indicated above, there will always be a relatively rich mixture in the combustion chamber 38. There is crankcase compression in both crankcases 14 and 32. The auxiliary crankcase 32 has an air inlet 74 provided with reed valves 76 for admission of air therethrough. Air is compressed in the auxiliary crankcase 32 and is admitted through the inlet ports 42. The fuel injector 62 may be a multi-jet injector which produces small jets for the cylinder 24 and, at higher power outputs of the engine, large jets are produced which pass through the transfer duct 41 into the main cylinder 6. Separate injectors (not shown in Figure 8) could be provided for supplying fuel to the respective cylinders. It will be observed in the arrangement of Figure 8 that the nozzle of the injector 62 will be protected by the auxiliary piston 26 when ignition takes place. This avoids excess wear on the nozzle.

Figures 9 and 10 illustrate a further modified engine of the invention. In this arrangement, the auxiliary piston 26 is not coupled to a crankshaft 30 but rather is mounted for reciprocating movement in the auxiliary cylinder 24 subject to pressure within the main cylinder 6 and also subject to resilient forces applied thereto by means of a rod 78. The rod 78 is biased downwardly by means of a compression spring 80. The spring 80 is preferably located in a guide tube 81 which is connected to the head 3. In Figure 9 the main piston 8 is at bottom dead centre and the pressure within the main cylinder 6 will be relatively low, the exhaust port 22 also being open. In this condition, the spring 80 forces the auxiliary piston 26 downwardly through the rod 78. This downward movement of the piston 26 enables a fixed charge of air fuel from the carburettor 34 to be drawn into the auxiliary cylinder 24 through inlet ports 42 above the piston 26. As the main piston 8 undergoes its compression stroke, pressure within the main cylinder 6 will increase to the point that the pressure exerted on the auxiliary piston 26 will overcome the resilient force of the spring 80 and the auxiliary piston 26 will move upwardly to its upward position as shown in Figure 10. The air fuel charge which has been admitted into the auxiliary cylinder 24 will then be transferred through transfer passage 82 into the auxiliary cylinder 24 beneath the piston 26 via ports 83. At or near top dead centre of the main piston 8, the spark plug 40 is fired in order to ignite the relatively rich mixture. In the particular configuration shown in Figures 9 and 10, the spark plug 40 is located directly in the transfer passage 82. It will be appreciated, however, that the spark plug 40 could be located in the combustion chamber 38 as before. In this configuration, the auxiliary piston 26 is provided with upper and lower sets of rings 84 and 86 so as to provide seals with the cylinder wall to prevent unwanted escape of air fuel and/or exhaust through the ports 42 or 83. It will be appreciated that in this configuration the compression spring 80 stores energy therein during the compression stroke and this energy will be released during the expansion stroke thereby contributing to useful output torque on the crankshaft 12. In a variation of this configuration, the compression spring 80 could be replaced by means of an hydraulic, air or fluid spring.

In a variation of the arrangement shown in Figures 9 and 10, the auxiliary pistons 26 could be arranged to reciprocate transversely above the main pistons. The rods 78 of adjacent auxiliary pistons could be coupled to one another so that they basically operate in anti-phase. The coupling between the auxiliary pistons for operation in anti-phase relationship to one another could be mechanical, pneumatic or hydraulic.

Figures 11 and 12 illustrate a further modification of the invention, Figure 11 showing a schematic axial view of the engine and Figure 12 schematically illustrating the relative positions of the cylinders in plan view. In this arrangement the auxiliary cylinder 24 and piston 26 are arranged at the side of the main piston and cylinder rather than being opposed as in the previous embodiments. This of course provides for an engine which is of lower height than the embodiments illustrated in Figures 1 to 10. The operation of the engine shown in Figures 11 and 12 is otherwise similar to that shown in Figure 1. However, it will be appreciated that the combustion chamber 38 is located to the side of the combustion chamber 46 of the main cylinder 6, the chambers 38 and 46 being connected by the transfer duct 41 which extends transversely between the chambers. The spark plug 40 projects into the combustion chamber 38 (or duct 41) so that during the compression stroke of the auxiliary piston 26, a correct air fuel mixture or rich air fuel mixture is located in the combustion chamber 38 adjacent to the spark plug 40 thereby ensuring reliable ignition when the spark plug 40 is operated. After ignition takes place, the combustion products rapidly move through the duct 41 into the combustion chamber 46 thereby causing combustion of the main air fuel charge admitted to the main cylinder 6.

Figure 11 shows a (1:1) timing chain 88 interconnecting the crankshafts 12 and 30 so that they rotate at the same rpm. In the arrangement illustrated in Figures 11 and 12, the cylinders 6 and 24 are inclined to one another. It would be appreciated of course that these could be arranged so as to be parallel to one another.

Figures 13 and 14 illustrate a further modification of the invention. In these drawings there is schematically shown in Figure 13 a multi-cylinder engine having a V-configuration, such as a V4, V6 or V8. Figure 14 schematically illustrates in plan view the location of the cylinders as well as the main and auxiliary crankshafts. The engine of Figure 13 can be best regarded as being of similar construction to the engine shown in Figures 11 and 12 except in a multi-cylinder arrangement having a V-configuration. It will be observed that all of the conrods 10 of the main cylinder 6 are coupled to the main crankshaft 12. Similarly, all of the conrods 28 of the auxiliary pistons 26 are coupled to the auxiliary crankshaft 30. In the illustrated arrangement, the auxiliary crankshaft 30 is located above the crankshaft 12. Each of the main cylinders 6 would be coupled to a respective crankcase chamber 85 for crankcase compression of the respective main cylinders. Similarly, each of the auxiliary cylinders 24 would be coupled to a respective crankcase chamber 87 for crankcase compression for the respective auxiliary cylinders. The crankshafts 12 and 30 are coupled together by means of a timing chain (not shown in Figure 13 or 14) for simultaneous rotation at the same rate. The operation of the engine is, nevertheless, analogous to the embodiments described previously. Each main cylinder has associated therewith an auxiliary cylinder for providing the reliable ignition.

The embodiment of Figures 13 and 14 would appear to offer significant advantages over conventional engines in terms of efficiency, power to weight ratio and production costs. It will be appreciated that many of the variations which have been discussed in relation to the previous embodiments can be incorporated into this multi-cylinder configuration.

Figures 15 and 16 illustrate a further modification of the invention. This arrangement is somewhat similar to that illustrated in Figures 11 and 12. The major difference is, however, that one or more inlet valves 90 are provided in the inlet ports 20 for controlling admission of compressed air fuel mixture from the crankcase 14 through a transfer duct 89 which extends from the crankcase 14 to communicate with the inlet port 20 of the main cylinder 6. The valves 90 improve the efficiency of the engine by avoiding fuel loss through the exhaust port 22 at or near bottom dead centre of the two-stroke cycle. In this configuration, it is convenient to operate the valves 90 by means of rocker arms 92 which are controlled by means of push rods 94. The push rods 94 are themselves operated by means of cam elements (not shown in Figures 15 and 16) which, in accordance with this embodiment of the invention, are located on the auxiliary crankshaft 30. The cam elements are arranged to open the inlet valves 90 shortly before bottom dead centre for instance as indicated by the lines 57 and 59 in Figure 2.

Figures 17, 18 and 19 illustrate a further modified form of the invention. This modified engine is similar to that shown in Figures 15 and 16 except that valves are provided for the exhaust ports 22 rather than the inlet ports. In the modified engine shown in these drawings, Figure 17 is a schematic axial view of the engine, Figure 18 is a fragmentary schematic view showing the location of the cylinders and Figure 19 is a fragmentary view showing part of the auxiliary crankshaft 30. In this arrangement, the exhaust ports 22 are provided with exhaust valves 98 which are actuated by means of the rocker arms 93. The auxiliary crankshaft 30 includes cam elements 97 which operate the rocker arms 93 through push rods 95. The cam elements 97 in this arrangement are arranged to open the exhaust valves 98 shortly before and after bottom dead centre as indicated by the line segment 56b in Figure 2.

Figures 20, 21 and 22 illustrate a further modified form of the invention. In this arrangement, the main piston 8 is arranged to operate on a four-stroke cycle whereas the auxiliary piston 26 is arranged to operate on a two-stroke cycle. It is thought that the four-stroke/two-stroke combination would probably be simpler and cheaper to manufacture than the two-stroke/two-stroke arrangements which have been described in the previous embodiments.

In order for this combination to work correctly, it is necessary to operate the auxiliary piston at half the rate of the main piston. This can be accomplished by connecting reduction sprockets 71 and 73 to the crankshafts 12 and 30, the timing chain 88 being coupled between the sprockets as shown. As in the previous embodiments, combustion commences in the auxiliary cylinder 24 and the combustion products are passed through the transfer duct 41 into the combustion chamber 46 of the main cylinder 6. This occurs when the auxiliary piston 26 is at or near top dead centre. The main piston 8 will also be at or near top dead centre but it will of course be possible to adjust the phase between the two cylinders in order to achieve optimum results. Thus combustion products from ignition in the combustion chamber 38 can be caused to flow through the duct 41 into the combustion chamber 46 of the main cylinder 6 at or near top dead centre of its compression stroke so that reliable combustion can take place regardless of the air fuel ratio in the main cylinder. In this configuration, the head 3 of the main cylinder 6 includes an inlet valve 90 and an exhaust valve 98. These are operated by respective rocker arms 92 and 93 and appropriate cam elements 96 and 97 are provided on the auxiliary crankshaft 30 for correct timing of the valves.

In the engine of Figures 20 and 21, the four-stroke part of it can be lubricated in the usual way, i.e. by a sump in the crankcase 14 but it is preferred to arrange to have the two-stroke parts of the engine lubricated by means of oil injection or by using oil in the fuel. Again, in this configuration, the auxiliary two-stroke engine can be operated with a controlled air fuel charge admitted thereto and the main engine can have variable fuel charges admitted thereto by means of the carburettor 16.

In the engine illustrated in Figures 20, 21 and 22, the main piston 8 operates under a four-stroke combustion cycle. It does, however, incorporate crankcase compression in the main crankcase 14, compressed air being delivered through the passage 55 to inlet ports 20 which will be opened shortly before bottom dead centre. The main advantage of this is to provide air from the crankcase into the cylinder towards the end of the exhaust stroke so as to purge exhaust gases through the exhaust valve 98, which will be open at this stage of the cycle. At the bottom of the induction stroke, some air may be admitted through the inlet ports 20 but the effect of this will be to provide a layer of air near the crown of the main piston 8. This again assists in maintaining a richer fuel near the main combustion chamber 46. It will be appreciated by those skilled in the art that a conventional form of four-stroke combustion cycle, that is to say without crankcase compression, could be provided, as will be described below. The arrangement of Figure 20 does have the disadvantage that normal lubricating techniques for four-stroke engines could not be utilised and techniques similar to those used on two-stroke engines would be required.

Figure 23 diagrammatically illustrates the timing sequence for the engine shown in Figures 20, 21 and 22. Figure 23 does not show opening and closing of the inlet ports 20 for clarity of illustration. The opening and closing of these ports does not, however, play a major role in the overall combustion cycle of the engine shown in Figure 20. The line 54 for displacements of the auxiliary piston is essentially the same as that shown in Figure 2. The line 100 shows various cycles for the main piston 8. Because it operates under a four-stroke cycle, there are two TDC points 51 and two BDC points 53 per cycle. The line segment 100a shows the expansion stroke of the main piston 8 followed by the exhaust stroke as indicated by the line segment 100b. The main piston 8 then executes an induction stroke as indicated by line segment 100c followed by a compression stroke as indicated by line segment 100d. Firing of the spark plug 40 occurs just before TDC point 50 so as to cause combustion of the correct air-fuel mixture in the auxiliary combustion chamber 38. This coincides with or is shortly before the TDC point 51 of the compression stroke of the main piston 8.

Figures 24A to 24P diagrammatically illustrate the various stages of the combustion cycles of both pistons of the two-stroke/four-stroke engine shown in Figures 20, 21 and 22. For clarity of illustration, Figures 24A to 24P do not show opening and closing of the inlet ports 20 but again these do not play a major role in the overall combustion cycle of the engine. In relation to the embodiment of Figures 20, 21 and 22, there is crankcase compression in both crankcases but, nevertheless, Figures 24A and 24P are still appropriate to illustrate the various stages of the combustion cycles. The timing of the arrangement is such that the auxiliary piston operates at half the rpm of the main piston. This is diagrammatically illustrated in Figures 24A to 24P where sixteen instants in a complete rotation of the auxiliary crankshaft 30 are indicated by the numbers 1 to 16. These instants coincide with sixteen instants in two complete rotations of the main crankshaft 12 which occur during the period in which the crankshaft 30 undergoes a single rotation. Further, both the main piston 8 and auxiliary piston 26 are at top dead centre in synchronism as diagrammatically illustrated in Figure 24A. As shown in Figure 24F, the main piston 8 is in an exhaust stroke when the inlet ports 42 of the auxiliary cylinder 24 will be opened. Thus, there will be a relatively high pressure at the inlet ports 42 and therefore compression will be required in order to force air stroke fuel through the inlet ports 42 into the auxiliary cylinder 24. Crankcase compression can take place in the auxiliary crankcase 32 or in the main crankcase 14, as described previously. The point 0 on the rotation diagram for the main crankshaft 12 is where ignition takes place and the expansion stroke occurs between points 1 to 4. The exhaust stroke occurs between points 4 to 8. The induction stroke occurs between points 8 to 12. The compression stroke occurs between points 12 to 16. During these two revolutions of the main crankshaft, the auxiliary crankshaft will undergo a single revolution, as indicated by the points 0 to 16 on its rotation diagram. It will be seen that at the rotation point 4 of the main crankshaft, the exhaust valve 98 opens and remains open through points 5, 6 and 7, as diagrammatically shown in Figures 24E to 24H. Towards the end of the exhaust stroke, the auxiliary piston 26 begins to expose the inlet ports 42, as diagrammatically shown in Figures 24G and 24H. At rotation points 8 and 9, as diagrammatically illustrated in Figures 24I and 24J, the inlet ports 42 are open for admission of a compressed correct mixture air fuel charge into the auxiliary cylinder 24. At rotation point 9, as indicated in Figure 24J, the inlet valve 90 of the main cylinder opens so as to allow induction of the main air fuel charge into the main cylinder. At rotation point 10, as diagrammatically shown in Figure 24K, the inlet ports 42 of the auxiliary cylinder will close. At rotation point 12, marking the end of the induction stroke, the inlet valve 90 will close as shown in Figure 24N. At this stage there will be some movement of air fuel mix from the main cylinder through the transfer duct 41 into the auxiliary cylinder 24. This occurs as a consequence of the relatively higher pressure which tends to be produced in the main cylinder. Because the auxiliary cylinder 24 and the main cylinder 6 are always in communication, however, the pressures therein will be substantially equal throughout the whole combustion cycle. Because of this it is preferred to supply an overly rich mixture to the auxiliary cylinder to compensate for the diluting effect of air or lean mixture passing through the duct 41 to the chamber 38. Accordingly, a substantially correct air fuel mix will be present in the combustion chamber 38 even at top dead centre of the auxiliary cylinder 24 as indicated by rotation point 0 or 16 of Figure 24A and ignition will take place at this point.

Because some of the air fuel mixture admitted to the main cylinder 6 passes into the auxiliary cylinder 24, the effective compression ratio of the main cylinder is reduced. This effect is diagrammatically illustrated in Figure 25 which will be discussed in more detail below.

Figures 26 and 27 illustrate a V-configuration of the engine which is diagrammatically shown in Figures 20 to 22. In this arrangement, the auxiliary crankshaft 30 is coupled to the conrods 28 of both banks of the auxiliary pistons 26. Further, the crankshaft 30 can be arranged to operate the push rods 94 and 95 for the inlet and exhaust valves 90 and 98 respectively for the main cylinders 6 as diagrammatically shown in Figure 27. It will be appreciated by those skilled in the art that the arrangement of the auxiliary cylinders 24 with respect to the banks of main cylinders 6 is a very convenient arrangement. It will be further appreciated that the use of the crankshaft 30, as diagrammatically illustrated in the upper part of Figure 27, is a very compact and efficient way of operating the various valves in the engine. In the arrangement illustrated in Figures 26 or 27, the auxiliary crankcases 32 could provide crankcase compression for the auxiliary cylinders but alternative arrangements could be provided to avoid the need for this crankcase compression.

Figures 28, 29 and 30 illustrate a further modified form of the invention. This arrangement is similar to that shown in Figures 20 and 21 except that the auxiliary cylinder and piston are located above the main cylinder piston as in the arrangement of Figure 1. Also in Figures 28, 29 and 30, the inlet ports 20 in main cylinder 6 are not required. This is another embodiment in which the main piston 8 undergoes a four-stroke combustion cycle whereas the auxiliary piston 26 undergoes a two-stroke cycle. Because the auxiliary cylinder and piston are located above the main block 4, it is theoretically possible to incorporate these components into a novel head 3 which could be retrofitted to an existing four-stroke engine. Alternatively, the new head could be fitted to a four-stroke block which has been manufactured in a conventional four-stroke vehicle engine plant without the need for substantial changes in the manufacturing plant to derive the benefits of the engines of the invention.

It will be seen from Figure 28 that the inlet and exhaust valves 90 and 98 are operated by means of respective rocker arms 92 and 93 which are in turn controlled by means of cam elements 96 and 97 mounted on the auxiliary crankshaft 30. As seen in Figure 30, the arrangement of the auxiliary cylinder and piston is generally the same as that shown in Figure 1 and therefore need not be described in detail. Figure 29 is a schematic plan view showing the location of the inlet and exhaust valves 90 and 98 as well as the spark plug 40. This engine functions essentially the same way as that shown in Figures 20 and 21 as described above. Crankcase compression is required in the crankcase 32 so that fuel will pass through the inlet ports 42 at bottom dead centre of the piston 26 because at that point the main piston 8 has begun its compression stroke as diagrammatically shown in Figure 24I.

Figure 31 illustrates a variation of the engine shown diagrammatically in Figure 30. In particular, the air fuel mixture from the carburettor 34 is inputted directly through input duct 99 to the inlet ports 42. In this arrangement, lubrication of the piston 26 and crankshaft 30 can be arranged by supplying oil from the sump of the main crankcase 14. In other words, because there is no crankcase compression in the crankcase 32, it is not necessary to provide a fuel-oil mix for lubrication as in the case of a conventional two-stroke engine (and as would be required for the arrangement of the auxiliary engine of Figure 30). It is, however, necessary to rearrange the timing between the pistons 8 and 26 in order to achieve correct operation of the respective engines. In particular, it is desirable to retard the crankshaft 30 relative to the timing of the main crankshaft 12. The main purpose for doing this is to arrange for the inlet ports 42 to be opened during an intake stroke of the main piston 8. This is diagrammatically illustrated in Figures 33A to 33P. Figures 33A to 33P correspond generally to Figures 24A to 24P and therefore need not be described in detail. It will be noted, however, that, as shown in Figure 33A, at top dead centre of the main piston 8, the auxiliary piston 26 has not yet reached top dead centre, that is to say it is approximately 90° retarded (as measured relative to the rotational positions of the main crankshaft 12). In other words, its crankshaft 30 needs to rotate through another 45° in order to achieve top dead centre. This is also apparent in Figure 32 where the TDC point 50 for the auxiliary piston 26 is 90° advanced relative to the TDC point 51 of the main piston 8. The exact angle can be varied according to the configuration but it is envisaged that the retard angle would normally be in the range 30° to 50° as measured relative to the cycle of the auxiliary piston 26 indicated by line 54.

The speed of operation of the two crankshafts is such that the main crankshaft 12 rotates at twice the rpm as the auxiliary crankshaft 30. As can be seen, however, in Figures

33I, 33J and 33K, the inlet ports 42 to the auxiliary cylinder 24 are open while the main piston 8 is undergoing its intake stroke. Thus fuel can be admitted through the carburettors 16 and 34 simultaneously because the inlet manifolds for both are substantially at atmospheric pressure. In other words there is no positive pressure in the auxiliary cylinder 24 which would tend to resist flow of air fuel from the carburettor 34. Accordingly, the arrangement of Figure 31 represents a significant and simple way to achieve substantial improvement in the running of an internal combustion engine, particularly since normal four-stroke lubricating methods can be used in both the two- and four-stroke parts of the engine.

Because some of the fuel air mixture admitted to the main cylinder 6 passes into the auxiliary cylinder 24, the effective compression ratio of the main cylinder is reduced, as in the case of the engine shown in Figures 20, 21 and 22. As indicated above, this effect is diagrammatically illustrated in Figure 25 which diagrammatically shows points on the rotation of the main and auxiliary crankshafts and the computed compression ratios calculated on the assumption that there was no communication path between them. Figure 25 is not drawn to scale because the stroke of the main piston 8 would normally be at least twice that of the small piston 26, assuming the main piston 8 was twice the diameter of the small piston 26. In the embodiments of the invention, however, there is communication between them and the net effect is that the compression ratio for them would be only slightly less than those computed for the large cylinder because it is normally about ten times the swept volume of the smaller cylinder. The semi-circles 120 and 122 represent the rotations of the crankshafts 30 and 12 respectively. Each has rotational points 1, 2, 3, 4 and 5 marked thereon. Point 1 represents bottom dead centre for the main piston 8 and point 5 represents top dead centre for the main piston 8. At the intermediate points 2 and 3, the theoretical compression ratios for the two cylinders are about the same (again assuming that there was no transfer duct 41 therebetween). At rotation points 4 and 5, however, there are significant differences as set out in Table 1 below.

TABLE 1

Rotation Point	Compression Ratio Auxiliary Cylinder 24*	Compression Ratio Main Cylinder 6*
2	1.17	1.09
3	1.54	1.53
4	2.3	3.46
5	4.72	9.3

* Assumes cylinders 6 and 24 not connected by the duct 41.

The actual values would, of course, depend on the relative bores and strokes of the cylinders. As a result of these considerations, it is probably desirable to further control the way in which air fuel charges are supplied to the respective cylinders. In summary, it would be desirable at full load to supply the correct air fuel mixtures to both the small and large cylinders. At half load it is desirable to supply 50% more fuel than the correct mixture to the small cylinder and correspondingly less fuel to the large cylinder. At idle it would be desirable to supply the correct mixture to the small cylinder and air only to the large cylinder. With modern fuel injecting equipment, such controls can readily be implemented and need not be described in detail.

The above mentioned fuel control techniques arise in motors where there is a two-stroke/four-stroke combination. These considerations are basically unnecessary in two-stroke/two-stroke combinations or four-stroke/four-stroke combinations when both crankshafts 12 and 30 rotate at the same speeds.

It will also be apparent from Figure 25 that firing of the charge occurs at approximately 20° before top dead centre of the main piston 8, that is at point 5 as shown in the drawing. At this stage, the auxiliary cylinder 24 has not yet reached its top dead centre and continues through to rotation points D and B as illustrated in the diagram. This serves to push any unburnt fuel or hot combustion products in the auxiliary piston through to the main combustion chamber. This effect also serves to improve overall combustion efficiency.

The auxiliary piston 26 preferably sweeps out substantially the entire volume of the auxiliary cylinder 24 which is possible because the contents of the cylinder are transferred into the auxiliary combustion chamber 38 and from there to the main combustion chamber 46. In the main cylinder a head space of say 10% of its swept volume would be left in the usual way.

In the engine configuration diagrammatically illustrated in Figures 33A to 33P, it is most advantageous to adjust the head volume of the main cylinder 6 so as to maintain the correct compression ratio, notwithstanding that the auxiliary piston is 45° behind. Thus at top dead centre of the main piston 8, the auxiliary piston 26 will not have reached top dead centre. Accordingly, it is advantageous to reduce the head volume at top dead centre of the main cylinder 6 so as to ensure that the correct compression ratio is achieved. Preferably this is in the range 6-10:1 and most preferably 8:1.

Figure 32 diagrammatically illustrates the displacement cycles for the main and auxiliary pistons where the piston 26 and 8 undergo two- and four-stroke combustion cycles and where there is no crankcase compression in the crankcase 32. The lines 54 and 100 represent the displacement cycles of the small piston 26 and large piston 8 respectively as before. These lines correspond generally to those of Figure 23 and therefore need not be described in detail. It will be observed that the TDC 50 of the auxiliary piston 26 is 90° advanced (relative to the cycle of main piston 8) relative to the TDC 51 of the main piston 8.

Figures 34 and 35 show a further modified form of the invention. This diagram schematically shows an engine in which there are a pair of main cylinders 6 mounted in line with a pair of auxiliary cylinders 24. The main pistons 8 and auxiliary pistons 26 are coupled to a single crankshaft 12. The configuration is such that adjacent pairs of main and auxiliary cylinders open into adjacent combustion chambers 38 and 46 by ducts 41 similar to the arrangement illustrated in Figures 11 and 12. The engine configuration shown in Figure 34 is capable of operating in a number of different modes. For instance, the main and auxiliary pistons could both execute two-stroke cycles or alternatively both could execute four-stroke cycles. In the illustrated arrangement, a two-stroke/two-stroke arrangement is illustrated. In a four-stroke/four-stroke arrangement, a new head design would be required which included inlet and exhaust valves in the usual way.

Figure 36 diagrammatically illustrates displacement cycles for the auxiliary and main pistons when both are operating in four-stroke combustion cycles. The line 100 for the main piston 8 has line segments 100a, 100b, 100c and 100d which correspond generally to those of Figure 23. The line 130 for the auxiliary piston 26 has line segments 130a, 130b, 130c and 130d corresponding to expansion, exhaust, induction and compression strokes respectively of the auxiliary piston 26.

Figure 37 illustrates a pilot engine device 101 which can be retrofitted to an existing engine. Generally speaking, the configuration of the auxiliary piston 26 and cylinder 24 is similar to that shown in Figures 9 and 10 except that the auxiliary cylinder 24 is located in a separate block 102 which is provided with a threaded spigot 104 which in use can be threaded into the spark plug hole 140 in the head 3 of an internal combustion engine or another hole (not shown) drilled and tapped for this purpose. The spring 80 is located in a guide tube 142, the lower end of which is mounted on the block 102. The spark plug leads of the engine are connected to the spark plugs 40 which are mounted in the block 102 so as to ignite the correct mixture fuel which would be located within the transfer passage 82. The carburettor 34 supplies air fuel mixture to the auxiliary cylinder, in the same way as described in relation to Figures 9 and 10. When combustion takes place in the auxiliary cylinder 24, the combustion products pass through the transfer duct 41 formed in the spigot 104 and are directly admitted into the combustion chamber 46 of the head 3 or main cylinder 6. Improved combustion thus takes place in the main combustion chamber as before. For retrofitting of a multi-cylinder engine, pilot engine devices 101 would be mounted in each of the spark plug holes 140 in the head of the engine, the spark plug leads being coupled to the spark plugs of the pilot engine devices 101. A single carburettor 34 could be provided and connected to the respective devices 101 by means of an inlet manifold (not shown). The devices 101 would enable some of the advantages of the invention to be obtained by retrofitting existing engines.

Many modifications will be apparent to those skilled in the art without departing from the spirit and scope of the invention.

CLAIMS:

1. A method of operating an internal combustion engine (2) including the steps of:

introducing a first charge of air or air fuel mixture into a main cylinder (6) in which a main piston (8) is mounted for reciprocating movement;

introducing a second charge of air fuel mixture into an auxiliary cylinder (24) in which an auxiliary piston (26) is mounted for reciprocating movement;

causing the auxiliary piston to compress the second charge;

igniting the compressed second charge; and

transferring burning fuel and/or combustion products produced from the second charge to the main cylinder whereby any fuel in the first charge is burnt.

2. A method as claimed in claim 1 wherein the step of introducing the second charge is such that the charge does not substantially change.

3. A method as claimed in claim 2 wherein including the step of changing the amount of fuel in the first charge in accordance with power out requirements of the engine.

4. A method as claimed in claim 1, 2 or 3 including the step of transferring said combustion products to the main cylinder when the main piston is at or near top dead centre (TDC) of its operating stroke.

5. A method as claimed in any one of claims 1 to 4 including the step of operating the auxiliary piston so that it operates under a two-stroke combustion cycle.

6. A method as claimed in any one of claims 1 to 5 including the step of operating the main piston so that it operates under a two-stroke combustion cycle.

7. A method as claimed in any one of claims 1 to 5 including the steps of operating the main piston so that it operates under a four-stroke combustion cycle, operating the auxiliary

piston so that it operates under a two-stroke combustion cycle and synchronising said operating cycles so that at least parts of the expansion strokes thereof occur at substantially the same time.

8. A method as claimed in any one of claims 1 to 4 including the steps of operating the auxiliary piston so that it operates under a two-stroke combustion cycle, operating the main piston so that it operates under a four-stroke combustion cycle and coupling the auxiliary piston (26) and the main piston (8) to an auxiliary crankshaft (30) and main crankshaft (12) respectively and coupling the crankshafts together so that the main crankshaft rotates at twice the rate of rotation as the auxiliary crankshaft.

9. A method as claimed in claim 8 including the step of causing the main piston to execute an induction stroke as part of its operating cycle in which said first charge is admitted to the main cylinder and wherein said first charge passes through said auxiliary cylinder.

10. A method as claimed in claim 9 including the step of supplying lubricating fluid to said auxiliary and main crankshafts, said fluid not being mixed with fuel in said first or second charges.

11. A method as claimed in any one of claims 1 to 10 including the step of causing the main piston to compress the first charge whilst the second charge is being compressed by the auxiliary piston.

12. An internal combustion engine (2) including:

at least one main cylinder (6) with a main piston (8) mounted for reciprocating movement therein;

at least one auxiliary cylinder (24) with an auxiliary piston (26) mounted for reciprocating movement therein;

fuel supply means (16,34,62,63) for supplying first and second charges to said main and auxiliary cylinders respectively;

ignition means (40) for igniting fuel in the second charge; and

transfer means (41) for transferring burning fuel and/or combustion products from combustion of the second charge to said main cylinder to thereby cause burning of any fuel in said first charge.

13. An engine as claimed in claim 12 wherein the auxiliary cylinder has a substantially lower swept volume than the swept volume of the main cylinder.

14. An engine as claimed in claim 13 wherein the ratio of the swept volumes is in the range 0.05:1 to 0.5:1.

15. An engine as claimed in claim 14 wherein said ratio is about 0.1:1.

16. An engine as claimed in any one of claims 12 to 15 wherein the engine includes timing means (88) which is operable to control the timing of the auxiliary piston relative to the main piston so that the combustion products from the ignition of the fuel in the auxiliary cylinder are transferred to the main cylinder at or near top dead centre (TDC) thereof.

17. An engine as claimed in any one of claims 12 to 16 wherein the fuel supply means includes first fuel supply means (16,62) and second fuel supply means (34,63) for supplying said first and second charges respectively.

18. An engine as claimed in claim 17 including control means for controlling the amount of fuel in the first charges.

19. An engine as claimed in any one of claims 12 to 18 wherein the main piston operates under a two-stroke combustion cycle.

20. An engine as claimed in any one of claims 12 to 19 wherein the auxiliary piston operates under a two-stroke combustion cycle.

21. An engine as claimed in any one of claims 12 to 18 wherein the main piston operates under a four-stroke combustion cycle and the auxiliary piston operates under a two-stroke combustion cycle.

22. An engine as claimed in any one of claims 12 to 21 including an auxiliary combustion chamber (38) into which the second charge is compressed prior to ignition.

23. An engine as claimed in claim 22 wherein the transfer means includes a transfer duct (41) which communicates the auxiliary combustion chamber (38) to a main combustion chamber (46) of the main cylinder.

24. An engine as claimed in claim 23 wherein the main and auxiliary pistons are generally opposed to one another and the transfer duct extends generally axially between said main and auxiliary cylinders.

25. An engine as claimed in claim 23 wherein the main and auxiliary pistons are located generally side by side and the transfer duct extends generally transversely relative to the main and auxiliary cylinders.

26. An engine as claimed in any one of claims 12 to 25 wherein the main piston is coupled to a main crankshaft (12) and the auxiliary piston is coupled to an auxiliary crankshaft (30).

27. An engine as claimed in claim 26 including timing means (88) for coupling the main and auxiliary crankshafts together for controlled relative rotation.

28. An engine as claimed in claim 21 wherein the main piston is coupled to a main crankshaft (12) and the auxiliary piston is coupled to an auxiliary crankshaft (30) and including timing means (88) for coupling the main and auxiliary crankshafts together so that the main crankshaft rotates at twice the rate of rotation as the auxiliary crankshaft.

29. An engine as claimed in claim 28 wherein the main piston has an induction stroke as part of its combustion cycle and the fuel supply means is operable to supply the first charge such that it passes through the auxiliary cylinder before being admitted to the main cylinder.

30. An engine as claimed in claim 29 wherein the auxiliary crankshaft is located in an auxiliary crankcase (32) in which there is no crankcase compression.

31. An engine as claimed in claim 30 including lubricating means for supplying lubricating oil to the auxiliary crankcase.

32. An engine as claimed in any one of claims 12 to 31 wherein there are a plurality of pairs of said main and auxiliary cylinders and pistons.

33. An engine as claimed in claim 32 wherein the main cylinders are disposed in a V-configuration.

34. An engine as claimed in claim 26 or 27 wherein there are a plurality of pairs of said main and auxiliary cylinders and pistons, the main cylinders are disposed in a V-configuration and wherein all of the main pistons are coupled to the main crankshaft and all of the auxiliary pistons are coupled to the auxiliary crankshaft.

35. An engine as claimed in claim 34 wherein the auxiliary crankshaft is located above the main crankshaft.

36. In an internal combustion engine (2) having a block (4) in which a number of main cylinders (6) are located and in which main pistons (8) are mounted for reciprocating motion in a four-stroke combustion cycle, a head (3) which includes a plurality of auxiliary pistons (26) which are mounted for reciprocating motion, fuel supply means (34,63) for supplying charges of air fuel mixture to said auxiliary cylinders, the arrangement being such that the auxiliary pistons compress said charges prior to ignition thereof and wherein combustion

products pass into main combustion chambers of the main cylinders.

37. In an internal combustion engine (2) having a block (4) in which a number of main cylinders (6) are located and in which main pistons (8) are mounted for reciprocating motion in a four-stroke combustion cycle, and including a head (3) having a threaded spark plug hole (140) opening to a main combustion chamber (46) of each of the main cylinders (6), a pilot engine (101) which includes:

an auxiliary piston (26) mounted for reciprocating movement in an auxiliary cylinder (24);

fuel supply means (34,62) for supplying a charge of air fuel mixture to said auxiliary cylinder ignition means (40) for igniting said charge after compression thereof by said auxiliary piston; and

a transfer tube (41) having threads (104) thereon which enable threaded connection of the tube to the threads of the spark plug hole (140), the arrangement being such that combustion products from the said charge pass through said tube and enter the main combustion chamber.

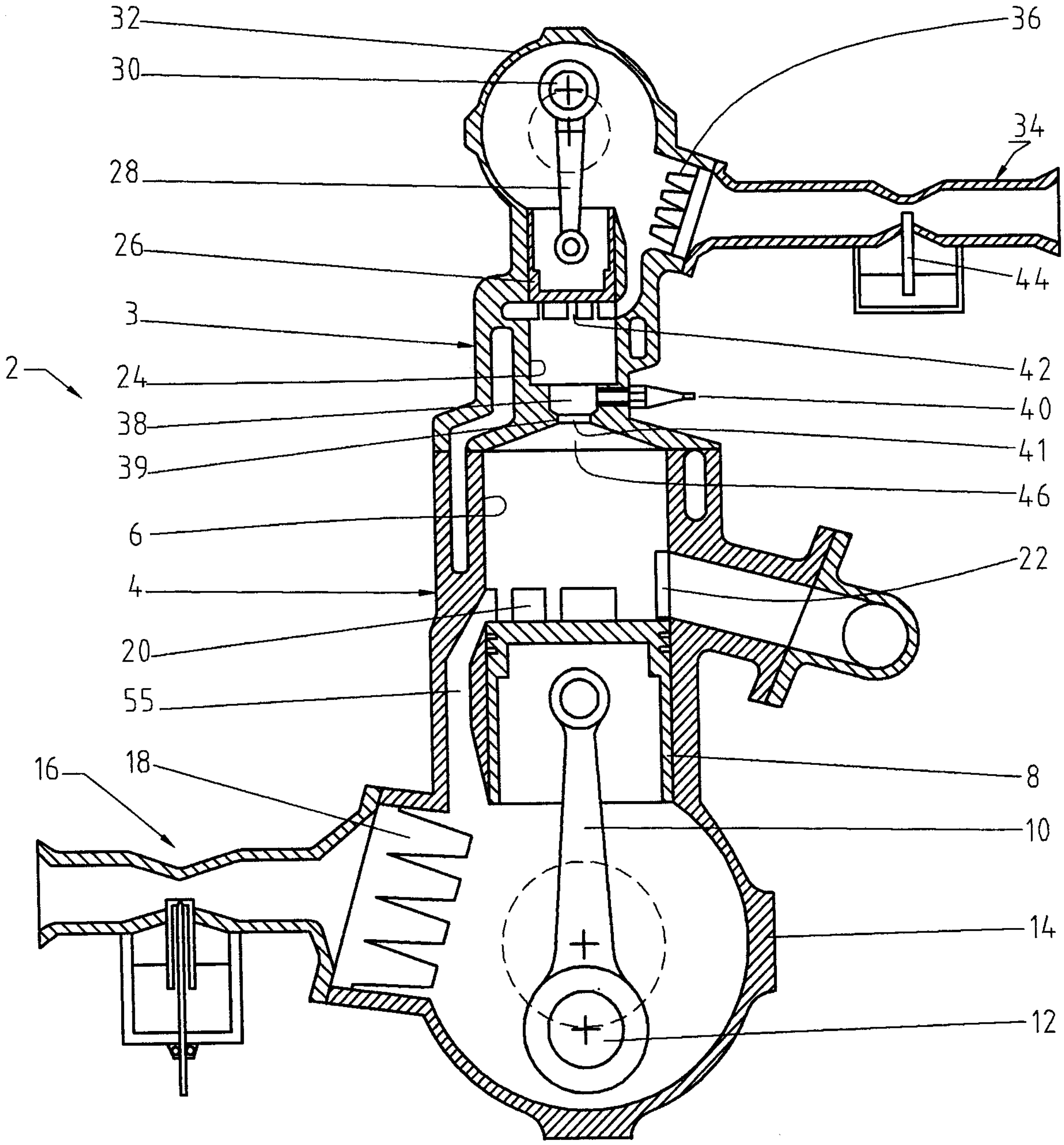


FIG 1

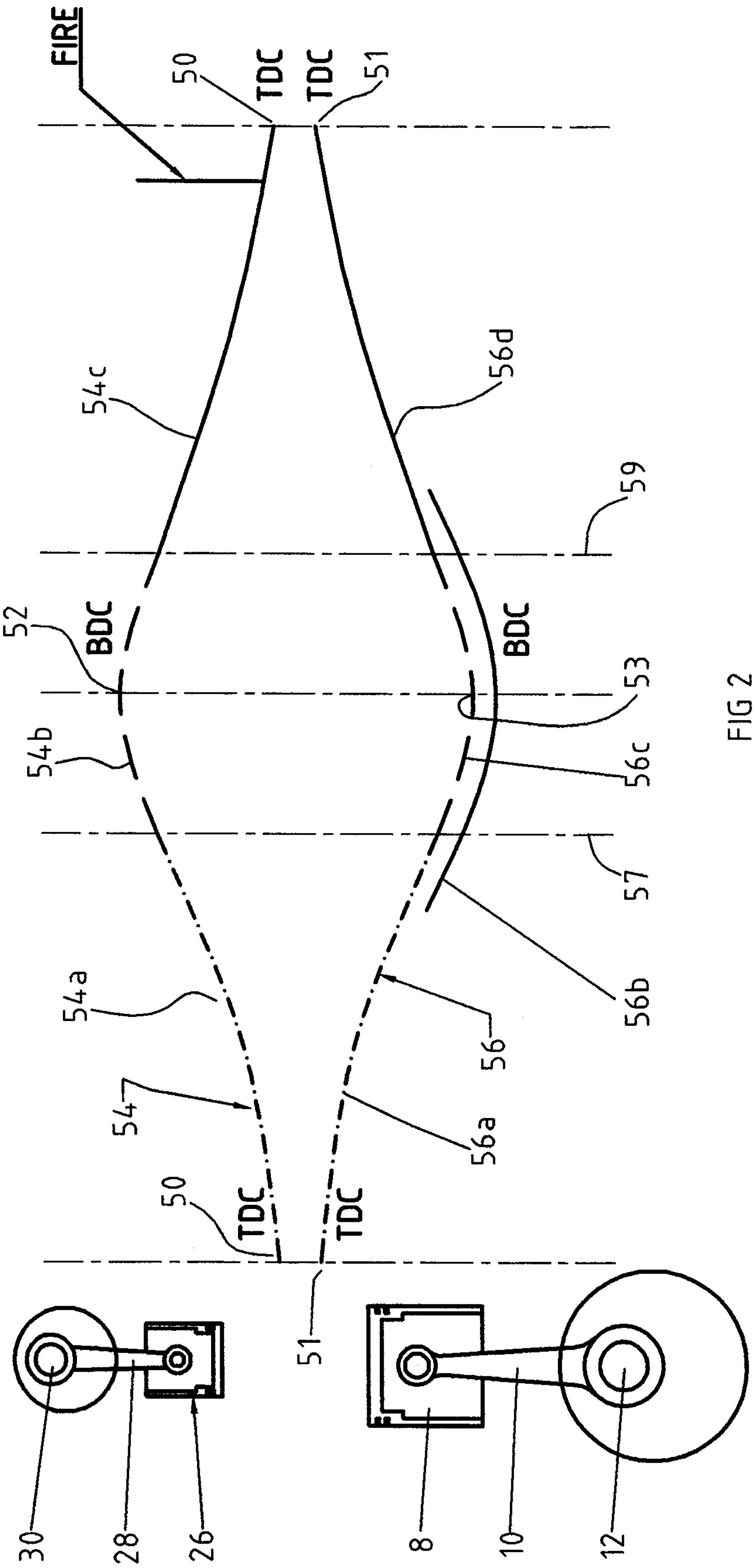
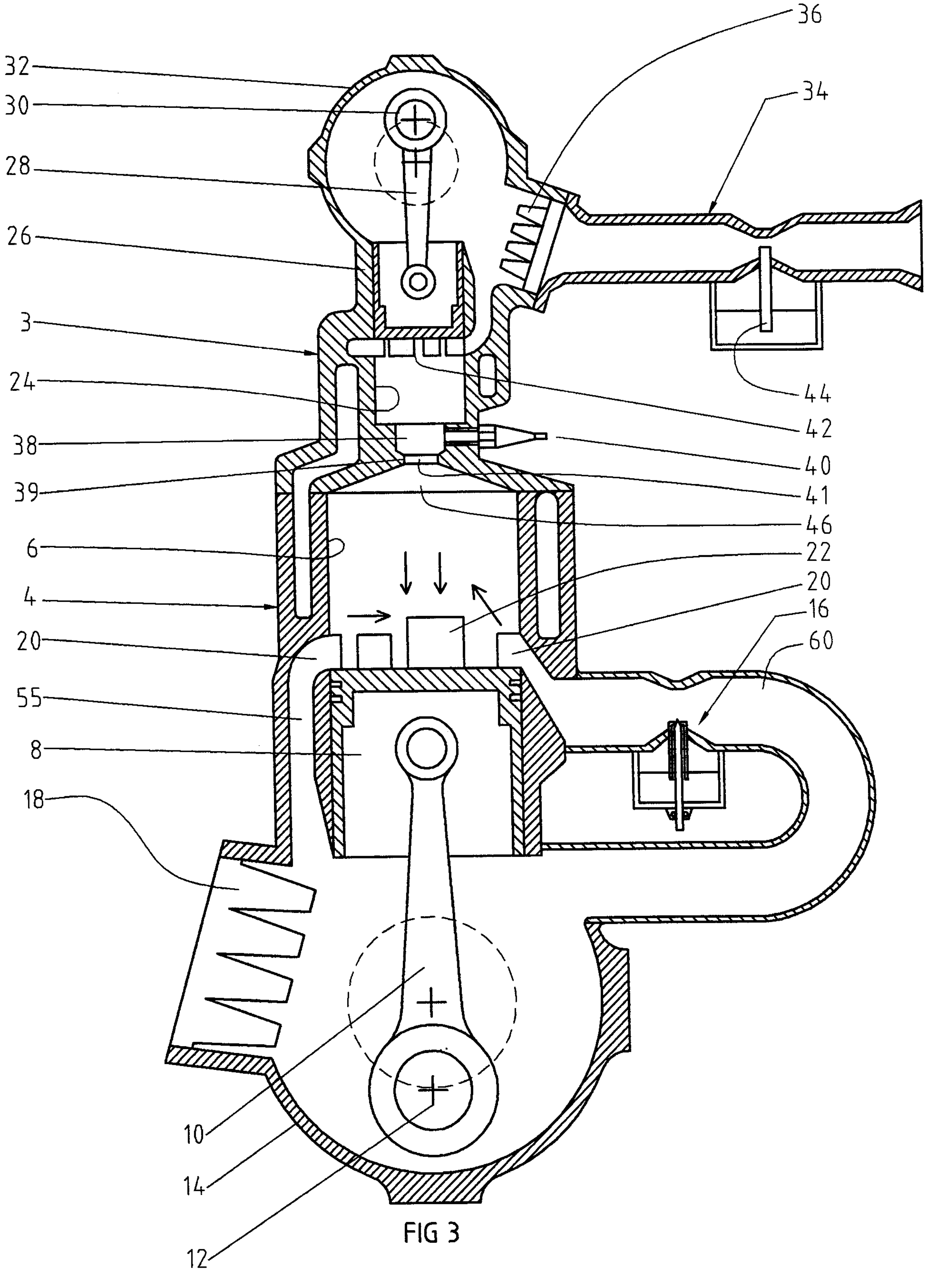

FIRE
50
TDC
TDC
51
54c
56d
59
52
BDC
BDC
53
54b
56c
57
54a
56b
56
54
56a
50
TDC
TDC
51
30
28
26
8
10
12

FIG 2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34
28
26
3
44
24
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39
41
46
6
22
4
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20
16
60
55
8
18
10
14
12

FIG 3

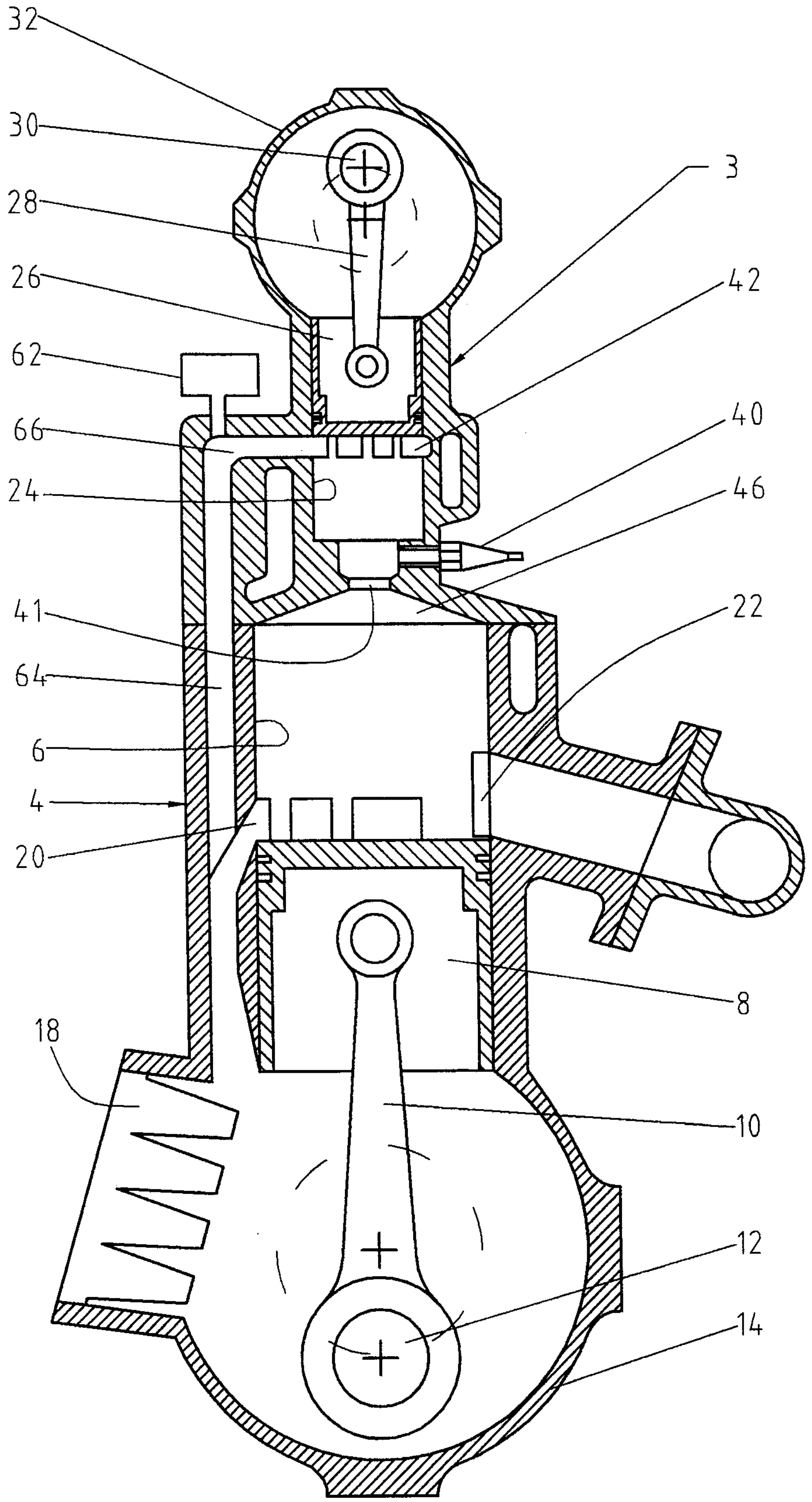


FIG 4

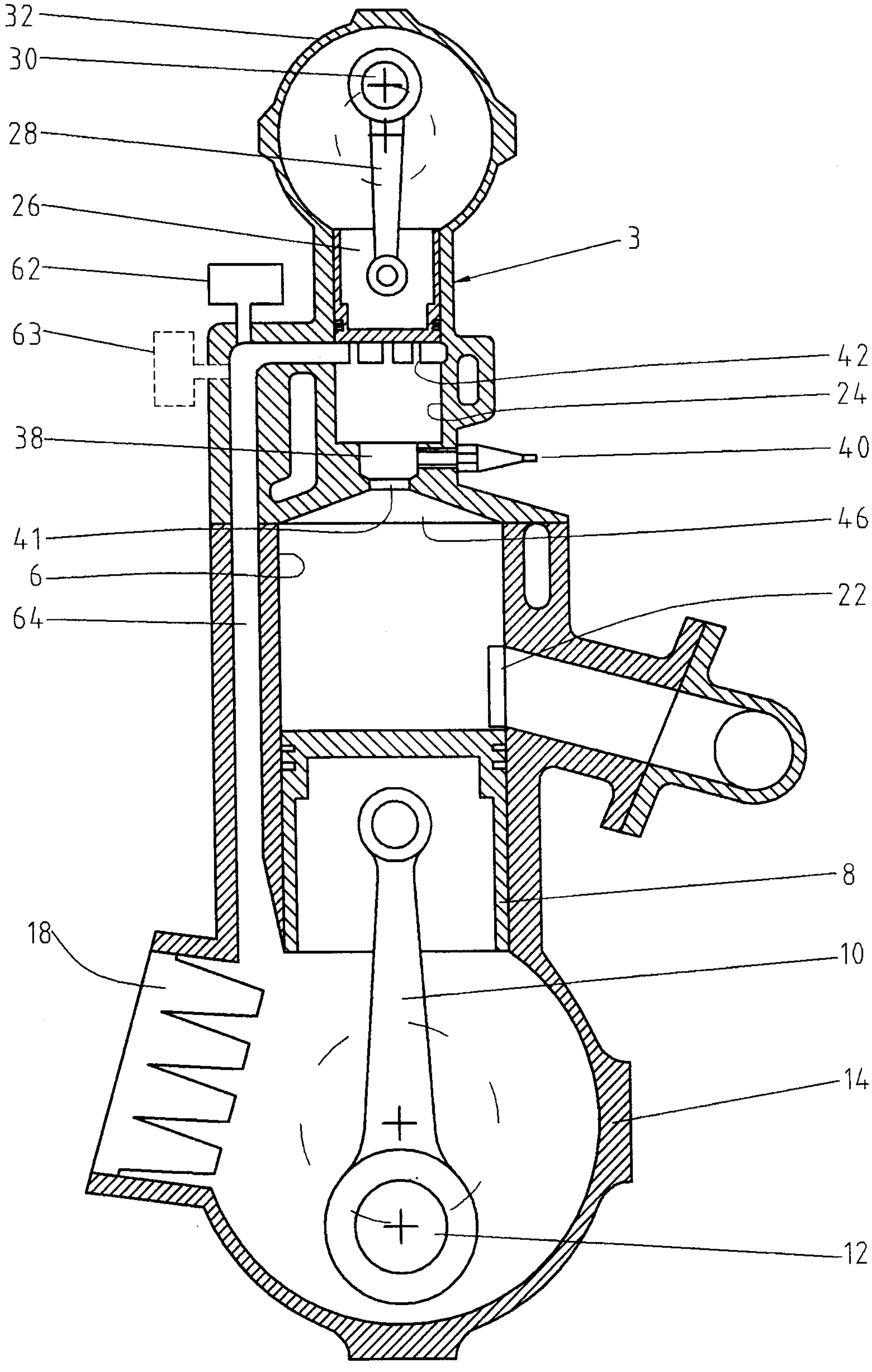


FIG 5

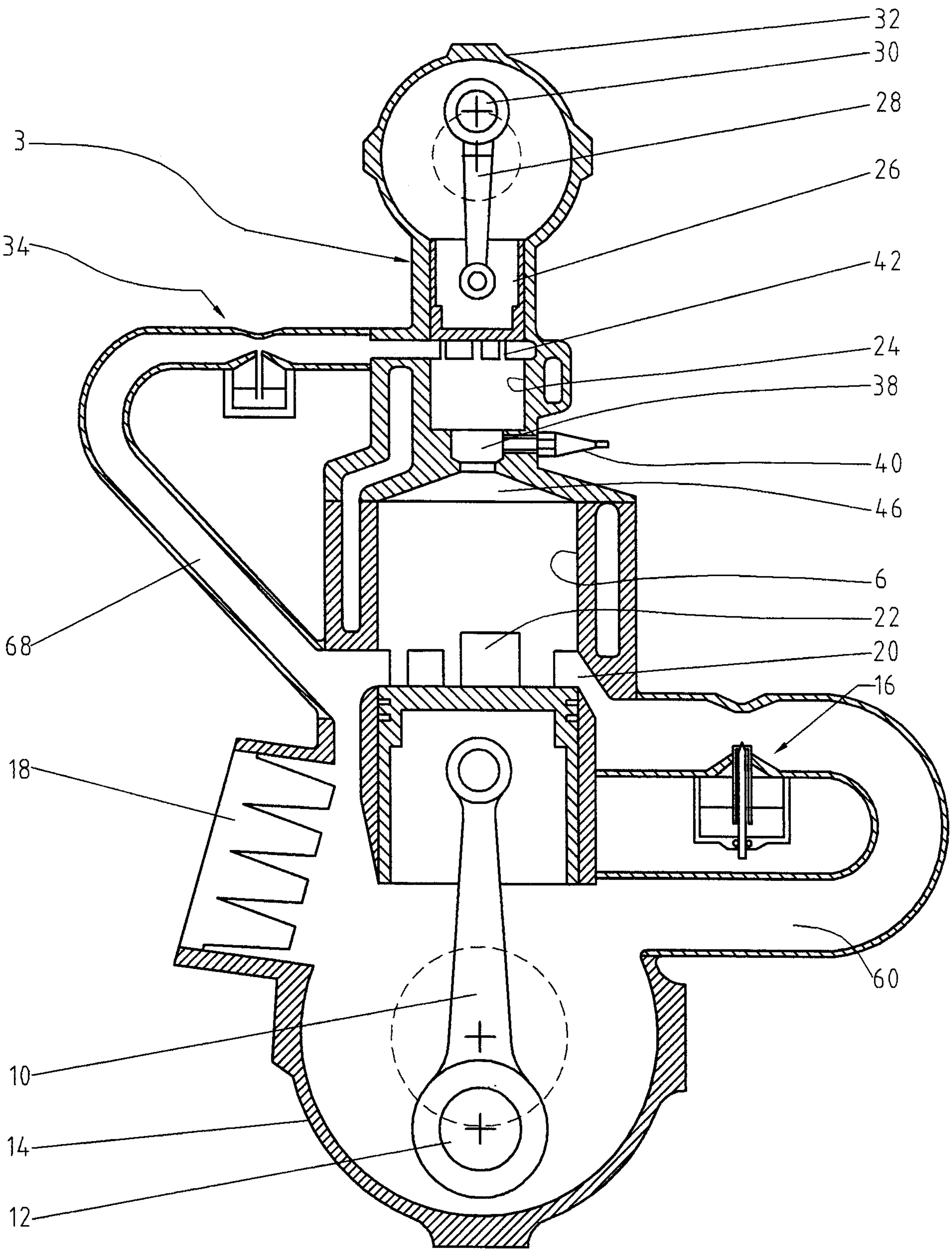


FIG 6

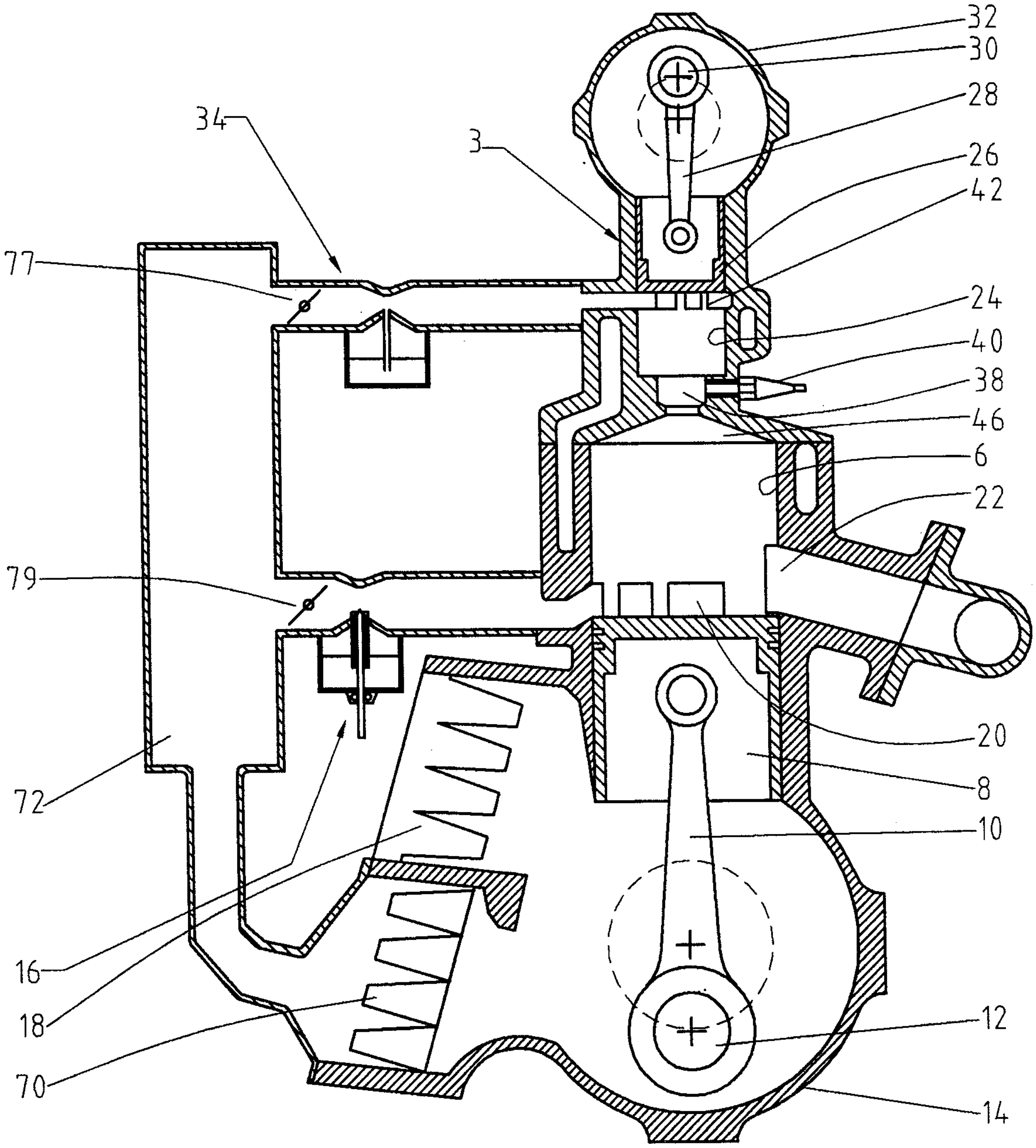


FIG 7

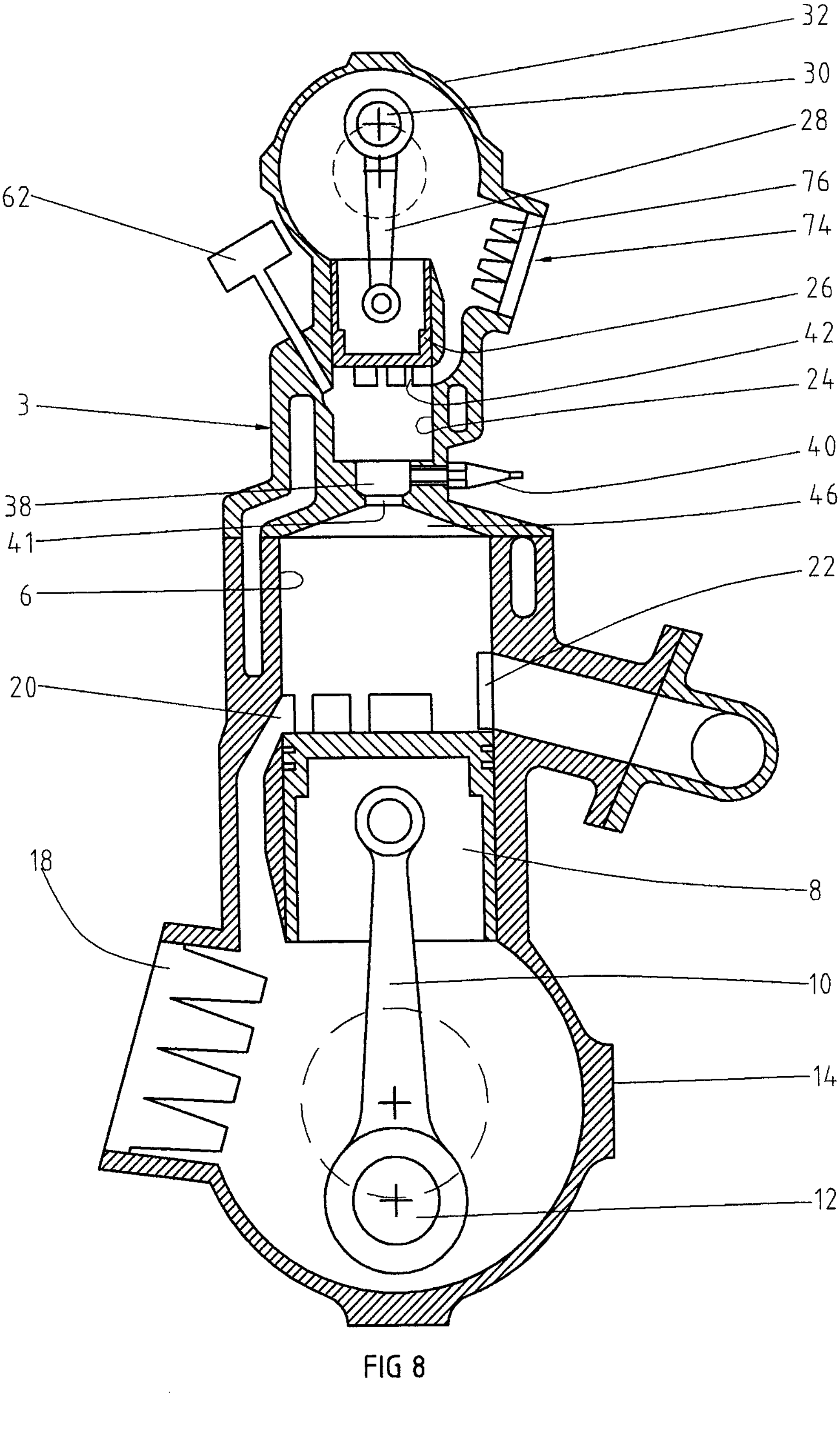
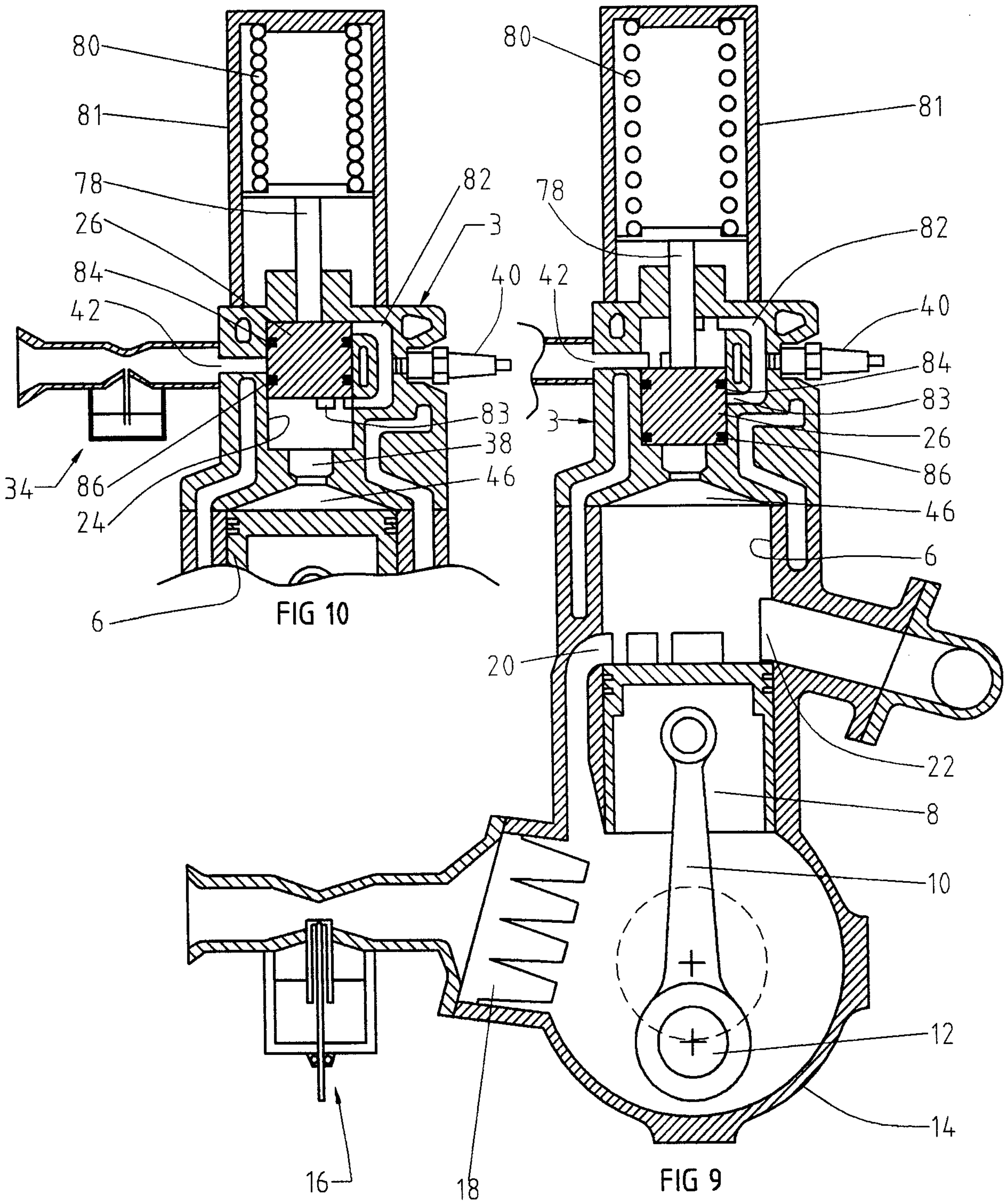


FIG 8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81
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26
84
42
34
86
24
82
3
40
83
38
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6
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81
78
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3
26
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46
6
20
22
8
10
12
14
16
18

FIG 10

FIG 9

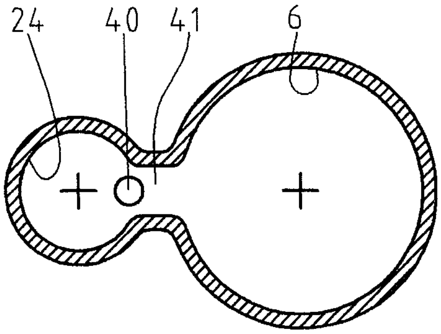


FIG 12

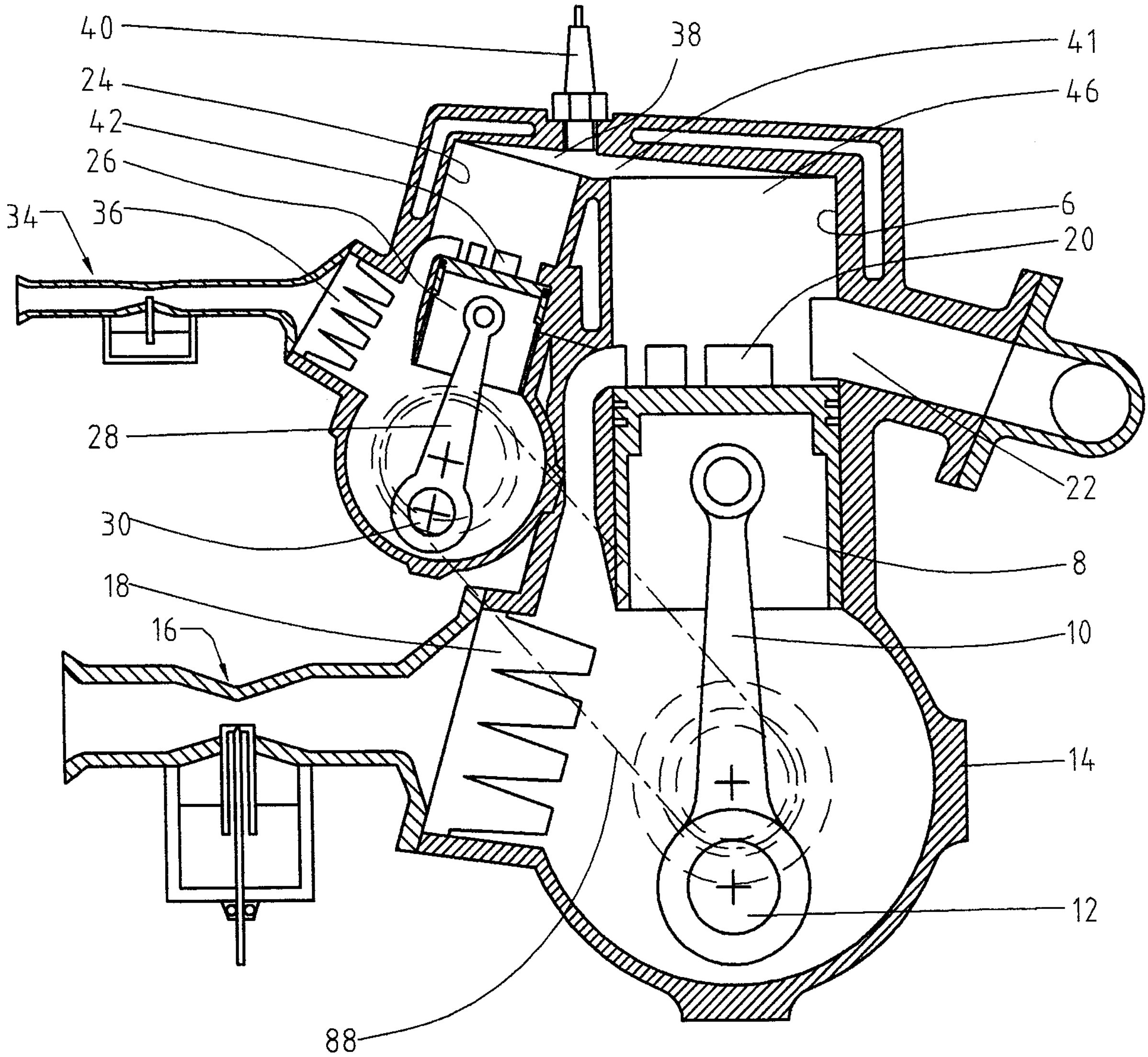


FIG 11

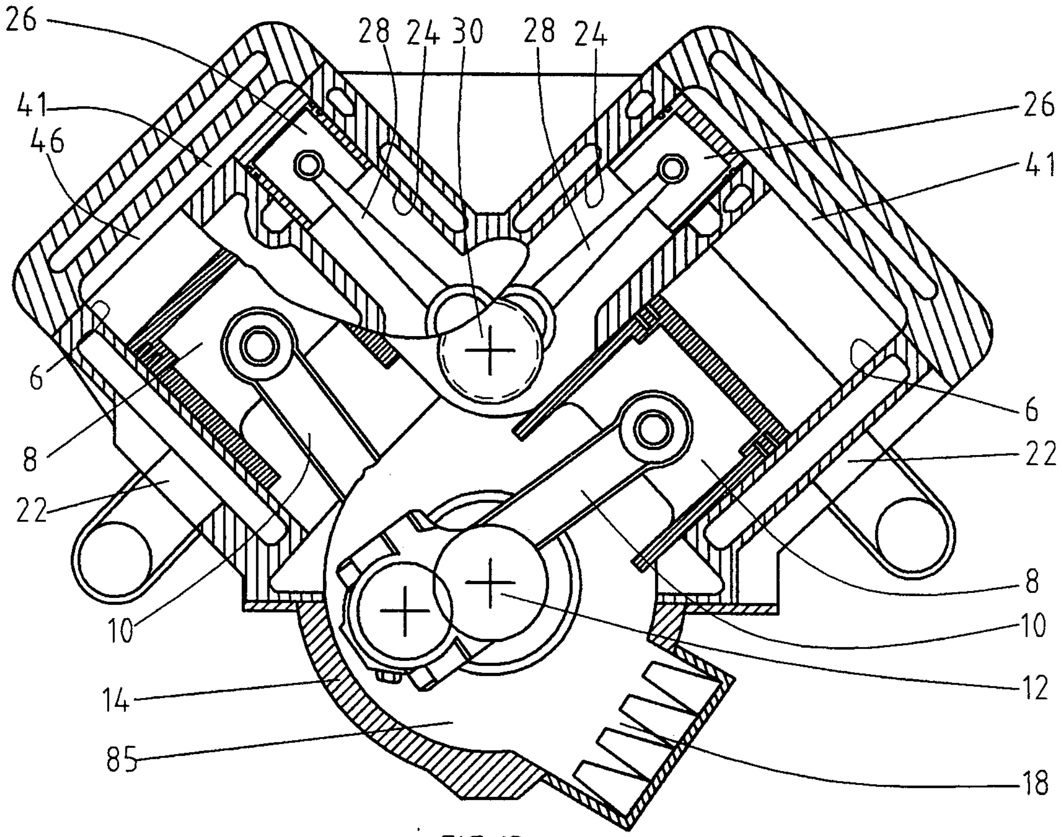
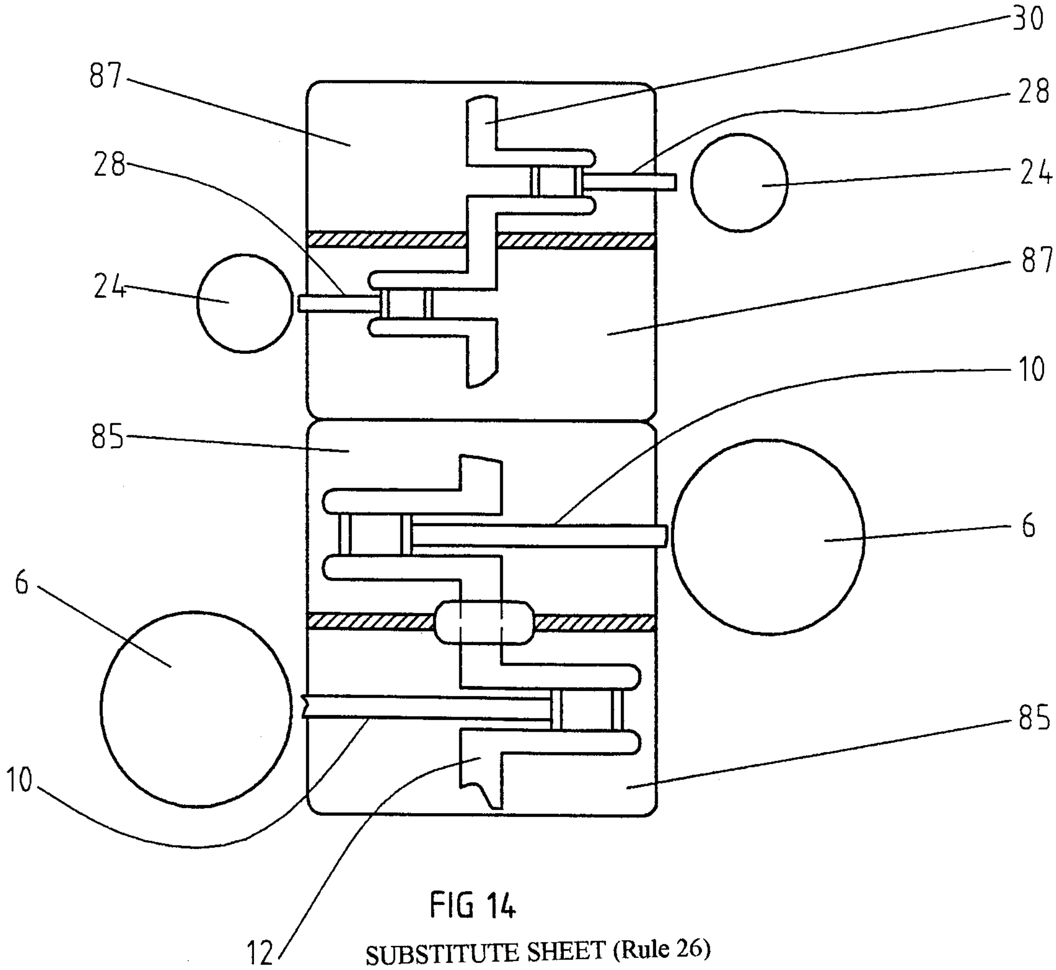

26
28
24
30
28
24
26
41
46
41
6
6
8
22
22
8
10
10
12
14
85
18

FIG 13


30
87
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28
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24
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10
85
6
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85
10
12

FIG 14

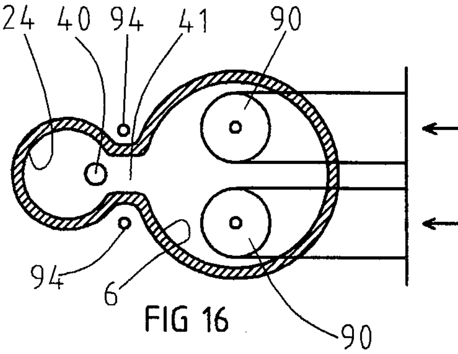
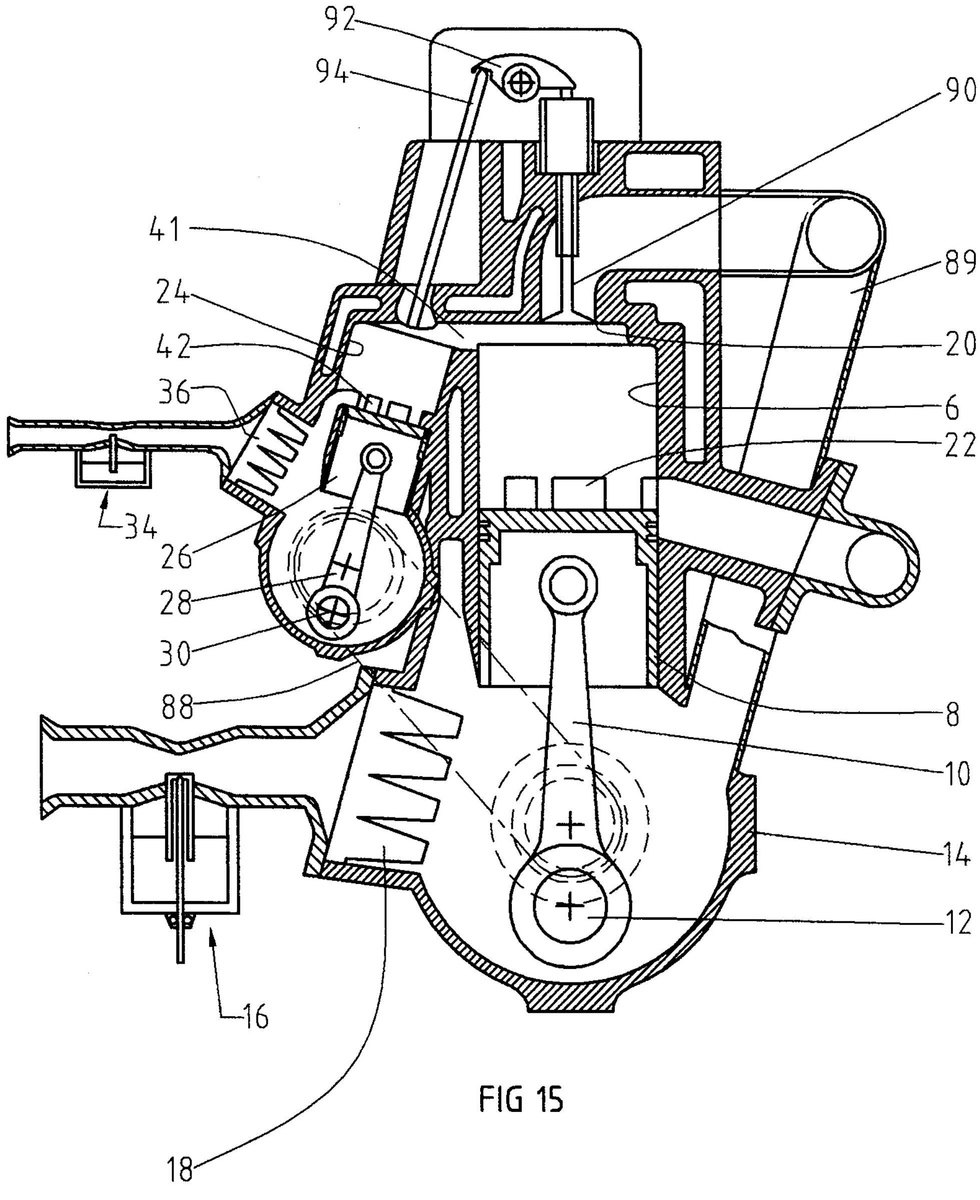

24
40
94
41
90
94
6
90

FIG 16


92
94
90
41
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42
20
36
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22
34
26
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8
88
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12
16
18

FIG 15

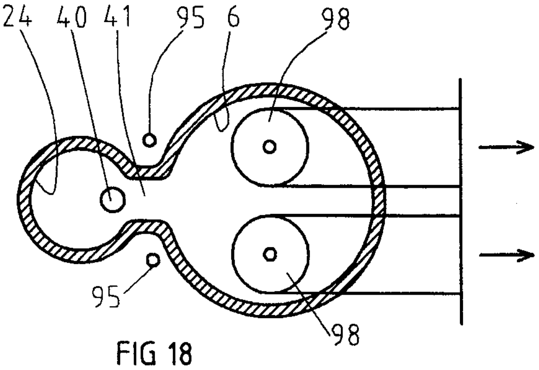


FIG 18

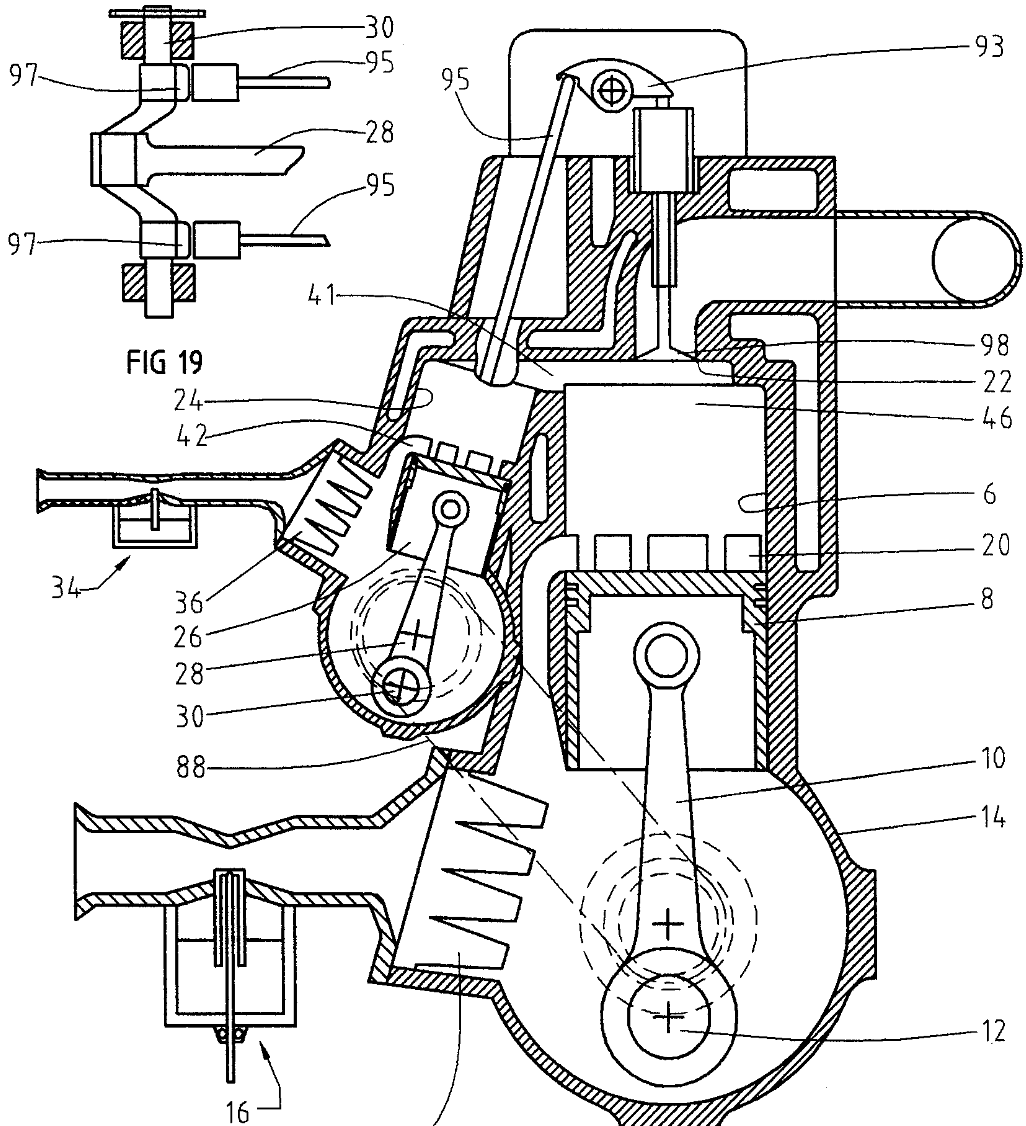


FIG 19

FIG 17

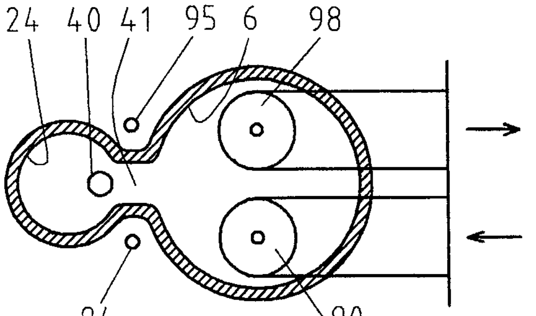


FIG 21

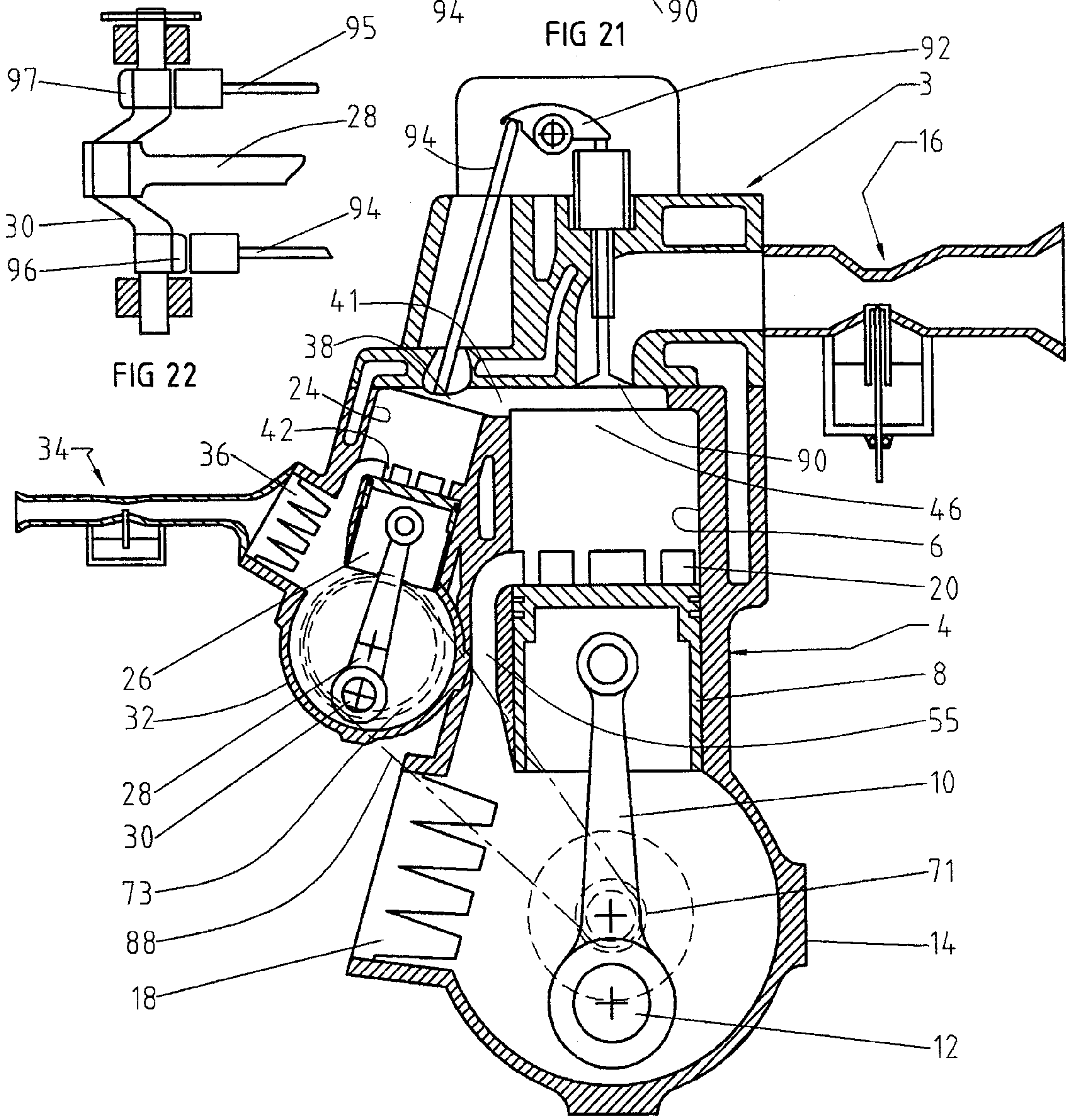


FIG 22

FIG 20

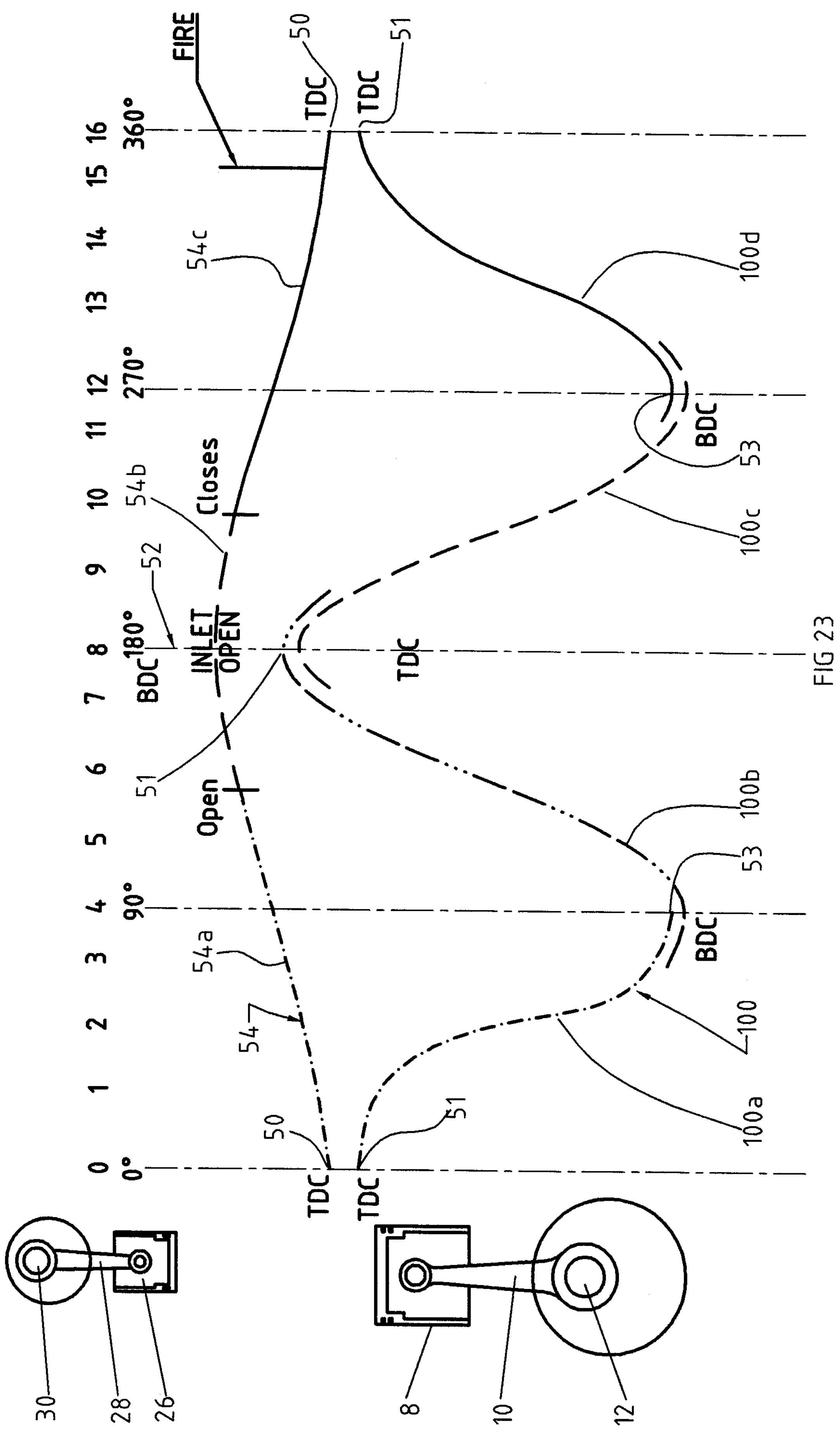

FIRE
50
51
TDC
TDC
360°
270°
180°
90°
0°
0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16
54c
100d
Closes
54b
52
BDC
53
100c
INLET
OPEN
TDC
BDC
51
Open
100b
53
54a
54
BDC
100
100a
50
51
TDC
TDC
30
28
26
8
10
12

FIG 23

b1

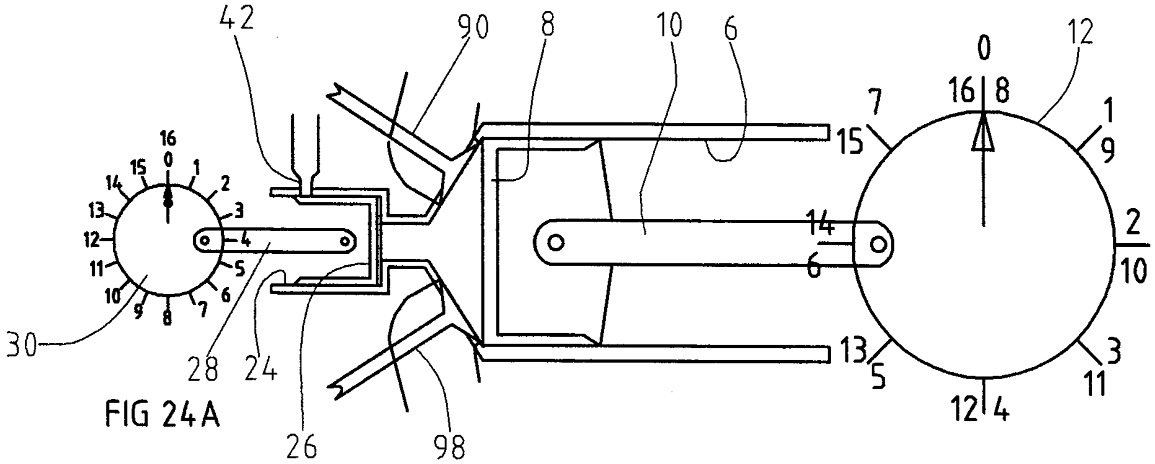


FIG 24A

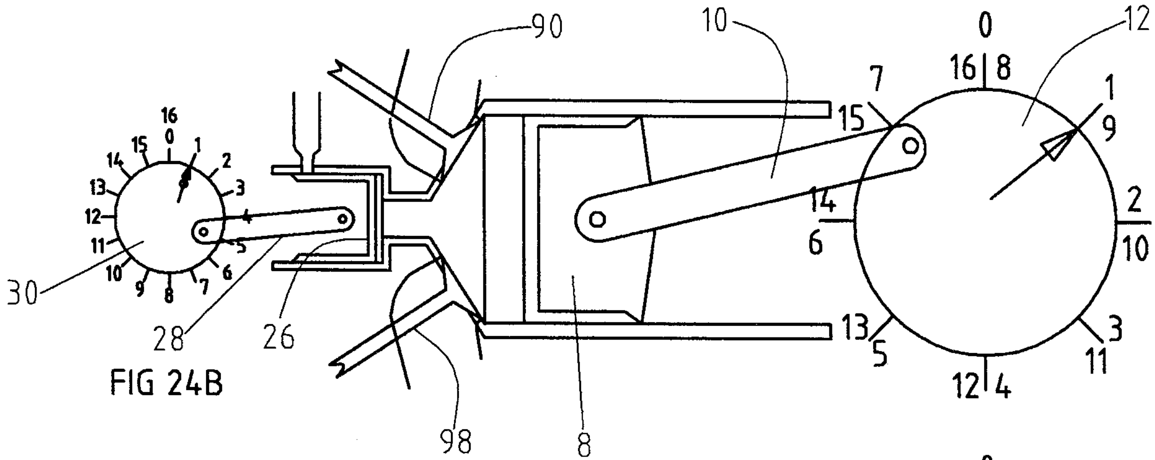


FIG 24B

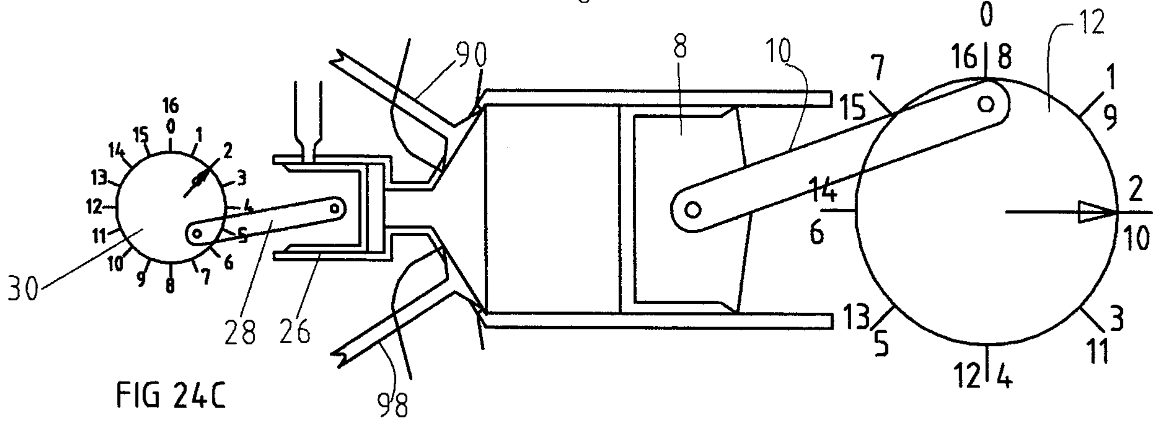


FIG 24C

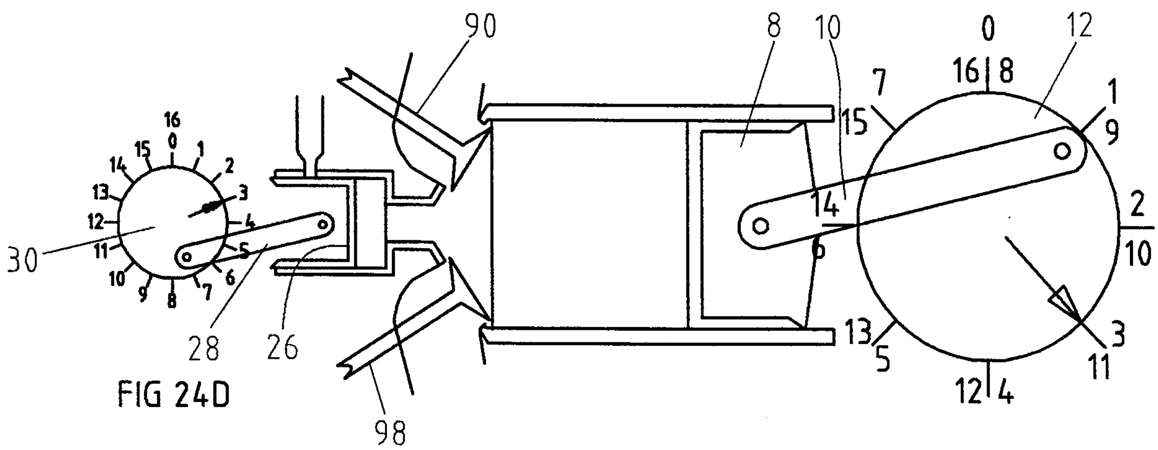


FIG 24D

17/31

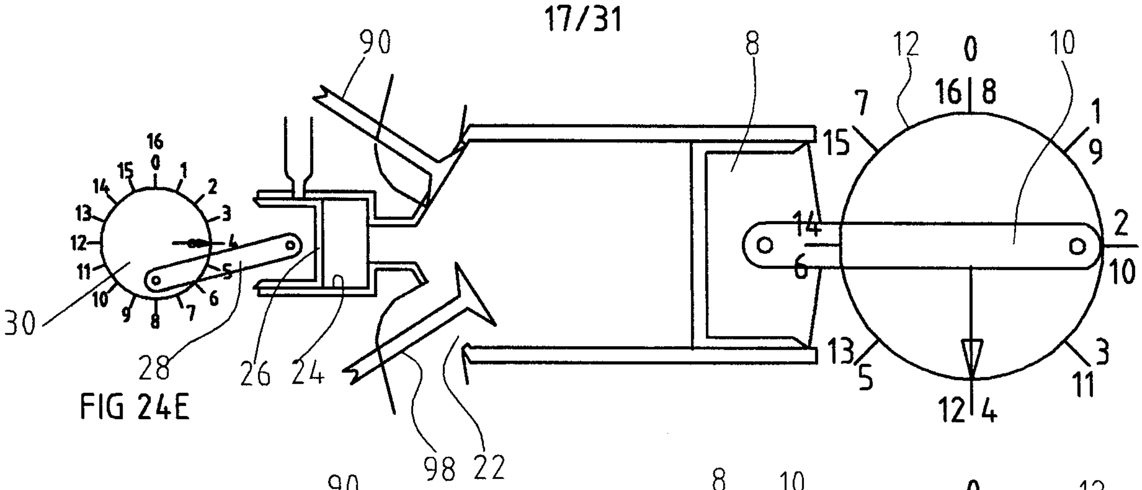


FIG 24E

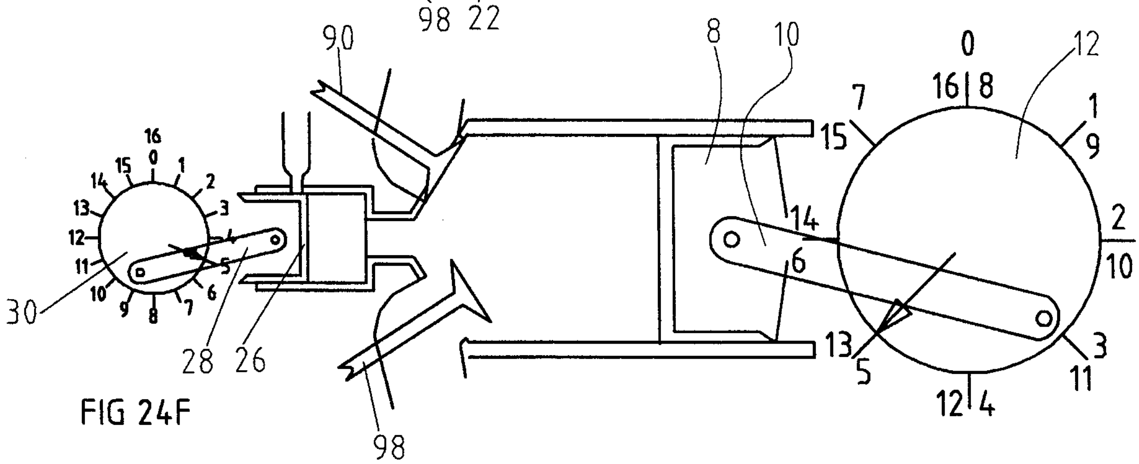


FIG 24F

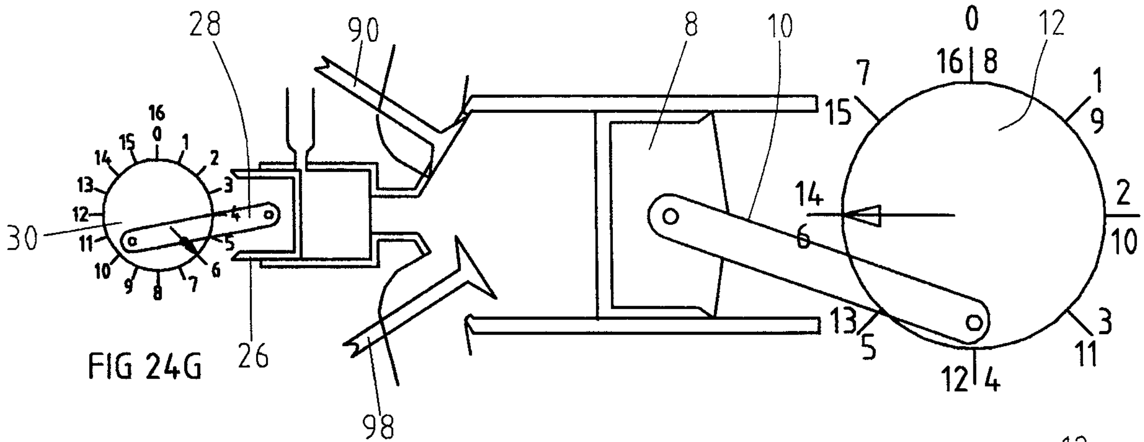


FIG 24G

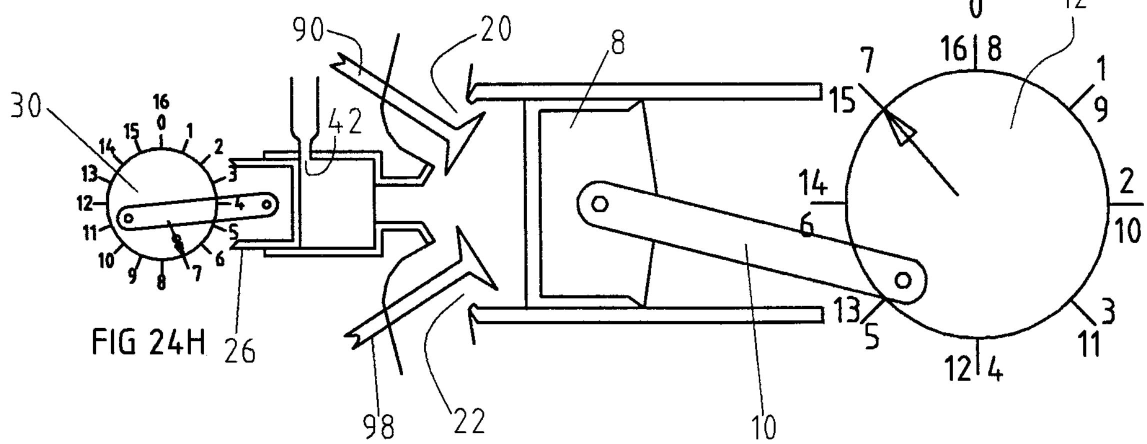


FIG 24H

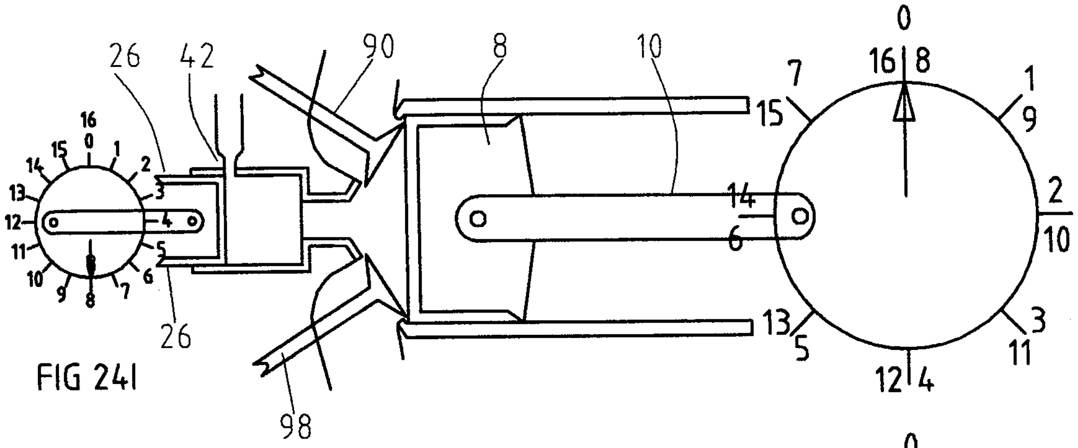


FIG 24I

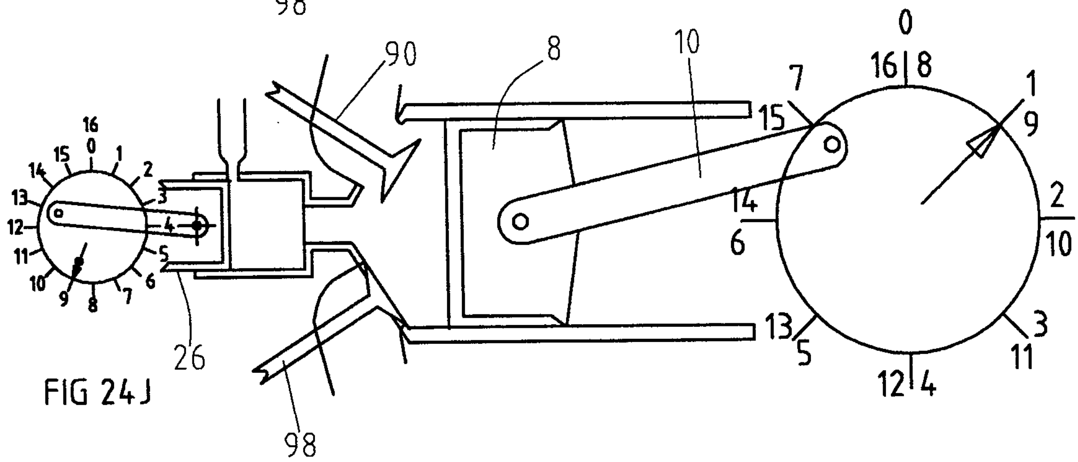


FIG 24J

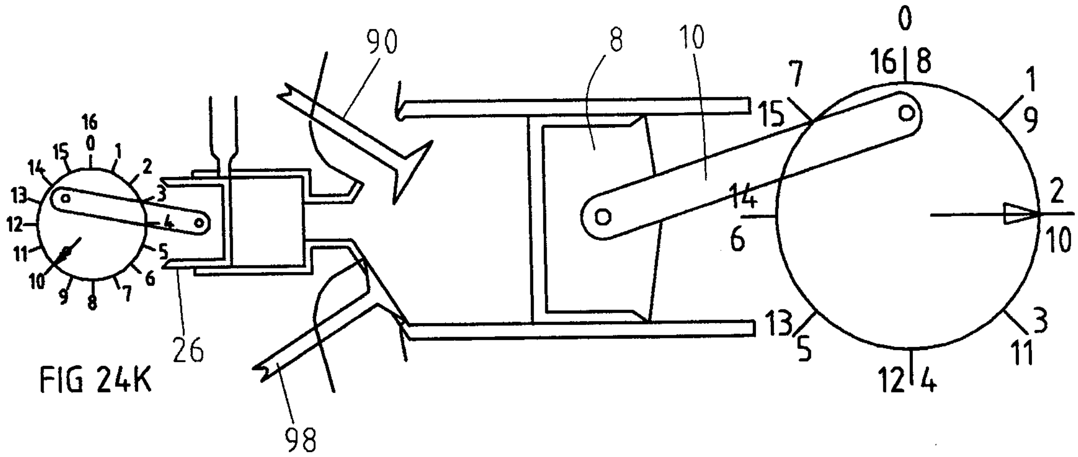


FIG 24K

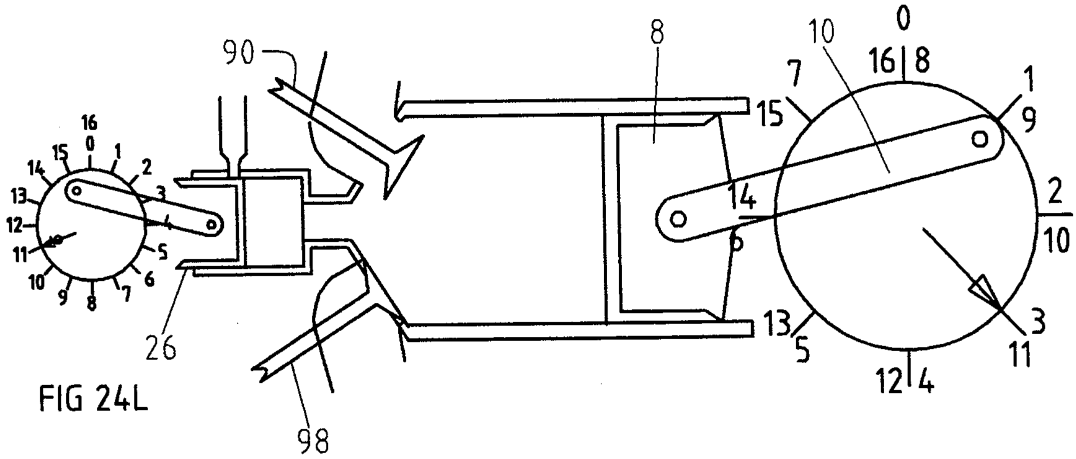


FIG 24L

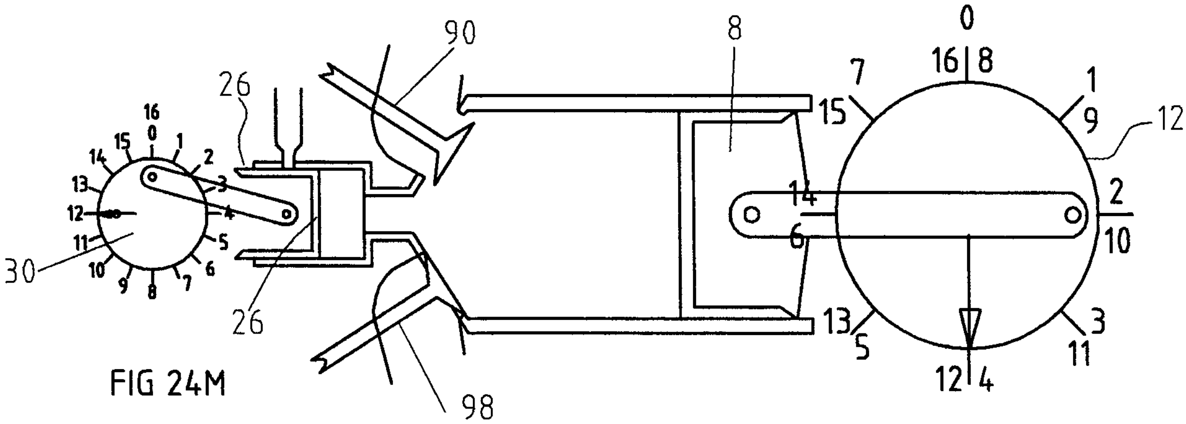


FIG 24M

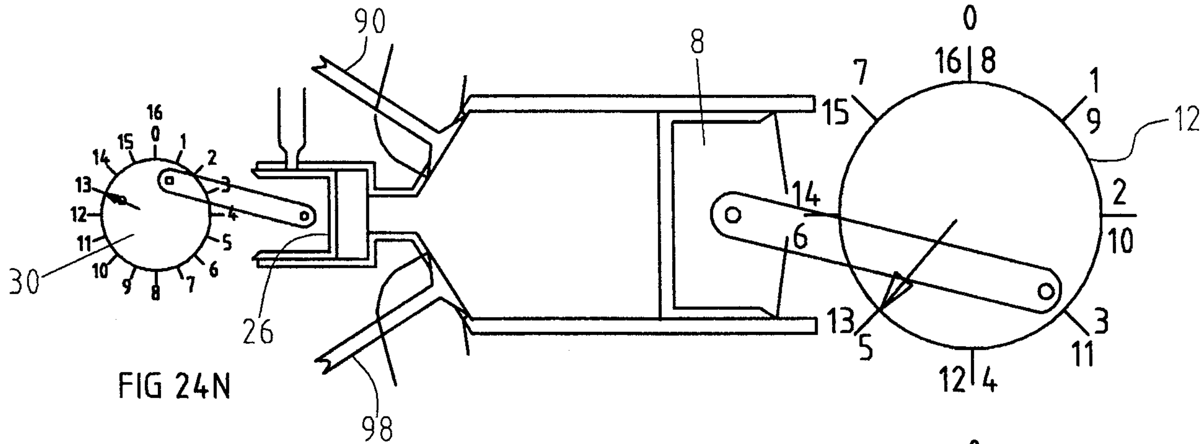


FIG 24N

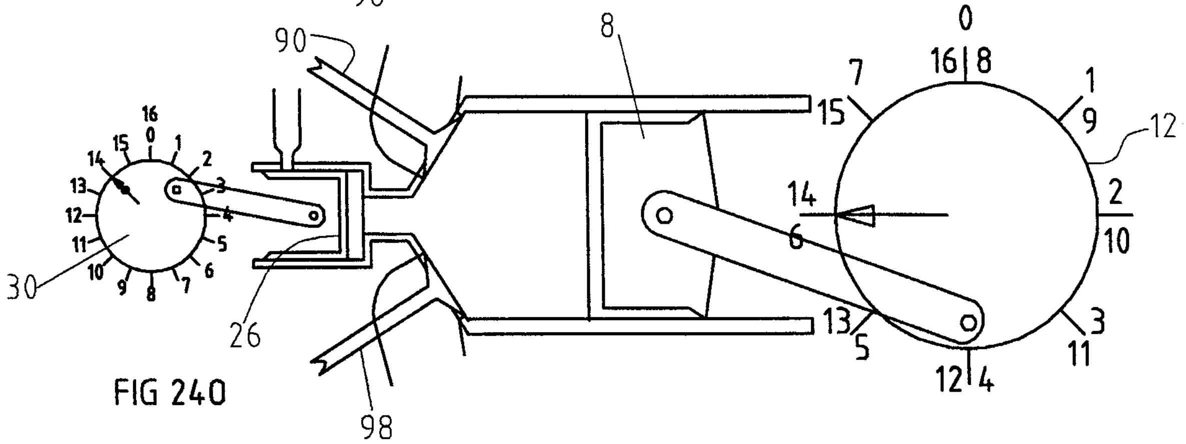


FIG 24O

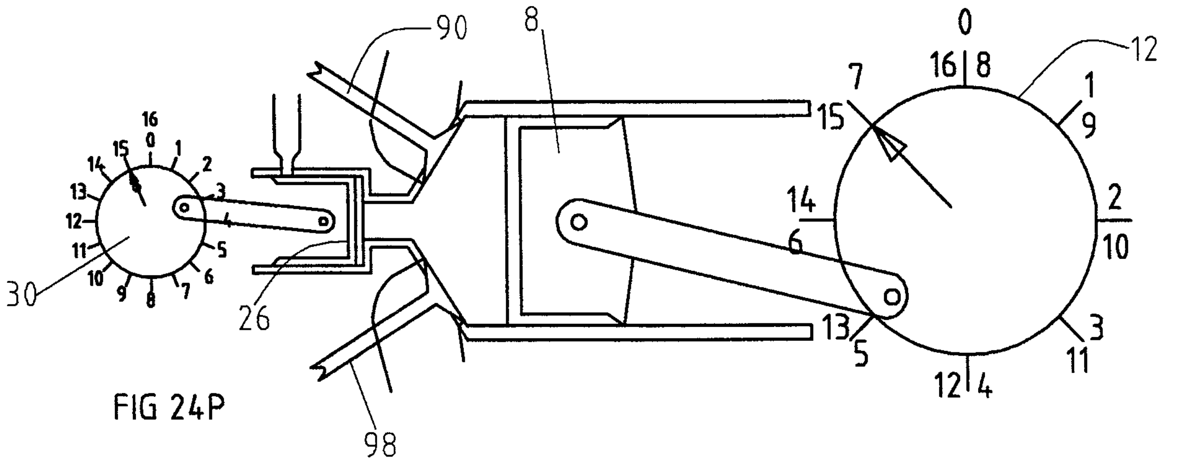


FIG 24P

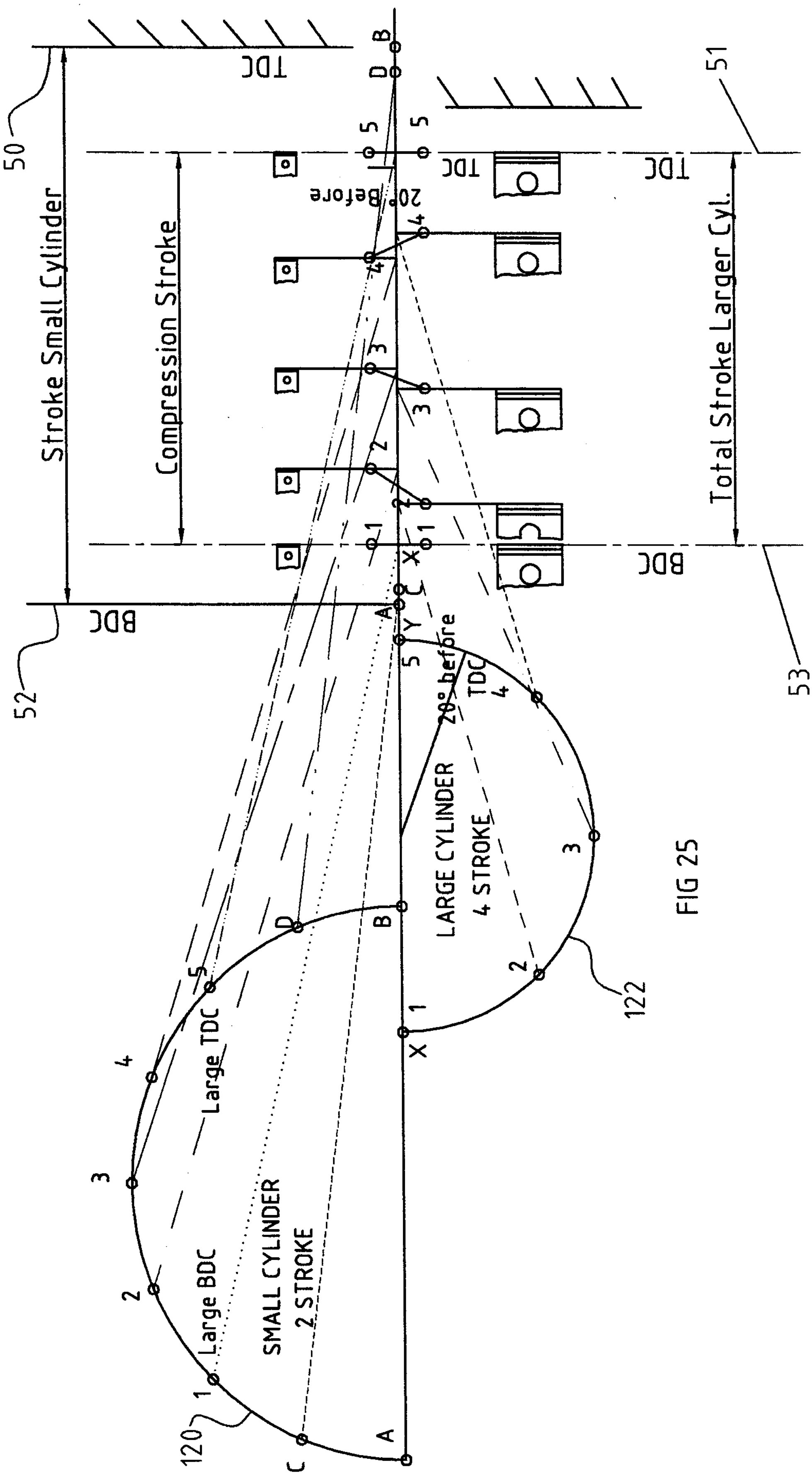
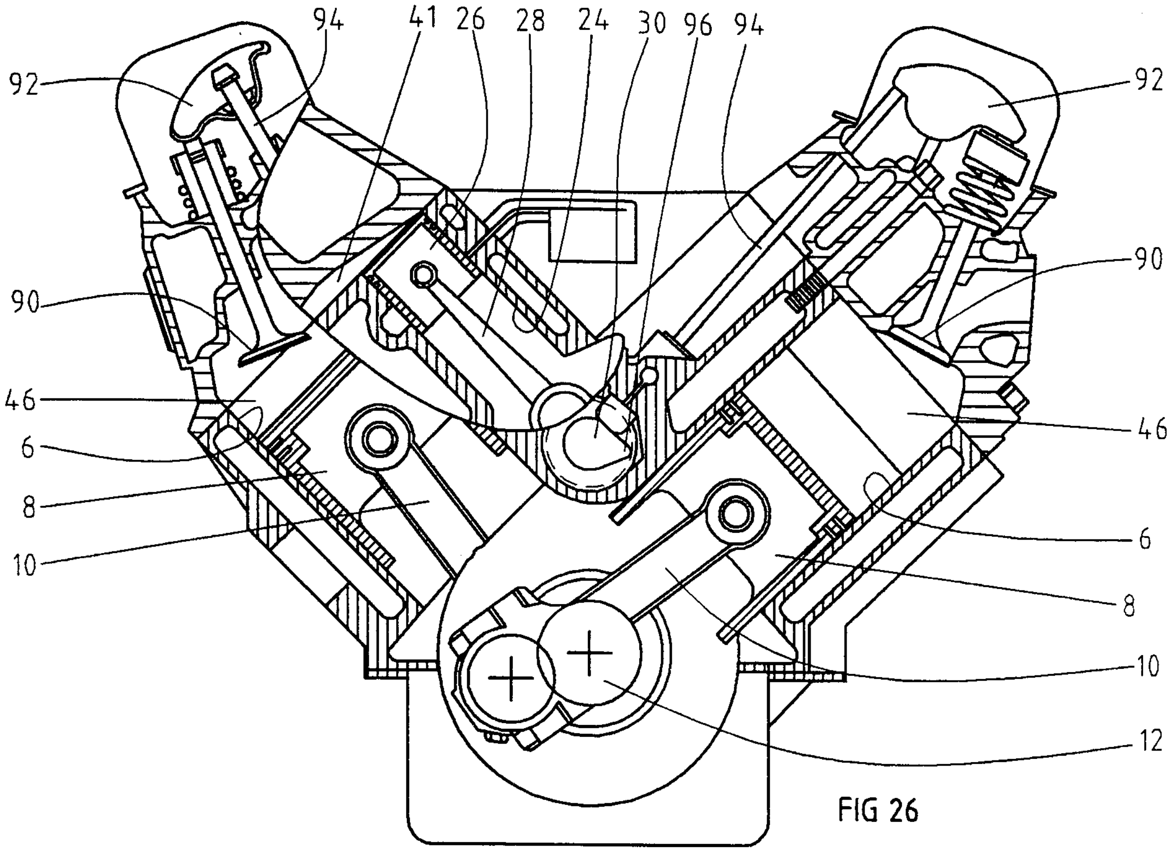

Stroke Small Cylinder
Compression Stroke
Total Stroke Larger Cyl.
TDC
BDC
20° Before
LARGE CYLINDER
4 STROKE
20° before
TDC
SMALL CYLINDER
2 STROKE
Large TDC
Large BDC
50
51
52
53
120
122

FIG 25


94
41
26
28
24
30
96
94
92
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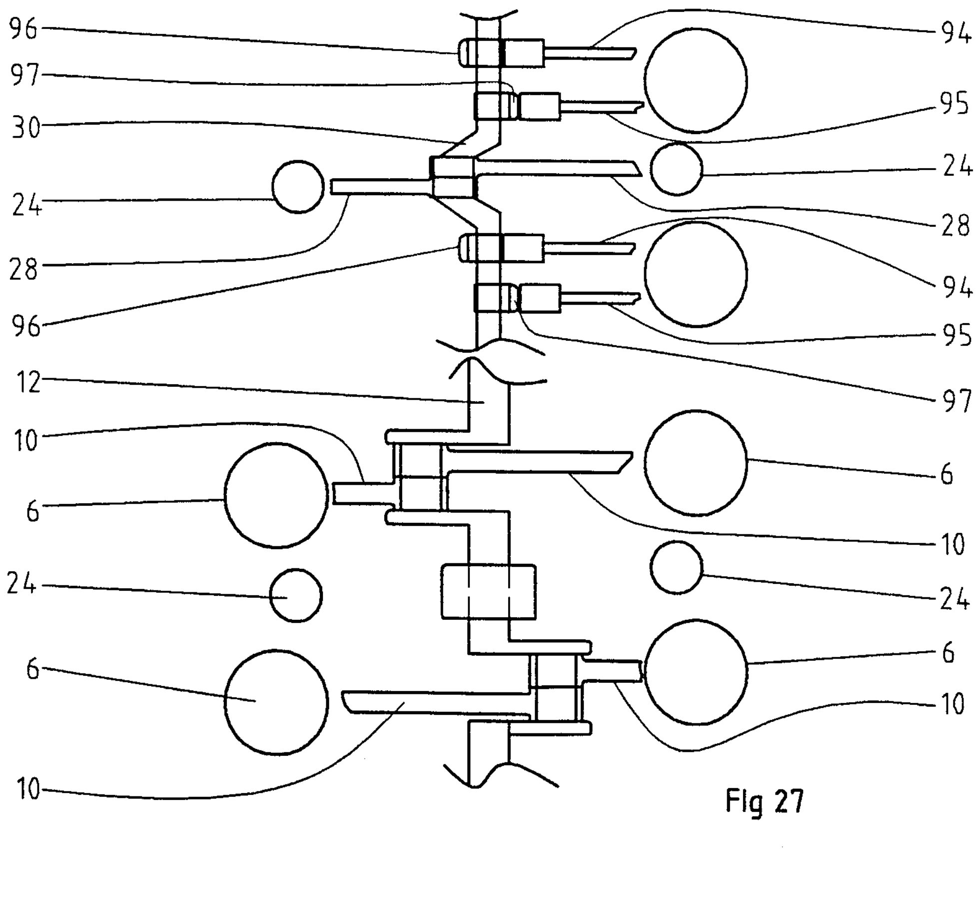
90
90
46
46
6
6
8
8
10
10
12

FIG 26


96
94
97
95
30
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24
28
28
96
94
95
97
12
10
6
6
10
24
24
6
6
10
10

Fig 27

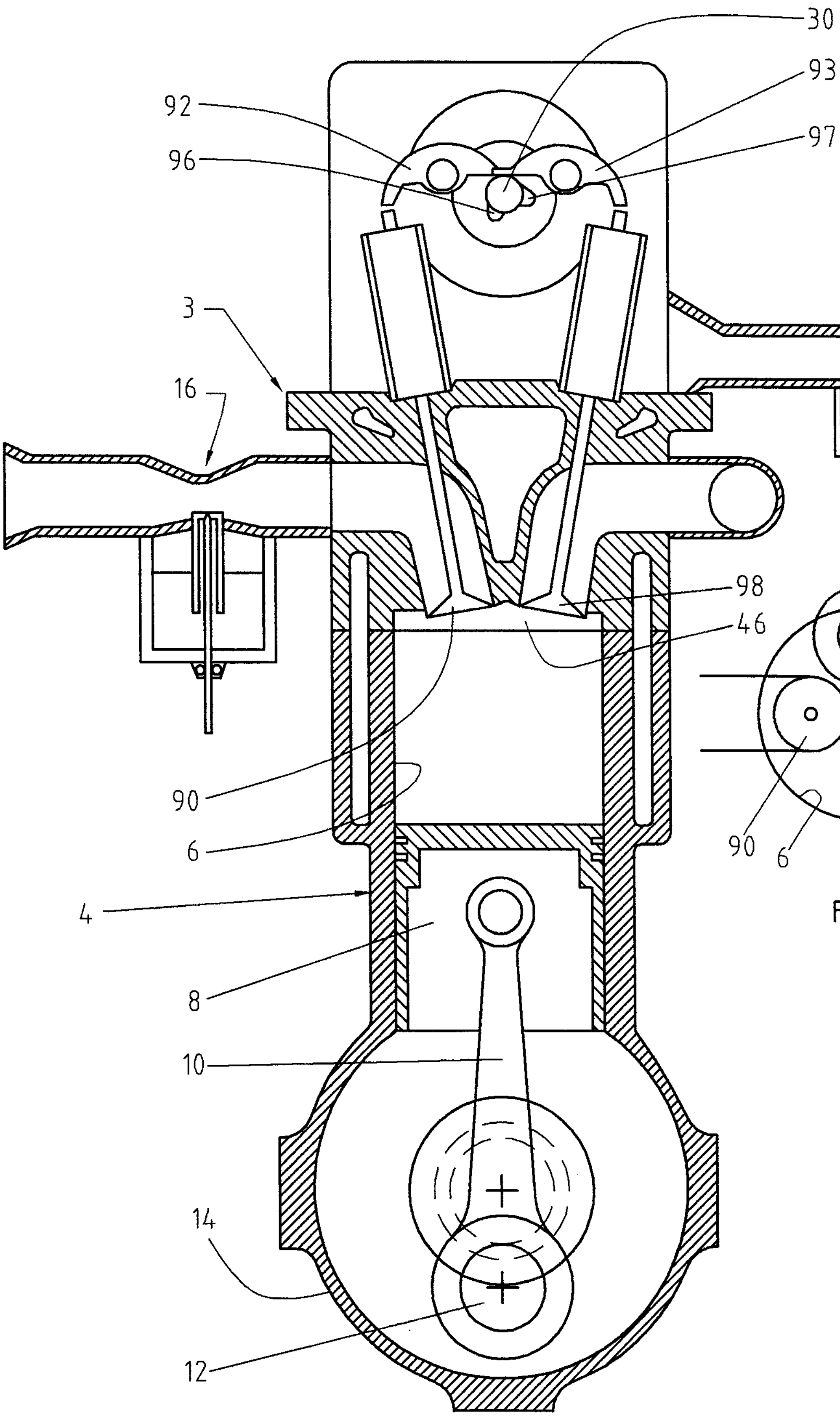

FIG 28

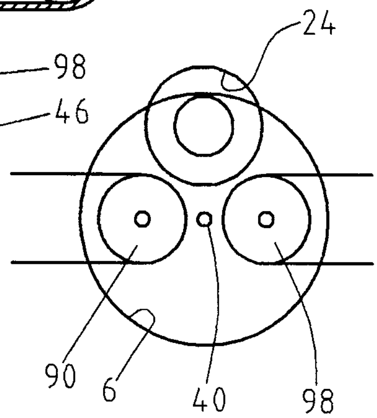

FIG 29

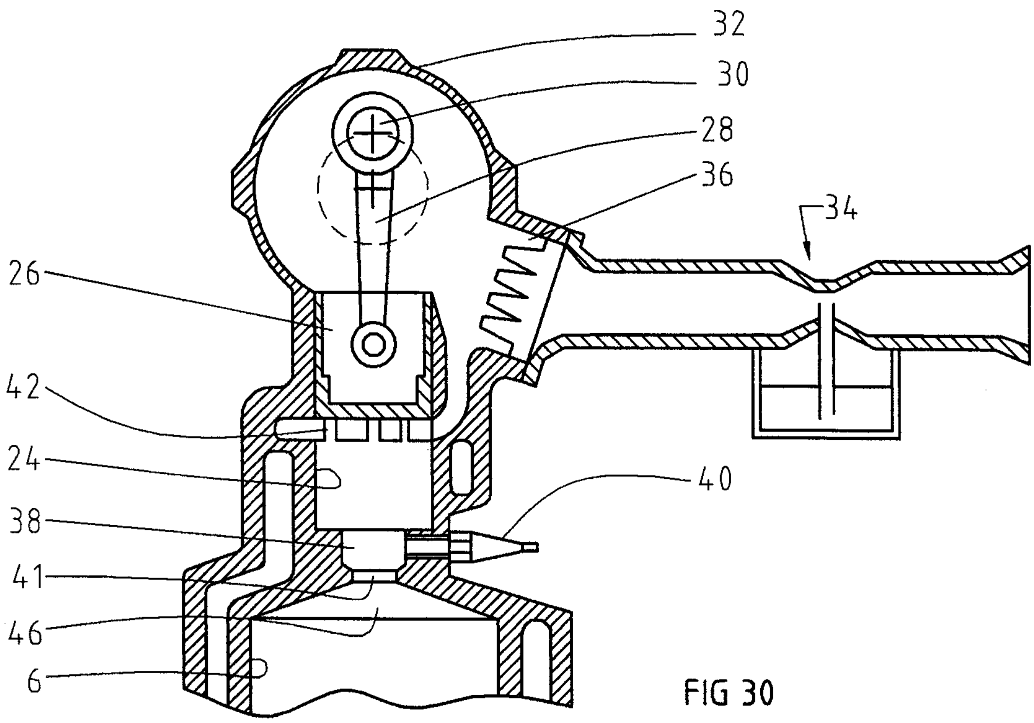

FIG 30

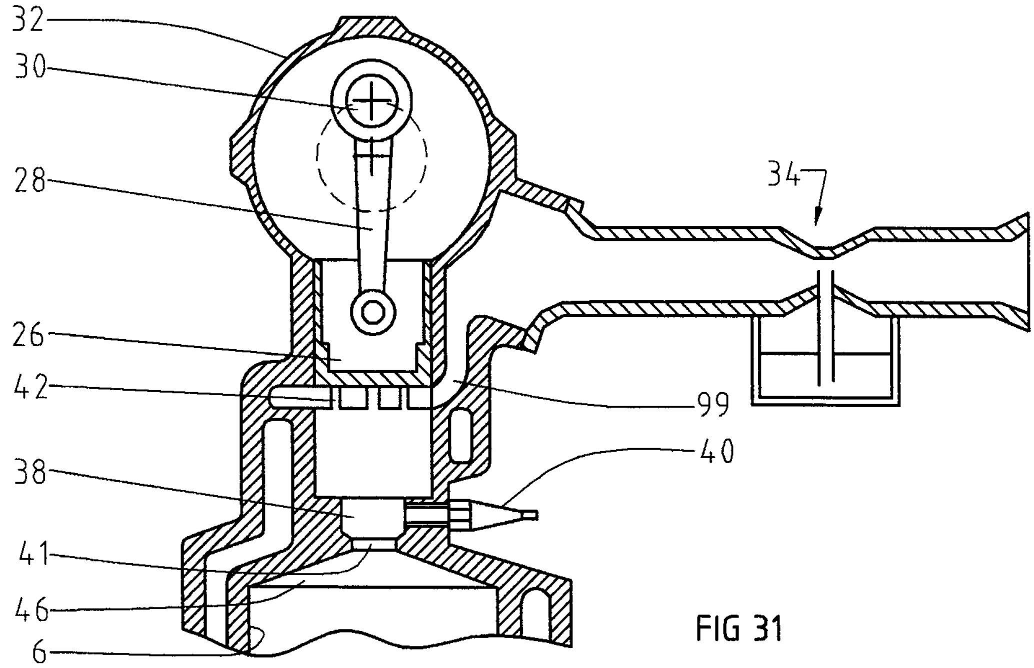

FIG 31

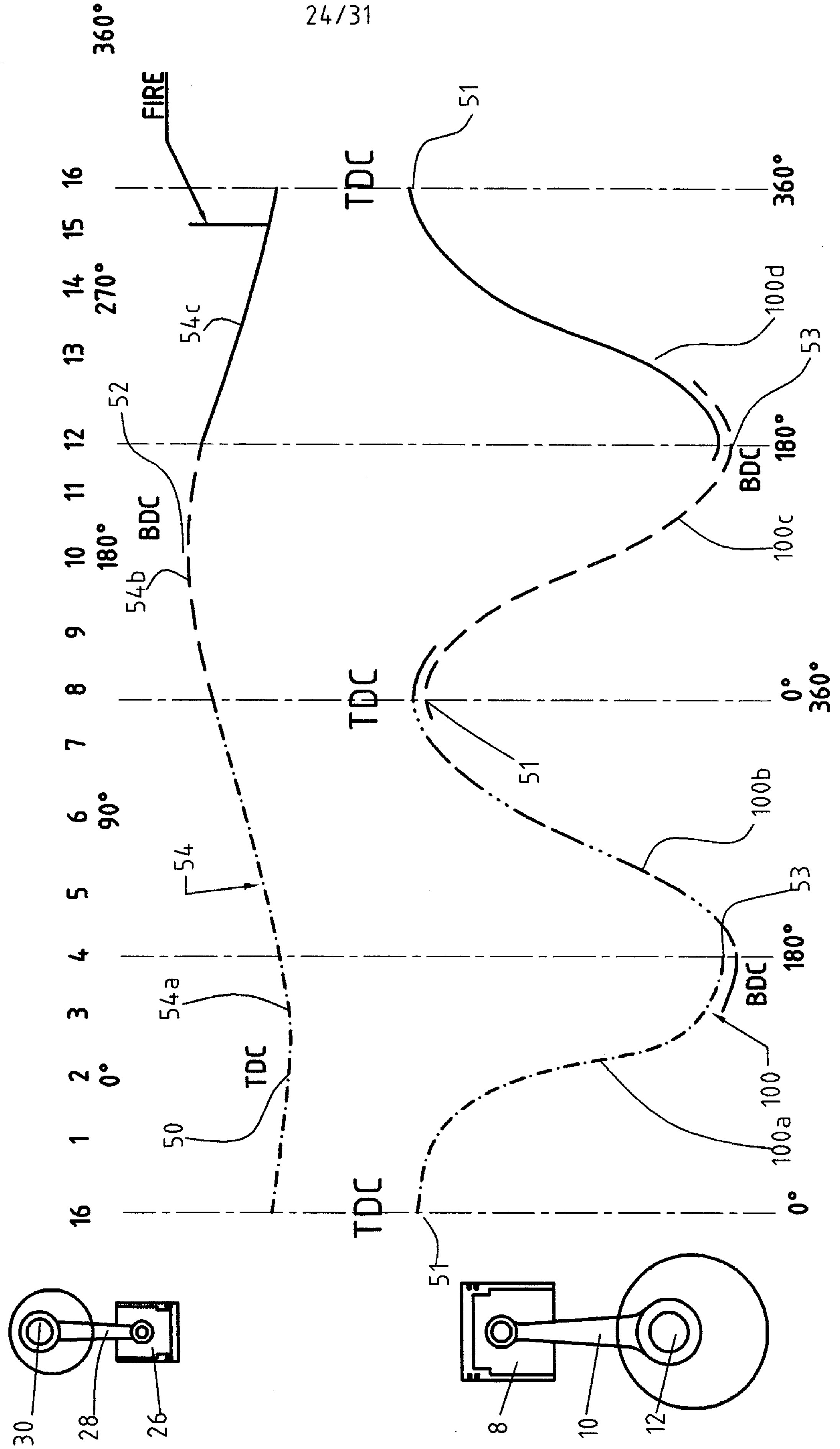
360°
FIRE
TDC
BDC
0°
90°
180°
270°
16
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
50
51
52
53
54
54a
54b
54c
100
100a
100b
100c
100d
8
10
12
26
28
30

FIG 32

25/31

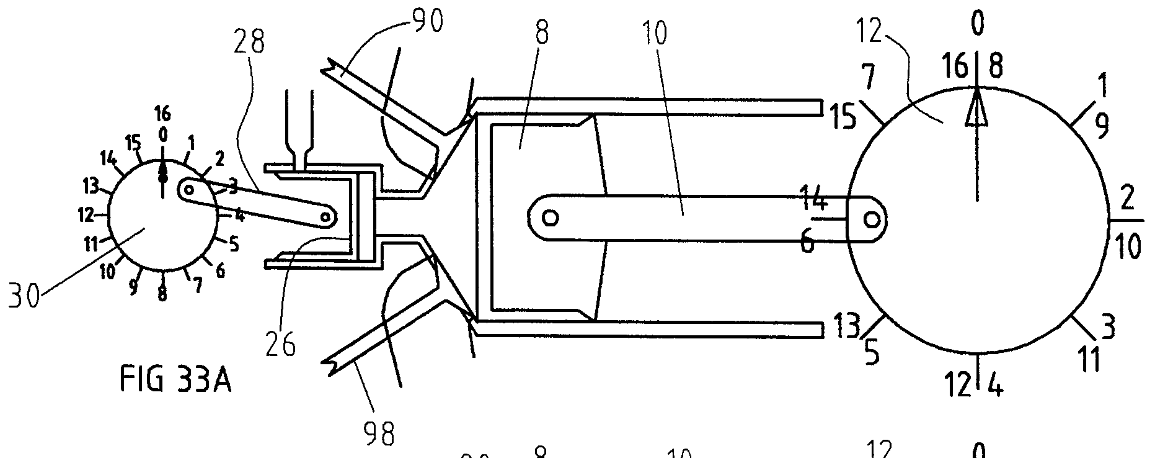

FIG 33A

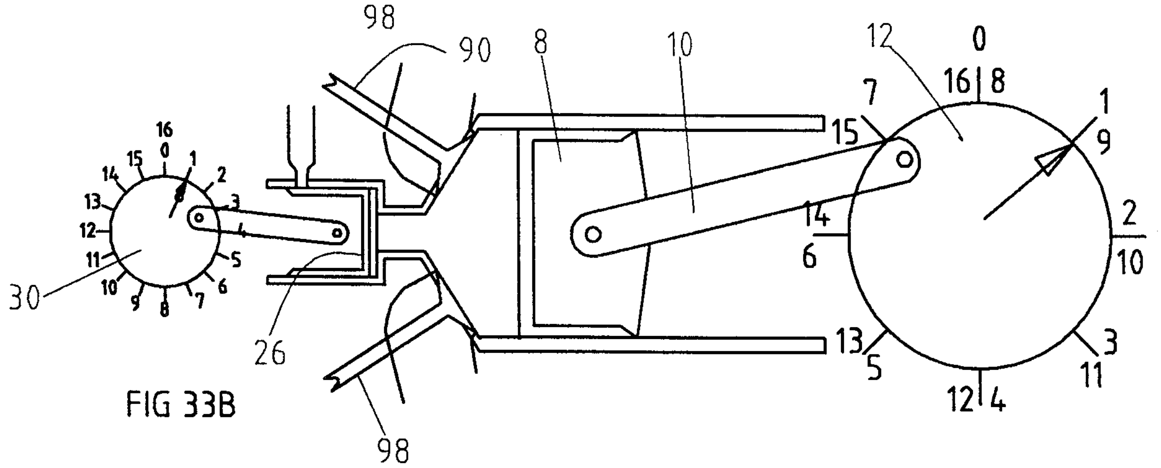

FIG 33B

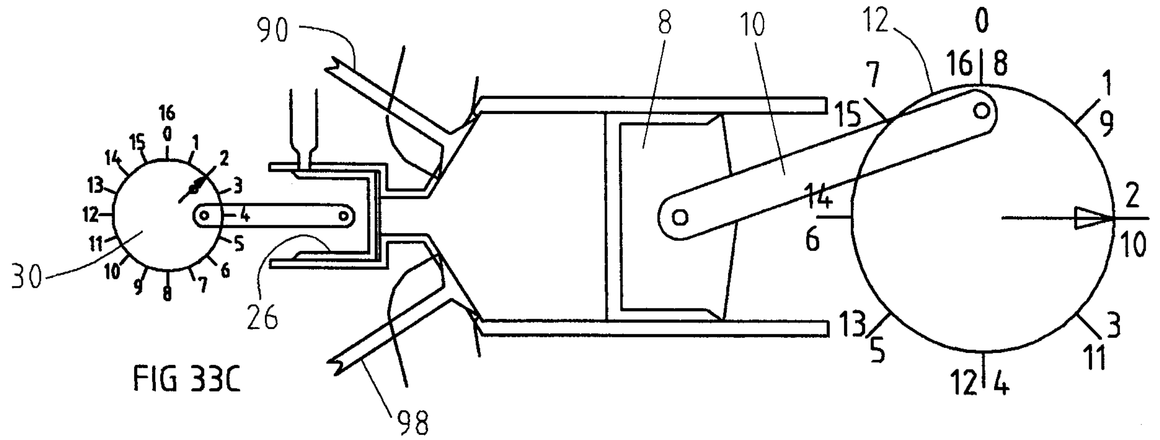

FIG 33C

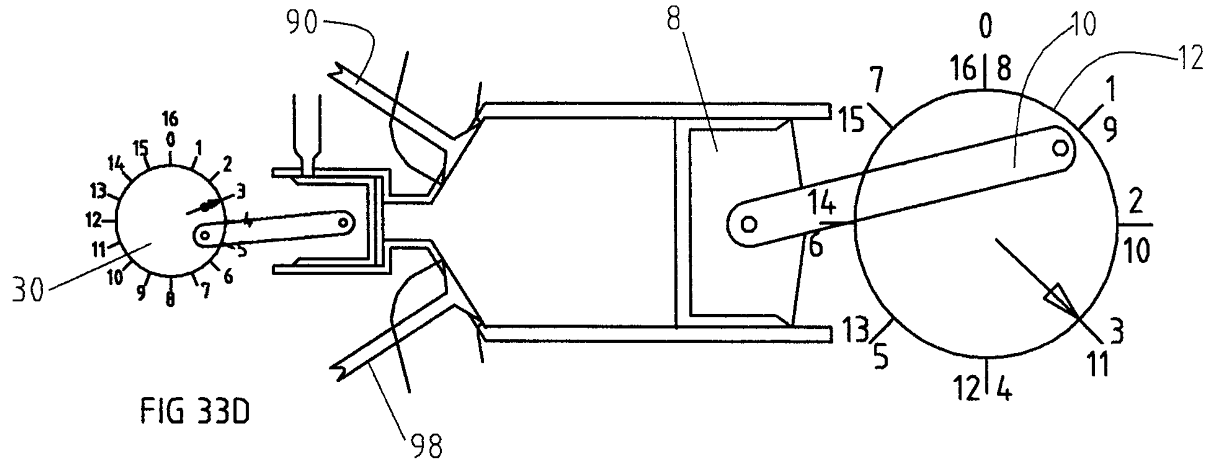

FIG 33D

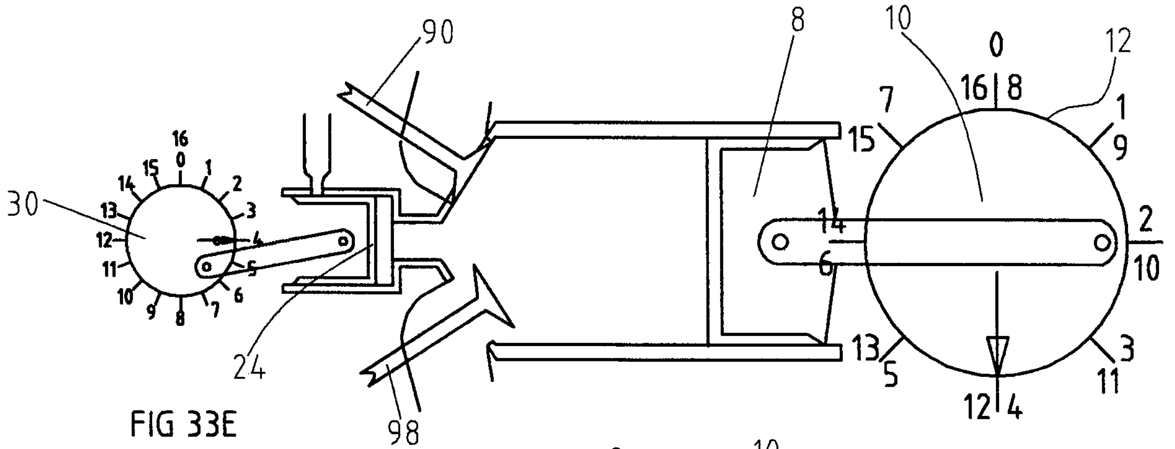

FIG 33E

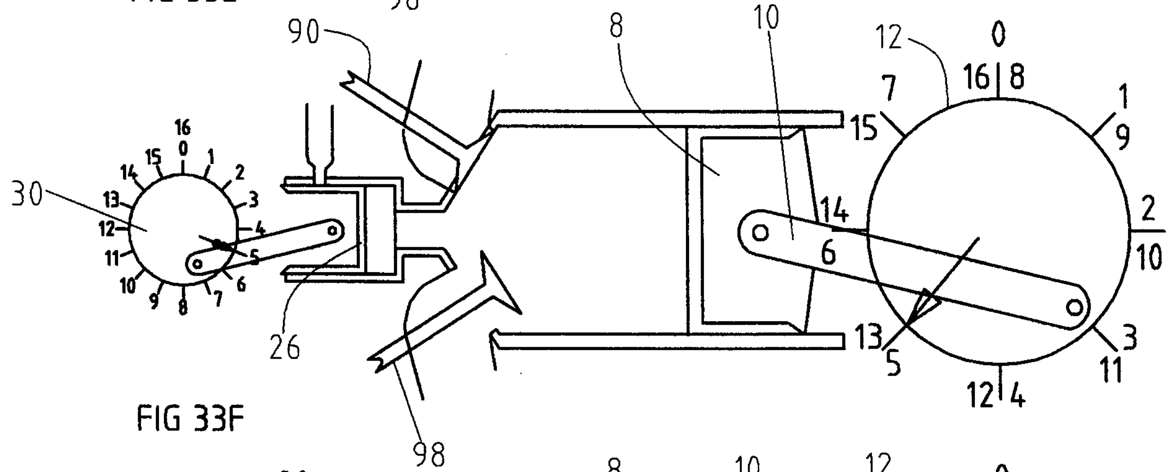

FIG 33F

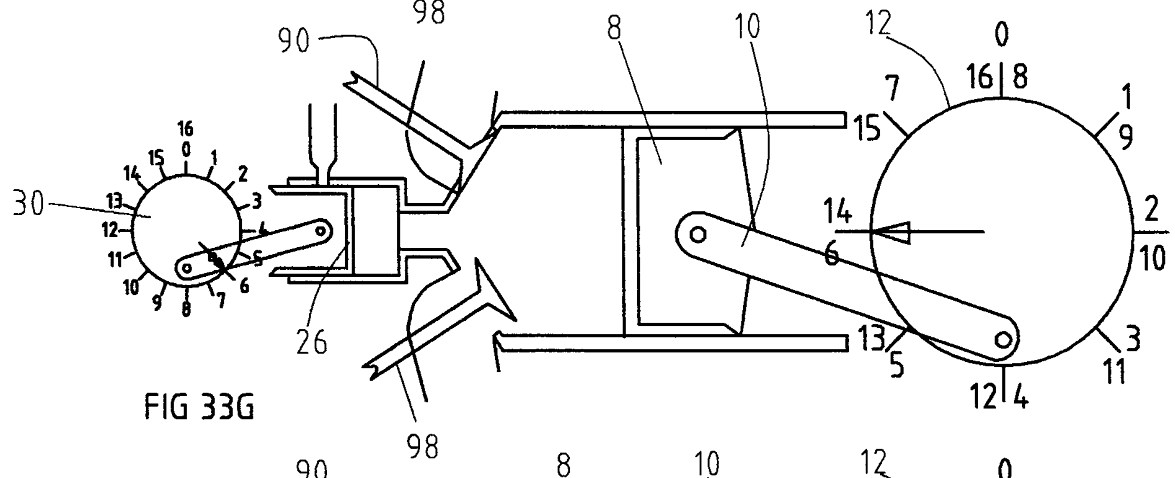

FIG 33G

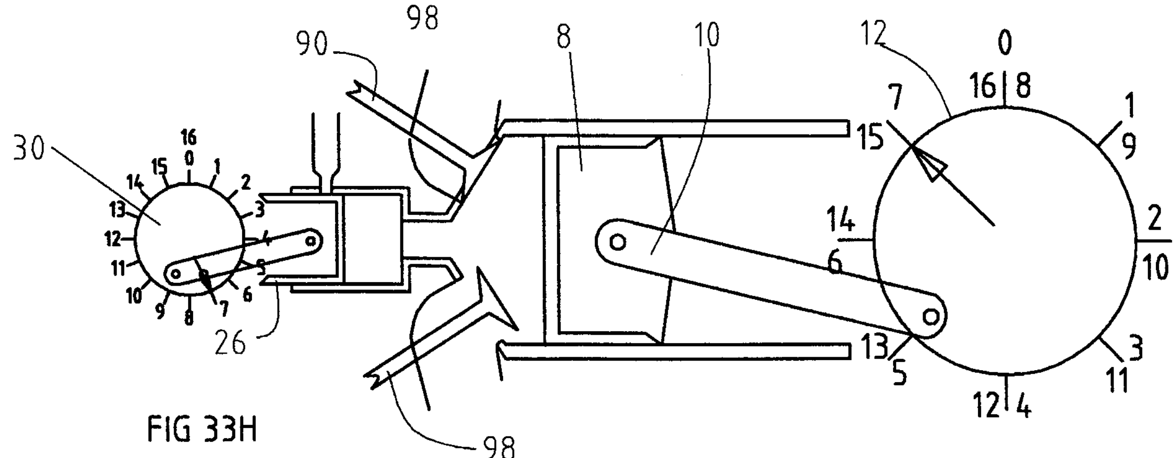

FIG 33H

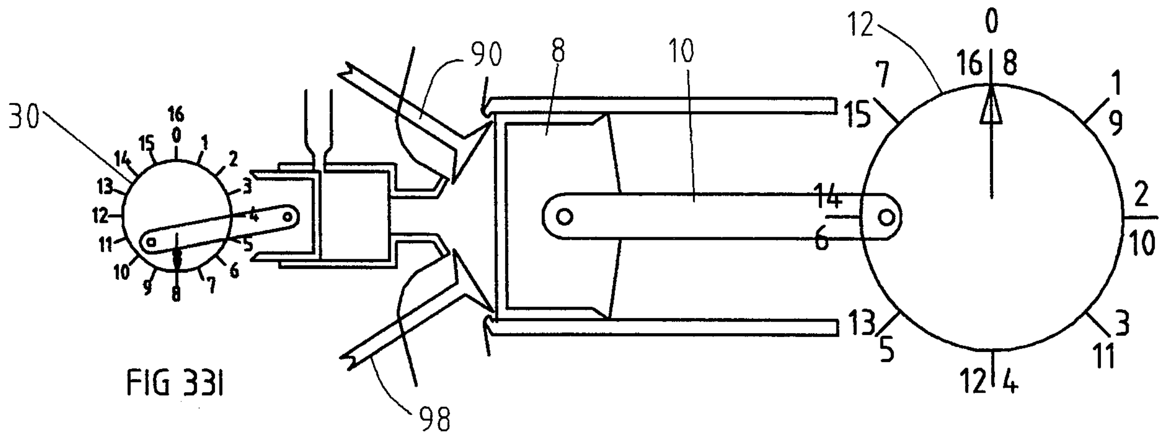

FIG 33I

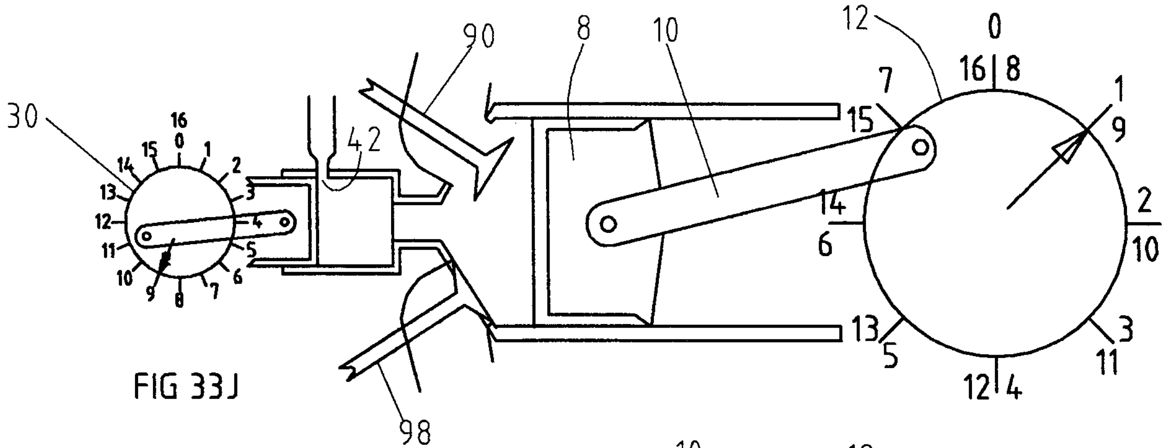

FIG 33J

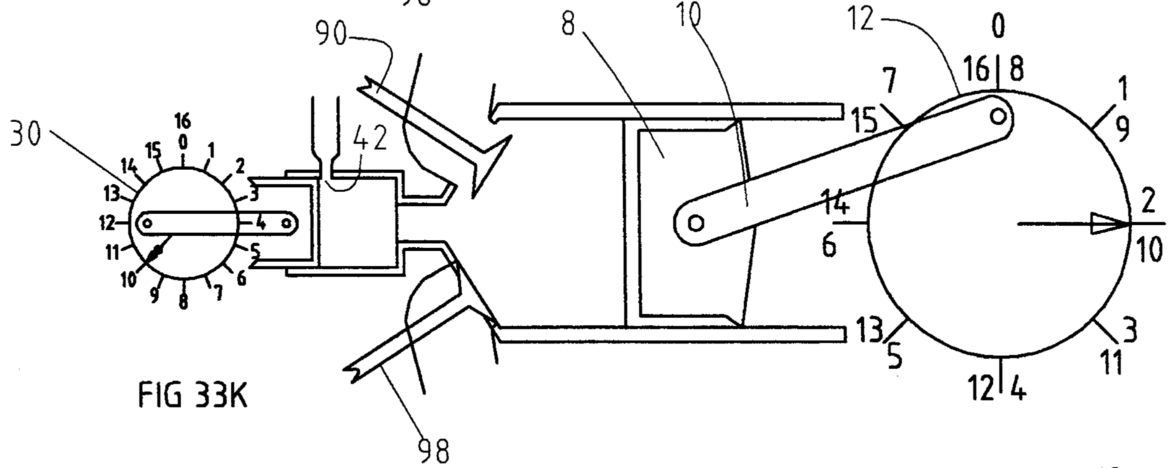

FIG 33K

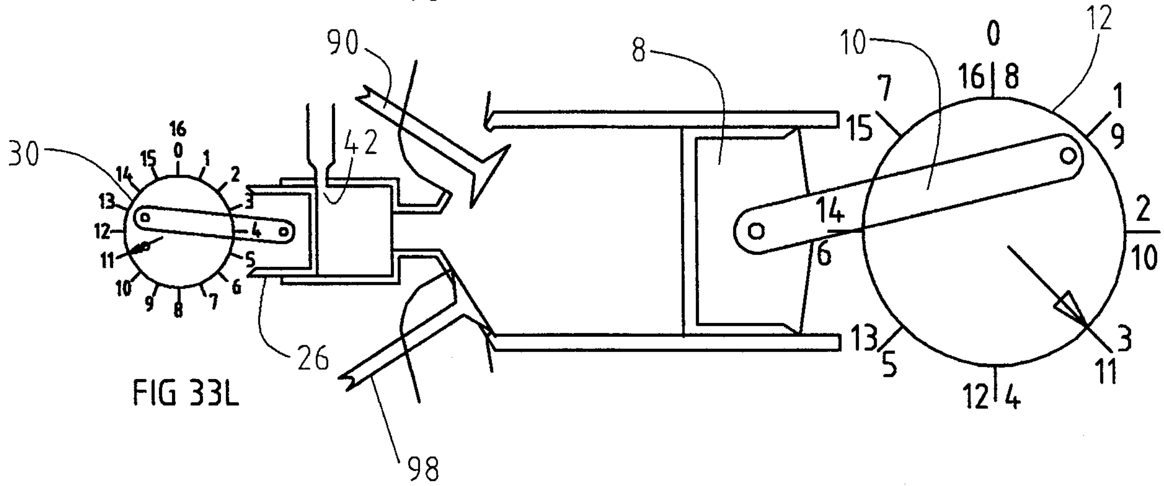

FIG 33L

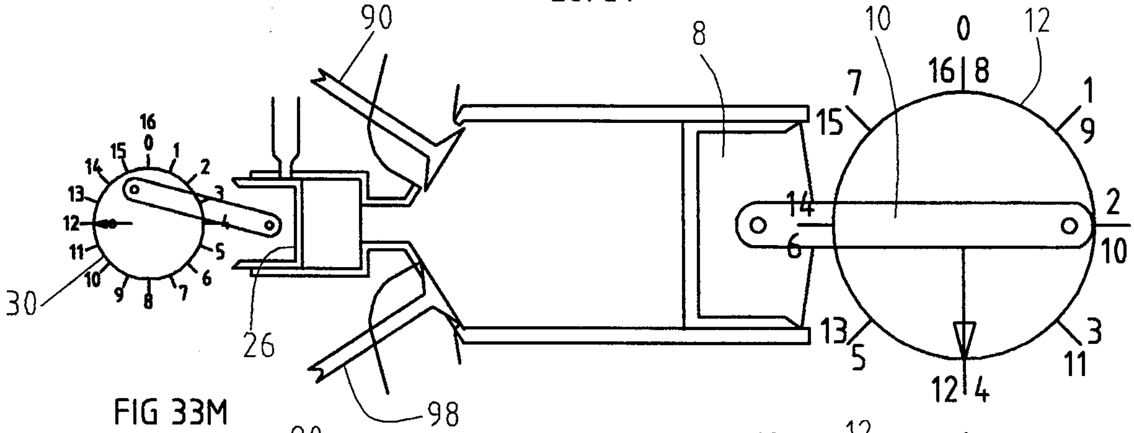

FIG 33M

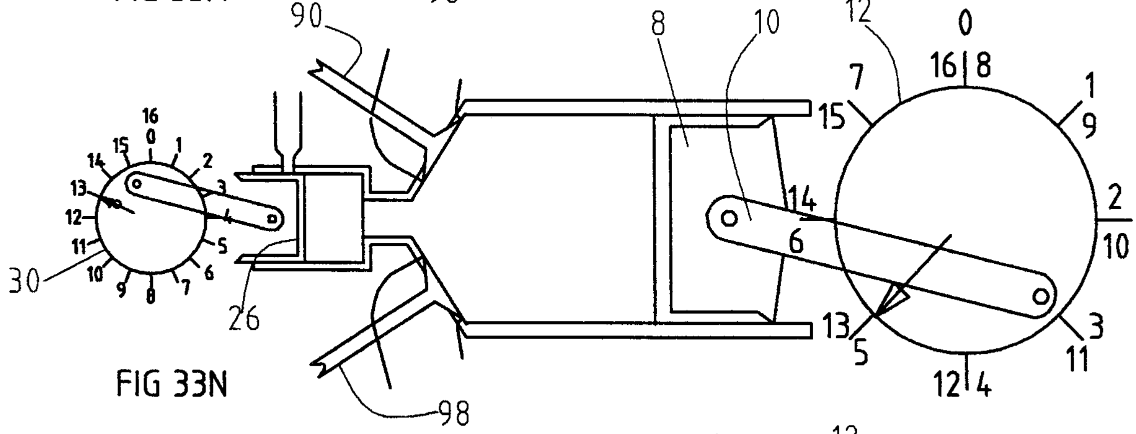

FIG 33N

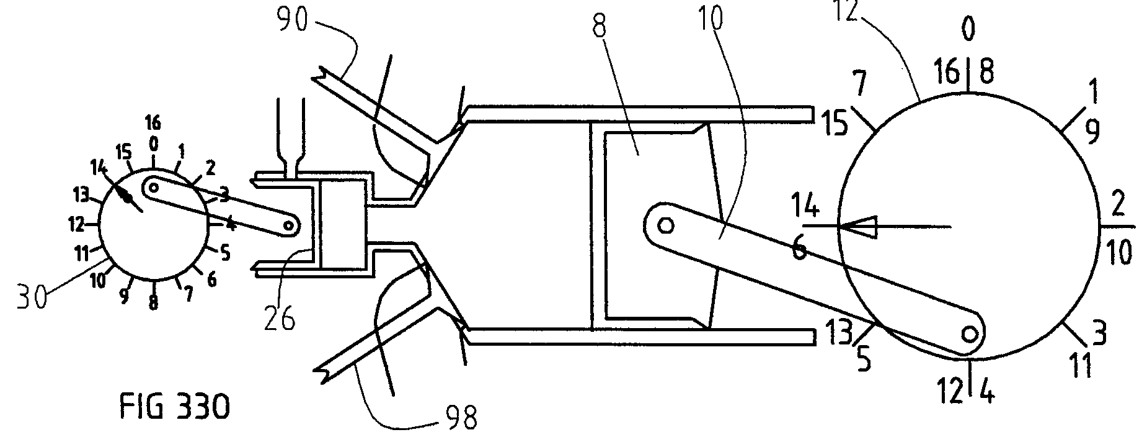

FIG 33O

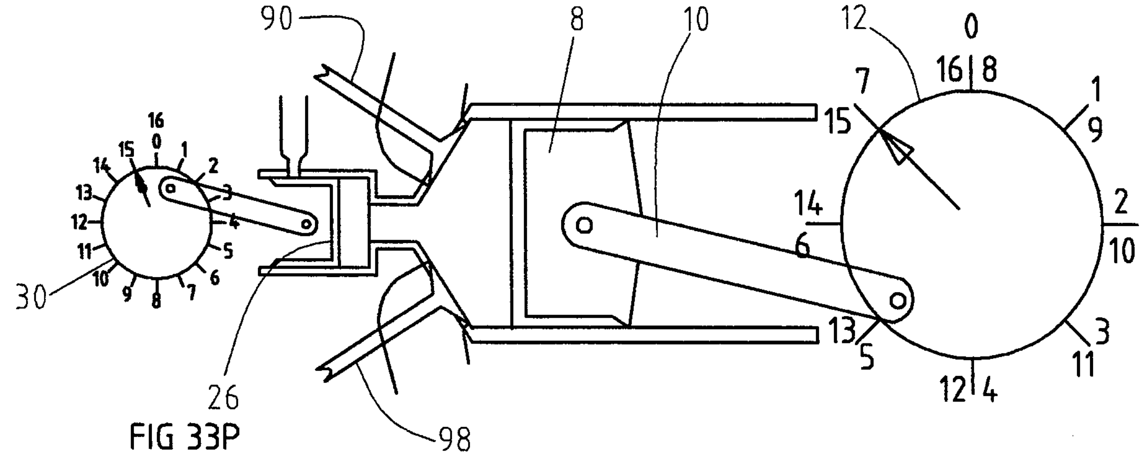

FIG 33P

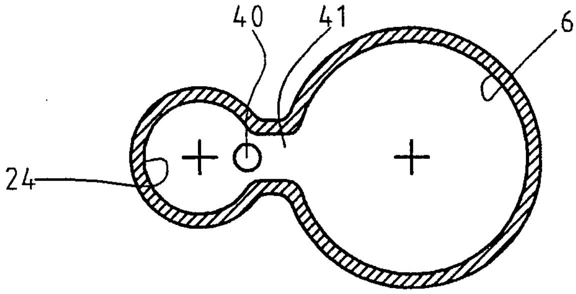

FIG 35

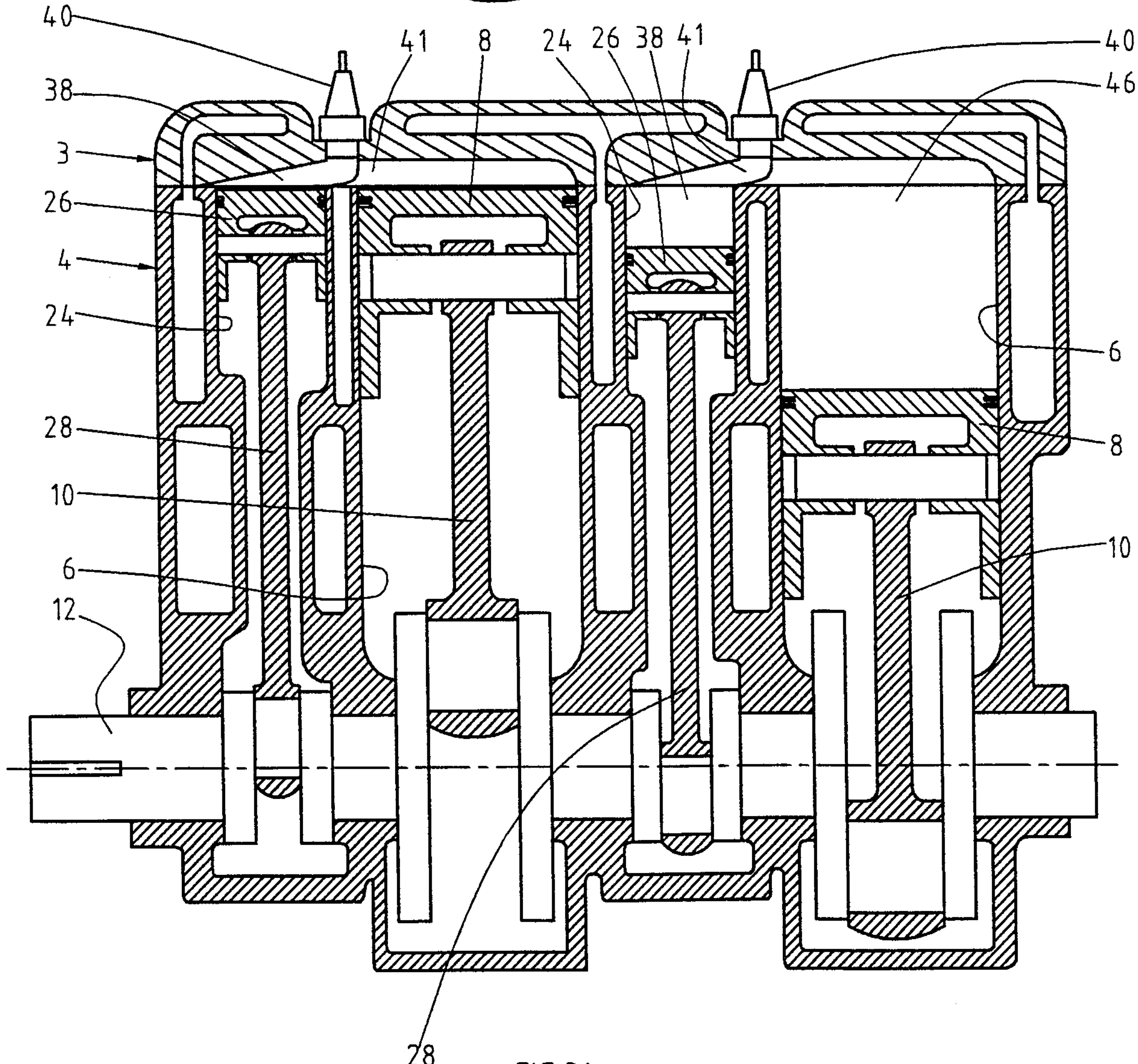

FIG 34

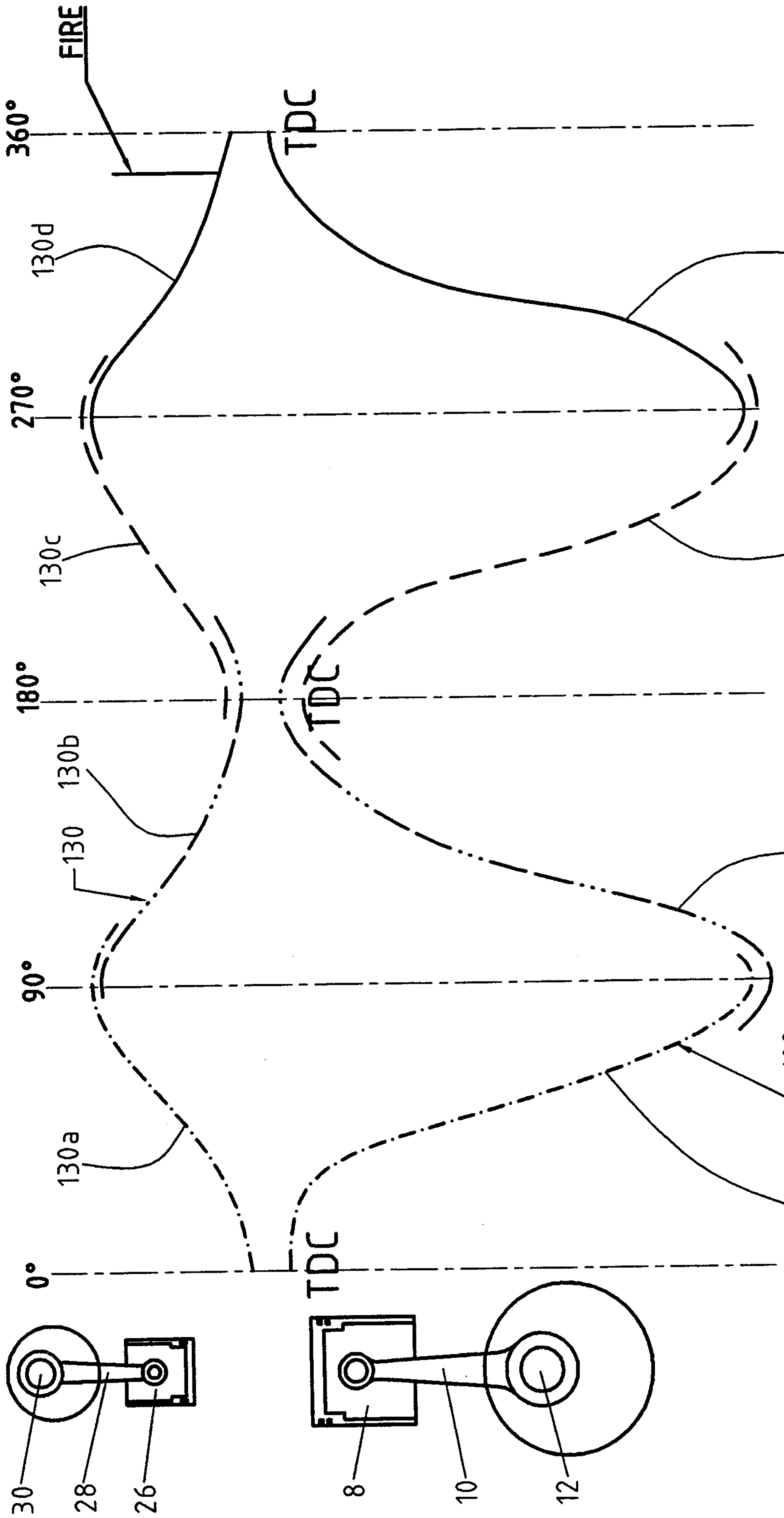
FIRE
360°
270°
180°
90°
0°
TDC
TDC
TDC
130d
100d
130c
100c
130b
130
100b
100
130a
100a
30
28
26
8
10
12

FIG 36

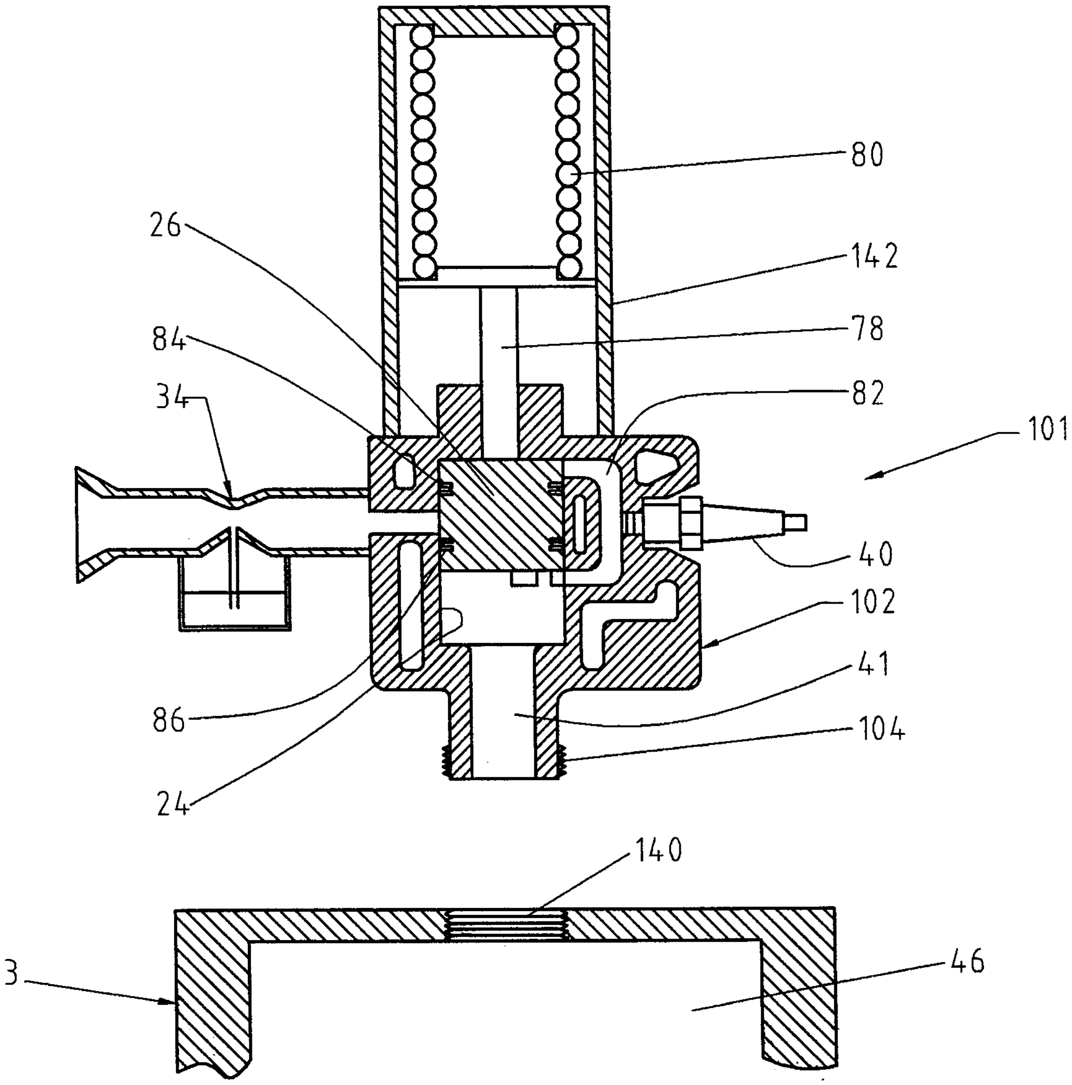

FIG 37

INTERNATIONAL SEARCH REPORT

International Application No.
PCT/AU 98/00613

A. CLASSIFICATION OF SUBJECT MATTER

Int Cl^{6}: F02B 19/06

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)
F02B 19/00, 19/06

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched
AU: F02B 19/06

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)
WPAT :]
:] F02B 19/00, 19/06
JAPIO :]

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	DT 2456062 A1 (PORSCHE) 12 August 1976 entire document	1-8,11-28,32-34,36
X	DE 2830274 A1 (VOLKSWAGENWERK) 24 January 1980 entire document	1-8,11-28,32,34,36
X Y	US 4981114 A (SKOPIL) 1 January 1991 entire document	1-8,11-28,32,34,36 37

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

[X] See patent family 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

Date of the actual completion of the international search	Date of mailing of the international search report
23 September 1998	-2 OCT 1998

Name and mailing address of the ISA/AU	Authorized officer
AUSTRALIAN PATENT OFFICE PO BOX 200 WODEN ACT 2606 AUSTRALIA Facsimile No.: (02) 6285 3929	**JAGDISH BOKIL** Telephone No.: (02) 6283 2158

INTERNATIONAL SEARCH REPORT

International Application No.
PCT/AU 98/00613

C (Continuation) DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	WO 94/25743 (NEGRE) 10 November 1994 entire document	1-8,11-28,32,34,36
X	US 2415506 A (MALLORY) 7 February 1945 entire document	1,2,4-8,11-13,16-26,32,36
X Y	EP 30832 A1 (MAY) 24 June 1981 entire document	1,2,4-8,11-28,32,34,36 37
Y	*GB 2089889 A (FORD MOTOR) 30 June 1982 page 1, lines 83-85	37
	* Please note that this document can be "mosaiced" with either US 4981114 or EP 30832.	

INTERNATIONAL SEARCH REPORT

Information on patent family members

International Application No.
PCT/AU 98/00613

This Annex lists the known "A" publication level patent family members relating to the patent documents cited in the above-mentioned international search report. The Australian Patent Office is in no way liable for these particulars which are merely given for the purpose of information.

Patent Document Cited in Search Report		Patent Family Member					
DT	2456062	NONE					
DE	2830274	NONE					
US	4981114	NONE					
WO	94/25743	AU	66608/94	EP	648314	FR	2704903
		US	5592904				
EP	030832	NONE					
GB	2089889	CA	1185853	DE	3150675	JP	57110716
		US	4384553				

END OF ANNEX