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(51) INT CL^6
F01L 7/02

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(56) Documents Cited
GB 0922073 A GB 0786105 A GB 0289565 A
GB 0285586 A GB 0173877 A US 5154147 A
US 4788945 A US 3945364 A

(58) Field of Search
INT CL^5 F01L 7/02 7/14

(54) **Four-stroke engine rotary valve gear**

(57) The valve casing 2 comprises three ports 12, 14 and 16 that are radially disposed relative to the axis of valve rotor rotation. A pair of rotors 1 (Fig. 16) may control communication between the inlet and exhaust (15, 17) and the cylinder (13) with the piston crown shaped to be adjacent the rotors at top dead centre. The rotor recess may provide communication between the inlet and exhaust at top dead centre on the power stroke.

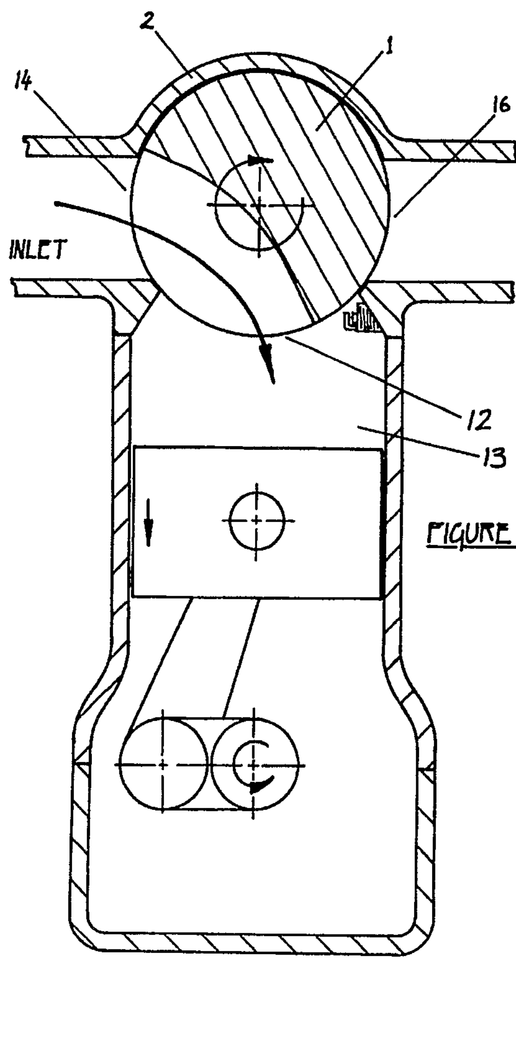

FIGURE 4

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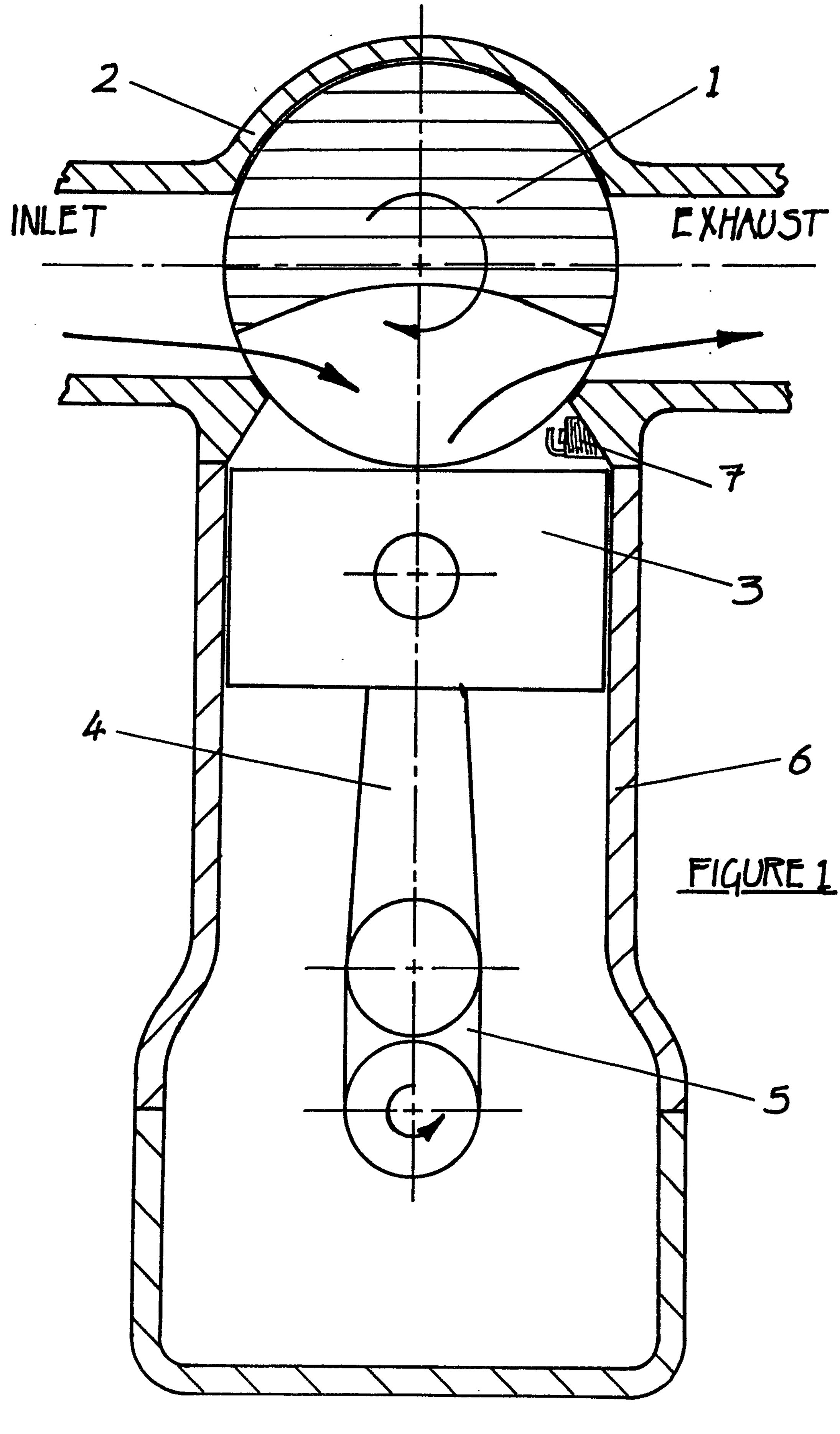
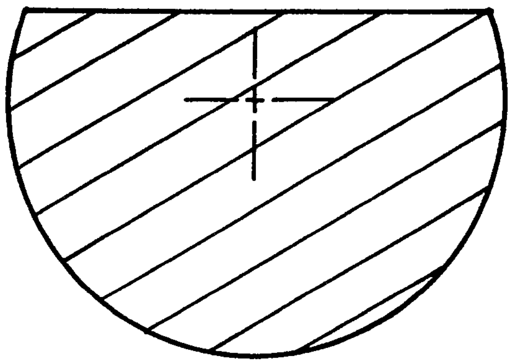


FIGURE 1



CROSS-SECTION

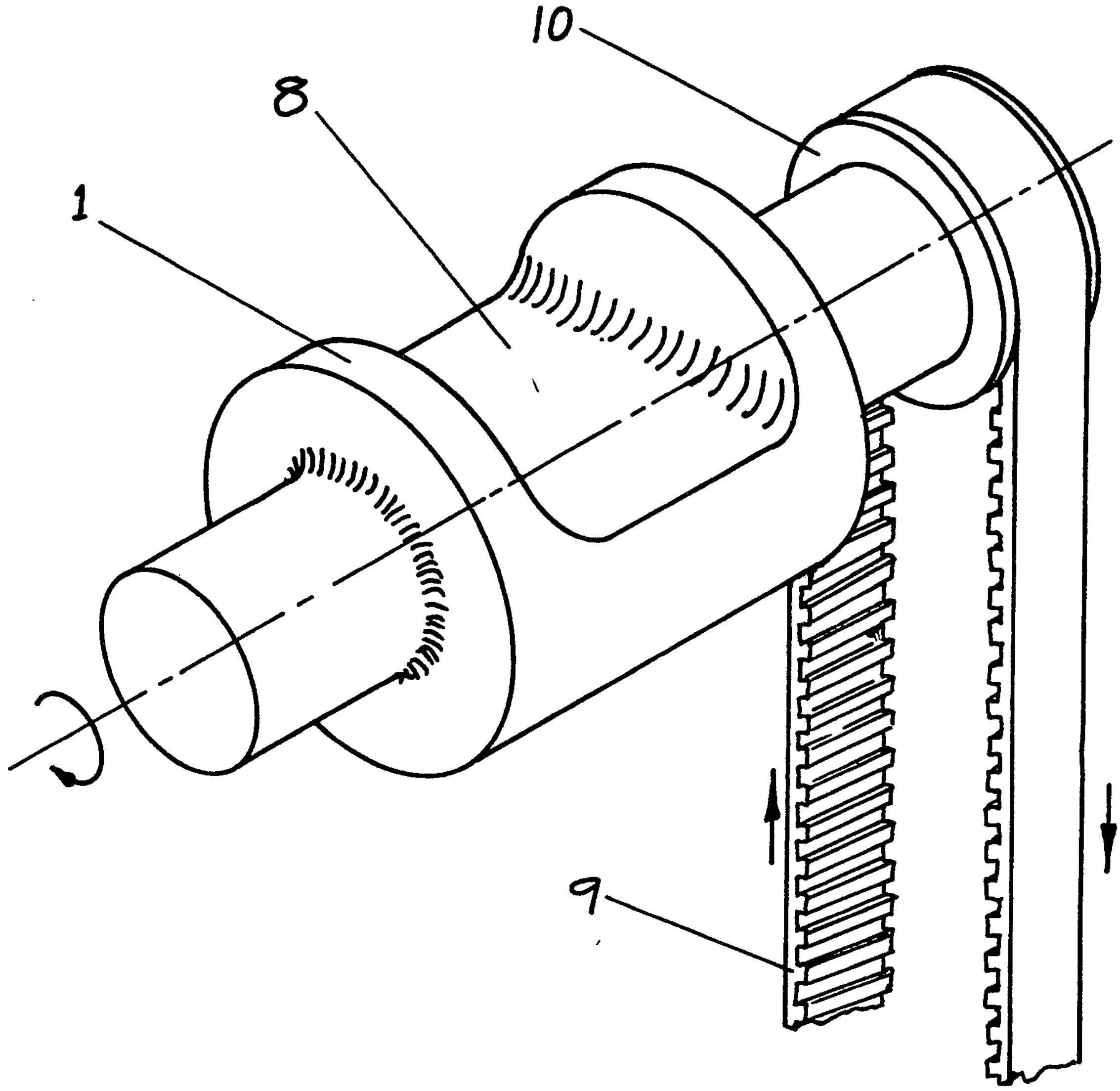


FIGURE 2

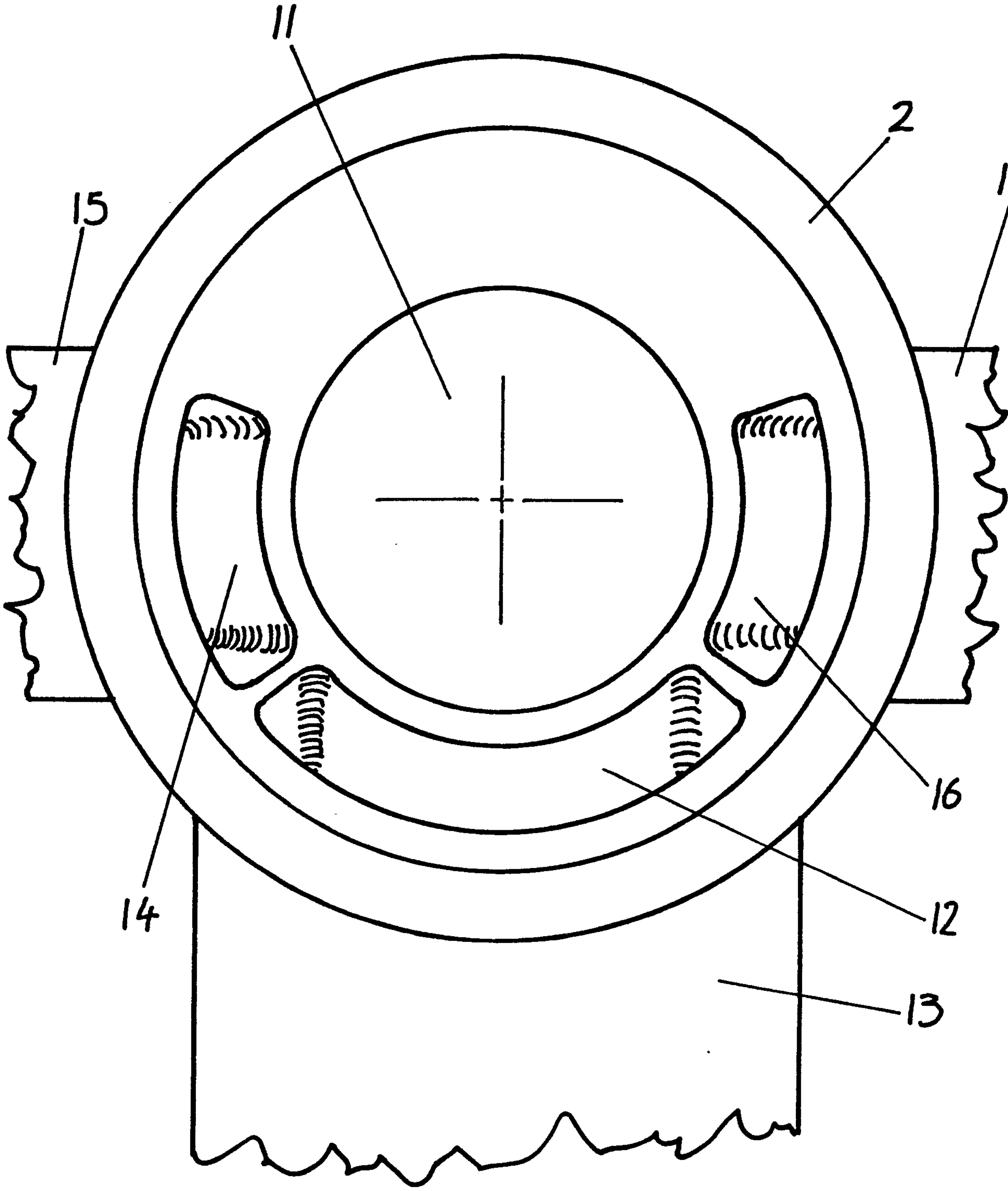


FIGURE 3

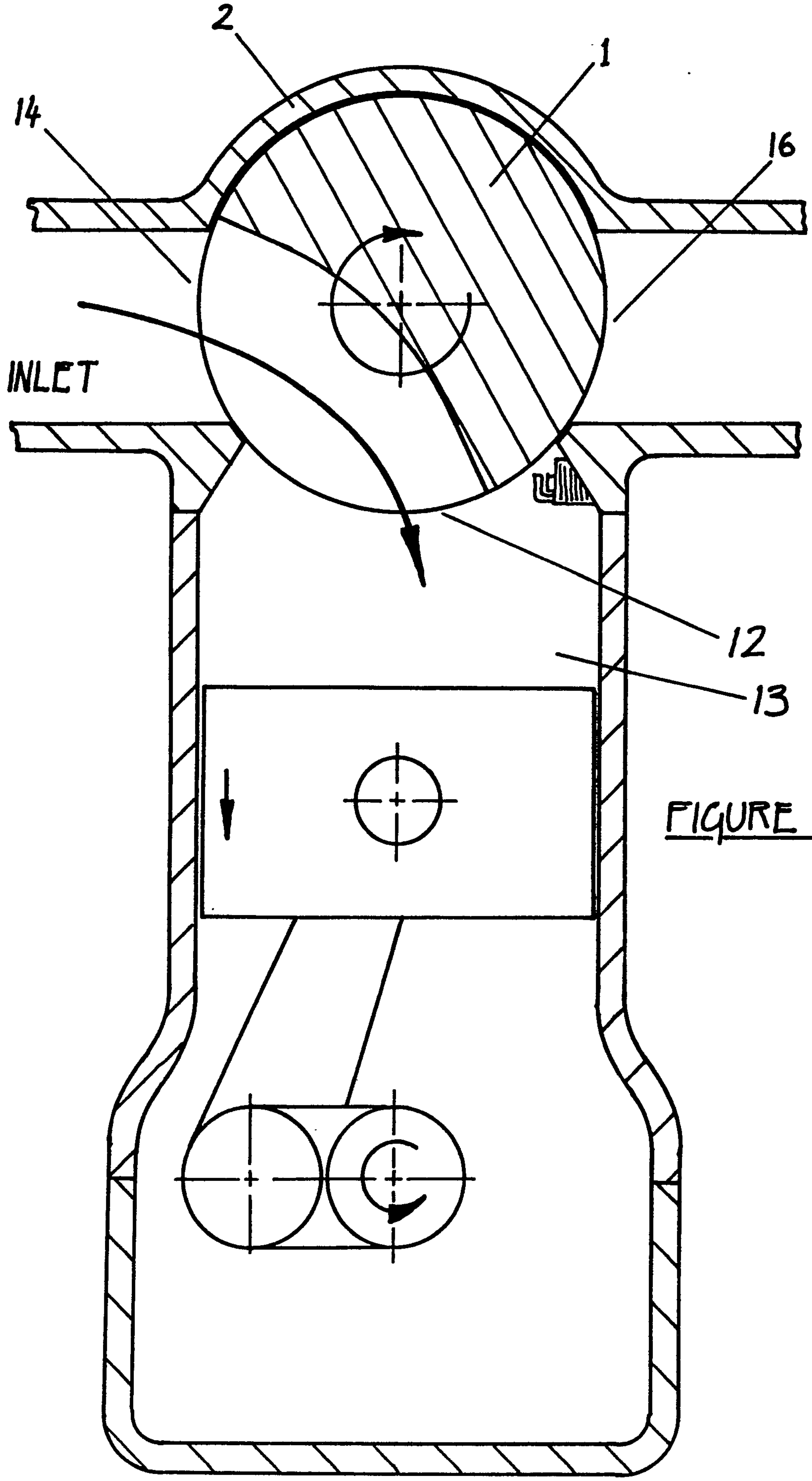


FIGURE 4

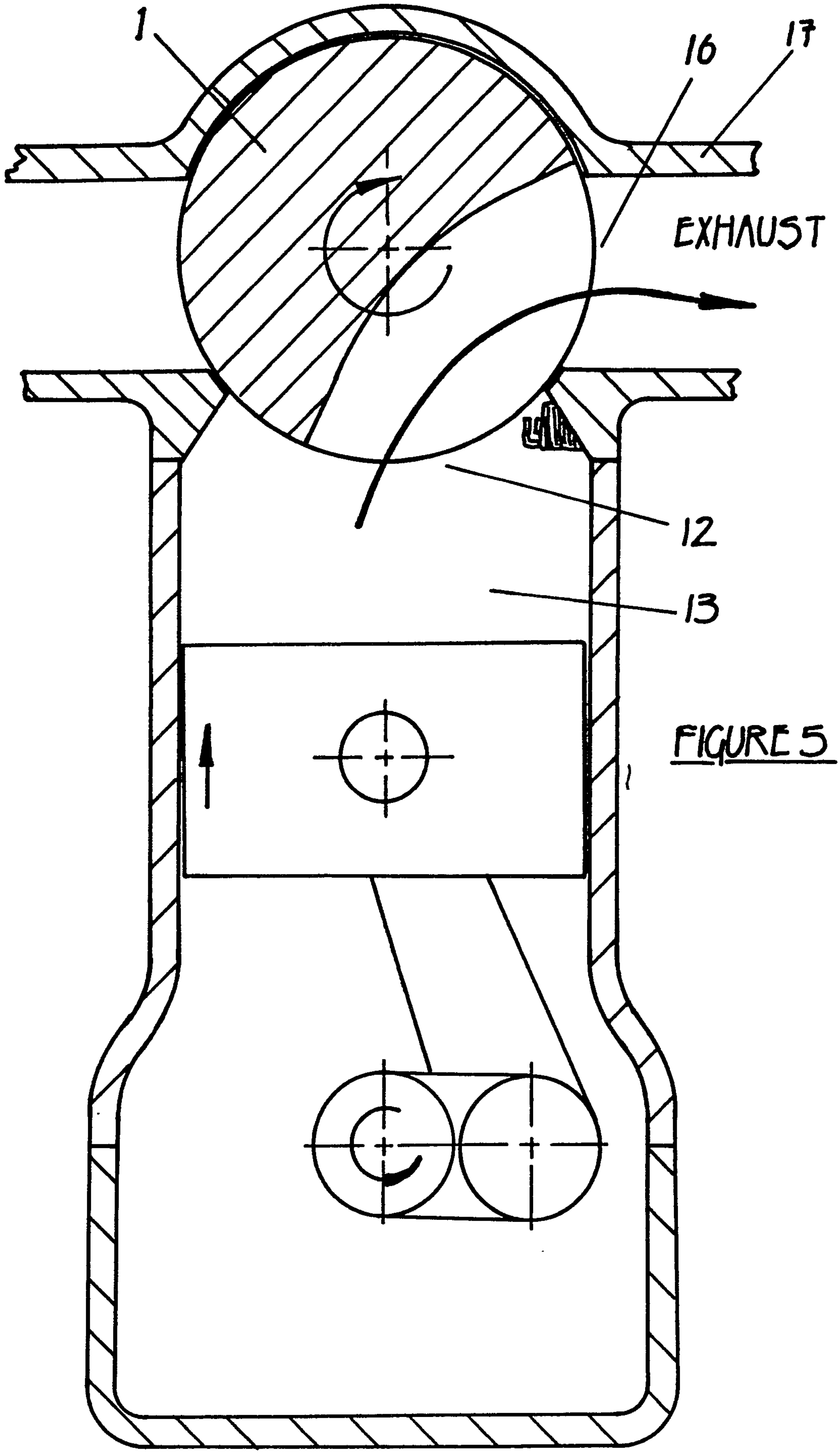


FIGURE 5

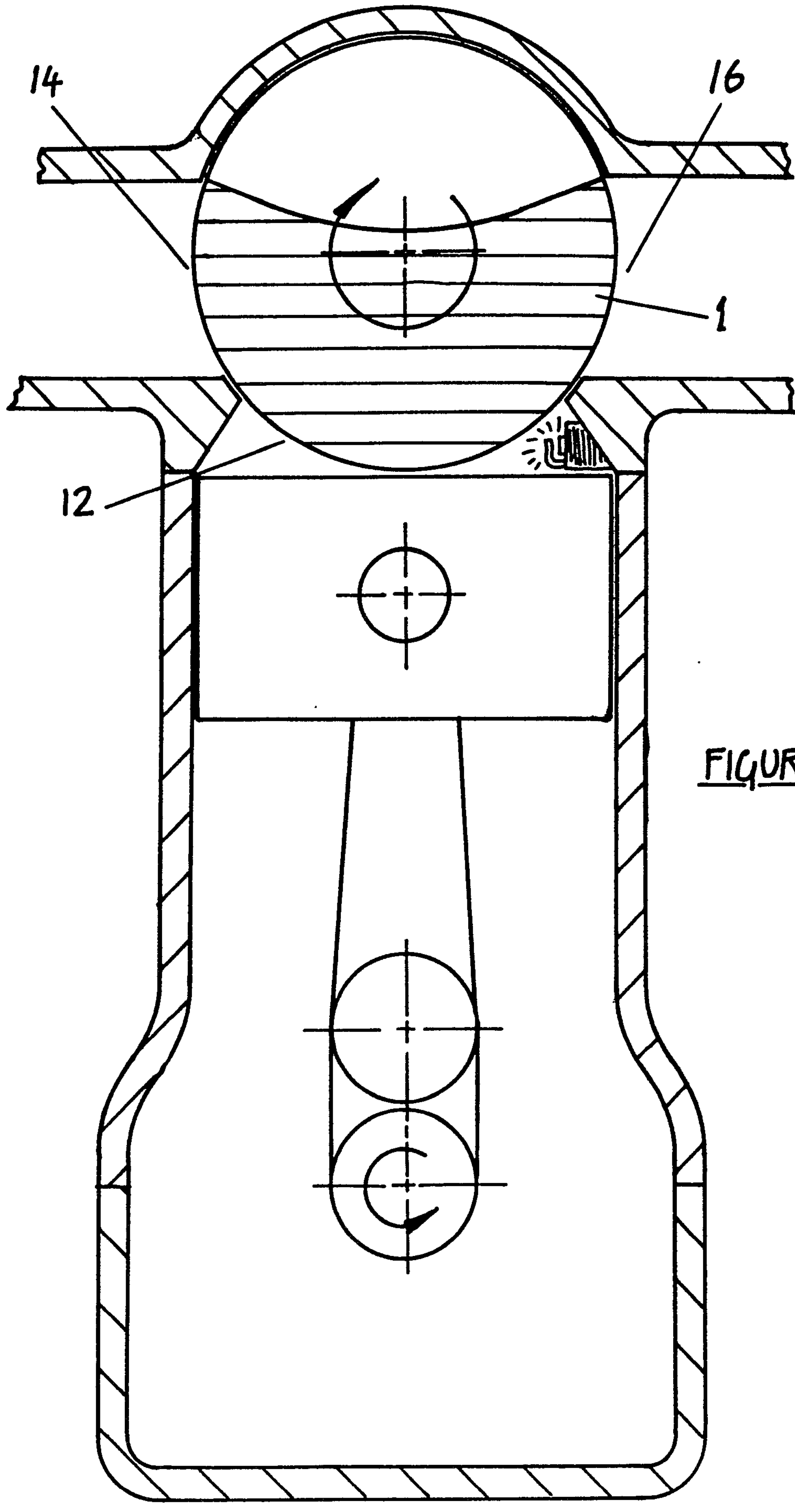


FIGURE 6

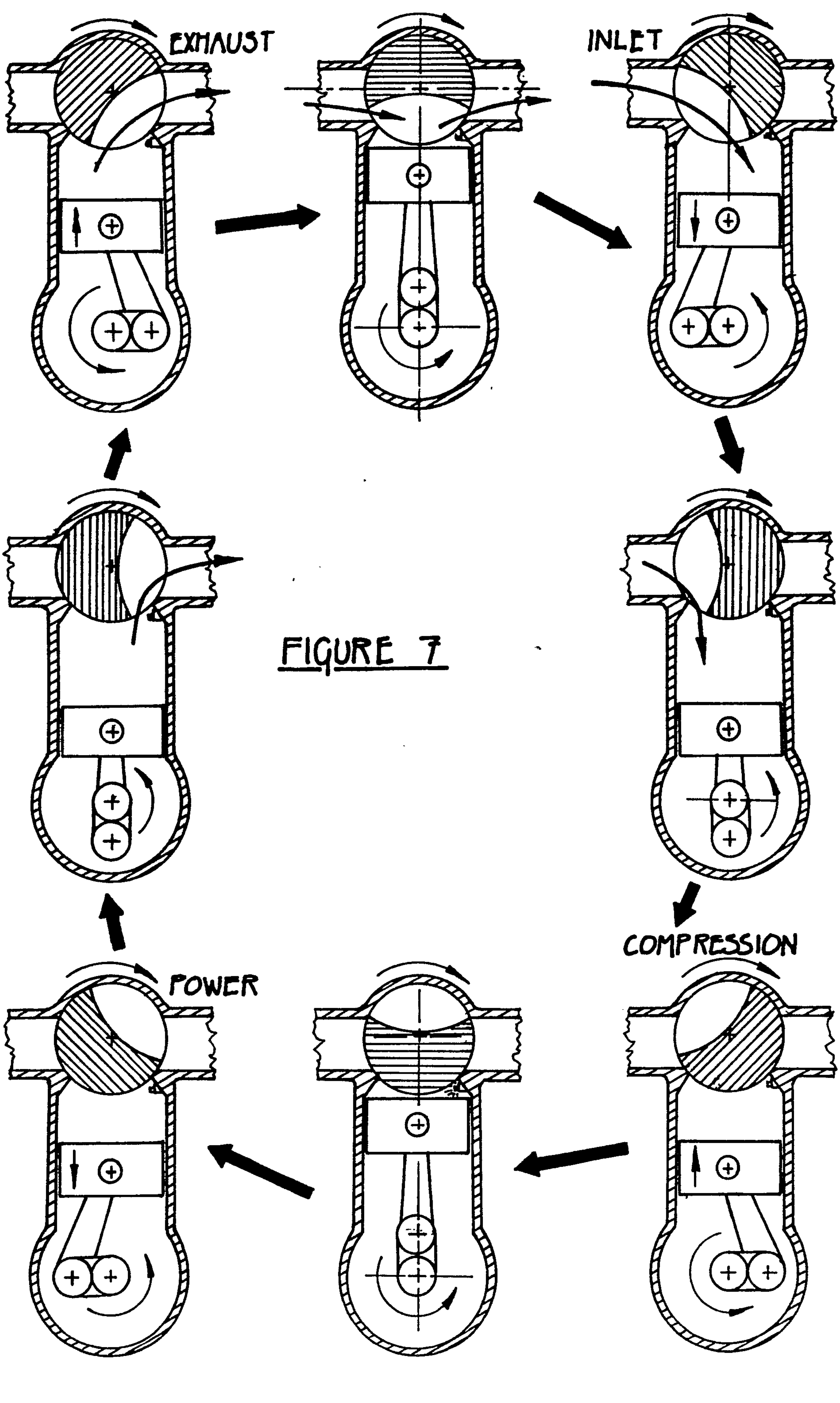


FIGURE 7

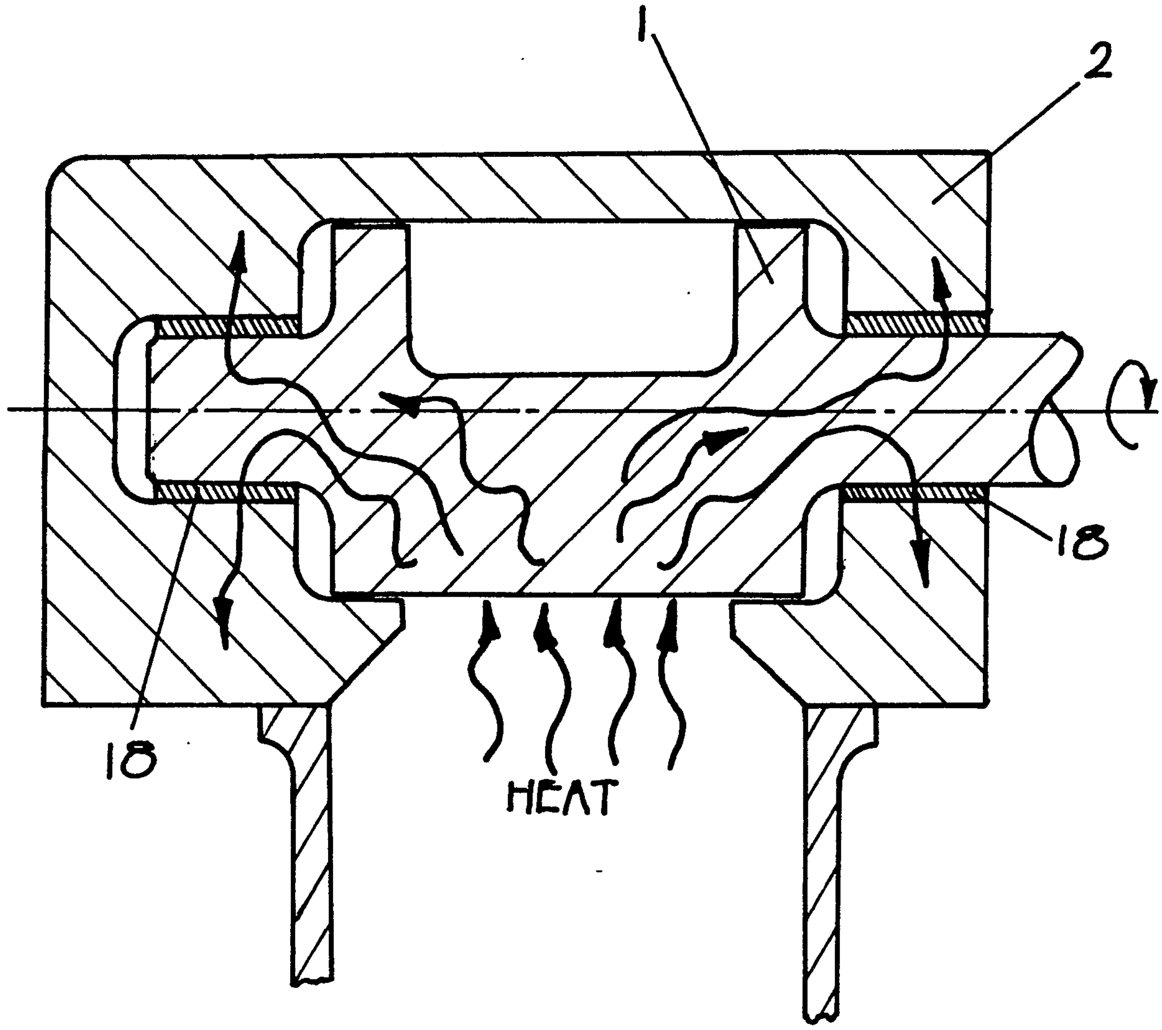


FIGURE 8

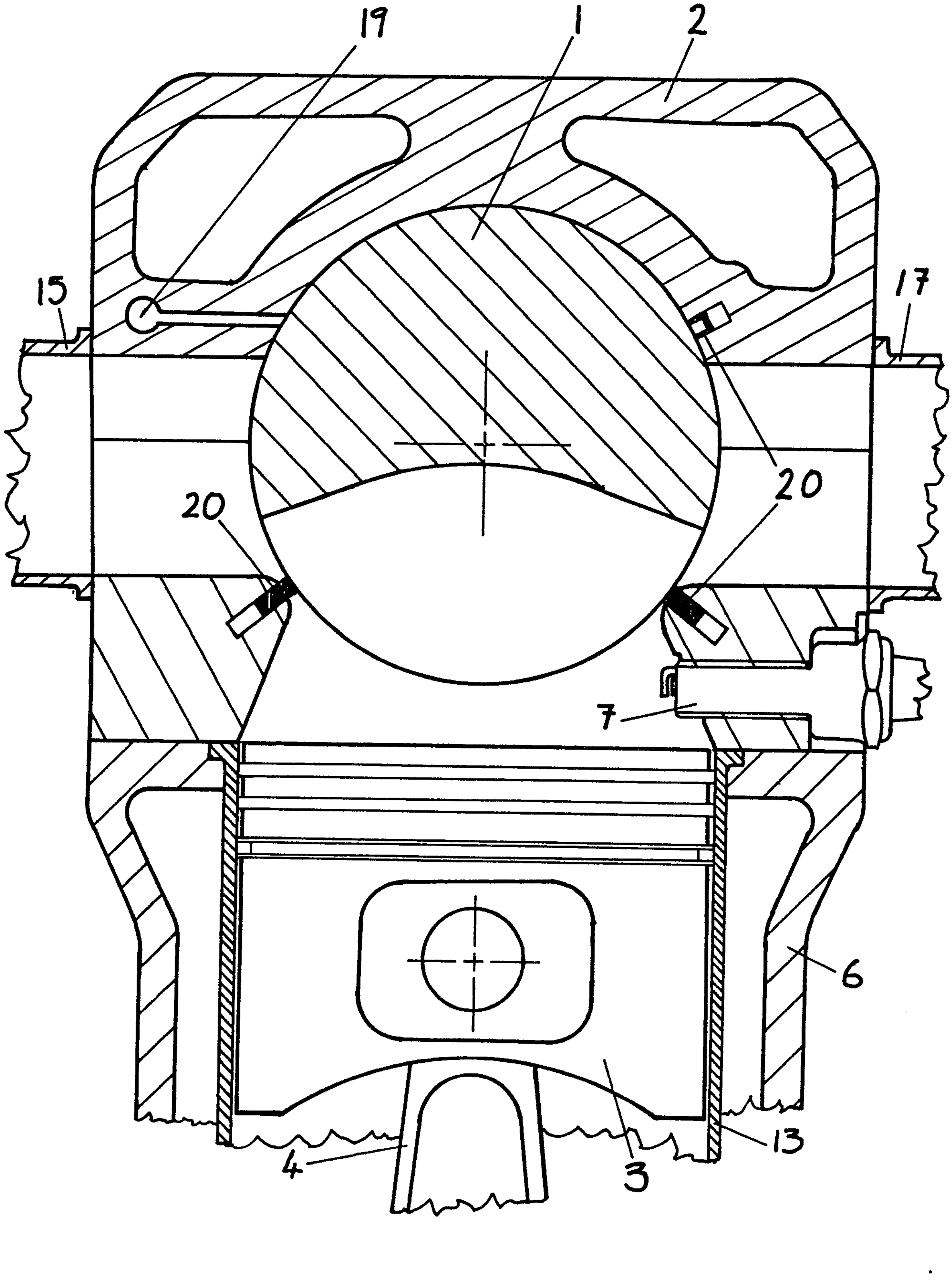


FIGURE 9

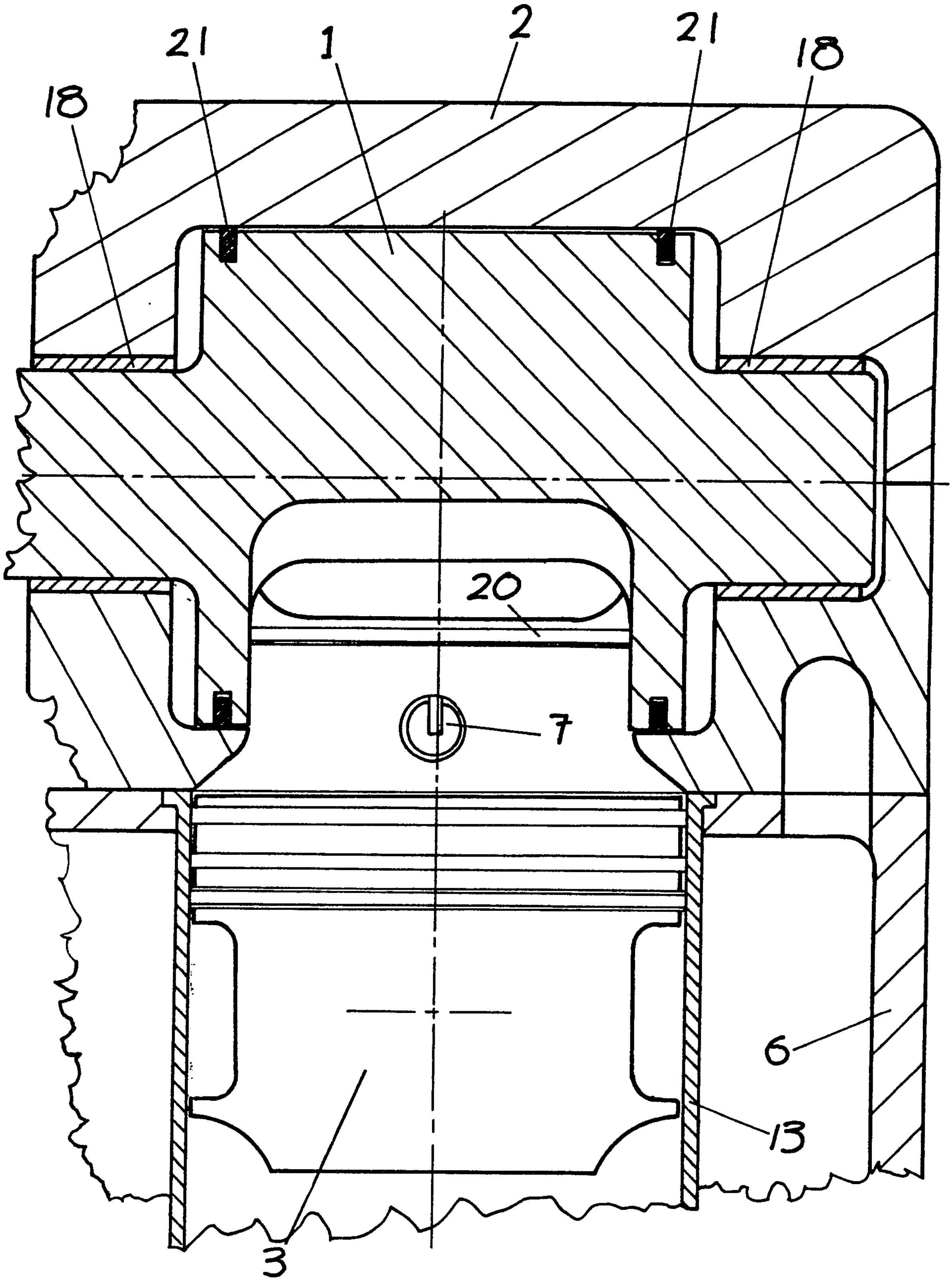
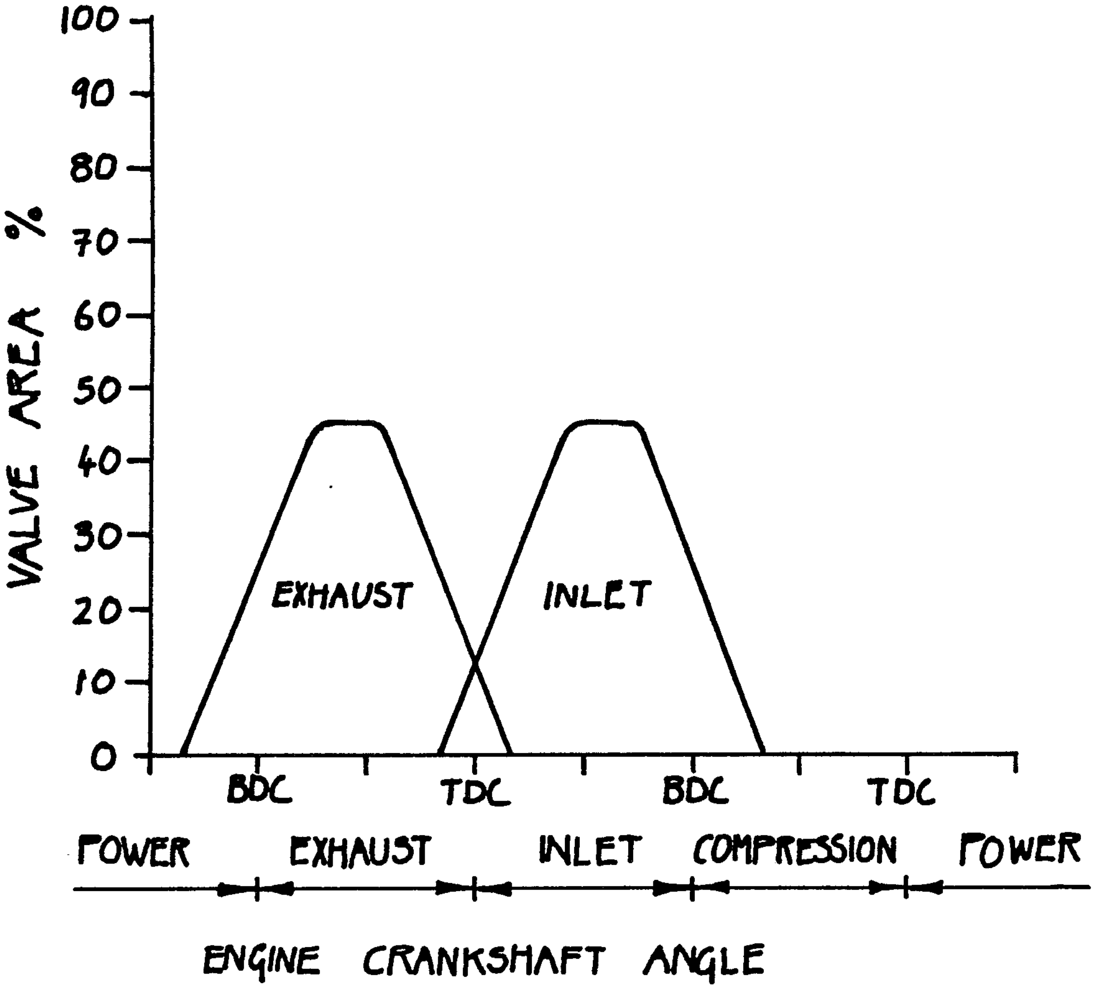


FIGURE 10



100
90
80
70
60
50
40
30
20
10
0
VALVE AREA %
EXHAUST
INLET
BDC
TDC
BDC
TDC
POWER
EXHAUST
INLET
COMPRESSION
POWER
ENGINE CRANKSHAFT ANGLE

FIGURE 11

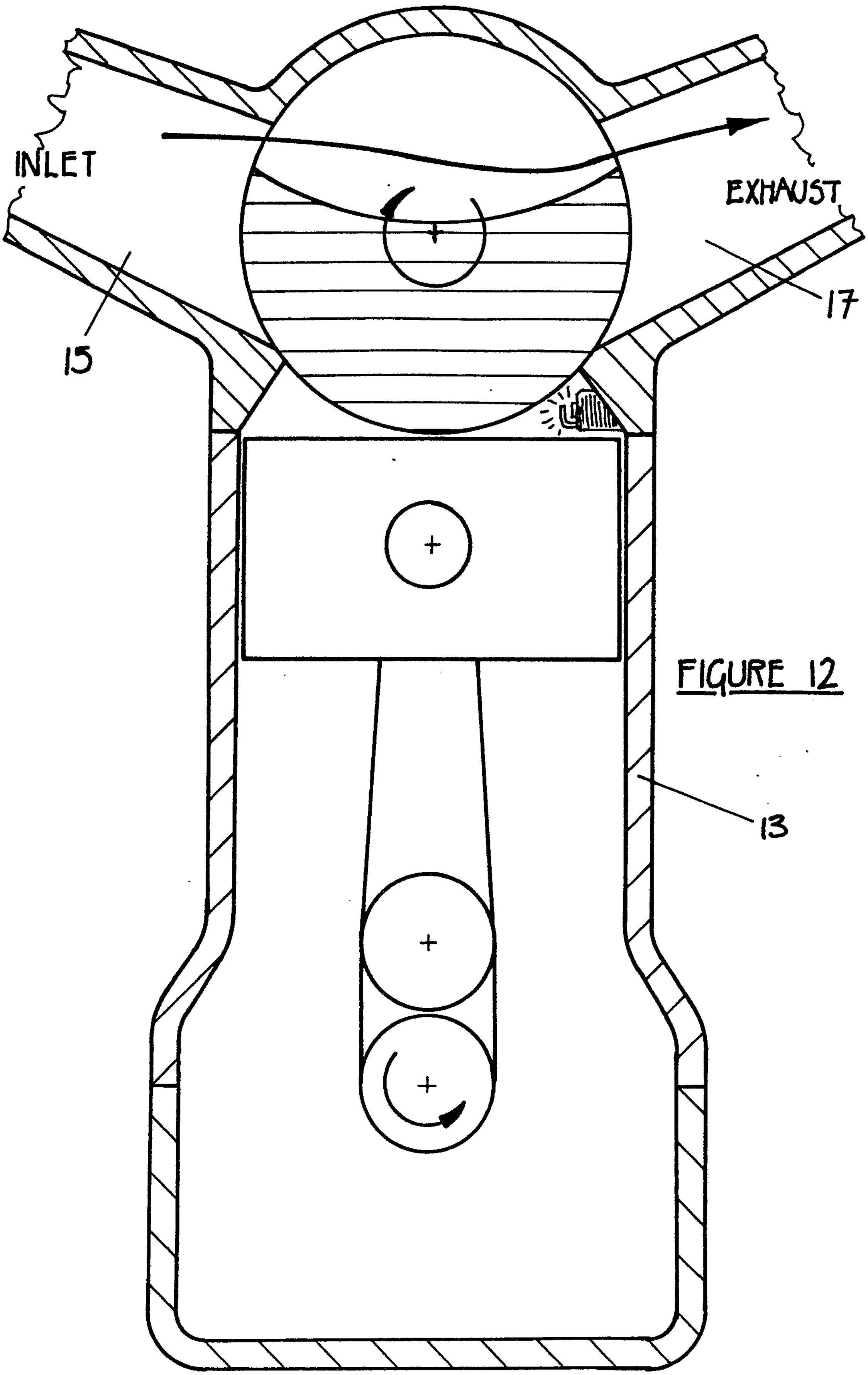


FIGURE 12

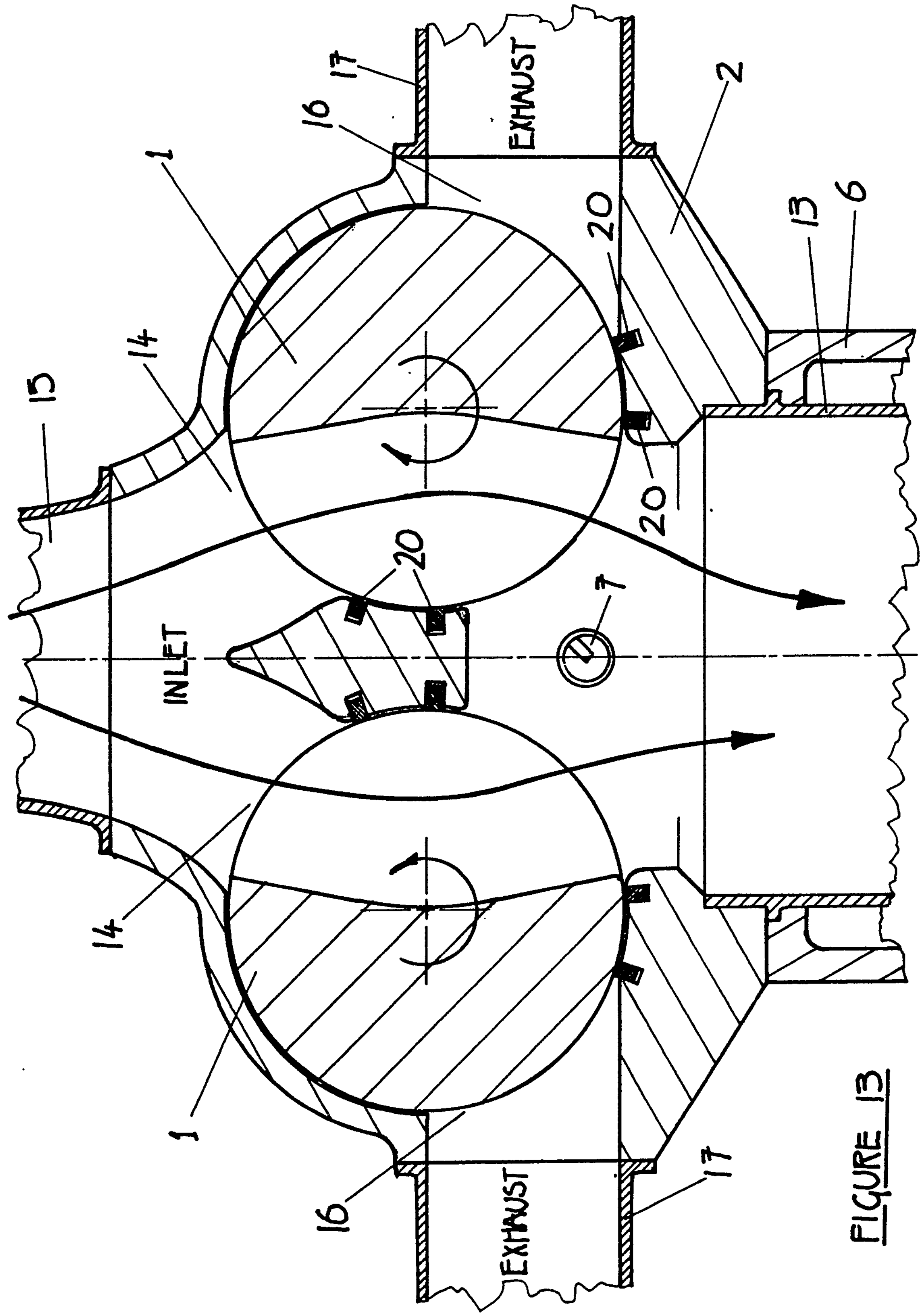
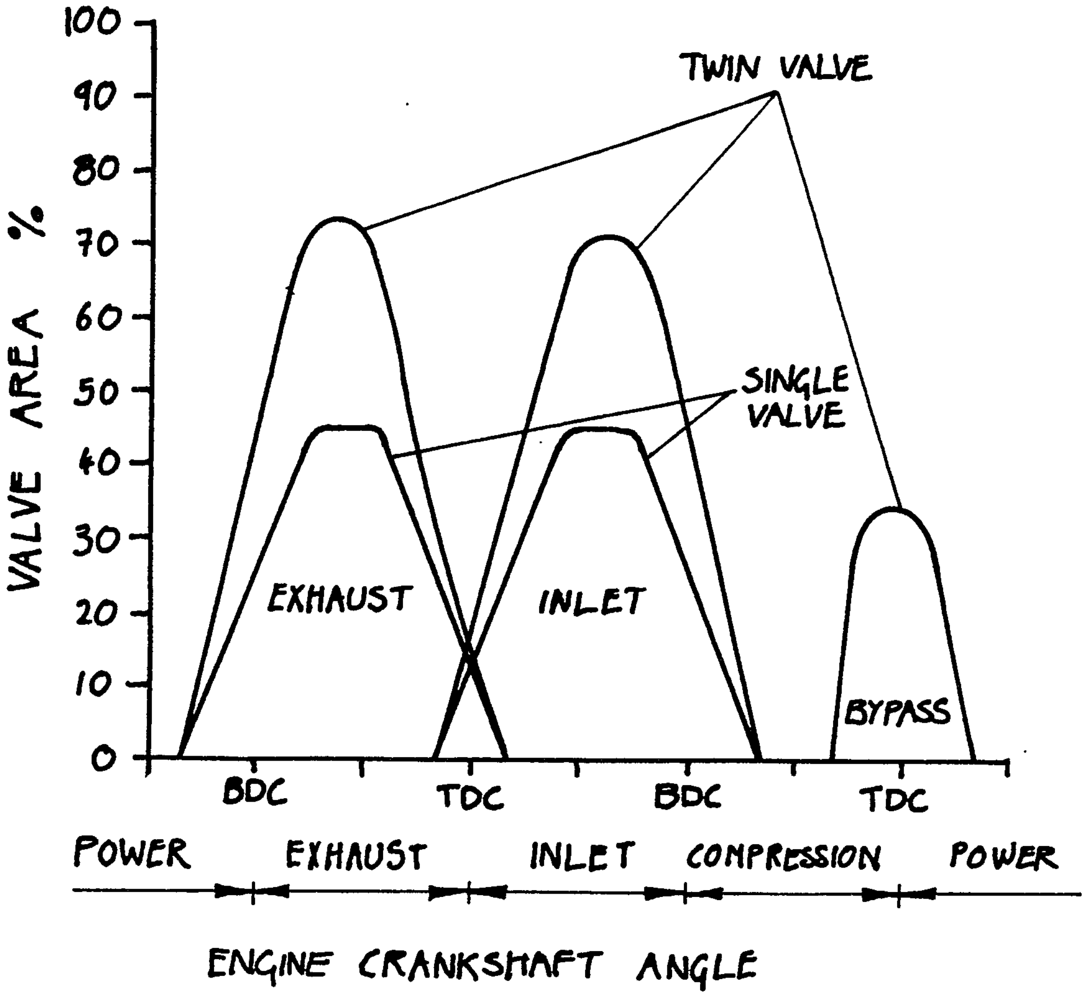


FIGURE 13



100
90
80
70
60
50
40
30
20
10
0
VALVE AREA %
TWIN VALVE
SINGLE VALVE
EXHAUST
INLET
BYPASS
BDC
TDC
BDC
TDC
POWER
EXHAUST
INLET
COMPRESSION
POWER
ENGINE CRANKSHAFT ANGLE

FIGURE 14

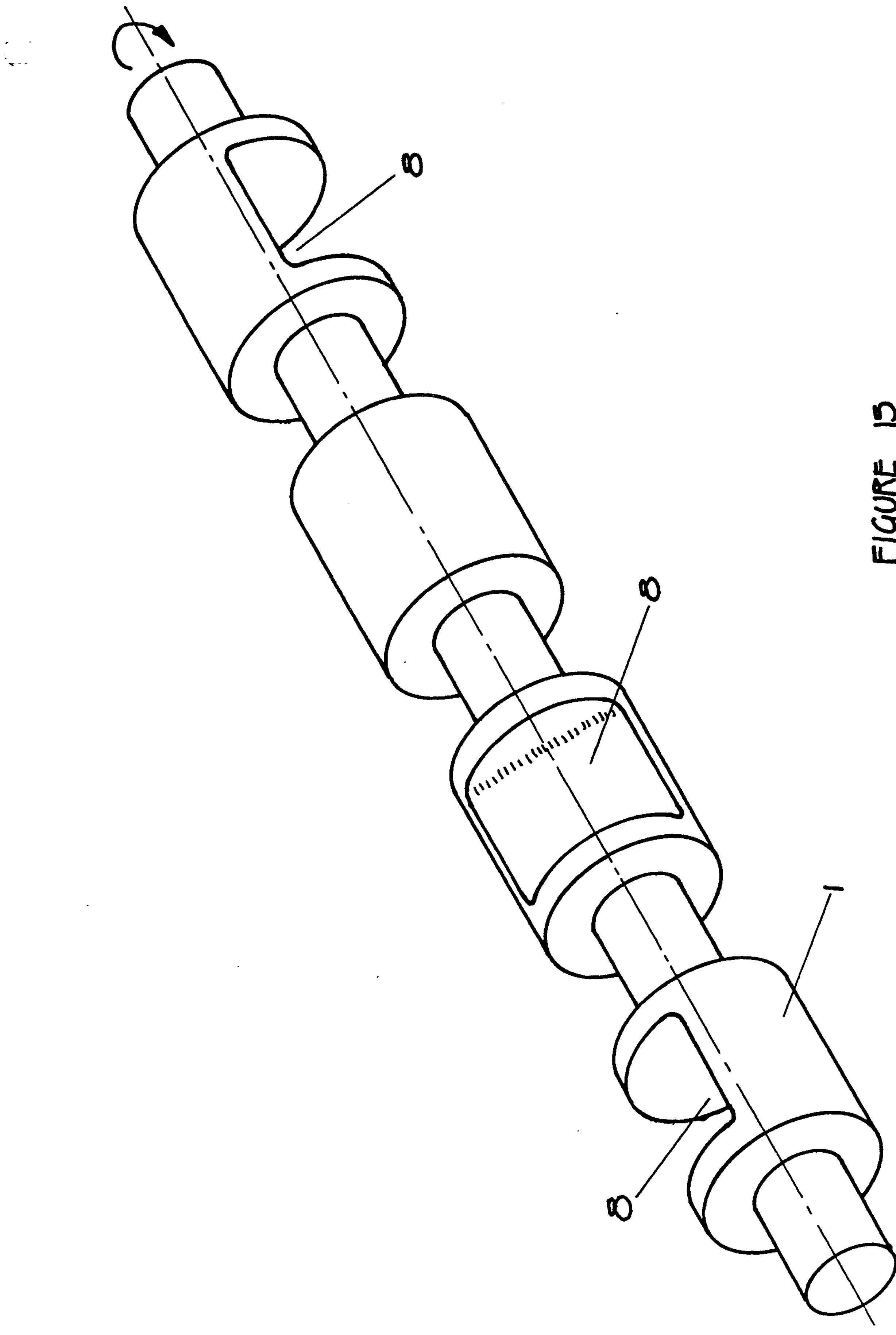


FIGURE 15

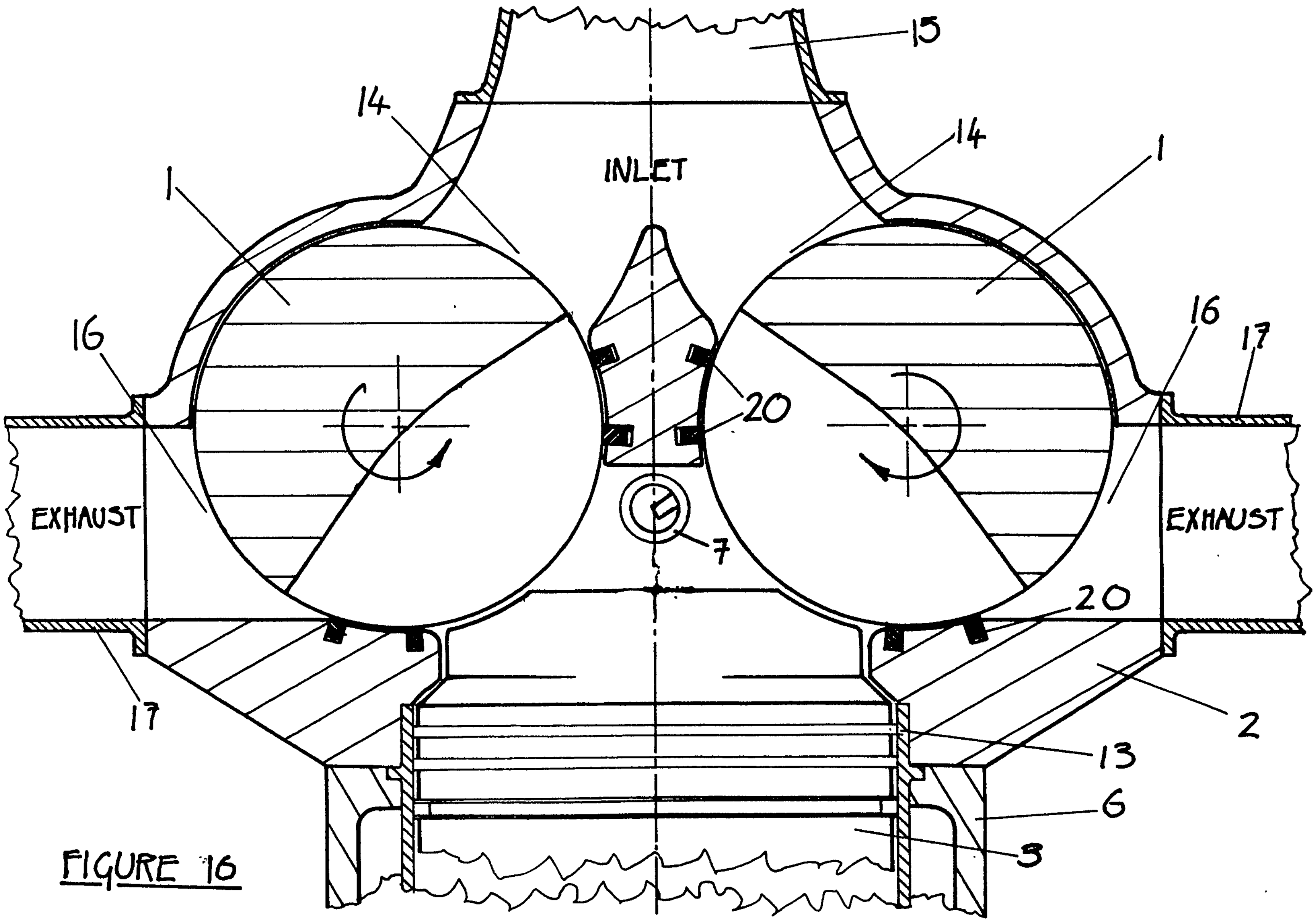


FIGURE 16

ROTARY VALVE GEAR FOR INTERNAL COMBUSTION ENGINES

This invention relates to valve gear for internal combustion engines that operate on the four-stroke thermodynamic cycle.

Four stroke spark or compression ignition engines are well known devices that rely, almost universally, on the use of reciprocating poppet valves to introduce fresh charge and release exhaust gases from an engine's cylinders. It is well known that he inertia of reciprocating poppet valves places a limit on the rotational speed an engine can attain, and hence may limit the power available from a given engine.

Furthermore the presence of a hot incandescent exhaust valve in the combustion chamber can cause pre-ignition or detonation of the fresh charge, with a consequent detrimental effect on engine performance and life. A hot exhaust poppet valve also limits the compression ratio of the engine, hence the engine's performance is limited.

Also poppet valves, even whilst fully open, present a considerable flow blockage, thus adversely affecting the free passage of gases into and out of the engine. Furthermore, valve loading demands dictate less than ideal valve opening rates which also adversely affects volumetric efficiency. Moreover, poppet valve gear is noisy, bulky and costly.

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According to the present invention there is provided a valve arrangement for a four-stroke internal combustion engine comprising a rotating valve rotor and a valve casing in which the valve rotor is constrained to rotate. The valve casing arrangement comprises three, radially disposed, gas ports which connect with the hole in which the valve rotor rotates. The valve rotor has a substantial recess in it's surface which, when rotated, can provide a gas passage between any two adjacent ports as demanded by the four-stroke cycle.

Preferably the valve casing is integral with the engine's cylinder head, with the axis of rotation at right angles to the cylinder axis, such that one port is an opening into the engine's cylinder, another forms the opening of the inlet duct and another forms the opening of the exhaust duct.

In preferred arrangements the valve rotor recess takes the form of a scallop and the geometry of the valve casing is such that a gas passage can only be formed between inlet and cylinder ports or cylinder and exhaust ports. These gas passages open and close in time with the cylinder breathing requirements of the four-stroke engine, when the valve rotor is rotated at one half the crankshaft rotational speed, such that fresh charge may enter the engine's cylinder during the inlet stroke when inlet and cylinder ports are connected; and exhaust gases may escape from the engine's cylinder, during the exhaust stroke, when the valve scallop forms a passage between cylinder port and exhaust port.

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During the compression and power strokes the non-recessed, solid portion, of the valve rotor lies over cylinder port thus preventing the flow of gas into or out of the cylinder.

Similarly during the exhaust stroke the inlet port is closed by the solid portion of the valve rotor, and during the inlet stroke the exhaust port is closed.

Since the valve rotor rotational speed is constant, at a given crankshaft rotational speed, valve inertia effects, that can limit engine rotational speeds, are eliminated.

In certain arrangements the engine's cylinder is fitted with two or more rotary valves of the afore mentioned type.

In various arrangements valve rotors may be chain, gear, shaft or belt driven.

In further embodiments variable valve timing mechanisms may be employed to optimise off design performance.

In certain arrangements the valve geometry may be such that the recessed portion of the valve rotor forms a gas passage between inlet and exhaust ports when the piston is at, or close to, top dead centre at the start of the power stroke.

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Fitting a correctly designed extractive exhaust system and matched inlet duct could induce ambient air to flow through the valve from inlet to exhaust duct, thus accelerating the air in the inlet duct. This acceleration of inlet duct air, prior to the opening of the inlet gas passage, could reduce gas inertia effects that are detrimental to engine volumetric efficiency.

Furthermore, this engine bypass flow will tend to cool the valve and will also replace air in the inlet duct that has been heated with fresh, cooler, air of higher density which will also improve engine power and efficiency.

Embodiments of the invention will now be described in more detail. The descriptions make reference to the accompanying diagrams in which:-

Figure 1 is a schematic cross-sectional diagram of a four-stroke spark ignition engine fitted with a rotary valve according to the present invention. The piston is at Top Dead Centre (TDC) prior to the inlet stroke.

Figure 2 is a three dimensional view of a valve rotor driven by a belt and pulley. A cross-section of the valve rotor, through the recess and normal to the axis of rotation, is also shown.

Figure 3 is a three dimensional view of a valve casing, looking along the axis of valve rotation, with the valve rotor removed.

Figure 4 is a cross-section through a rotary valve engine during the inlet stroke.

Figure 5 is a cross-section through a rotary valve engine during the exhaust stroke.

Figure 6 is a cross-section through a rotary valve engine at TDC at the start of the power stroke.

Figure 7 illustrates rotary valve engine operation at eight points around the four-stroke cycle. Cross-sections are shown every 90 degrees of crank rotation.

Figure 8 is a longitudinal cross-section through a rotary valve that is coplanar with the axis of piston travel. The figure illustrates the flow of heat from the hot surface of the valve rotor to the valve casing/cylinder head, via plain journal bearings in which the valve rotor rotates.

Although not shown in Figure 9, the valve rotor may also be provided with internal passages or cavities through which a suitable cooling medium may be arranged to flow.

Figure 9 is a cross-section of the top half of a rotary valve engine normal to the axis of valve rotation.

Figure 10 is a cross-section of the top half of a rotary valve engine along the axis of valve rotation.

Figure 11 is the valve timing diagram for the engine configuration depicted in figures 9 and 10.

Figure 12 is a cross-section through a rotary valve engine wherein a gas passage exists between inlet and exhaust ducts when the engine is at, or close to, TDC at the start of the power stroke.

Figure 13 shows a cross-section through an engine fitted with two rotary valves.

Figure 14 shows the valve timing diagram for the twin rotary valve engine, depicted in figure 13, compared with that of a single rotary valve engine of equal cylinder size.

Figure 15 shows the valve arrangement for an in line four cylinder engine.

Figure 16 shows a cross-section through an engine fitted with two rotary valves and a "squish" piston.

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The general arrangement of a four-stroke spark ignition engine fitted with a rotary valve, according to the present invention, is shown in figure 1.

The arrangement comprises a valve rotor 1 that is free to rotate in the valve casing/cylinder head 2. Also shown are a piston 3, connecting rod 4, crankshaft 5, engine block 6 and spark plug 7 all of conventional design. The means of driving the valve rotor 1 and other minor details are not illustrated in figure 1.

The valve rotor 1 comprises a substantially cylindrical body with a recess in it's surface. In preferred arrangements the recess has the form of a scallop such that the cross-section of the valve rotor 1, normal to axis of rotation, is roughly "D" shaped, rather than circular, in the vicinity of the recess.

Figure 2 shows a three-dimensional view of a valve rotor 1 showing the scalloped recess 8 and drive mechanism, in this case a belt 9 and pulley 10 arrangement.

In preferred arrangements the valve casing is integral with the engine's cylinder head, with the axis of rotation at right angles to the engine,s cylinder 13 axis.

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Figure 3 is a view of the valve casing/cylinder head 2, looking along the axis of rotation, with the valve rotor 1 removed. The valve casing/cylinder head 2 has a substantially cylindrical hole 11 in which the valve rotor 1 rotates. Three separate gas ports connect with the hole 11.

The first port, the cylinder port 12, is an opening into the engine's cylinder 13. The second port, the inlet port 14, forms the engine end of the inlet duct 15. The third port, the exhaust port 16, forms the engine end of the exhaust duct 17.

When the valve rotor 1 is rotated in the casing 2 the recessed portion of the valve 8 forms a gas passage between adjacent ports.

Figure 4 shows the valve rotor 1 rotated to such a position, during the inlet stroke, that a gas passage is formed between inlet port 14 and cylinder port 12 allowing a fresh charge of air/fuel mixture to enter the engine's cylinder 13. It will be evident that the fully open gas passage presents a close to ideal duct with minimal obstruction to gas flow compared to a comparable poppet valve arrangement.

Figure 5 shows the valve rotor 1 rotated to a position during the exhaust stroke such that a gas passage is formed, between cylinder port 12 and exhaust port 16, enabling exhaust gases to escape from the cylinder 13 into the exhaust duct 17.

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Figure 6 shows the valve rotor 1, at the beginning of the power stroke, wherein the solid portion of the valve rotor 1 closes the cylinder port 12 from the inlet port 14 and the exhaust port 16.

The engine cross-sectional diagrams of figure 7 illustrate the cyclic operation of a four-stroke rotary valve engine according to the present invention.

The four-stroke cycle requires that an exhaust gas passage should exist during one stroke of the cycle; that during the subsequent stroke an inlet gas passage should exist; and that during the greater part of the following two strokes there should be no gaseous exchange into or out of the cylinder volume 13.

In order for the described rotary valve to meet these demands it is necessary for the valve rotor 1 to be rotated at half the engine crankshaft 5 rotational speed. In preferred embodiments the valve rotor 1 drive may be by chain, gear,shaft or belt.

The maximum rotational speed a conventional poppet valve engine can attain is often limited by inertia effects associated with the reciprocating mass of the poppet valves.

With the rotary valve concept the direction of valve rotor 1 rotation is not reversed and the magnitude of rotational speed is constant at a given engine crankshaft 5 speed. Therefore valve rotor 1 inertia effects are eliminated as regards limiting engine crankshaft rotational speed, thus offering the potential for improved engine performance.

The use of this type of rotary valve in a four-stroke engine should allow much lower valve temperatures to be maintained, compared with the high temperatures typical of conventional exhaust poppet valves, since the valve rotor 1 which is heated during the power and exhaust strokes is also cooled directly by the flow of cool air over it's surface during the inlet stroke. Alternatively higher combustion chamber temperatures could be tolerated.

Also, since the valve rotor 1 mass is likely to be far in excess of that of a comparable poppet valve, it should provide an effective heat sink for those parts of it's surface that are exposed to hot gases. Furthermore, when the valve rotor 1 is set to rotate in plain journal bearings 18, the exceptionally high surface area that is available for heat transfer to the valve casing/cylinder head 2, for all four cycle strokes, should ensure that the maximum valve rotor 1 temperature is little in excess of the cylinder head 2 temperature. Figure 8 shows the heat flow from the heated valve rotor 1 surface to the cylinder head 2 via journal bearings 18.

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One advantage of a valve rotor 1 that operates at close to cylinder head 2 temperatures is that gas and lubrication sealing are greatly simplified due to the geometry of the valve rotor 1, and running clearances, being relatively insensitive to differential thermal expansions and distortions during transient and steady state thermal conditions. Cool valves are also attractive as regards obtaining stable combustion, at high compression ratios, without risk of pre-ignition or detonation. Potential benefits also exist as regards the demands a cool valve places on fuel quality and the quality of valve material itself.

Figures 9 and 10 show, in some detail, an embodiment of the invention wherein the exhaust gas passage opens 60 degrees before Bottom Dead Centre (BDC) and closes 30 degrees after TDC, and the inlet gas passage opens 30 degrees before TDC and closes 60 degrees after BDC. The valve rotor 1 is set to rotate in plain bearings 18 that are fed with oil, preferably from oil galleries 19. Oil scraper/gas seals 20 are positioned to reduce the thickness of lubricating oil that is exposed to the combustion chamber and to aid gas sealing.

Figure 10 shows where piston ring type seals 21 may be positioned to aid oil and gas sealing. Oil and gas sealing could also be achieved by many other means not described herein.

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Figure 11 shows the valve timing diagram for the engine illustrated in figures 9 and 10. Plotted against the x-axis is "Engine crankshaft angle". Plotted against the y-axis is a valve area parameter, "Valve area %", which is equal to the minimum geometric cross-sectional area of the gas passage, normal to the mean gas streamline, non-dimensionalised against the engine's cylinder cross-section thus:

$$\text{Valve area \%} = \frac{\text{Gas passage flow area}}{\text{Cylinder cross-section}} \times 100$$

In certain embodiments the valve geometry is such that a gas passage exists between inlet duct 15 and exhaust duct 17 when the engine is at, or close to, TDC at the start of the power stroke, as illustrated in figure 12. This gas passage could be used to recirculate exhaust gases and reduce engine emissions.

Alternatively a tuned extractive exhaust and matched inlet duct system could be employed to generate a pressure drop from inlet duct 15 to exhaust duct 17 near the start of the power stroke. There are potential performance benefits to be realised as this pressure drop will induce a flow of inlet duct 15 air to bypass the engine's cylinder 13 and flow into the exhaust duct 17 as illustrated in figure 12.

The acceleration of stagnant, or near stagnant, air that is resident in the inlet duct 15, prior to the opening of the inlet gas passage will cause the net inlet charge momentum to be greater than would otherwise be the case at the time of inlet gas passage opening. This will reduce gas inertia effects that have a detrimental impact on cylinder 13 filling and volumetric efficiency.

Furthermore this bypass flow will tend to cool the valve rotor 1 and will also have the effect of removing heated inlet charge air from the inlet duct 15. The heated, low density, air will be replaced by cooler, higher density, air again improving performance.

In a forced induction engine the bypass flow may tend to reduce the magnitude of pressure and mass flow fluctuations at the compressor outlet, associated with the cyclic filling of the cylinder 13 , thus allowing the compressor to run closer to it's characteristic surge line.

In certain arrangements two or more rotary valves may be incorporated into a cylinder head 2 design in order to maximise valve areas and optimise spark plug 7 location along with other potential benefits. Figure 13 shows a cross-section through such an engine, in this case one fitted with two valve rotors 1. Figure 14 shows the valve timing diagram for this engine compared with a single rotary valve engine of equal cylinder ize.

In further embodiments the valve rotors 1 can be incorporated into four-stroke engines with multiple cylinders 13 arranged in banks of any number. Figure 15 shows the valve rotor 1 configuration for one such engine, an in line four cylinder.

In still further embodiments the valve rotor 1 may have more than one recess in it's surface, such that the valve cross-section between any two adjacent recesses is circular, thus enabling seals to be supported along their length, or the valve to valve more than one inlet port or more than one exhaust port in any one cylinder.

In certain embodiments, the engine's piston is of a shape designed to optimise combustion performance using the phenomena of "squish" by taking advantage of the fact that the engine's piston 3 can come extremely close to the valve rotor 1 without risk of mechanical contact occurring. Figure 16 shows an engine with two valve rotors 1 and an appropriate design of piston 3, at TDC, illustrating just one of the many piston/combustion chamber design options available.

The above embodiments describe only spark ignition four-stroke engines, but it will be appreciated that the concept could be readily applied to compression ignition engines.

Furthermore this valve concept may have applications altogether remote from those described herein.

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CLAIMS

ROTARY VALVE GEAR FOR INTERNAL COMBUSTION ENGINES

1 A valve arrangement for a four-stroke internal combustion engine comprising a rotating valve rotor and a valve casing in which the valve rotor is constrained to rotate. The valve casing comprises three, radially disposed, gas ports which connect with the hole in which the valve rotor rotates. The valve rotor has a substantial recess in it's surface which, when rotated, can form a gas passage between any two adjacent ports.

2 A valve arrangement as claimed in claim 1 wherein the valve casing is integral with the engine's cylinder head, such that one port opens directly into the engine's cylinder, another is the engine exhaust port and another is the engine inlet port.

3 A valve arrangement as claimed in claims 1 or 2 wherein the valve rotor recess has the form of a scallop, such that the local valve rotor cross-section, normal to the axis of valve rotation, is roughly "D" shaped.

4 A valve arrangement as claimed in any of the preceding claims wherein the valve geometry is such that a gas passage can only be formed between inlet and cylinder ports or cylinder and exhaust ports.

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5 A valve arrangement as claimed in any of the preceding claims wherein the valve rotor is gear, shaft, chain or belt driven.

6 A valve arrangement as claimed in any of the preceding claims wherein the valve rotor is rotated at one half the rotational speed of the engine's crankshaft.

7 A valve arrangement as claimed in claim 6 wherein the valve rotor drive mechanism incorporates a variable valve timing device.

8 A valve arrangement as claimed in any of the preceding claims wherein the valve rotor is constrained to rotate in plain journal bearings.

9 A valve arrangement as claimed in any of the preceding claims wherein piston ring type seal are used to aid oil and/or gas sealing.

10 A valve arrangement as claimed in any of the preceding claims wherein scrapers are used to aid oil and/or gas sealing.

11 A valve arrangement as claimed in any of the preceding claims wherein the valve geometry is such that a gas passage may exist between inlet and exhaust ports.

.../17

12 A valve arrangement as claimed in claim 11 wherein exhaust and inlet duct tuning is utilised to induce a gas flow from inlet to exhaust duct.

13 A valve arrangement as claimed in any of the preceding claims that is used in conjunction with a forced induction system, be it mechanically or exhaust gas driven.

14 A valve arrangement as claimed in any of the preceding claims wherein the valve arrangement is used in conjunction with a specially shaped engine piston, designed to come into very close proximity with the valve rotor when the piston is at top dead centre.

15 A valve arrangement as claimed in any of the preceding claims wherein two or more valve rotors may be incorporated into a single cylinder head.

16 A valve arrangement as claimed in any of the preceding claims wherein the valve rotor has more than one recess in it's surface such that it may be used to valve more than one cylinder, when cylinders are arranged in in-line banks.

17 A valve arrangement as claimed in any of the preceding claims where the valve rotor has more than one recess, to enable seals to be supported, or more than one inlet port or exhaust port to be valved in any one cylinder.

.../18

18 A valve arrangement as claimed in any of the preceding claims wherein the valve rotor is provided with internal passageways or cavities through which a suitable cooling medium may be arranged to flow.

19 A valve arrangement substantially as described herein with reference to and illustrated in the accompanying diagrams.

.../19

Patents Act 1977 **Examiner's report to the Comptroller under Section 17** **(The Search report)** -19-	Application number GB 9317598.2
Relevant Technical Fields (i) UK Cl (Ed.L) F1B (ii) Int Cl (Ed.5) F01L 7/02, 7/14	Search Examiner R J DENNIS Date of completion of Search 13 OCTOBER 1993
Databases (see below) (i) UK Patent Office collections of GB, EP, WO and US patent specifications. (ii)	Documents considered relevant following a search in respect of Claims :- 1 TO 19

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P: Document published on or after the declared priority date but before the filing date of the present application.

E: Patent document published on or after, but with priority date earlier than, the filing date of the present application.

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Category	Identity of document and relevant passages		Relevant to claim(s)
X	GB 0922073	(BUDACHS)	1 to 6 at least
X	GB 0786105	(SBAIS)	1 to 6, 11 and 18 at least
X	GB 0289565	(MELLERSH-)	1 to 6 at least
X	GB 0285586	(SITWELL)	1 to 6, 10, 11 and 18
X	GB 0173877	(ACHER)	1 to 6, 8, 17 and 18
X	US 5154147	(MUROKI)	1 to 6, 9, 10, 11 and 17
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X	US 3945364	(COOK)	1 to 14 at least
	The above specifications are examples of many similar		

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