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GB 1195060 A EP 0098543 A1 WO 91/19889 A1

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(54) **Multi-cylinder two-stroke engine charge intake system**

(57) Each valve-controlled cylinder head intake port is connected to at least one intake duct 22 connected to a plenum 24 common to such intake ducts of other engine cylinders and to at least one intake duct 32 connected to a plenum 34 common to such intake ducts of the other cylinders. A compressor 40 pressurises at least the plenums 34 whereby, during periods when the intake valves 14 are closed, charge is stored in the ducts 22 and the plenums 24 after entering the ports through the ducts 32 and displacing gases present in the ports. Fuel is supplied by injectors 60 or carburation to the ducts 22 or plenums 24 and, by control of intake and exhaust throttle valves 46, 48 and a direct compressor connection with a proportioning valve to the plenums 24, stratification of charge in the cylinders and the extent of scavenging may be controlled. The intake ports provide swirl to concentrate mixture from the ducts 22 around the spark plug 16.

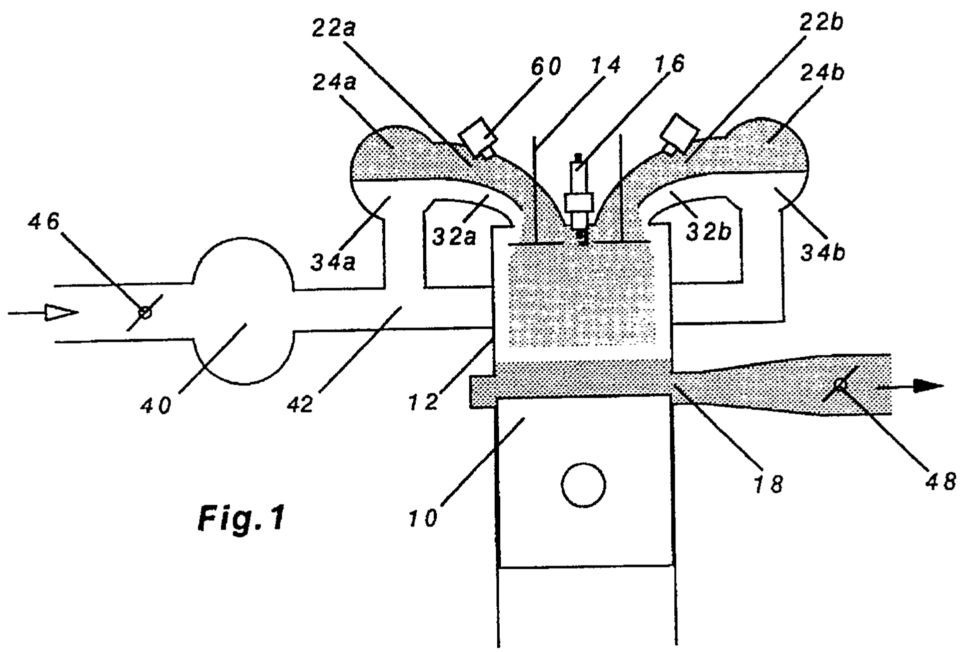

Fig.1

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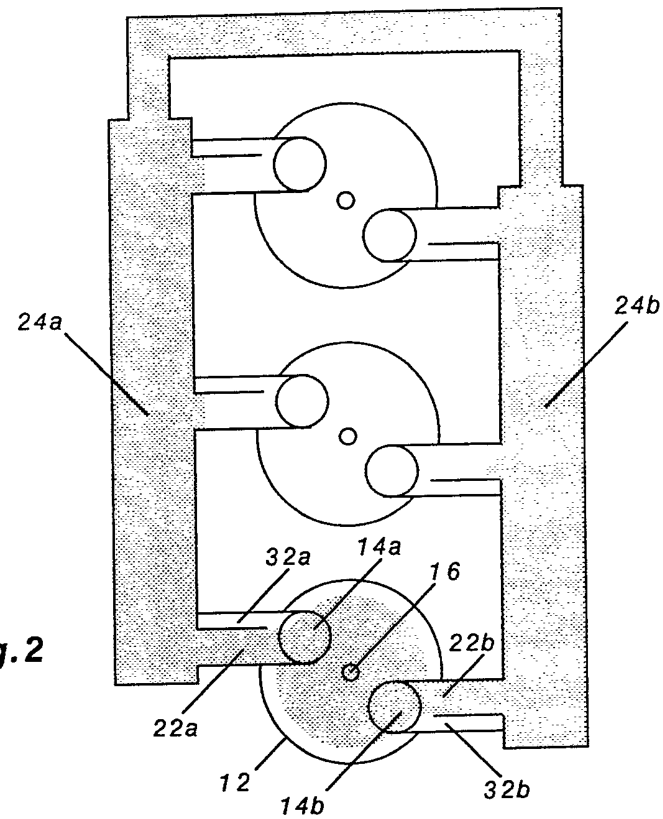
24a
24b
32a
14a
16
22b
22a
12
14b
32b

Fig. 2

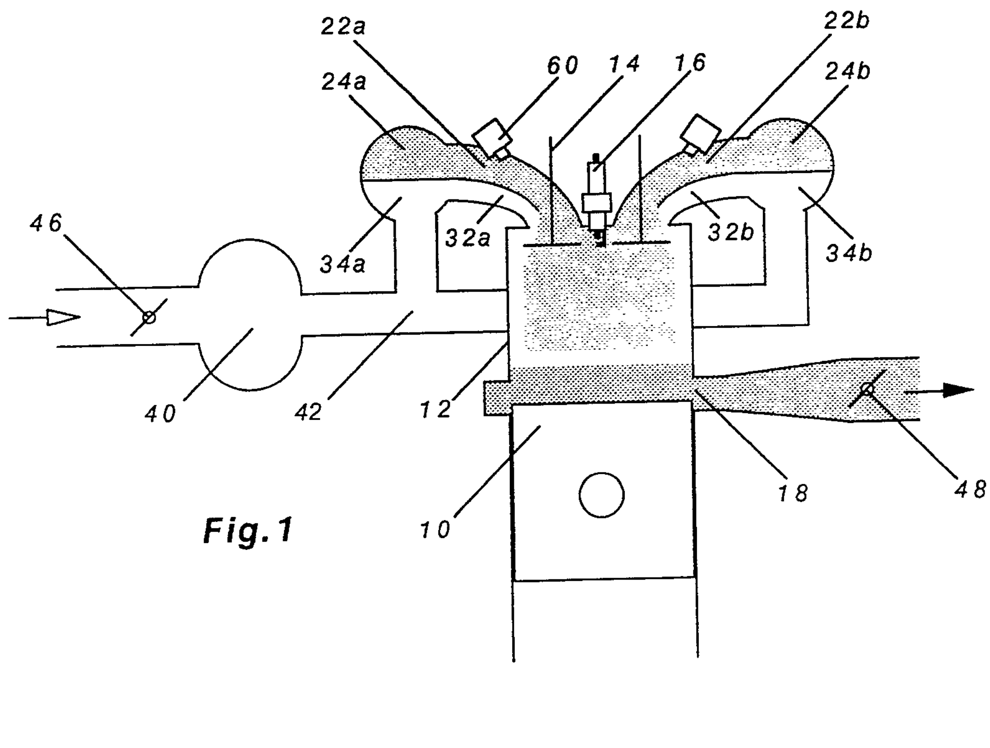
22a
22b
60
14
16
24a
24b
46
34a
32a
32b
34b
40
42
12
18
48
10

Fig. 1

Intake Manifold System for a Two Stroke Engine

The present invention relates to an intake manifold system for a multi-cylinder two stroke spark-ignition internal combustion engine.

The invention seeks to provide an intake system that improves control over the distribution of fuel and air within the charge in the engine combustion chamber.

According to the present invention, there is provided an intake manifold system for a multi-cylinder two stroke spark-ignition internal combustion engine of the type having at least one intake port at the top of each cylinder and at least one exhaust port in the cylinder wall leading to an exhaust manifold and uncovered by the piston when the latter is near the bottom of its stroke, each intake port being connected at one end to the combustion chamber by an intake valve and at the other end to a first pressurised intake duct that is connected to a first plenum common to the first intake ducts of other engine cylinders, wherein each intake port is further supplied by a second pressurised intake duct terminating near and being directed towards the closed end of the intake port whereby during the periods that the intake valve is closed gas introduced under pressure through the second intake duct by an air blower or compressor is stored in the first intake duct, the gas entering through the second intake duct flowing around the end of the intake port and displacing gases present in the intake port into the first intake duct.

A reverse flow intake manifold system for use in a four stroke engine is described in co-pending GB Pat. Appln. No. 9403290.1 and a further improvement of the system is described in Pat. Appln. No. 9404156.3.

In the above proposals, air is drawn into the manifold by the negative pressure created by the pistons undergoing the intake strokes of the four-stroke engine cycle. The present invention is based on the realisation that the same system can effectively be applied to a two-stroke engine in which the air is introduced under pressure into the intake manifold. The high pressure can either be created by an externally driven compressor or blower or by crankcase compression. As in the above mentioned applications, such a reverse flow intake manifold system can prove an effective way of stratifying the charge within the combustion chamber, thereby avoiding the need for direct fuel injection into the combustion chamber.

Various configurations have been described in the prior art for the positioning of the intake and exhaust ports in a two-stroke engine, each having its merits and drawbacks. the commonly used configuration uses an exhaust port and an inlet port at the bottom of the cylinder, both controlled by the piston. At the end of the power stroke, the piston first uncovers the exhaust port and after the initial blow down the inlet port is opened. The fresh intake charge is blown in and flows in a loop pushing ahead of it the exhaust gases into the exhaust port. This is conventionally termed loop-scavenging.

This construction has no moving parts apart from the piston but there is significant mixing between the intake charge and the exhaust gases during the scavenging process and during the compression period because of the strong tumbling charge motion. The consequence of this mixing is that charge stratification cannot be easily achieved if the intake charge is pre-mixed with fuel. Another problem that occurs is termed "short-circuiting" and is caused by direct flow of the intake charge from the intake port to the exhaust port, because of the close positioning of the ports.

Using a pre-mixed charge in such a configuration leads to high hydrocarbon emissions and high fuel consumption.

To avoid these problems, extensive work has been carried out recently on a two-stroke engine with loop-scavenging in which these problems are avoided by blowing only air into the intake port and introducing the fuel under high pressure directly into the combustion chamber after the exhaust port has been closed and the charge is trapped. Such a proposal achieves charge stratification by careful timing of the injection and spark timing but requires complex and costly control systems to maintain emissions within permissible levels. Even though emission requirements may be met under ideal conditions, it is difficult to guarantee that the engine will comply with emission regulations after prolonged use and when not in perfect tune.

The almost universal adoption of the loop-scavenged engine as the standard configuration for an automotive two-stroke engine is historical in nature and stems from the fact that its power to volume and power to weight ratios are the highest of the possible configurations. The prior art does however describe other configurations, amongst them the one used in the present invention in which air is blown in from the top of the combustion chamber through an intake valve and drives the burnt charge out through exhaust ports positioned around the base of the cylinder. This is conventionally termed uniflow scavenging as the burnt gases are displaced downwards along the length of the cylinder. Here there is little mixing between the intake charge and the burnt gases and there is little possibility of short-circuiting. Consequently, the emissions are very low but performance suffers because the intake valve restricts the breathing of the engine. Because such an engine has a lower power to weight ratio than a loop-scavenged engine and more moving parts, it has hitherto been disregarded in automotive applications. Nevertheless it still offers a power to weight advantage when compared with a four stroke engine (twice the firing frequency) and has fewer moving parts (no exhaust valves).

The present invention combines a uniflow scavenged two-stroke engine with a reverse flow intake manifold system to achieve reduced exhaust emissions and improved fuel economy without resorting to complex injection equipment and control strategies. A reverse flow intake system can achieve a stratified charge in a simple manner to ensure that the fuel remains concentrated in the vicinity of the spark plug under all conditions while the uniflow scavenged engine ensures minimal mixing of the burnt gases with the fuel in the intake charge. In this way, it is easier to ensure compliance with emission regulations even after prolonged use.

The power to weight ratio of this engine is potentially greater than that of an equivalent four-stroke engine but it can be improved still further by supercharging or turbocharging to counteract the flow restriction of the intake valves. It is believed that in today's climate in which compliance with emission regulations and fuel economy have become more important than engine performance, it is better to start with a close to ideally scavenged engine with potentially low emissions and then take steps to improve its performance, rather than to start with a high performance but less than ideally scavenged engine and attempt to lower its emissions.

Furthermore a uniflow scavenged stratified charge engine does not cause any fresh charge to be short-circuited to the exhaust system, at least not during part load operations. This gives higher exhaust gas temperatures and permits fueling calibrations that maintain a stoichiometric exhaust gas composition, thus allowing the use of a three-way catalytic converter for efficient after-treatment of all three exhaust pollutants (HC, CO and NOx) under urban driving conditions.

By using a reverse flow intake system, a major proportion of the intake charge is blown into the intake manifold and stored in the first intake duct before it is introduced into the combustion chamber. This has various advantages that stem from being able to control reliably the mixture stratification in different parts of the combustion chamber.

A problem encountered with fuel injected engines is that the fuel is not well mixed in the charge under all conditions. If fuel is injected into the intake port, it tends to forms pools and this allows liquid fuel to pour straight into the combustion chamber and mix with the burnt gases.

In the present invention, during the time when the intake valve is closed, the intake port continues to be scavenged by air blown in through the second intake duct. This maintains the intake port dry and creates in the first intake duct a column containing a stratified mixture of fuel and air.

Depending on the geometry of the intake manifold, in particular the length of the first intake duct, this column can either be stored ready to be introduced back into the same cylinder when the intake valve next opens or the mixture may enter the first plenum to mix thoroughly with the mixture blown from other cylinders. If the charged is stored as a stratified column in a long duct, then when the intake valve opens, the column is transferred into the cylinder and determines the charge stratification within the cylinder. On the other hand if the first ducts are short, not only is better charge preparation achieved in each cylinder but homogeneity between cylinders is improved.

The ability to store a stratified column allows the invention to accentuate the inhomogeneity in the cylinder and thereby permit the engine load to be regulated at least in part by modifying the amount of fuel introduced into the cylinder (rather than the amount of fuel and air mixture) while introducing an excess of air into the engine.

In an extreme case, it is possible to concentrate all the fuel at one end of cylinder as a homogeneous easily ignitable mixture while filling the lower end of the cylinder with air or burnt gases into which the flame cannot propagate. Such a divided charge may be used to achieve very lean overall fuel-air mixture or very high overall burnt gas dilution. If air forms the lower part of the divided charge and a very rich fuel-air mixture forms the upper part, then such division can create incomplete combustion in the upper part of the combustion chamber while still delivering an overall stoichiometric mixture to a catalytic converter or an afterburner. The completion of combustion of such an exhaust gas mixture in the exhaust system can assist in raising the catalytic converter to its light off temperature.

The geometry of the two ducts in the intake port can be designed to promote swirl in the combustion chamber. Also, because the second duct carries only air while the fuel is contained in the air stored in the first duct, it is possible to achieve radial stratification by directing the air from the second duct towards the outer circumference of the cylinder.

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

Figure 1 is a schematic section through a uniflow scavenged two-stroke engine incorporating a reverse flow intake system, and

Figure 2 is a schematic plan view of the engine shown in Figure 1.

The illustrated two-stroke internal combustion engine comprises three pistons 10 reciprocating in cylinders 12 having spark plugs 16. Each cylinder has an exhaust port 18 uncovered by the piston as it approaches bottom dead centre and a pair of intake ports controlled by intake valves 14a and 14b. A butterfly throttle 48 is optionally provided in the exhaust manifold.

Air is supplied to the engine by a blower 40 that is connected to a pair of plenums 34a and 34b by a pipe 42. The blower 40 draws ambient air from an intake pipe which incorporates a butterfly throttle 46. The air from the plenums 34a and 34b flows through a duct 32a and 32b towards the end of the intake port at all times. At such times as the intake valves 14a and 14b are closed, this air flows around the intake port and doubles back into ducts 22a and 22b that lead to two further plenums 24a and 24b. For ease of description the ducts 22a and 22b will hereinafter be referred to as the first intake ducts 22 and the ducts 32a and 32b as the second intake ducts 32.

The first plenums 24 and first intake ducts 22 act as a pressurised reservoir. When an intake valve of a cylinder is closed, air blown in through the second intake duct of that intake port doubles back and travels along the first intake duct in the reverse direction towards another cylinder that has an open intake valve. It is for this reason that the system is referred to as a reverse flow intake system. The air need not reach the other cylinder but is stored as pressurised air in the first intake duct and the first plenum. When eventually the intake valve of that cylinder opens, air enters not only directly from the second intake duct 32 but also from the first intake duct 22.

Because a column of air is prepared and stored in this way in the first intake duct 22 while the intake valve is closed, one can accurately control the distribution of fuel along that column to determine eventually the charge stratification within the combustion chamber. This is because when the intake valve 14 is opened, the port geometry is designed to ensure that the charge is not mixed along its length as it is transferred from the first intake duct 22 to the combustion chamber. It is essential from this point of view to prevent tumble but, as illustrated in Figure 2, it is desirable to promote swirl. Swirl results in the column residing in the combustion chamber as a double helix but still retaining axial stratification.

A further advantage of swirl is that it enables radial stratification within the cylinder also to be achieved in that, as shown by the shaded region in Figure 2, the fuel laden air from the first intake duct 22 can be concentrated at the centre of the cylinder 12 while the pure air blown in by the second intake duct 32 can adhere to the circumference of the cylinder.

The fuel distribution along the column stored in the first intake duct 22 can be determined by the timing of fuel injector 60 disposed in the intake port. If the injector injects fuel immediately after the intake valve 14 is closed, then that fuel will be transported up the first intake duct 22 and will remain concentrated in the region of the charge that is subsequently introduced last into the combustion chamber. This achieves charge stratification with the bulk of the fuel concentrated around the spark plug. Furthermore the charge that is first blown into the cylinder when the intake valve 14 opens is virtually void of fuel so that any mixing with the burnt gases will not cause fuel to be discharged from the exhaust ports 18.

The exhaust ports 18 are preferably angled circumferentially to assist the discharge of the swirling burnt gases. This helps to maintain a continuous swirling motion within the cylinder 12 carried over from one cycle to the next, thus reducing the relative swirling velocity between the in-coming charge and the retained burnt gases and minimising any mixing between the two.

At full load, one can over-scavenge with excess air. In this way, all the burnt gases are expelled followed by some of the air containing no fuel that is first blown into the cylinder. The final trapped charge containing all the fuel-air mixture then occupies the entire cylinder volume. Under part load operation, however, the throttle 46 or 48 may be partially closed to limit the scavenging of the burnt gases and restrict the volume of the intake charge blown into the cylinder. In this case charge stratification will still permit ignition in that the part of the charge near the spark plug 16 will contain a uniform ignitable mixture which is separated by a layer of air from a substantial layer of retained burnt gases from the previous cycle.

Such operation is illustrated in Figure 1 in which the lighter shaded region represents the fuel-air mixture surrounding the spark plug 16, the unshaded region represents the stratified air dilution layer and the darker shaded region represents the burnt gases being displaced towards the exhaust ports 18 by the swirling intake charge.

This layered structure prevents the fuel from entering the region of the burnt gases where, in the absence of oxygen, it would be unable to burn and would result in hydrocarbon emissions. If the stratified air dilution layer is reduced to a minimum, it is further possible to operate the engine with a slightly rich fuel-air mixture in the part of the charge surrounding the spark plug 16 and maintain an overall stoichiometric composition in the exhaust gases. In this case a three-way catalytic converter may be used in the exhaust system to oxidise the unburnt hydrocarbons and carbon monoxide as well as reduce the oxides of nitrogen.

The engine part load operation may also be controlled by regulating the speed of the air blower or air compressor 40, for example by a variable speed ratio drive. This reduces the volume of the intake charge surrounding the spark plug 16 without relying on the throttle 46 or 48.

As described in the previous applications mentioned above, that relate to the use of a reverse flow intake system in a four stroke engine, the length of the first intake ducts 22 will determine whether the intake charge to each cylinder is stored separately from those of the other cylinders within its own intake duct or whether the charges for all the cylinders are allowed to mix.

Furthermore while the first plenums 24 have been described and illustrated as being sealed chambers with all the air being blown in from the second plenums 34, they may be connected to the pressurised air source 40. This can be used to prevent accumulation of fuel within the plenums 24 and to allow fuel and air mixture to be directly introduced at the end of the columns remote from the combustion chamber instead of injecting fuel near the intake ports. The fuel may be introduced in this case into the first plenums by fuel injection of by means of a carburettor. Additionally a vaporiser or atomiser may be used to improve the mixture preparation so that only a gaseous mixture will reach the engine cylinders.

In the latter case, the pressurised air source 40 may be connected to both the first and the second plenums 24 and 34 by a proportioning valve that varies the ratio of the reverse flow air to the direct flow air into the respective ends of the stored columns. Part load operation may then be controlled by means of the proportioning valve regulating the volume of the fuel-air mixture delivered last along the columns into the combustion chamber 12.

The cross sectional flow area of the second intake duct should be sufficient to allow charging of the first intake duct during the times that the intake valve is closed. As the intake period accounts for approximately one quarter to one third of the engine cycle, the second intake duct should occupy substantially the same proportion of the cross sectional flow area of the intake port, although this could be reduced if a higher pressure is supplied from the air blower 40.

CLAIMS

1. An intake manifold system for a multi-cylinder two stroke spark-ignition internal combustion engine of the type having at least one intake port at the top of each cylinder and at least one exhaust port in the cylinder wall leading to an exhaust manifold and uncovered by the piston when the latter is near the bottom of its stroke, each intake port being connected at one end to the combustion chamber by an intake valve and at the other end to a first pressurised intake duct that is connected to a first plenum common to the first intake ducts of other engine cylinders, wherein each intake port is further supplied by a second pressurised intake duct terminating near and being directed towards the closed end of the intake port whereby during the periods that the intake valve is closed gas introduced under pressure through the second intake duct by an air blower or compressor is stored in the first intake duct, the gas entering through the second intake duct flowing around the end of the intake port and displacing gases present in the intake port into the first intake duct.

2. An engine as claimed in claim 1, wherein each intake port is designed to induce swirl within the combustion chamber.

3. An engine as claimed in claim 1 or 2, wherein the second intake duct is positioned to the side of each intake valve nearer to the circumference of the cylinder.

4. An engine as claimed in claim 3, wherein the through flow cross-section of the second intake duct to each intake port is between one quarter and one third of the effective through flow cross-section of the intake port.

5. An engine as claimed in any preceding claim, wherein the total volume of each intake port and each first intake

duct of the reverse flow intake manifold system connected to the intake port exceeds 75% of the cylinder intake charge such that all the flow from the second intake duct delivered during the part of the engine cycle of the cylinder when the intake valve is closed is stored in the first intake duct, and is blown into the combustion chamber during the intake period of the engine cycle of the cylinder when the intake valve is open.

6. An engine as claimed in any one of claims 1 to 4, wherein the total volume of each intake port and each first intake duct of the reverse flow intake manifold system connected to the intake port is less than 40% of the cylinder intake charge such that a substantial proportion of the flow from the second intake duct is discharged into the first plenum of the reverse flow intake manifold system and is blown into the adjacent cylinders.

7. An engine as claimed in any one of the preceding claims wherein substantially no air is supplied to the cylinders through the first plenum, substantially all the intake air to the engine is supplied from the second intake ducts via a second plenum connected to the air blower or air compressor.

8. An engine as claimed in claim 7, wherein a fuel injector is provided in each intake port of each cylinder to direct a fuel spray towards the closed end of the intake port.

9. An engine as claimed in claim 5, wherein the first plenum and the second plenum are both connected to the pressurised air supply delivered by the air blower or compressor, a proportioning valve being provided to vary the relative flows supplied to the first and the second plenums.

10. An engine as claimed in claim 9, wherein fuel is supplied to the first plenum.

11. An engine as claimed in claim 10, wherein means are provided in the first plenum for vaporising or finely atomising the fuel.

12. An engine as claimed in any preceding claim, wherein a throttle is provided to regulate the air flow delivered by the air blower or air compressor in order to vary the scavenging efficiency of the engine.

13. An engine as claimed in any preceding claim, wherein a throttle is provided to regulate the exhaust flow discharged from the engine in order to vary the scavenging efficiency of the engine.

14. An engine as claimed in any preceding claim, wherein engine load is controlled by regulating the speed of the air blower or compressor relative to the engine speed.

Patents Act 1977
Ex iner's report to the Comptroller under Section 17
(The Search report)

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(ii) Int Cl (Ed.5) F02B 25/04, 25/20, 25/22; F02M 35/10

Databases (see below)
(i) UK Patent Office collections of GB, EP, WO and US patent specifications.

(ii)

Search Examiner
R J DENNIS

Date of completion of Search
14 APRIL 1994

Documents considered relevant following a search in respect of Claims :-
1-14

Categories of documents

X: Document indicating lack of novelty or of inventive step.

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

&: Member of the same patent family; corresponding document.

Category	Identity of document and relevant passages		Relevant to claim(s)
Y	GB 1195060	(DEUTSCHE)	1 at least
Y	EP 0098543 A1	(BMV)	1 at least
Y	WO 91/19889 A1	(LOTUS)	1 at least

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