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(56) Documents Cited:
GB 1385687 A **GB 0969591 A**

(58) Field of Search:
INT CL **F02B**
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(54) Abstract Title: **A Rotary Piston Internal Combustion Engine Cooling Arrangement**

(57) A rotary piston internal combustion engine of the Wankel type having a housing with a two lobed epitrochoidal inner peripheral surface 14 a shaft 8 journalled in end housings 3 and 4, a rotor 9 eccentrically mounted on the shaft and geared to rotate at one third the speed of said shaft whereby working chambers are formed between the flanks of the rotor and the housings which vary in volume as the rotor rotates. The rotor is cooled by a generally closed circuit system wherein pressurised gasses are circulated by an external pump 27, via connecting passages in the end housings 19, 17 and external connecting ducts 23,28 and through an external cooling heat exchanger 24, the source of the gasses and the pressurisation being blow-by gasses from the high pressure working chambers which have leaked passed the rotor side seals into the rotor cooling passages.

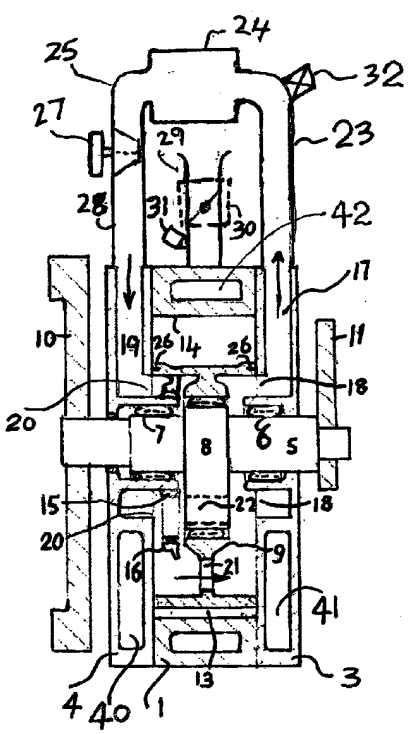


FIG. 1

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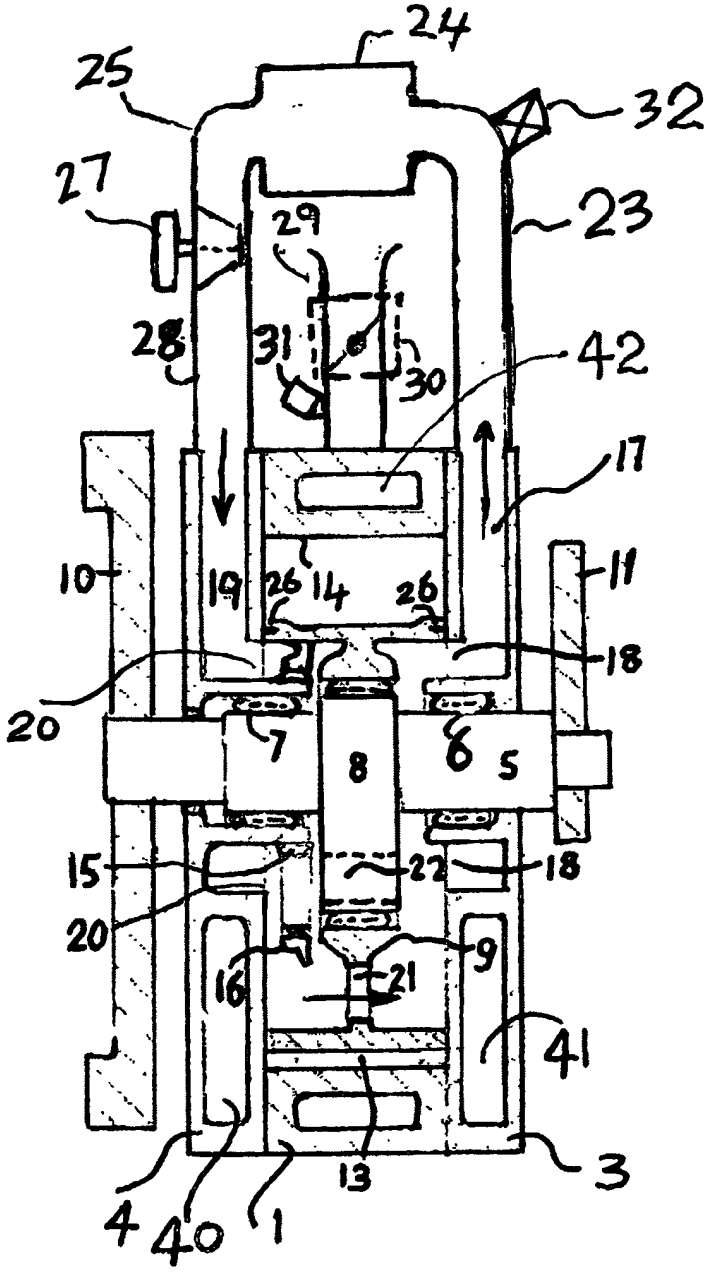

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FIG. 1

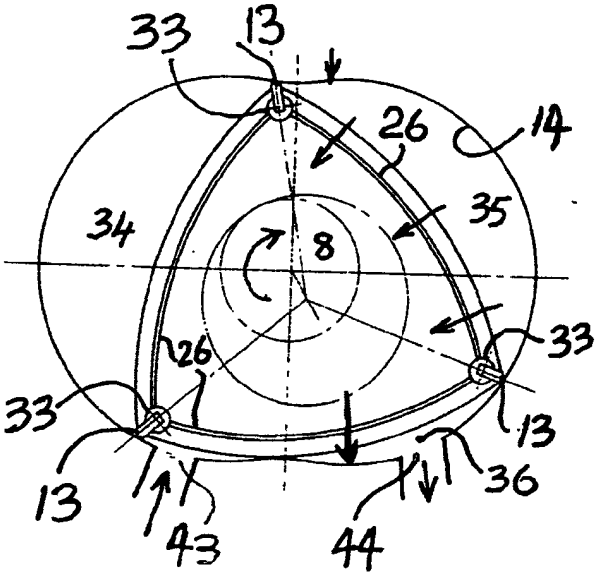


FIGURE 2

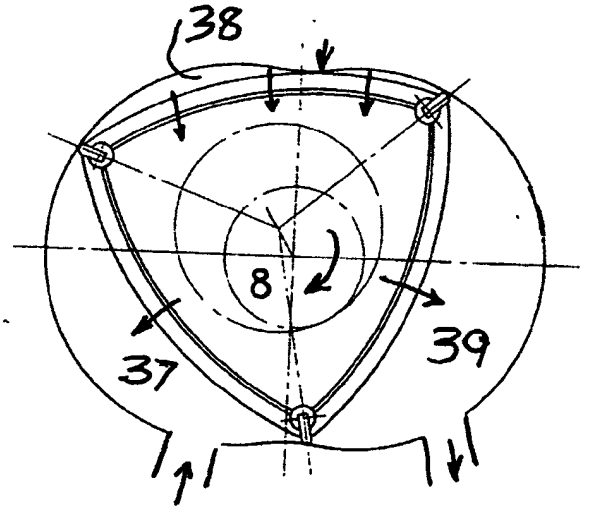


FIGURE 3

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Rotary Piston Internal Combustion Engine

The present invention relates to rotary piston internal combustion engines.
More particularly, but not exclusively, the invention relates to a so called Wankel engine in which a rotary piston, or so called rotor, rotates within a cavity formed by a housing or so called rotor housing in combination with end housings or so called end plates, the rotor outer periphery and the inner walls of the cavity being so shaped that working chambers are formed between the rotor and said walls and which vary in volume as the rotor rotates, the cavity being provided with inlet and exhaust ports. In the best known example of an engine of the kind referred to said cavity comprises a stationary rotor housing having a two lobed epitrochoidal shaped bore and a rotor of substantially triangular shape but with convex arcuate flanks, seals or so called apex seals in the apices of the rotor maintaining sealing contact with the peripheral bore of the rotor housing and seals or so called side seals in the sides of the rotor maintaining sealing contact with two axially spaced end plates and the rotor rotating in a planetary manner within the cavity.
One known method of cooling the rotor of an engine of the kind referred to is to circulate oil through internal passages in the rotor. Disadvantages of this system include considerable design complexity to adequately seal the oil within the rotating and orbiting rotor, the space required for the oil seals prohibits use of engine geometry of the more compact kind; and some power and thermal efficiency is lost through friction of the oil seals combined with the so-called cocktail-shaker losses of the oil inside the rotor.
A known alternative method of cooling the rotor is to use air as the coolant.
This has advantages of simple construction, more compact engine geometry, and low mechanical friction losses,
In one version one or more passageways are formed in the rotor which form part of the induction passage to the working chambers of the engine such that air plus fuel is drawn through and thereby cools the rotor. A disadvantage of this system is that the mixture is heated during its passage through the rotor and hence the volumetric efficiency and the power output of the engine is impaired.
To partially overcome this disadvantage, alternative systems are known where only the induction air is drawn through the rotor, the air then passing through a plenum chamber and / or an intercooler before the fuel is added and the mixture then inducted into the

working chamber. Volumetric efficiency and power output is thereby increased relative to the previous example.

Nevertheless the power is lower than the oil cooled rotor type of engine because of the pressure loss associated with the complex induction system; and the inability to cool the air down to ambient levels with a practical size of intercooler.

Rotary piston engines of this type are set forth in Patent Nos GB 1385687 and GB 1386811.

A further known type of rotary piston engine with air cooled rotor uses means to force ambient air through the rotor and then eject it direct to atmosphere and not be inducted by the engine, the air that is inducted to the working chamber being a separate supply of air at ambient pressure and temperature. Alternative known means of forcing the air through the rotor passages include use of an engine-driven centrifugal fan, or use of an exhaust ejector, or use of ram air only, such ram air resulting from the velocity of the machine in which the engine is mounted through the atmosphere

A rotary piston engine of this type is set forth in Patent No EP 0273653.

By these means these types of engine will give a higher power output than the oil cooled rotor type of engine as a result of the higher mechanical efficiency of the air cooled rotor type combined with the same level of volumetric efficiency as the oil cooled rotor type.

A disadvantage of the type of engine that ejects the spent rotor cooling air to atmosphere is that the oil which lubricates the bearings and rubbing surfaces inside the rotor is used for only one pass through the rotor and is then inevitably ejected with the cooling air. In some applications of the engine, such as the powering of unmanned air vehicles, this emission of wet oil particles is acceptable. But in most ground applications it is unacceptable. Another disadvantage is that the oil consumption is relatively high because separate oil supplies have to be provided for both the rotor internals and the epitrochoidal surface upon which the apex seals are sliding.

The higher oil usage means that the aircraft must take off with a higher weight of oil or the aircraft range is reduced

To counteract these problems, in another known type of rotary piston engine, the ejected rotor cooling air plus oil is passed through an oil separator and the proportion of oil so captured is available for re-use. The disadvantages are that it is difficult to make the oil separator 100% efficient in collecting the oil without the pressure loss to the cooling air passing through the oil separator being high, thereby requiring the engine-

driven rotor cooling fan to require a high power input. For this reason of high air pressure loss it is not practical to combine the use of an efficient oil separator with the ram air or the exhaust ejector systems of forcing the air through the rotor

In all the rotary engines with air cooled rotors described above, the cooling of the rotor is such that the systems require careful design to avoid the rotor overheating in adverse conditions of operation.

There exist potential applications of these engines where known methods of cooling of the rotor by air are not suitable and the oil cooled rotor system has to be employed

For example, the use of a rotary piston engine with an air cooled rotor would not be appropriate with turbocharging or other means of forced induction when the heat input to the rotor is considerably increased relative to a naturally aspirated engine.

This situation is particularly true for the important application of turbocharging of engines fitted to aircraft which are required to fly with good performance at higher altitudes. The purpose and effect of turbocharging is to maintain high power in a less dense atmosphere which results in continuing high level of heat input to the rotor which coincides with a reduced ability of the less dense air to cool the rotor.

Three advantages may be achieved with the present invention.

The first is the provision of an engine of the kind referred to in which no wet oil particles are emitted; the second is the provision of a rotor cooling system with greater cooling capacity such that the engine is suitable for all types of usage including being fitted with a turbocharger; and the third is the provision of improved lubrication of the rotor internals whilst achieving a lower level of oil usage.

To this end, according to the invention, a rotary piston internal combustion engine comprises a housing having a two lobed epitrochoidal inner peripheral surface, end casings for the housing, a shaft journalled in the end casings, a three flanked rotary piston within the housing mounted on the shaft eccentrically with respect thereto and geared to rotate at one third speed of said shaft whereby working chambers are formed between the rotor flanks and the inner surface of the housing, the rotor having side seals which engage with the inner side faces of the end casings, internal passageways being formed in the rotor which align with a passageway or passageways in each end casing and which form part of a cooling circuit which includes a cooling heat exchanger, a circulating pump and connecting ducting, the circulating medium effecting the cooling consisting of blow-by gasses which have leaked from the working chambers past the side seals of the rotor to the internal passages.

It should be understood that the side seals which are fitted in the sides of the rotor are intended to prevent any gas leakage from taking place between the working chambers of the engine and the inside of the rotor. In practice, the sealing efficiency is not 100% and some blow-by of gas takes place.

The volume of blow-by gas varies with engine speed and load but it is known that it amounts to around 1% to 2% of the induction airflow for this type of engine.

This invention utilises this blow-by of gas to advantage.

Before the engine is first started the closed-circuit cooling system for the rotor will enclose only air at ambient pressure. When the engine operates, some blow-by air-fuel mixture from the compression working chamber and some blow-by combustion gas from the expansion working chamber will leak past the side seals of the rotor and enter the closed circuit system so the pressure in that system will rise. The gas which enters the system will be a mixture of air and fuel, nitrogen, carbon dioxide and other partially combusted gases. The greater proportion of the mixture of gases which enter the closed circuit system may be spent combustion gases because it can be expected that there will be a greater volume of blow-by gas from the high pressure expansion working chamber than there will be from the lower pressure compression working chamber.

This gas leakage will rapidly pressurise the rotor cooling system to a value significantly above the outside ambient pressure and, if desired, to a pressure of several bar above the ambient pressure when the engine is operated at high load. The maximum pressure reached will normally be controlled by a pressure relief valve, the relief pressure value being selected to give the level of pressurisation required. The location and design of the pressure relief valve will ensure that no oil is lost through this valve.

The gas being released from the pressure relief valve will normally be ducted to the engine induction system as is normal for the blow-by gas which is also emitted in reciprocating engines.

The result of the pressure in the interior cooling ducts of the rotor becoming higher than ambient will be to cause some reverse blow-by past the rotor side seals from the rotor cooling passages outwards to the induction and exhaust working chambers of the engine where the pressure in those chambers will now generally be lower than the pressure inside the rotor.

It may be that at light engine load, when the inward blow-by volume is not high, that the outward reverse flow is sufficiently high to result in the pressure in the closed-circuit system being lower than the pressure selected for the relief valve to operate. This

is of no consequence because under these low load conditions the heat input to the rotor is low and there is then no requirement for a higher pressure to be achieved than does actually occur.

It will be understood that with this closed–circuit rotor cooling system (totally sealed other than the internal leakages which pass the side seals from and to the working chambers of the engine, together with the gas vented from the pressure relief valve as previously described) that no wet oil particles are emitted to atmosphere. The first advantage of the invention may thereby be achieved.

It will also be understood that the higher pressure and resulting higher density gas which is passing through the rotor cooling passages has greater potential for removing heat than the previously known systems employing ambient pressure air. The pressure value of the gas in the cooling circuit may be chosen to be up to several bar higher than the ambient pressure depending on the application of the engine. For example, an engine fitted with a turbocharger may utilise a higher level of pressurisation; whilst a non turbocharged engine with a light duty cycle may utilise a lower level of pressurisation.

In addition to the pressure and density advantages, the cooling capability of the gas may be further improved relative to that of air because a proportion of the gas will consist of carbon dioxide which gas has a density greater than air and thereby a greater cooling capability

The second advantage of the invention may thereby be achieved.

It will also be understood that the lubricating oil which is supplied from a metering pump to a point close to the entry of the cooling ducts in the rotor, will be carried by the circulating gas to lubricate the internal rubbing surfaces of the rotor. The greater proportion will then pass out of the rotor, through the external ducting, through the heat exchanger and pump and will then be returned to the entry point of the rotor cooling ducts to recommence a further circuit. After the first starting of the engine there will be a gradually increasing quantity of oil passing through the rotor and this rate will be higher than the rate being supplied from the oil metering pump. A stable quantity of oil which is circulating through the system will be achieved when the quantity of oil escaping outwards and passing the rotor side gas seals into the working chambers is equal to the quantity supplied from the oil metering pump, there being no other route for oil to leave the rotor cooling system.

The oil which passes into the working chambers will move outwards on the side plate inner surfaces and thereby on to the trochoidal bore surface of the rotor housing thereby lubricating the apex seals before finally passing out of the exhaust port with the hot exhaust gases and being burned or vapourised.

This system of enabling the same oil to pass through the rotor several times instead of just once with the known systems will result in a higher standard of engine lubrication, and will permit the flow setting of the oil metering pump to be reduced thereby resulting in lower oil consumption.

The third intended advantage of the invention may thereby be achieved.

In combination, these three improvements to the known air cooled rotor type of rotary combustion engine may result in this type being superior to the oil cooled rotor type in respect of power output, fuel consumption, weight, and manufacturing cost whilst achieving a similar low rate of oil usage.

The present invention will now be described with reference to the accompanying drawings which illustrate one embodiment of the engine.

Fig 1 is a schematic cross section through a rotary piston engine according to the invention.

Fig. 2 is a diagrammatic side view of the rotor.

Fig.3 corresponds to Fig.2 but with the shaft having rotated through 180 degrees.

The engine comprises a rotor housing 1 having a two-lobed epitrochoidal inner surface or bore 14 and end plates 3 and 4. A shaft 5 is rotatably journalled via rolling element bearings 6 and 7 in the end plates 3 and 4 respectively and has an eccentric journal 8 on which is rotatably mounted a rotor 9. A flywheel 10 and a balance weight 11 are mounted on opposite ends of the shaft 5. The rotor 9 has a sealing strip 13 disposed at each of its three apices, each sealing strip 13 maintaining sealing contact between the apices of the rotor and the bore 14 as the rotor rotates in planetary manner within the bore. The rotor rotates at a speed of one third that of the shaft as is controlled by an externally toothed stationary gear 15 meshing with an internally toothed gear 16 which is mounted in the rotor.

The end plate 4 has an inlet passage 19 which has openings 20 disposed to either side of the stationary gear 15, being so shaped as to be contained within the inner envelope of rotation of the side seals 26 on the rotor which effect a seal between the sides of the rotor and the end plates 3 and 4.

The other end plate 3 is similarly provided with openings 18 and an outlet passage 17.

Cooling gas can flow through passage 19 and thence from openings 20 to openings 18 through the interior of the rotor there being apertures 21 in the web of the rotor to allow airflow therethrough

Additionaly one or more apertures 22 may be formed in the journal 8 to allow a parallel airflow therethrough and thereby cooling the shaft.

A duct 23 connects passage 17 to a cooling heat exchanger 24. A duct 25 connects the exit of the heat exchanger to a circulating pump 27 and a duct 28 connects the exit from the pump to passage 19.

A pressure relief valve 32 is shown in duct 23 but may be located in other parts of the duct system.

An induction air pipe 29, and a carburettor 30 or an alternative fuel injection nozzle 31 supply the induction chamber of the engine. Alternatively the engine may be supplied with fuel by means of a direct injection system into the working chambers not shown.

This rotor cooling system is additional to the normal main cooling system for the rotor housing and end plates. Cooling passage 42 in the rotor housing and 40 and 41 in the end plates contain liquid coolant for a conventional cooling system, not shown. Alternatively, the rotor housing and end plates may be fitted with cooling fins and be air cooled.

Fig 2 illustrates the gas leakage paths across the side seals between the working chambers and the passages inside the rotor. Apex seals 13 slideably engage with the epitrochoidal bore 14. Side seals 26 make sealing engagement with the apex seals 13 via so called corner bolts 33. Inlet port 43 and exhaust port 44 are shown.

Working chamber 34 is at the beginning of the compression stroke and the pressure in the chamber at this instant will be a similar value to the pressurised cooling passages inside the rotor and little or no leakage across the seal will take place. Working chamber 35 is part way through the expansion stroke, the pressure in the chamber is high and there will be gas leakage to the cooling passages inside the rotor.

Working chamber 36 is just completing the exhaust stroke. The chamber pressure will be relatively low and there may be some reverse gas leakage from the pressurised rotor cooling passages to chamber 36.

Fig 3 shows working chamber 37 part way through the induction stroke, the chamber pressure will be low and there will be some leakage from the rotor cooling passages to chamber 37. Working chamber 38 is close to the end of the compression stroke, the pressure is high and there will be leakage of air-fuel mixture to the rotor cooling

passages. The situation for working chamber 39 which is part way through the exhaust stroke is similar to that of chamber 36 in Fig. 2.

The heat exchanger 24 may be of any suitable type which produces the desired cooling effect. The cooling capacity does not have to be high because the quantity of heat rejected to the rotor in this sort of engine is equivalent to only about 10% of the shaft power output. Typically the heat exchanger is an air to air type where the ambient air is the coolant However, in a marine engine the coolant may more preferably be water.

The pump 27 may be any suitable type but generally would be a centrifugal fan type mechanically driven from the engine or by an electric motor or by a turbine powered by the exhaust gases of the engine.

Although Fig 1 shows the system with duct 19 being the entry and duct 17 being the exit for the cooling gases to the rotor, the system could as an alternative reverse the flow of cooling gases through the rotor by using duct 17 as the entry and duct 19 as the exit.

If it so happened that side seals for this type of engine were developed such that the sealing quality was so high that insufficient gas leaked past these seals to satisfactorily pressurise the rotor cooling circuit to the desired level, then a small depression could be formed in either or both side plates, the depression being traversed by the side seals shortly before the exhaust port opens thereby intentionally allowing additional leakage from the high pressure expansion chamber to take place and the desired pressurisation thereby being achieved.

Whilst the invention described with reference to Fig. 1 shows a single rotor engine, it will be apparent that it is equally applicable to engines of the kind referred to having two or more rotors the gas cooling flows for the multiple rotors being arranged to be in parallel rather than in series.

Claims

1. A rotary piston internal combustion engine comprising a housing having a two lobed epitrochoidal inner peripheral surface, end casings for the housing, a shaft journalled in the end casings, a three flanked rotary piston within the housing mounted on the shaft eccentrically with respect thereto and geared to rotate at one third speed of said shaft whereby working chambers are formed between the rotor flanks and the inner surface of the housing, the rotor having side seals which engage with the inner side faces of the end casings, internal passageways being formed in the rotor which align with a passageway or passageways in each end casing and which form part of a cooling circuit which includes a cooling heat exchanger, a circulating pump and connecting ducting, the circulating medium effecting the cooling consisting of blow-by gasses which have leaked from the working chambers past the side seals of the rotor to the internal passages.

2. A rotary piston engine as in Claim 1 wherein the blow-by gasses in the rotor internal passages operate at a pressure higher than the external ambient air pressure.

3. A rotary piston engine as in Claim 1 or 2 wherein the maximum pressure of the blow-by gasses in the rotor internal passages is controlled by a preset or programmable pressure relief valve.

4. A rotary piston engine as in any preceding claim wherein the engine is lubricated from an oil-metering pump supplying oil into the blow-by gasses in the rotor internal passages, and thereby circulates with the cooling gasses.

5. A rotary piston engine as in claim 4 wherein the initial entry of the lubricating oil is via the journal bearing in the upstream end casing.

6. A rotary piston engine as in claim 4 or 5 wherein the circulating oil eventually escapes past the rotor side seals into the working chambers of the engine and thereby lubricates or contributes to the lubrication of the rotor apex seals.

7. A rotary piston engine as in any of claims 4 to 6 wherein a second outlet from the oil-metering pump supplies additional oil to the epitrochoidal surface to give additional lubrication to the apex seals.

8. A rotary piston engine as in any of claims 4 to 7 wherein any oil which gathers in a well in the circulating system outside the rotor passages is returned to an oil tank or the oil pump entry by means of the positive pressure in the cooling system

Application No: GB0802603.1 **Examiner:** Mr Peter Mason

Claims searched: 1-8 **Date of search:** 17 June 2008

Patents Act 1977: Search Report under Section 17

Documents considered to be relevant:

Category	Relevant to claims	Identity of document and passage or figure of particular relevance
A	-	GB 1385687 A (BIRMINGHAM SMALL ARMS CO LTD) See whole document.
A	-	GB969591 A (CURTISS WRIGHT CORPORATION) See whole document.

Categories:

X	Document indicating lack of novelty or inventive step	A	Document indicating technological background and/or state of the art.
Y	Document indicating lack of inventive step if combined with one or more other documents of same category.	P	Document published on or after the declared priority date but before the filing date of this invention.
&	Member of the same patent family	E	Patent document published on or after, but with priority date earlier than, the filing date of this application.

Field of Search:

Search of GB, EP, WO & US patent documents classified in the following areas of the UKC^{X} :

Worldwide search of patent documents classified in the following areas of the IPC

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The following online and other databases have been used in the preparation of this search report

Online: EPODOC, WPI

International Classification:

Subclass	Subgroup	Valid From
F02B	0055/06	01/01/2006