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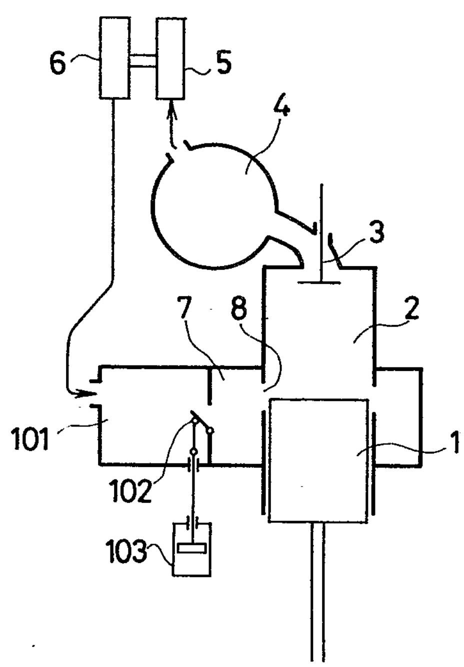
(54) **Air-charging control system for two-cycle diesel engine.**

(57) An air-charging control system for the two-cycle diesel engine of the exhaust-turbo supercharged type comprises a plurality of air-charging control valves (102) each of which is installed in the passage between an air-charging chamber (101) and each of scavenging chambers (7) for engine cylinders so as to close the passage during part of the period in which the scavenging ports (8) of the associated cylinder are open, including at least the closing point of the scavenging ports, and to open the passage for the remainder of the period thereby admitting the air to the associated scavenging chamber, and drive means (103) for opening and closing the air-charging control valves (102).



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its downward stroke, scavenging air from the chamber 07 enters the cylinder 02 through the ports, forcing burned gases clearly out of the cylinder into the exhaust manifold 04 via the exhaust valve 03.

The piston 01 then ascends again to cover the scavenging ports 08, followed by closing of the exhaust valve 03, too. The air trapped inside the cylinder 02 is compressed by the rising piston 01 and is burned together with fuel injected, developing a high pressure to give an expansive work to the piston 01 in its expansion stroke. The work is taken out as power output.

In FIG. 2 is shown schematically an existing turbocharged two-cycle diesel engine equipped with a piston-underside pump. The numerals 01 to 08 are used to designate the members like those in FIG. 1.

A piston-underside chamber 09 communicates directly with the space underneath the piston. A scavenging reservoir 010 is provided in communication with the blower 06 to receive compressed air. Between this scavenging reservoir 010 and the scavenging chamber 07 is installed a check valve 011 which permits a unidirectional flow of compressed air from the reservoir to the chamber.

Another check valve 012 is disposed between the scavenging reservoir 010 and the piston-underside chamber 09 to permit only the flow from the reservoir 010 to the latter.

TITLE
see front page

This invention relates to improvements in the air-charging control system of a two-cycle engine.

FIGURE 1 is a schematic illustration of a conventional two-cycle diesel engine of the exhaust-turbo supercharged type. In the figure are shown a piston 01, a cylinder 02, an exhaust valve 03, and an exhaust manifold 04 which communicates with the cylinder 02 through the exhaust valve 03. The reference numeral 05 indicates a turbine of the exhaust-turbo supercharger communicated with the exhaust manifold 04, and the numeral 06 indicates a blower of the supercharger adapted to run on the same shaft as with the turbine 05. A scavenging chamber 07 is formed around the cylinder 02 to receive compressed air from the blower 06. In the wall of the cylinder 02 are formed scavenging ports 08, which are closed and opened as the piston 01 moves up and down to control the communication between the scavenging chamber 07 and the cylinder 02.

After the combustion stroke of the piston 01, the exhaust valve 03 opens to release exhaust gases from the cylinder 02 into the exhaust manifold 04. The high-temperature, high-pressure exhaust energy drives the turbine 05 and therefore the blower 06 on the common shaft, with the consequence that air is compressed by the latter and led into the scavenging chamber 07.

As the piston 01 uncovers the scavenging ports 08 on

FIG. 3 graphically represents changes in the pressure inside the cylinder (full line), scavenging pressure (broken line), and exhaust pressure (alternate long and short dashes line) during the upward or scavenging stroke of the piston 01 in the conventional engine of FIG. 1.

After the piston 01 has completed its expansion stroke, the exhaust valve 03 opens to release the exhaust gases under a high pressure. As a consequence, the pressure inside the cylinder 02 decreases to a level below the scavenging pressure before the scavenging ports 08 are opened. This permits copious supply of scavenging air from the chamber 07 into the cylinder while the scavenging ports 08 are open, or over the period from SPO to SPC.

As indicated in FIG. 4, which is a pressure-stroke volume diagram of a cylinder cycle, delaying the timing for opening the exhaust valve 03, or EVO, from the point A to the point B increases the effective expansion stroke of the engine accordingly, enabling additional power corresponding to the hatched area in the figure to be taken out. This means more power output from a given amount of fuel and hence a lower fuel consumption by the engine.

In the ordinary engine shown in FIG. 1, however, retardation of the timing for exhaust valve opening, EVO, will bring EVO so close to the timing SPO for opening the scavenging ports 08, as indicated in FIG. 5, that the

Still another check valve 013, provided between the piston-underside chamber 09 and the scavenging chamber 07, permits only the flow from the former to the latter.

Compressed air from the blower 06 is conducted into the scavenging reservoir 010 and thence, during the upward stroke of the piston 01, is drawn by suction into the piston-underside chamber 09 through the check valve 012 to relieve the partial vacuum formed by the rising piston.

On the next stroke for expansion the piston 01 moves downward to force the air from the piston-underside chamber 09 through the check valve 013 into the scavenging chamber 07, thus increasing the pressure inside the latter. Further downward movement of the piston 01 uncovers the scavenging ports 08, admitting the air under a high pressure from the chamber 07 into the cylinder 02 through the ports 08 so as to drive the burned gases out of the cylinder via the exhaust valve 03 into the exhaust manifold 04. The admission of scavenging air into the cylinder 02 reduces the pressure inside the scavenging chamber 07. As the pressure drops below the level inside the reservoir 010, scavenging air is supplied from the reservoir 010 to the scavenging chamber 07 by way of the check valve 011 to be ready for the next run of cylinder scavenging.

The engines of the prior art described above have disadvantages now to be explained.

07 is elevated and, even though the timing for opening the exhaust valve 03, EVO, is delayed from the point in FIG. 3 to the point in FIG. 6, the scavenging pressure is higher than the pressure inside the cylinder 02 at the point SPO where the scavenging ports 08 open. Hence there is no such exhaust blowback as in FIG. 5.

However, since the piston 01 works also to compress the air inside the piston-underside chamber 09, the ordinary work to be done by the piston is decreased by the compression work. As a consequence, the fuel consumption by the engine cannot be saved by retarding the timing for opening the exhaust valve and thereby extending the effective expansion stroke.

Although the disadvantages of the conventional two-cycle diesel engines of the uniflow scavenged type having exhaust valves have so far been explained, exactly the same is true of the engines of the loopflow scavenged type in which the exhaust valves are replaced by exhaust ports formed above the scavenging ports and also of the reverse-uniflow type in which each cylinder has a scavenging valve on top and exhaust ports in the lower part.

The present invention has for its object the provision of an air-charging control system for a two-cycle diesel engine of the exhaust-turbo supercharged type capable of saving the fuel consumption in the light of the foregoing, scavenging ports 08 will open while the pressure inside the cylinder 02 is still higher than that inside the scavenging chamber 07. Consequently, during the period hatched in FIG. 5, a phenomenon known as exhaust blowback will take place, in which the residual exhaust gases in the cylinder 02 flows back through the scavenging ports 08 into the scavenging chamber 07.

The blowback can choke the scavenging ports 08 with carbides in the residual exhaust gases during the scavenging period, thus seriously affecting the reliability of the engine. The burned gases so blown back are again forced into the cylinder during the remainder of the scavenging period, largely decreasing the efficiency of scavenging the cylinder 02 and increasing the fuel consumption due to aggravation of combustion performance with insufficient air supply. In other words, the attempt of extending the effective expansion stroke by delaying the timing for opening the exhaust valve gives a rather undesirable result of larger fuel consumption.

In order to prevent the blowback of exhaust gases, the other engine of the prior art shown in FIG. 2 is modified so that, as the piston 01 moves downward, the air in the piston-underside chamber 09 is compressed and forced into the scavenging chamber 07. As is clear from FIG. 6, the scavenging pressure (broken line) in the scavenging chamber characterized in that:

(1) The scavenging air space is partitioned into separate chambers, one for each cylinder;

(2) Air under a high pressure from the blower of the exhaust-turbo supercharger is led to an air-charging chamber;

(3) Air-charging control valves for the individual cylinders are installed between the air-charging chamber and the respective scavenging chambers for the cylinders; and

(4) Each of the air-charging control valves for the cylinders remains closed during the whole period in which the scavenging ports or valve of the associated cylinders is open, or at least the part of the period including the closing point of the scavenging ports or valve, but is open for the rest of the period to admit air from the air-charging chamber into the scavenging chamber of the cylinder.

In the arrangement according to the invention, it is possible to keep the pressure inside the air-charging chamber sufficiently higher than the exhaust pressure, so that the scavenging pressure is raised by the high-pressure air charged at the beginning of the scavenging period to preclude the exhaust blowback despite retardation of the timing for opening the exhaust valve. This makes possible the saving of fuel consumption by the engine.

The invention will be better understood from the following detailed description when taken in conjunction with the accompanying drawings, wherein:

FIG. 1 is a schematic illustration of a conventional two-cycle diesel engine of the exhaust-turbo supercharged type;

FIG. 2 is a schematic view of a conventional two-cycle diesel engine of the exhaust-turbo supercharged type having a piston-underside pump;

FIG. 3 is a diagram showing changes in the pressure in the cylinder and the scavenging and exhaust pressures during the scavenging-exhaust stroke of a piston in the conventional engine;

FIG. 4 is a pressure-stroke volume diagram of a cycle in the cylinder;

FIG. 5 is a diagram showing changes in the pressure inside the cylinder and the scavenging and exhaust pressures with retardation of timing for opening the exhaust valve of the engine shown in FIG. 1;

FIG. 6 is a diagram corresponding to FIG. 5 but showing changes in the engine of FIG. 2;

FIG. 7 is a schematic view, in vertical section, of a two-cycle diesel engine of the exhaust-turbo supercharged type incorporating an embodiment of the system of the invention;

FIG. 8 is a transverse sectional view of the engine shown in FIG. 7; and

FIG. 9 is a diagram showing changes in the pressure inside the cylinder and the scavenging and exhaust pressures during the scavenging-exhaust stroke of a piston in the engine of FIG. 7.

Referring to FIGS. 7 and 8, the numeral 1 designates a piston, 2 a cylinder, 3 an exhaust valve, and 4 an exhaust manifold into which burned gases from the cylinder 2 is conducted through the exhaust valve 3. The turbine 5 of an exhaust-turbo supercharger receives part of the gases from the exhaust manifold 4. The blower 6 of the supercharger is driven in parallel with the turbine 5 on the same shaft. A scavenging chamber 7 is formed around the cylinder 2 as one of separate compartments for the respective cylinders. Scavenging ports 8 are formed in the surrounding wall of each cylinder 2 to provide and shut off communication between the scavenging chamber 7 and the cylinder 2 as the piston 1 moves up and down.

An air-charging chamber 101 common to the cylinders 2 is adapted to receive compressed air from the blower 6.

One such charging chamber 101 may be provided for each blower 6 of the exhaust-turbo supercharger as well as for the total number of cylinders as in the embodiment being described.

Indicated at 102 are air-charging control valves for the individual cylinders, each installed between the air-charging chamber 101 and the scavenging chamber 7 of each cylinder. Each air-charging control valve 102 is driven by an actuator 103 to close and thereby cut off the communication between the air-charging chamber 101 and the scavenging chamber 7 of the associated cylinder, for either the whole period in which the scavenging ports 8 of the associated cylinder are open or at least the part of the period including the point of time when the ports 8 close, and to open so as to charge air under a high pressure from the charging chamber 101 into the scavenging chamber 7 of the associated cylinder for the remainder of the period.

The actuator 103 is timed with the crank angle of the engine to open or close the air-charging control valve 102. Although an oil hydraulic cylinder is shown in FIG. 7, the actuator may take the form of an electric or cam-type drive means.

The air-charging control valve 102 may be provided for each of the cylinders 2, or alternatively only certain cylinders may be equipped with such valves.

In the latter case the air-charging chamber 101 is kept in communication with the scavenging chambers 7 of the remaining cylinders 2 not equipped with the valves.

Next, the operation of the embodiment of the foregoing construction will be described below.

Air under a high pressure is forced from the blower 6 into the air-charging chamber 101 and thence is charged into the scavenging chamber 7 through the air-charging control valve 102 as the valve is opened by the actuator 103 during the period in which the scavenging ports 8 remain covered by the piston 1 on its compression and expansion strokes. After the pressure inside the scavenging chamber 7 has risen to be equal to the pressure in the air-charging chamber 101, the charging control valve 102 is closed by the actuator 103.

As the piston 1 moves downward and uncovers the scavenging ports 8, the air under pressure from the scavenging chamber 7 enters the cylinder 2 through the ports 8, driving the residual exhaust gases clearly out of the cylinder through the exhaust valve 3 into the exhaust manifold 4.

The pressure inside the scavenging chamber 7 drops as the air volume decreases due to the ingress of scavenging air into the cylinder 2. The pressure inside the cylinder 2 declines, too, until it and the scavenging pressure are both reduced to levels substantially equal to the exhaust pressure immediately before the piston 1 that has passed its bottom dead center and rebounded covers the scavenging ports 8.

In FIG. 9 are graphically shown changes in the pressure inside the cylinder (full line), scavenging pressure (broken line), exhaust pressure (alternate long and short dashes line), and pressure inside the air-charging chamber (dotted line) during the scavenging-exhaust stroke of the engine incorporating the present embodiment. Since the charging pressure can be set higher than the exhaust pressure as will be explained later, the scavenging pressure at the time the scavenging ports are opened, SPO, is kept high. When the timing for opening the exhaust valve 3 is delayed to increase the effective expansion stroke of the piston 1 so as to reduce the fuel consumption of the engine, the pressure inside the cylinder after the opening of the scavenging ports 8 is lower than the pressure inside the scavenging chamber 7, and there is no possibility of the residual exhaust gases flowing back through the scavenging ports 8 into the chamber 7.

For the above purpose the quantity of scavenging air per cycle, G_s, is, where the overall opening time and area of the scavenging ports 8 are adequate, approximately expressed as

$$G_s \propto (P_b - P_{el}) \cdot V_s$$

where P_b is the pressure inside the air-charging chamber, P_{el} is the exhaust pressure, and V_s is the volume of the scavenging chamber.

The greater the difference between the charging and exhaust pressures, or the larger the volume of the scavenging chamber, V_s, the more is the quantity of air available for scavenging.

On the other hand, the energy balance between the turbine 5 and the blower 6 of the exhaust-turbo supercharger determines the level of the charging chamber pressure P_b with respect to the exhaust pressure P_{el}. The pressure P_b relative to the pressure P_{el} can be raised as the supercharger efficiency increases, or as the scavenging air quantity decreases and the exhaust temperature increases.

In ordinary engines the difference between the scavenging and exhaust pressures depends largely on the flow resistances of the fluids past the scavenging ports and the exhaust valve. Since the two pressures are practically at the same level, an enhanced efficiency of the supercharger merely increases the scavenging air flow rate and lowers the exhaust temperature; the scavenging pressure level relative to the exhaust pressure remains almost unchanged.

In this embodiment of the invention, by contrast, choice of scavenging chambers 7 of an appropriate volume, V_s, permits an increase of the charging pressure P_b with an improved supercharger efficiency. Also, as compared with the arrangements of the prior art, the energy balance of the supercharger according to the invention provides a smaller quantity of air and a higher exhaust temperature, with a consequent increase in the proportion of exhaust energy to be recovered by the turbine 5 of the exhaust-turbo supercharger.

In brief, this embodiment of the invention obtains a high scavenging pressure, at the time the scavenging ports are opened, by means of the turbine 5 and the blower 6 of the supercharger using the exhaust energy, not by the compression work of the piston in the piston-underside chamber as in FIG. 2.

While an embodiment of the invention has been described as incorporated in a two-cycle diesel engine of the uniflow scavenged type in which each cylinder is provided with scavenging ports and an exhaust valve, it should be clear to those skilled in the art that the invention is equally applicable, with exactly the same construction, actions, and effects, to the two-cycle diesel engines of the loop flow type in which each cylinder has exhaust ports, in place of the exhaust valve, above the scavenging ports, and of the reverse uniflow type in which the cylinder has a scavenging valve on top and exhaust ports in the lower part.

With the construction so far described, the system of the invention presents the following advantages:

(1) Since a pressure higher than the exhaust pressure can be used for scavenging at the time the scavenging ports are opened, the timing for opening the exhaust valve may be delayed without the possibility of the exhaust gases

flowing backward from the cylinder through the scavenging ports into the scavenging chamber. Hence, there occurs no deterioration of the combustion cycle due to any choking of the scavenging ports or a drop of scavenging efficiency. Thus, retardation of the timing for opening the exhaust valve allows an increase in the effective expansion stroke of the piston, leading to less fuel consumption by the engine.

(2) The high scavenging pressure does not affect the fuel economy of the engine in any way, because it is achieved by the recovery of exhaust energy due, for example, to the enhanced efficiency of the exhaust-turbo supercharger and also to the elevated exhaust temperature with decreased air consumption, not by the compression work of the piston as in the ordinary arrangement of the piston-underside chamber type.

What is claimed is:

1. In an air-charging control system for a two-cycle diesel engine having an exhaust-turbo supercharger which is driven by exhaust gases from the engine, scavenging chambers formed in the same number as, and respectively for, the engine cylinders, scavenging ports or valves providing communication between said scavenging chambers and said cylinders, and an air-charging chamber for receiving air under pressure from said exhaust-turbo supercharger, the improvement which comprises a plurality of air-charging control valves each of which is installed in the passage between said air-charging chamber and each said scavenging chamber to close said passage during part of the period in which the scavenging ports of the associated cylinder are open, including at least the closing point of said scavenging ports, and to open said passage for the remainder of said period so as to admit the air to the associated scavenging chamber, and drive means for opening and closing said air-charging control valves.

2. A system according to claim 1, wherein said air-charging control valves are provided in the same number as said cylinders.

3. A system according to claim 1, wherein said air-charging control valves are installed between only limited numbers of said scavenging and air-charging chambers.

FIG.1

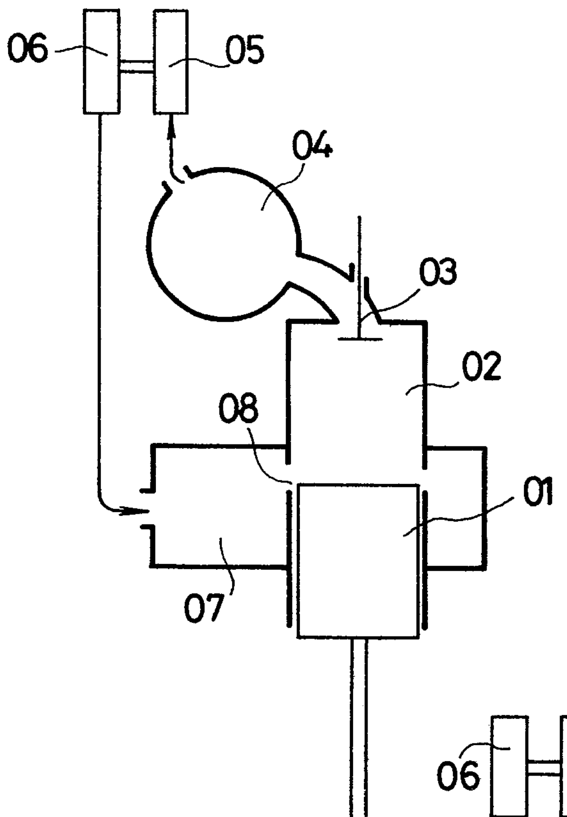


FIG.2

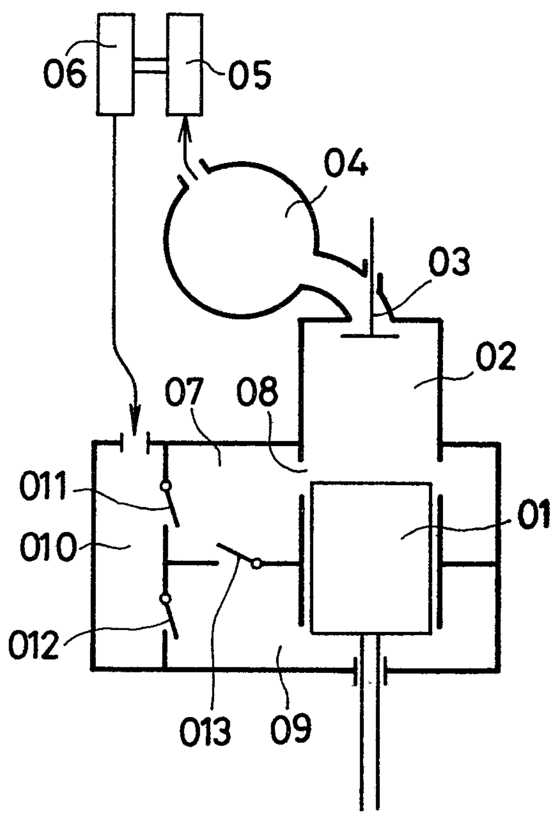


FIG.3

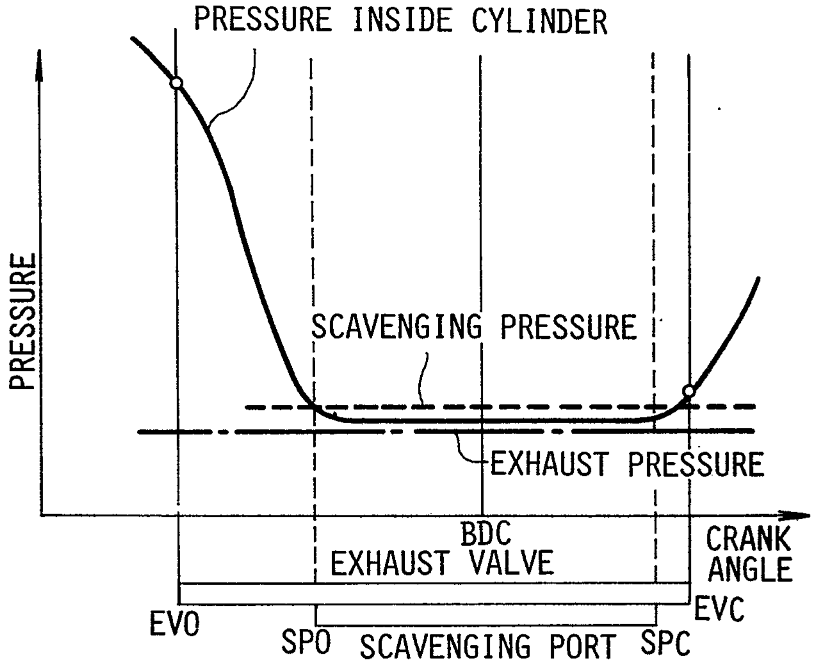

PRESSURE INSIDE CYLINDER
PRESSURE
SCAVENGING PRESSURE
EXHAUST PRESSURE
BDC
CRANK ANGLE
EXHAUST VALVE
EVO
EVC
SPO
SCAVENGING PORT
SPC

FIG.4

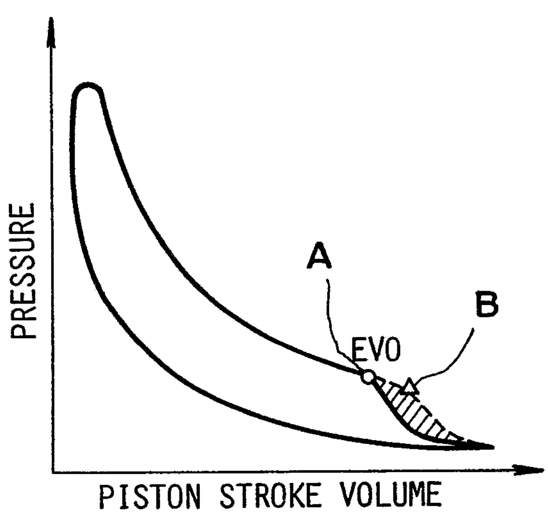

PRESSURE
A
B
EVO
PISTON STROKE VOLUME

FIG.5

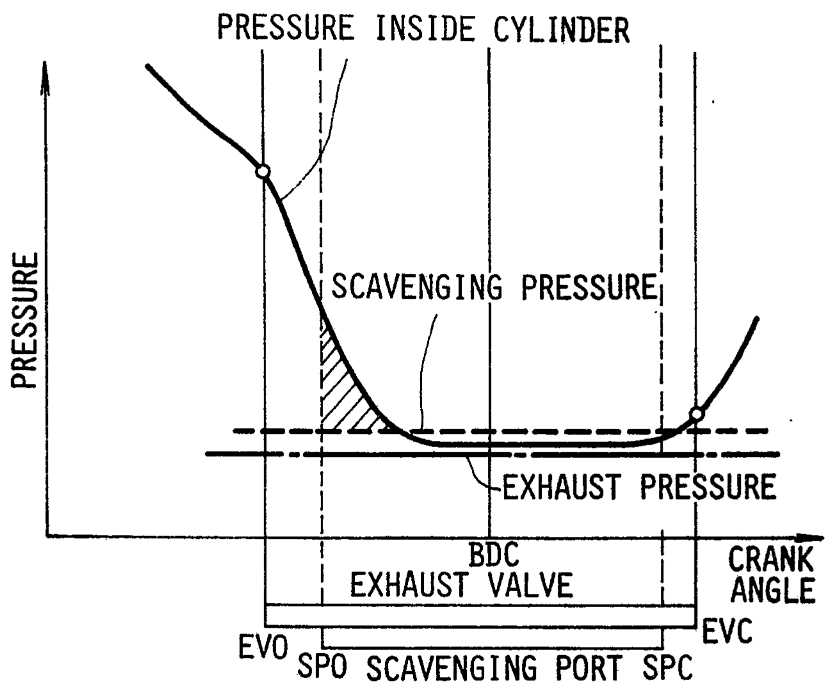

PRESSURE INSIDE CYLINDER
PRESSURE
SCAVENGING PRESSURE
EXHAUST PRESSURE
BDC
CRANK ANGLE
EXHAUST VALVE
EVO
EVC
SPO
SCAVENGING PORT
SPC

FIG.6

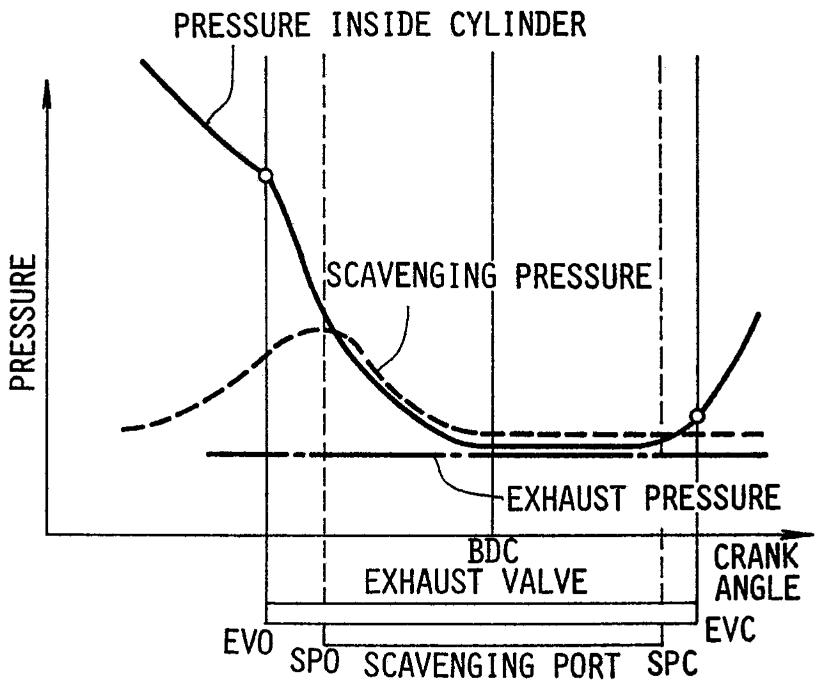

PRESSURE INSIDE CYLINDER
PRESSURE
SCAVENGING PRESSURE
EXHAUST PRESSURE
BDC
CRANK ANGLE
EXHAUST VALVE
EVO
EVC
SPO
SCAVENGING PORT
SPC

FIG.7

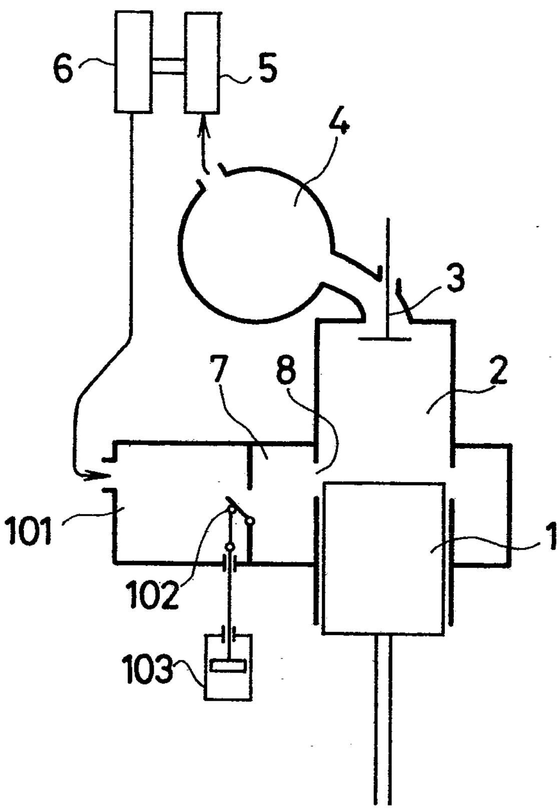


FIG.8

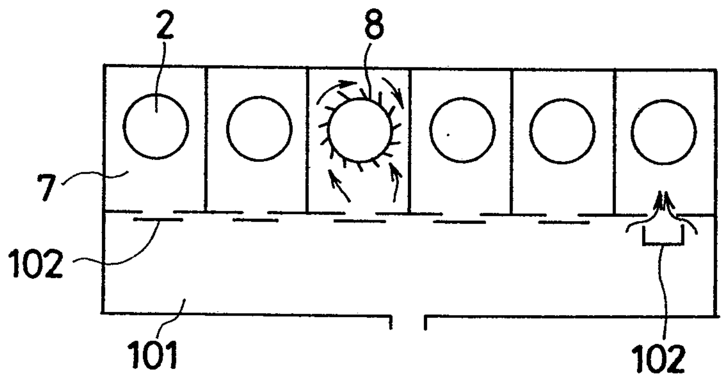
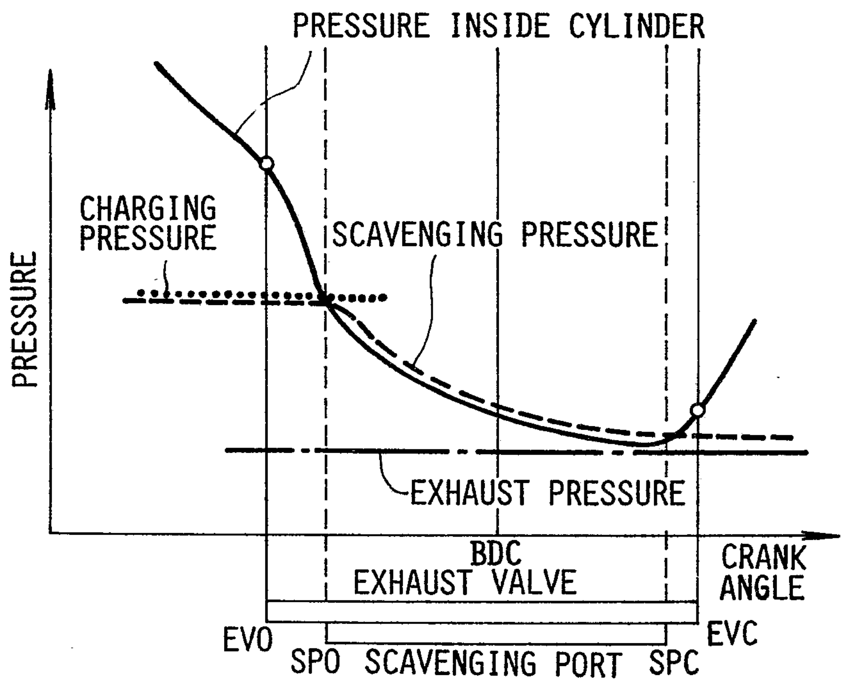


FIG.9


PRESSURE INSIDE CYLINDER
CHARGING PRESSURE
SCAVENGING PRESSURE
PRESSURE
EXHAUST PRESSURE
BDC
CRANK ANGLE
EXHAUST VALVE
EVO
EVC
SPO
SCAVENGING PORT
SPC

0060802

European Patent Office

EUROPEAN SEARCH REPORT

Application number

EP 82 73 0026

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int. Cl. 3)
A	GB-A-1 145 945 (SULZER) * page 1, lines 9-32; page 2, lines 11-52 * ---	1	F 02 B 33/44
A	GB-A- 737 340 (BURMEISTER & WAIN) * page 1, lines 10-26; page 3, lines 85-130; page 4, lines 1-2; 31-64 * ---	1	
A	FR-A-1 155 449 (S.O.B.E.M.) * page 1, left-hand column, paragraphs 1,3; right-hand column, last paragraph; page 2, left-hand column, paragraphs 1,2,3; right-hand column, last 2 paragraphs; page 3, left-hand column, paragraphs 1,2 * & US - A - 2 989 840 ---	1	
			TECHNICAL FIELDS SEARCHED (Int. Cl. 3)
A	CH-A- 342 407 (B.B.C.) * page 1, lines 36-55, 71-74; page 2, lines 1-20 * ---	1	F 02 B F 02 D
A	GB-A- 765 950 (SULZER) * page 1, lines 9-19; 92-94; page 2, lines 1-10; 91-100; page 3, lines 53-126 * -----	1	

The present search report has been drawn up for all claims

Place of search	Date of completion of the search	Examiner
THE HAGUE	17-06-1982	JORIS J.C.

CATEGORY OF CITED DOCUMENTS

X : particularly relevant if taken alone
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