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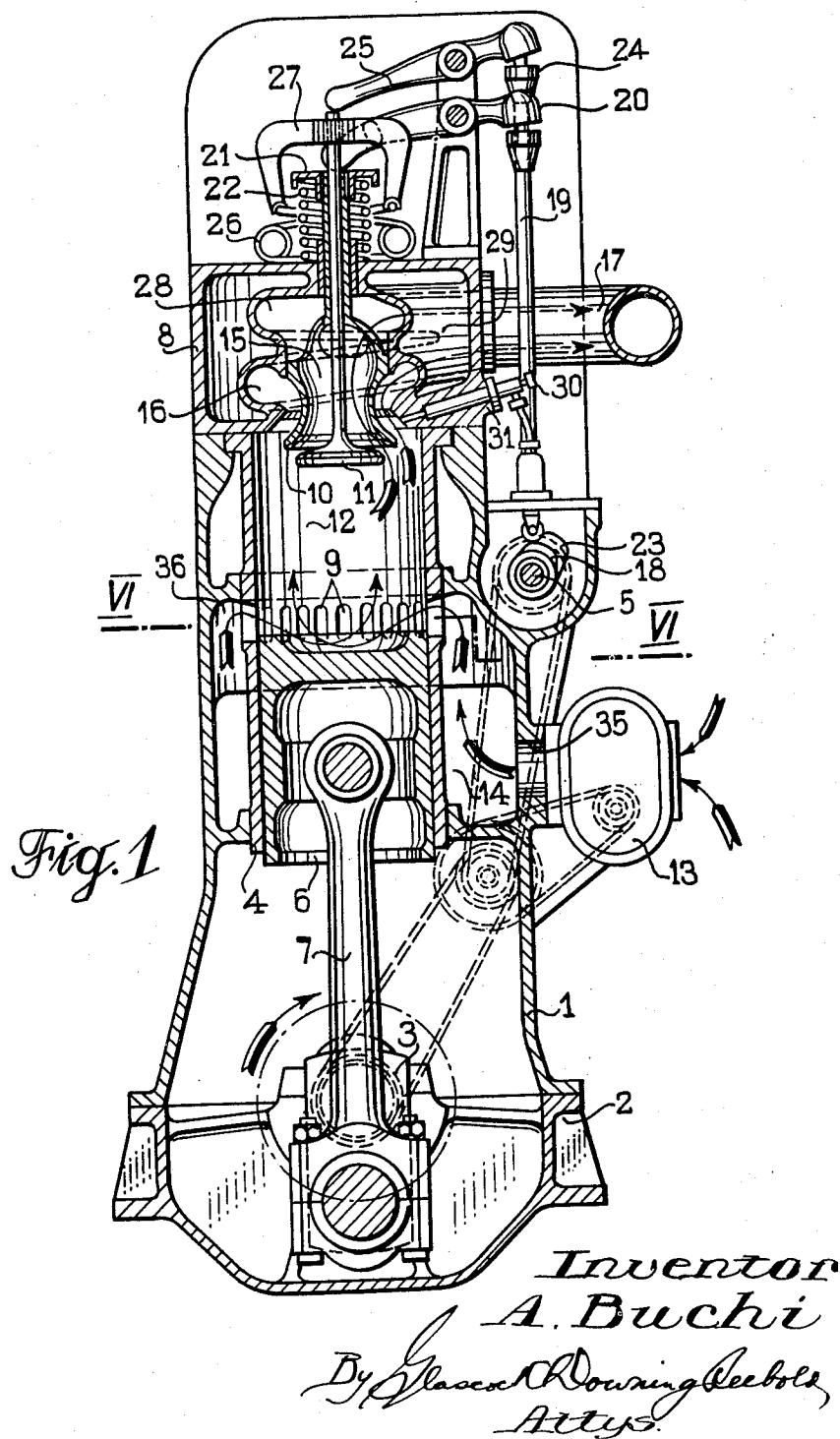
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TWO-STROKE INTERNAL COMBUSTION ENGINES

Filed Aug. 4, 1958

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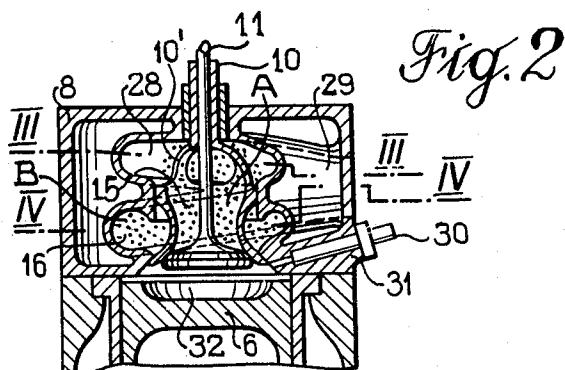
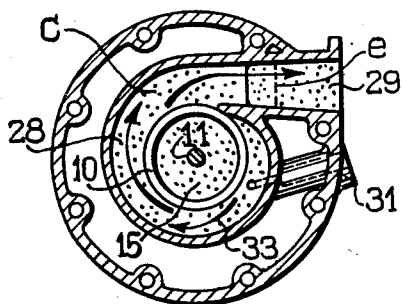
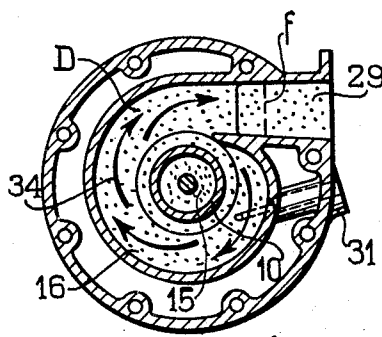
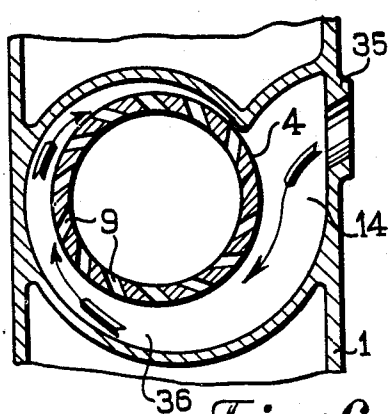
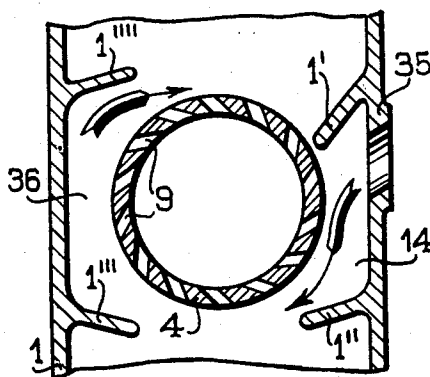
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TWO-STROKE INTERNAL COMBUSTION ENGINES

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4 Sheets-Sheet 2



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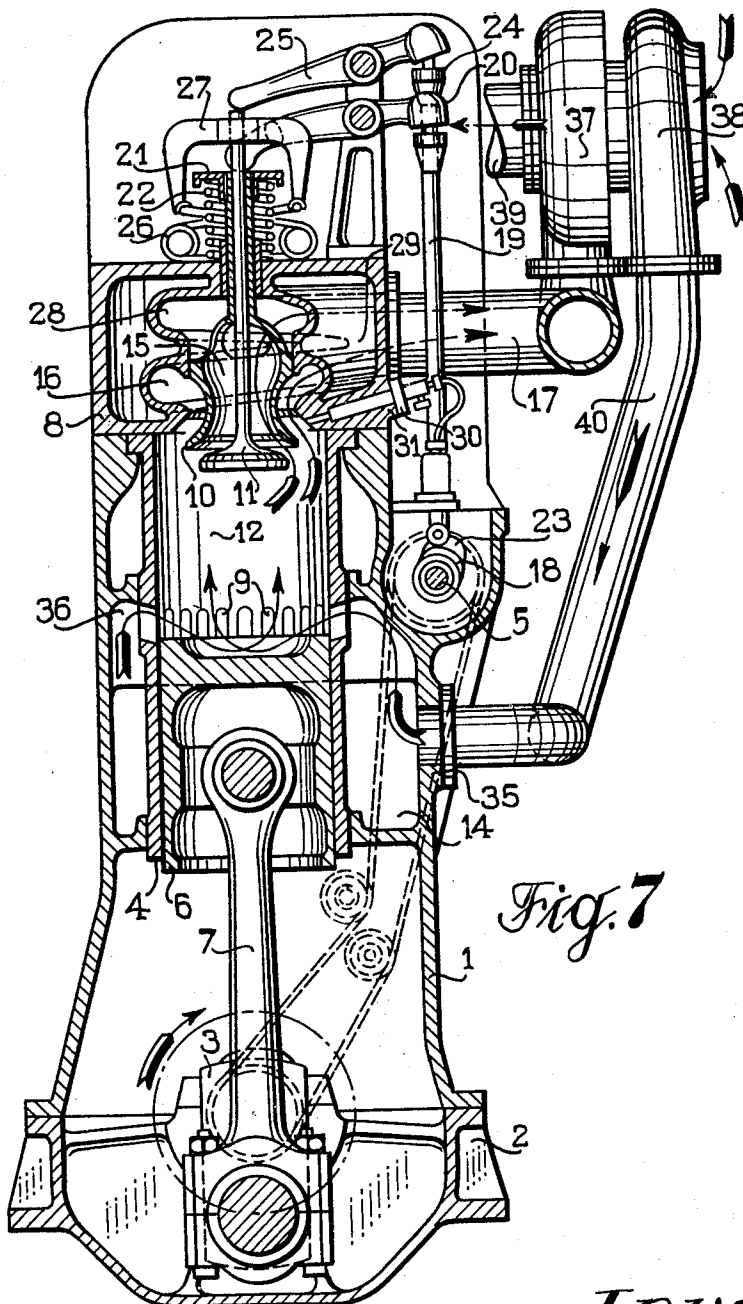
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TWO-STROKE INTERNAL COMBUSTION ENGINES

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4 Sheets-Sheet 3



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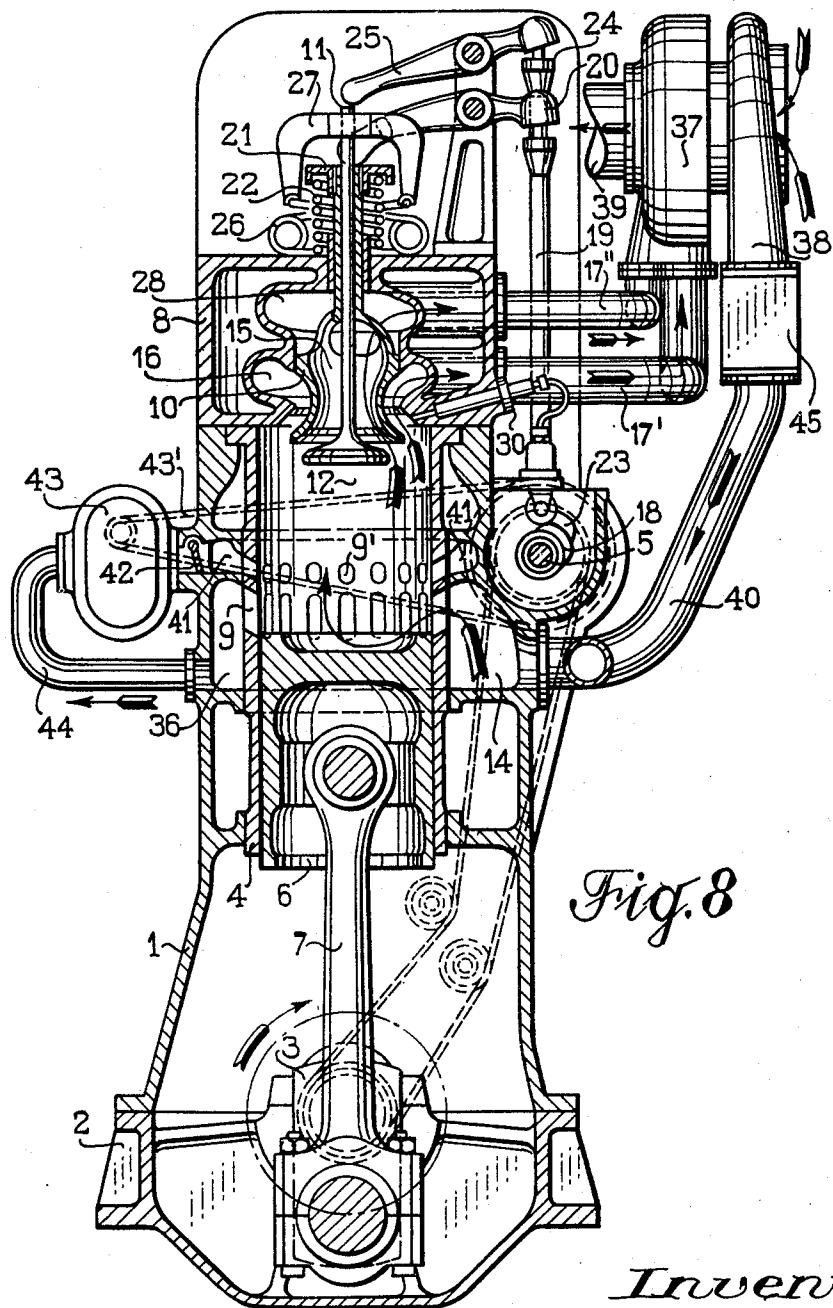
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TWO-STROKE INTERNAL COMBUSTION ENGINES

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4 Sheets-Sheet 4



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2,962,009

TWO-STROKE INTERNAL COMBUSTION ENGINES

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6 Claims. (Cl. 123—65)

This invention relates to two-stroke internal combustion engines wherein air inlet ports are provided near the dead center position of the piston and the invention is characterized in that the exhaust gases, and at least some of the scavenging air expelling the exhaust gases, leave the engine cylinder through at least two exhaust valves disposed concentrically one inside another in the cylinder head.

Preferably the combustion chamber comprises a recess in the crown of the piston.

It may be arranged so that the inner exhaust valve closes on the outer exhaust valve, and if required, both valves may be located substantially centrally in the cylinder head.

Practical applications of the present invention will now be described, by way of example only, with reference to the accompanying drawings whereof:

Fig. 1 is a longitudinal section of a two-stroke internal combustion engine according to the present invention the exhaust valves being open,

Fig. 2 is a fragmental sectional view showing the cylinder head and the exhaust valves closed,

Fig. 3 is a section on the line III—III of Fig. 2;

Fig. 4 is a section on the line IV—IV of Fig. 2;

Fig. 5 is a section on the line V—V of Fig. 1;

Fig. 6 is a view similar to Fig. 5 and showing an alternative construction;

Fig. 7 is a view similar to Fig. 1 showing an alternative construction of a two-stroke internal combustion engine according to this invention, and

Fig. 8 is a longitudinal section of yet another two-stroke internal combustion engine in accordance with the invention.

Referring to Fig. 1, the engine comprises a cylinder block 1, a baseplate 2, a crankshaft 3, a cylinder 12 having a cylinder liner 4, a cam-shaft 5, a piston 6, a connecting rod 7 and a cylinder head 8. The liner 4 is provided near the inner dead center position of the piston with inlet ports 9 and the cylinder head 8 has two concentric exhaust valves 10 and 11 and valve 11 moves inside valve 10. Scavenging and supercharging air is delivered to the cylinder 12 by a compressor 13 which is driven by the engine through a branch 35, chamber 14 and inlet ports 9. The exhaust gases leave the cylinder 12 between valves 10 and 11, enter space 15, pass through ports 10' in the wall of valve 10 and then enter chamber 28. The gases also leave the cylinder 12 between valve 10 and cylinder head 8 and enter chamber 16. From chambers 16 and 28 the exhaust gases flow through pipe 17 to atmosphere.

The exhaust valve 10 is actuated by an exhaust valve cam 18, push-rod 19, rocker arm 20, valve retainer 21 and spring 22, while the exhaust valve 11 is actuated by an exhaust valve cam 23, push rod 24, rocker arm 25, hairpin spring 26 and stirrup 27. Cams 18, 23 are on the common cam shaft 5.

Fig. 2 shows that the chamber 28 merges into a cham-

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ber 29 extending, as can be seen in Fig. 1, to the exhaust pipe 17.

The exhaust gases after passing through the ports 10' (Fig. 2) of valve 10 are delivered into a spiral part 5 of the chamber 28 which subsequently merges into the chamber 29.

The gases in the cylinder 12 and in chambers 16 and 28 rotate around the cylinder axis as indicated at 33, 34 respectively in Figs. 3 and 4.

A fuel injector 30 extends through a connector 31 in the cylinder head 8 and is directed towards a recess 32 in the crown of the piston 6. Preferably fuel is injected tangentially of cylinder 12 towards recess 32 in the direction of rotation of the gases in the cylinder and in the 15 chambers 16 and 28 i.e. as indicated by the arrows 33, 34 respectively in Figs. 3 and 4.

Fig. 5 shows the design of the inlet ports 9 in the cylinder liner 4. The ports 9 open tangentially into the cylinder chamber to impart to the air entering the cylinder a rotation preferably in the same direction as that indicated by the arrows 33 and 34 in Figs. 3 and 4. The wall of the cylinder block has webs 1', 1'', 1''' and 1'''' 20 so directed and shaped, and the entry branch 35 for the delivery of charging and scavenging air to the cylinder being so disposed that the rotation of the scavenging and charging air in the direction indicated is further assisted.

In the modified arrangement shown in Fig. 6, the scavenging and charging air enters the chamber 14 (Fig. 1) and passes around a chamber 36 having a spiral bounding wall before the same enters the inlet ports and only thereafter passes through the tangential ports 9 in the cylinder liner 4. This step is further to increase eddy- 25 ing in the cylinder.

In Fig. 6 the reference 35 denotes a flange about an opening for the scavenging and supercharging air delivered by the compressor 13 into the space 14 and thence into the space 36.

The only difference between the design shown in Fig. 7 and that shown in Fig. 1 is that in Fig. 7, the engine exhaust gases from the two exhaust valves are conveyed together into an exhaust gas turbo-compressor 37, 38. The exhaust gases pass from the exhaust gas conduit 17 into the turbine 37 and thence through conduit 39 to atmosphere or into a heat exchanger or the like. The scavenging and supercharging air passes from the compressor 38 through conduit 40 and connection 35 into the space 14, then into the chamber 36 and through the inlet ports 9 into the cylinder 12.

On the other hand, in the embodiment illustrated in Fig. 8, the gases (and possibly at least some of the scavenging air) are conveyed through the inner exhaust valve into the exhaust-gas conduit 17' separately to a turbine or stage of a turbine or to a segment of an exhaust gas turbine 37, while the gases or scavenging air which issue through the outer exhaust valve 10 and then through the annular chamber 16 flows through the exhaust-gas conduit 17' into another turbine or to another stage or into another entry segment of the turbine 37. In this embodiment, for instance, the inner exhaust valve 11 can be opened first and its exhaust gases can be delivered under by the higher pressure then prevailing in the cylinder through the chamber 28 to the turbine 37 or to a stage or to an entry nozzle segment of the turbine 37. The gases issuing through the outer exhaust valve can be conveyed subsequently, that is, when the gas pressure prevailing in the cylinder is lower, through the conduit 17' to another turbine or to another stage or to another nozzle entry segment of the turbine 37. The force required to open the inner exhaust valve is therefore less than if the two valves were to open simultaneously, and also the force for opening the outer exhaust 70

valve is reduced because, due to its later time of opening, the pressure in the cylinder 12 has already dropped, for instance, as low as the scavenging pressure.

Another feature of the design shown in Fig. 8 is that two rows of inlet ports 9 and 9' are provided in the cylinder liner 4. The bottom ports 9 serve mainly to scavenge the cylinder and the top ports 9' serve mainly to introduce the charge at a substantially increased pressure. Since the ports 9' must be closed as late as possible by the piston 6 for efficient auxiliary scavenging (i.e. when the exhausting operation has at least partly terminated) but since the exhausting operation begins while the ports 9' are still open, at least the entry chamber 41 to the ports 9' must be provided with non-return valve elements 42 to prevent exhaust gases from passing into the auxiliary scavenging compressor 43. The auxiliary scavenging ports 9' can be provided over only a part of the cylinder periphery but can open tangentially into the cylinder 12 to increase rotation therein. In the example illustrated, the auxiliary scavenging compressor is assumed to be a rotary blower 43 driven off the cam-shaft 5 through a chain 43' or by other means. Advantageously, this blower can draw air in from the space 14 or 36 before the inlet port 9 through the conduit 44, that is, it derives its air from the delivery conduit 40 of the turbo-compressor 38. Hence, the compressor 43 does not need a special suction device with air filter and sound attenuator, and when the turbo-compressor is running at high speeds the auxiliary supercharging pressure of the compressor 43 can be increased considerably above the scavenging air pressure. Of course, the delivery conduit of the compressor 38 can be provided with a supercharging air cooler 45, and a similar cooler (not shown) can be provided between the compressor 43 and the entry chamber 41 to the ports 9'. The compressor 43 can be of any other type, for instance, a centrifugal compressor. Through the bottom inlet ports 9 scavenging air can be supplied by a compressor driven mechanically or by other means, and through the top ports 9' substantially only scavenging air can be supplied by an exhaust gas turbo-compressor. The scavenging ports 9, auxiliary supercharging ports 9' and their respective supply chambers 36 and 41 can be so shaped that the air entering the cylinder is directed towards the recess in the piston in order to scavenge the piston combustion chamber satisfactorily and to cool the piston crown. The control of the exhaust valves can be so designed that they either open simultaneously or, and preferably, the inner exhaust valve opens before the outer exhaust valve. The design can also be such that the inner valve closes on the outer valve only when the outer valve has already been closed.

In Fig. 2 the lines A and B indicate the scavenging and filling of at least the spaces 15 and 16 downstream of the exhaust valves 10 and 11 when the same are closed. In Figs. 3 and 4 the lines C and D indicate the filling of the chambers 28 and 16 in the outlet conduit downstream of the valves. The lines e and f in Figs. 3 and 4 respectively denote that the exhaust conduits 29 are scavenged and filled with cool air. If there is a return of exhaust gases, for instance, from a turbine, at least the spaces 15 and 16 between and outside the exhaust valves 10 and 11 and the exhaust conduits 28, 29 (which have been scavenged with relatively cool scavenging air) remain filled with such air during the compression, combustion and expansion phases, for instance, as far as the line indicated at e.

With the arrangements described, the cylinder head is of simple design while the flow of exhaust gases from the cylinder, the admission of scavenging and charging air to the cylinder and the flow of gases through the cylinder is symmetrical of the cylinder axis. The charging and scavenging air flow centrally of the cylinder and around and through the exhaust valves.

Since the valves close seriatim a large area of dis-

charge opening can be provided without requiring large forces to open the valves.

The inner exhaust valve may readily be arranged to have a smaller surface area exposed to the pressure of the gases in the cylinder than the corresponding surface area of the outer valve. Consequently the force required to open the inner valve will be correspondingly reduced and since the inner valve is opened first against the higher pressure then prevailing in the cylinder the force to open the inner valve may be less than for the outer valve while maintaining the force to open the larger outer valve below a specified limit because the pressure in the cylinder when the outer valve opens is less than when the inner, smaller exhaust valve is opened.

With the concentric arrangement of two or more valves, the weight of each valve will be less than the weight of a comparable single exhaust valve. The area of each valve which is exposed to the pressure in the cylinder is also less than for a single exhaust valve and as a result, the forces to open each of the valves according to this invention is reduced as compared with a single valve. Again, the scavenging air will flow around both valves so that more effective cooling thereof is assured. By suitable selection of the area of the discharge opening for each valve, the relative flows of scavenging air around each valve can be varied with consequent effect on cooling and adequacy of scavenging having regard to the order in which the valves are opened.

The provision of means for producing rotation of the gases in the cylinder and a rotation of the gases downstream of at least one of the exhaust valves ensures uniformity of gas flow in the cylinder in the region of the exhaust valves (i.e. upstream and downstream of the valves) and the rotating gases are advantageous for exhausting, scavenging and supercharging, for cooling the walls over which the scavenging air flows and for dual distribution and atomization.

The quantity of scavenging air delivered to the cylinder is sufficient not only to scavenge the cylinder but also between and around the exhaust valves as far as the connection to the exhaust-gas conduit from the cylinder head with the consequence that the spaces around the valves, and along to the conduit remain filled with relatively cool air during the following compression, combustion and expansion phases of the engine. If the quantity of scavenging air is sufficient, the further advantage is attained that the air remaining in said spaces is sufficient to ensure that the exhaust gases from the conduit can not re-enter the spaces around the valves.

The provision of the recess in the crown of the piston reduces heat loss as compared with a recess in the cylinder head. Moreover the scavenging and cooling air being directed into the piston recess assures scavenging of the recess.

By providing a pair of exhaust valves, the exhaust gases for each valve may be conveyed by a separate exhaust-gas-conduit (one for each valve) for delivery to a pair of turbines or to two separate stages or segments of a single turbine. However, without material modification of the engine, the gases from both valves can enter a single exhaust-gas conduit.

The fuel distribution in the combustion chamber is adequately ensured since the fuel is injected tangentially of the cylinder within which the scavenging and charging air rotates so that satisfactory combustion results with relatively large fuel nozzles and low injection pressures. The fuel is injected towards the recess in the crown of the piston so that the stream of injected fuel is not directed against the cooled walls of the combustion chamber, e.g. the cylinder wall and head.

When the scavenging air and charging air is introduced through separate inlet ports the two supplies of air may be delivered from separate compressors so that a part of the delivery of one compressor is supplied to the other compressor whereby the delivery pressure of the second

compressor is increased without increasing (or even when reducing) the size of the second compressor.

The provision of two compressors enables an increased quantity of charging and scavenging air to be supplied readily to the engine at high loads while ensuring sufficient air for combustion at starting and low loads. This is especially so when a mechanically driven compressor and a turbo-compressor are provided.

What I claim:

1. In a two-stroke internal combustion engine, a cylinder, a piston means reciprocable in the cylinder, a cylinder head, said cylinder having air inlet ports located adjacent the inlet dead center position of the piston means, an exhaust pipe means for said cylinder, duct means in the cylinder head leading from the cylinder to the exhaust pipe means and including an outlet into said pipe means, first valve means in the cylinder head between the cylinder and the duct means operative to discharge high pressure exhaust gases concentrically of the cylinder into the duct means, said air inlet ports being such that upon movement of the piston to inlet dead center position, air in an amount in excess of that required to scavenge flows into said cylinder, and second valve means in the cylinder head concentrically arranged respecting said first valve means and operative in timed sequence relative to said first valve means so as to discharge the remaining exhaust gases in said cylinder, and air in a quantity sufficient to build up air in said duct means upstream of said outlet to prevent exhausted gases in the pipe means entering said duct means.

2. A two-stroke internal combustion engine as claimed in claim 1, in which said air inlet ports introduce the air tangentially of the said cylinder and said duct means including wall means defining a chamber, and said outlet extending tangentially relative to said chamber to impart rotational movement to the gases in the same direction as the air introduced into the cylinder.

3. A two-stroke internal combustion engine as claimed in claim 1, in which said outlet is disposed eccentrically with respect to the axis of said first valve means to increase the rotation of the said exhaust gases in said duct means.

4. A two-stroke internal combustion engine as claimed in claim 1, in which the piston means is provided with a recess in the crown thereof and the air inlet portions direct the inflowing air towards said crown.

5. A two-stroke internal combustion engine as claimed in claim 1, in which said duct means includes wall means defining two superposed chambers adapted to communicate with said first and second valves, respectively and said outlet extending tangentially to and communicating with said chambers for imparting rotational movement to the gases and air in said chambers.

6. In a two-stroke internal combustion engine, at least

one cylinder, a piston means reciprocable in the cylinder, a cylinder head, said cylinder having air inlet ports located adjacent the inlet dead center position of the piston means, the cylinder head having a circular opening therein above and concentric with the axis of the cylinder with the circumference of the opening including means defining a seat, means mounted in the cylinder head for axial movement toward and away from the cylinder to open and close said opening, said last-named means including a tubular outer valve component and an inner valve component mounted concentrically of the outer valve component with said outer and inner valve components constituting an exhaust valve, said tubular outer valve component having a perimetric portion adapted to rest on said seat and further having a seat thereon for the inner valve component, means defining superposed chambers in the cylinder head surrounding the outer and inner valve components, means defining communication paths between the inner valve component and the uppermost of said chambers and the outer valve component and the lowermost of said chambers, respectively, and at least one outlet extending tangentially relative to and communicating with said chambers, an exhaust pipe means to which said outlet leads, and means operably connected with said inner and outer valve components for opening said valve components in timed sequence whereby said inner valve component moves from its seat on the outer valve component while the outer valve component is on its seat in the cylinder head to discharge high pressure gases from the cylinder between the valve components into the uppermost chamber and thereafter the outer valve component moves from its seat to discharge the remaining gases and sufficient air into the lowermost chamber so that at least the outlet is filled with air to prevent gases exhausted from said cylinder into exhaust pipe means entering said outlet.

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