

March 7, 1939.

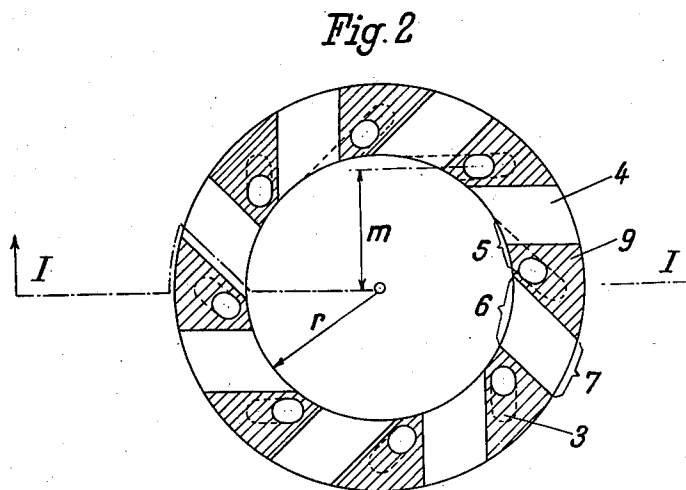
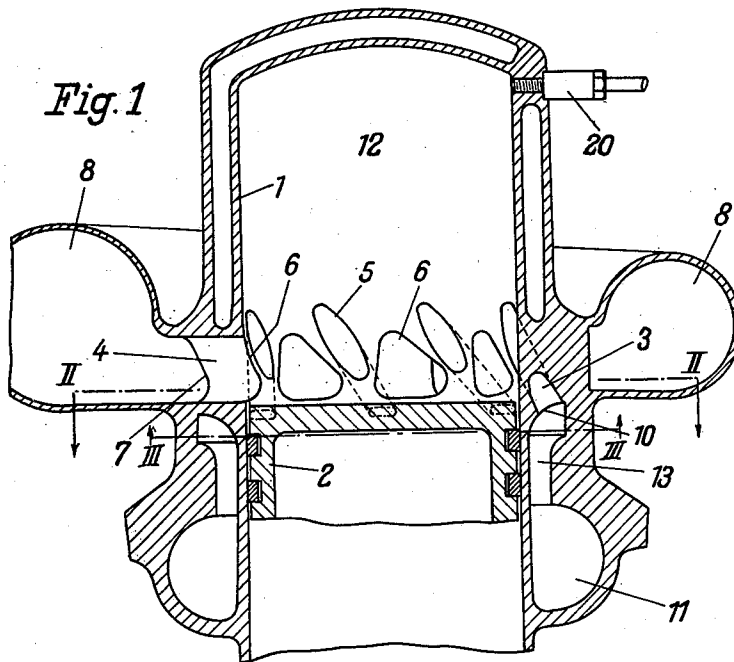
A. SCHEIBE

2,149,793

INTERNAL COMBUSTION ENGINE

Filed Sept. 29, 1936

2 Sheets-Sheet 1



Inventor:
ALFRED SCHEIBE
BY *Karl Kisch*
ATTY

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Fig. 4.

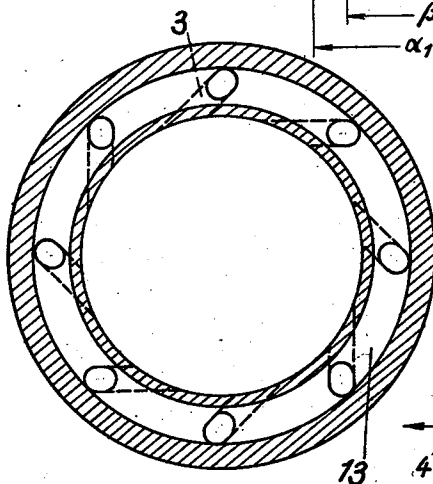
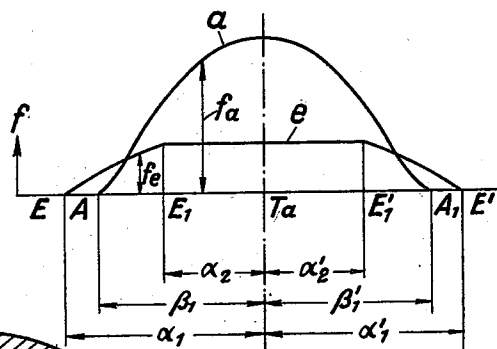
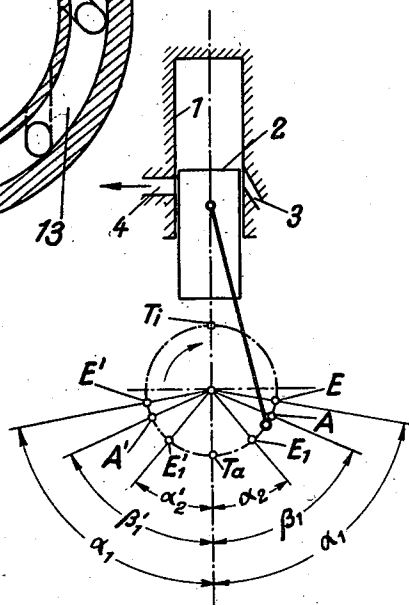


Fig. 3.

Fig. 5.



Inventor:

ALFRED SCHEIBE
By *Karl H. Schaefer*
ATTY

UNITED STATES PATENT OFFICE

2,149,793

INTERNAL COMBUSTION ENGINE

Alfred Scheibe, Dessau-Ziebigk, Germany, assign-
or to Junkers Flugzeug-und-Motorenwerke
A. G., Dessau, Germany

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10 Claims. (Cl. 123—65)

My invention relates to internal combustion engines, more especially of the single-piston two-cycle type. The invention is particularly concerned with the means whereby the fresh gases
5 are fed to and the gases of combustion exhausted from the cylinder of the engine.

It is an object of my invention to provide an arrangement of the intake and exhaust ports which results in a higher efficiency of the engine
10 than similar arrangements hitherto suggested.

In single-piston two-cycle engines as a rule two groups of ports are arranged in the wall of the engine cylinder, which are controlled by the piston. One of these groups of ports serves for
15 introducing the fresh gases under a certain pressure above atmospheric, while the other serves for exhausting the gases of combustion. As a rule the exhaust ports extend somewhat farther, in the direction of stroke, towards the combustion chamber than the intake ports, so that when the piston executes its outward stroke, it will first
20 uncover the exhaust ports. In consequence thereof, until the intake ports are uncovered, a great part of the total quantity of exhaust gases can escape from the combustion chamber and the pressure of the exhaust gases still left in the cylinder therefore decreases quickly, so that on the intake ports being uncovered, there
25 is no danger that exhaust gases might enter the fresh gas supply. On its inward stroke the piston first closes the intake ports and only thereafter the exhaust ports. In the period of time between the closing of the intake and the exhaust ports the combustion chamber still communicates with the exhaust pipe through the
30 still open exhaust ports. This involves the drawback that on the exhaust ports being closed and compression started, the fresh gases in the working cylinder cannot be acted upon by a pressure higher than the pressure prevailing in the exhaust pipe, so that it is not possible to simply
40 charge up the cylinder.

My invention is based on the conception that a faultless operation of two-cycle engines with
45 a recharging of the working cylinder is possible also in the case where merely intake and exhaust ports controlled by the working piston are provided and the intake ports extend farther towards the dead space than the exhaust ports. In order
50 to render this possible, it is necessary that after the exhaust ports have started to open, the exhaust gases expand very quickly and that the fresh gases enter the working space with high velocity. To attain this end, I form the exhaust
55 conduits with a far greater total cross-sectional

area of passage and preferably with several times a cross-sectional area, for instance the two- to fourfold, than the intake ports. These intake ports are of uniform shape and size and are so
5 designed as to form simple and plain paths of flow in both directions, containing no members, such as nozzle inserts, valves etc., which might form a hindrance to the gas flow in one direction.

In the drawings affixed to the specification and forming part thereof an embodiment of my invention is illustrated diagrammatically by way
10 of example.

In the drawings—

Fig. 1 is an axial section on the line I—I in Fig. 2 and
15 Fig. 2 is a cross-section on the line II—II in Fig. 1.

Fig. 3 is another cross-section on the line III—III in Fig. 1.

Fig. 4 is a diagram illustrating the size of the entrance and exhaust surfaces uncovered by the
20 piston on its working and return stroke, while

Fig. 5 is the corresponding crank gear diagram.

Referring to the drawings and first to Figs. 1, 2 and 3, 1 is the inner wall of the engine cylinder and 3 and 4 are the intake and exhaust conduits, respectively, ending in the inner wall of the cylinder near the outer end uncovered by the piston during its return stroke. The inner ends
25 of these conduits form the intake and exhaust ports 5 and 6, respectively, all of which are controlled by the piston 2, exclusively. The intake conduits 3 have the form of simple plain borings, the axes of which cross the cylinder axis in oblique direction, being spaced from it by a distance m which is smaller than the radius r of the cylinder cavity, so that the fresh gas enters the cylinder space in about a tangential direction and with a velocity component directed towards the dead space. The exhaust conduits 4
30 are simple perforations provided in the cylinder wall, which lead to a receiver 8 encircling the cylinder. The borings forming the intake conduits 3 extend across the parts or bridges 9 of the cylinder wall between the exhaust conduits 4 and their openings 10, which communicate with the fresh gas receiver 11, are spaced farther from the combustion chamber 12 than the outer openings 7 of the exhaust conduits 4 in the exhaust
35 gas receiver 8, while their openings (inlet openings 5) in the inner wall 1 of the cylinder extend farther towards the combustion chamber 12 than the openings 6 of the exhaust conduits 4. Thus the inlet openings 5 are uncovered by
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the working piston 2 on its outward stroke earlier and on its inward stroke are closed later than the exhaust openings 6.

This arrangement of the intake and exhaust conduits offers particular advantages. The arrangement of the intake conduits in the bridges left between the exhaust conduits allows, the fresh gases entering through the intake conduits to exert a cooling effect of the bridges, the outer walls of which are acted upon by the hot exhaust gases. The alternating arrangement of intake and exhaust conduits causes the working cylinder to assume, during the operation of the engine, the same temperature all over its circumference, in contradistinction to the usually adopted arrangement of the intake conduits on one and the exhaust conduits on the other side, whereby the temperatures are unequally distributed, so that there arises the danger of heat stresses and unduly great changes of form. Since the intake ports 5 are partly situated between the exhaust ports 6, the space available in the inner cylinder wall for the accommodation of the ports is particularly well utilized, more especially in connection with the contour shown in Fig. 1 of the exhaust ports which corresponds substantially to a triangle with strongly rounded corners. Here the magnitude of the cross-sectional areas of passage of the exhaust conduits 4 is several times that of the cross-sectional area of passage of the intake conduits 3. The peculiar design and arrangement of the intake conduits further allows the fresh gases traversing these conduits with high velocity to enter the cylinder in a uniform circumferential distribution and substantially in tangential direction, however with a velocity component directed axially to the combustion chamber, so that these fresh gases after leaving the conduits, move towards the combustion chamber under the form of a multiple thread screw line extending in close proximity to the inner cylinder wall. In consequence thereof the outer zone of the cylinder chamber is first filled with fresh gases and the remainder of the exhaust gases is forced towards the exhaust ports through the central part of the cylinder space. When the piston has reached its outer dead center position, the ratio of the uncovered surfaces of the exhaust and intake ports is greatest and the pressure in the cylinder has then reached its minimum. During the inward stroke of the piston the pressure then rises rather quickly since the exhaust ports are gradually covered, while the fresh gas supply at first continues as before and continues also after the exhaust ports have been closed, until ultimately the piston has also closed the intake ports. The interior of the cylinder is now filled with a body of fresh gases which owing to its inertia continues circulating therein and is placed under increased initial pressure. This high circulation velocity of the cylinder charge is particularly advantageous in internal combustion engines, in which the dead space forming a direct extension of the working chamber has the form of a plain revolution space devoid of any secondary spaces, while the fuel is injected, for instance by means of an injection nozzle 20, directly into the dead space, since under these circumstances the air of combustion is carried into particularly intimate contact with the injected fuel.

In the embodiment of my invention illustrated in the drawings a conduit 13 of annular cross-section is provided between the main fresh gas receiver 11 and the outer openings 10 of the intake

conduits 3. This conduit 13 has a comparatively small cross-section and a length such that the total passage of gas flow from the fresh gas receiver 11 up to the intake conduit openings 5 in the inner wall of the cylinder is considerably longer than the length of the exhaust conduits 4 which should be as short as possible. I thus obtain that the exhaust gases which, on the intake openings being uncovered, enter the intake conduits 3, can only enter the narrow connecting conduit 13, however not the main fresh gas receiver 11, so that they can substantially only displace the fresh gases in these narrow conduits 3 and 13 without being able to mix with them in the large main receiver 11.

In the diagram of Fig. 4 the surfaces f , uncovered during the working and return stroke of the piston 2, of the intake and exhaust openings are plotted in dependency from the crank angle. In the point E and at an angular distance α_1 from the outer dead center point T_a begins the uncovering of the intake ports 5 during the working stroke of the piston. The uncovered surface of these ports, which is represented by the ordinates f_a of the line e , gradually increases until in the point E_1 at an angular distance α_2 from the outer dead center point T_a the intake ports are uncovered altogether. They remain altogether open while the piston continues on its outward stroke up to the dead center point T_a and subsequently returns until it reaches the point E'_1 at an angular distance α'_2 from the dead center point T_a . From the point E'_1 on the free surface of the intake ports is again gradually reduced by the inwardly travelling piston until it has reached the point E' at an angular distance α'_1 from the outer dead center point T_a , at which the intake ports 5 are altogether covered by the piston.

The uncovering of the exhaust ports 6 starts in the point A at an angular distance β_1 before the dead center point T_a , which distance is smaller than the angular distance α_1 , but greater than the angular distance α_2 . This uncovering continues up to the dead center point T_a and after this point has been overstepped, the exhaust ports are again gradually covered by the piston, until, in the point A' at an angular distance β'_1 from the dead center point T_a they are covered altogether.

The surfaces of the exhaust ports (represented by the ordinates f_a of the line a) which are uncovered at any particular moment, are considerably larger over the greatest part of the exhaust angle $\beta_1 + \beta'_1$ than the uncovered surfaces f_a of the intake ports. There thus results a very quick increase of the cross-sectional areas of exhaust f_a from the uncovering point A on, so that now the exhaust gases can escape very quickly from the cylinder, while shortly thereafter (in the point E_1) the intake ports are already uncovered altogether, so that also the scavenging and charging can now take place freely.

Tests made with an engine of this kind have shown that the arrangement of the intake and exhaust ports above described results in particularly favorable conditions of operation. This is in contrast to the apparently obvious danger that, if the intake ports are already uncovered before the exhaust gases have started escaping, a considerable quantity of these latter gases, being still acted upon by a considerable pressure, would enter the fresh gas receiver and that in consequence thereof the engine would develop only a relatively low output. That this does not happen, can be explained as follows: When be-

ginning to uncover the intake ports, the piston still travels at a relatively high velocity and at the same velocity the volume of the working chamber increases; on the other hand the piston at first only uncovers small parts of the surfaces of the intake ports. Thus the bulk of the exhaust gases at first remains in the working chamber and only a small part passes into the fresh gas supply through the still partly closed intake ports.

On continuing its stroke the piston then also uncovers the exhaust conduits and since the cross-sectional area of passage of these ports is considerably greater than that of the intake conduits, the exhaust gases in the cylinder will now expand quickly and consequently scavenging and recharging with the fresh gases can now follow. The comparatively small cross-sectional area of the intake conduits results in a high velocity of the inflowing fresh gases, which in this case is favorable, since a fast flowing current of scavenging gas will not be dispersed as easily as a slowly flowing one, but will penetrate the working chamber up to the inner end and will thus scavenge the exhaust gases very completely. The holding together of the scavenging gas current is favored also by the fact that the intake conduits open in the cylinder wall in about tangential direction so that the scavenging gas current will flow in contact with the cylinder wall.

This inventive conception is of particular importance for the provision of powerful high speed two-cycle single-piston engines, for the particular means, which were hitherto believed to be indispensable for enabling such engines to be recharged, formed a hindrance to the design of high speed engines, since, in contradistinction to the control members in a four-cycle engine, they are required to move in synchronism with the piston. Therefore the two-cycle engine provided merely with piston-controlled intake and exhaust ports is the ideal high speed motor as far as construction from the designers point of view is concerned. Hitherto, however, nobody had succeeded to increase the output per unit of stroke-volume in the two-cycle engine to a similar extent as has long been attained in high speed four-cycle engines by pre-compressing the charge.

The present invention therefore provides a solution of the important problem of providing an engine which in view of the particularly great simplicity of design is excellently suited for high speed operation and which at the same time allows recharging, which results in a particularly high output per unit of stroke-volume.

I wish it to be understood that I do not desire to be limited to the exact details of construction shown and described for obvious modifications will occur to a person skilled in the art.

I claim:

1. In an internal combustion engine of the single-piston two-cycle type the combination of a cylinder and a piston reciprocable in said cylinder, the wall of said cylinder being formed

near its outer end, uncovered by the piston on its outward stroke, with intake and exhaust ports, each kind of ports being arranged in a circle, all said intake ports being substantially alike and having the form of simple plain passages in both directions of gas flow, their openings in the inner wall surface of said cylinder extending farther in the direction towards the dead space than those of the exhaust ports, the exhaust ports having a total cross-sectional area of passage considerably exceeding that of the intake ports, said piston being the sole means controlling the passage of air and gas through said ports.

2. The engine of claim 1, in which the circles of intake and exhaust ports are so arranged that the inner openings of the intake ports extend into the spaces between the inner openings of the exhaust ports.

3. The engine of claim 1, in which an intake port is arranged between each pair of adjoining exhaust ports.

4. The engine of claim 1, in which the intake ports extend through the bridges left between the pairs of adjoining exhaust ports.

5. The engine of claim 1, in which the inner openings of the intake ports end short of the part of the cylinder wall uncovered by the piston at the end of its outward stroke, while extending into the part of the cylinder wall situated between the pairs of adjoining exhaust ports.

6. The engine of claim 1, in which the straight intake ports extend substantially tangentially to the cylindrical space in the cylinder and obliquely to the cylinder axis with their openings in the inner cylinder wall in closer proximity to the combustion chamber than their openings in the outer cylinder wall.

7. The engine of claim 1, in which the intake and the exhaust ports cross each other within the cylinder wall with the openings of the exhaust ports in the outer wall surface located in closer axial proximity to the combustion chamber than the openings of the intake ports in said outer wall surface.

8. The engine of claim 1, comprising an exhaust gas receiver and a fresh gas receiver, both encircling the cylinder, the exhaust gas receiver being located in closer axial proximity to the combustion chamber than the fresh gas receiver.

9. The engine of claim 1, comprising an exhaust gas receiver and a fresh gas receiver, both encircling the cylinder, the exhaust gas receiver being spaced from the inner exhaust port openings as little as possible, while the inner intake port openings are spaced considerably farther from the fresh gas receiver.

10. The engine of claim 1, in which a connecting conduit of annular cross-section having a relatively small cross-sectional area of passage is inserted between the fresh gas receiver and the outer openings of the intake ports.

ALFRED SCHEIBE.