Sept. 24, 1946.

E. S. L. BEALE

2,408,030

TWO STROKE PISTON-CONTROLLED ENGINE

Filed June 30, 1942

2 Sheets-Sheet 1

Fig. 1.

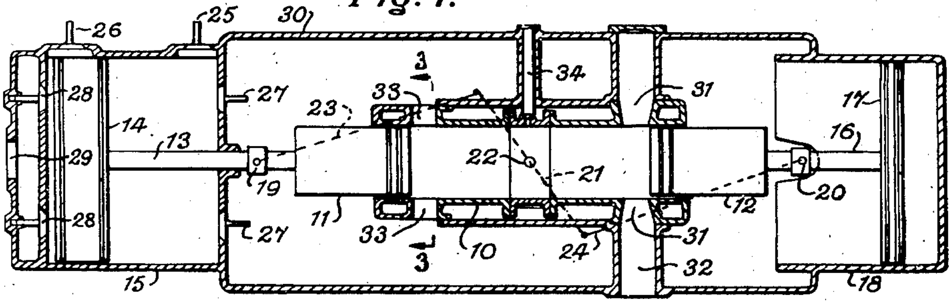


Fig. 3.

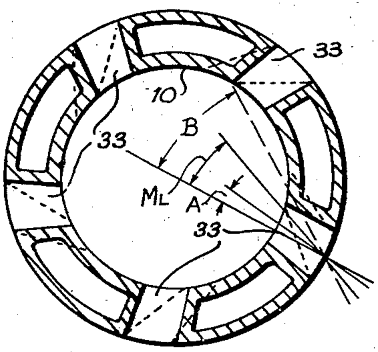


Fig. 2.

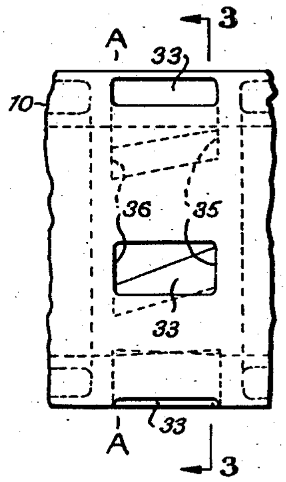


Fig. 4.

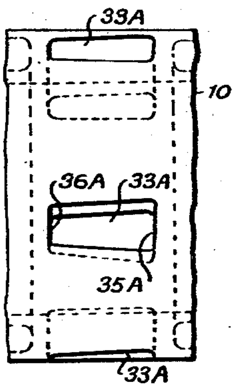
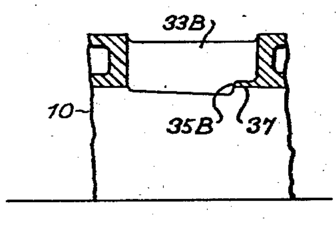


Fig. 5.



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TWO STROKE PISTON-CONTROLLED ENGINE

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Fig. 6.

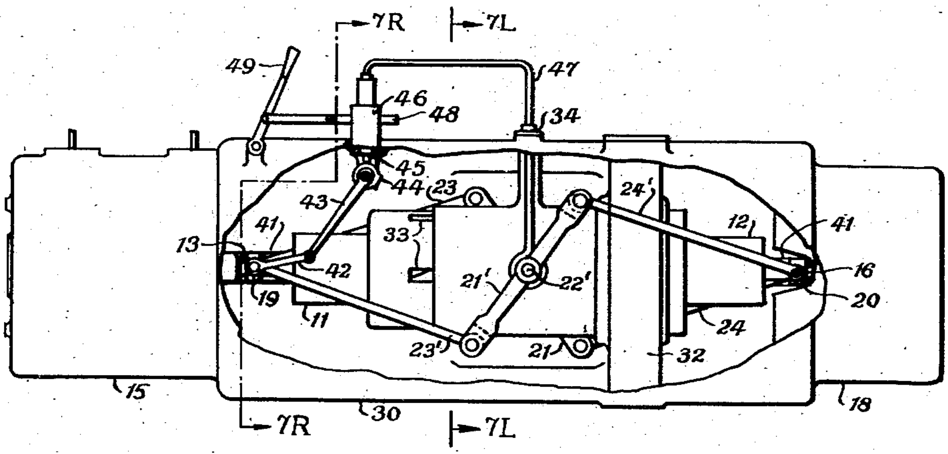


Fig. 7.

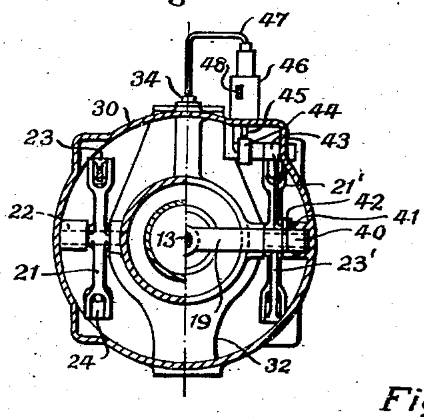


Fig. 8.

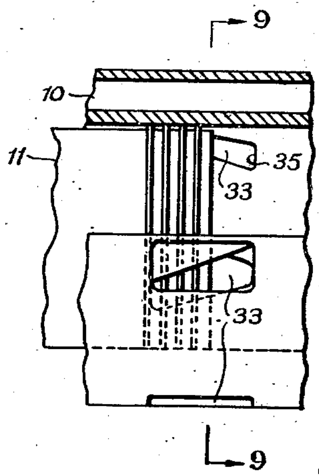
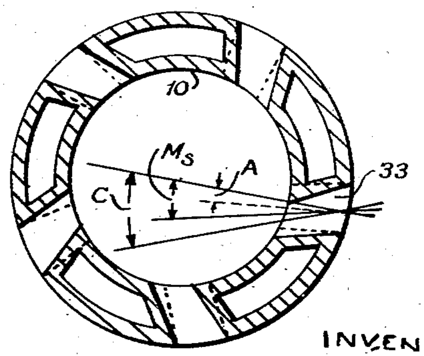


Fig. 9.



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UNITED STATES PATENT OFFICE

2,408,030

TWO-STROKE PISTON-CONTROLLED ENGINE

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Application June 30, 1942, Serial No. 449,078
In Great Britain June 26, 1941

8 Claims. (Cl. 123—51)

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The present invention relates to control of the rate of swirl of the charge in the working cylinder of a two-stroke internal-combustion engine of the kind in which the charge is admitted through one or more ports controlled by a working piston, and in which the position occupied by this piston at the end of its "out" stroke and in consequence the maximum depth of port openings are variable. The invention is especially but not exclusively concerned with compression-ignition free-piston engines in which the stroke of the piston varies with the load.

With internal-combustion compression-ignition engines in which the air charge to the working cylinder is admitted through piston-controlled ports cut in the cylinder walls, it is well known that if so-called "tangential" ports are used, that is ports the directions of which have components tangential to the cylinder, the air charge in the cylinder will be given a rotary movement. This rotary movement is generally known as swirl and will persist throughout the working cycle; when a suitable rate of swirl is obtained a considerable improvement occurs in the combustion of the fuel.

The rate of swirl is controlled by the velocity of the air charge through the inlet ports, and by the angle which the ports make to the radial direction. In the ordinary crankshaft engine, where the piston stroke is constant, the air charge is also constant over the whole range of load. Thus at any given engine speed the rate of swirl of the air charge is constant for all loads, since the same volume of air passes through the same ports in a given time.

In the case of a free-piston engine the piston stroke is not mechanically limited and varies with the load, and in particular there is a substantial variation of the position of the pistons at the end of their "out" strokes. With such engines it is desirable to obtain swirl of the air charge, and also it is desirable for this swirl to be nearly constant for all running conditions and independent of the position of the piston at the end of its stroke.

As the area of port opening available for the entry of the air charge into the cylinder is controlled by the piston, this effective port area is variable and dependent on the position of the piston at the end of its "out" stroke. The quantity of scavenge air available may also be variable and dependent on the stroke of the piston.

The rate of swirl in the cylinder is nearly proportional to the velocity of the air passing through the ports and also increases as the angle of the

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ports to the radial direction is increased, i. e. as the ports are directed away from the centre of the cylinder. If the ports are shaped in the usual way, that is with a constant angle along the whole of their depth, the velocity of the air passing through the ports is reduced as the area of port opening is increased, i. e. as the stroke of the piston is increased, and the rate of swirl decreases correspondingly.

According to this invention, in an engine of the kind specified, the shape of said port or ports is so varied along their depth that their mean directional effect on the charge flowing through them into the working cylinder varies with the depth of port opening in such a manner as to compensate, at least in part, for variation in the rate of swirl resulting from the change in the area of port opening due to variation in the position of said piston at the end of its "out" stroke. Thus the angle of said port or ports to the radial direction may be increased along their depth in the direction of the "out" stroke. Alternatively, or in addition the width of said port or ports may be decreased along their depth in the direction of the "out" stroke.

Embodiments of the invention will be described by way of example, as applied to an air-compressor of the free-piston type, with reference to the accompanying diagrammatic drawings, in which:

Fig. 1 is a sectional elevation of the compressor,

Fig. 2 is an elevation of a part of the apparatus shown in Fig. 1, to a larger scale,

Fig. 3 is a section on the line 3—3 in Figs. 1 and 2,

Fig. 4 is an elevation corresponding to Fig. 2, but showing a modified construction,

Fig. 5 is a section of a further modification,

Fig. 6 is a side elevation of the compressor shown in Fig. 1, with part of the casing broken away,

Fig. 7 is a section, the left-hand half being on the line 7L—7L and the right-hand half on the line 7R—7R in Fig. 6,

Fig. 8 is a part-sectional elevation of the part of the working cylinder shown in Fig. 1, with a working piston in it, and

Fig. 9 is a section on the line 9—9 in Fig. 8.

The air compressor shown in Fig. 1 has a working cylinder 10 in which operate two opposed working pistons 11 and 12. The piston 11 is connected by a rod 13 to a compressor piston 14 operating in a compressor cylinder 15. The piston 12 is connected by a rod 16 to a piston 17 operating in a cylinder 18. The rods 13 and 16 are rigid respectively with cross pins 19 and 20 of the synchronizing linkage which constrains the two piston assemblies 11, 14 and 12, 17 to move at equal speeds in opposite directions. This linkage includes at one side of the compressor a lever 21 pivoted about a transverse axis at 22 and pivotally connected, at its ends respectively, by rods 23 and 24 to the pivot pins 19 and 20. A similar lever 21, pivoted on a pin 22′, and pair of rods 23′ and 24′ (Figs. 6 and 7) are provided at the other side of the compressor, being so disposed that the two levers 21 and 21′ move oppositely.

On the in strokes of the working pistons 11 and 12 the compressor piston 14 draws in air through inlet valves such as 26 to the space on its left and at the same time delivers air on its right through delivery valves 27 to a scavenge-air receiver 30. On the out strokes of the working pistons the compressor piston 14 draws in air through inlet valves such as 25 to the space on its right and delivers compressed air on its left through delivery valves 28 through a discharge port 29 to a compressed-air receiver (not shown). The piston 17 and cylinder 18 co-operate to form a compensating cushion which assists in moving the piston assemblies through their in strokes.

Exhaust ports 31 in the working cylinder 10 are controlled by the piston 12 and communicate with an exhaust manifold 32. Scavenge ports 33 in the cylinder 10 are controlled by the piston 11 and open directly into the space within the scavenge air receiver 30. A fuel-injection nozzle is denoted by 34. The ends of the crosshead pins 19 and 20 are fitted with slippers, such as 40 in Fig. 7, which slide in guide channels 41 on the interior of the wall of the scavenge air receiver 30.

The crosshead pin 19 is connected by a link 42 and a crank 43 to an oscillatory cam 44 cooperating with the actuating member 45 of a fuel-injection pump 46, the lobe of the cam being so set that it actuates the pump 46 to cause a delivery of fuel through a pipe 47 to the injection nozzle 34 as the pistons approach their inner dead point. The pump 46 is provided with a slidable regulating member 48 coupled to a handle 49, whereby the quantity of fuel injected per cycle can be varied.

In this machine the inner dead point of the pistons is very nearly constant with varying load, for the following reasons. The diameters and clearance volumes of the compressor cylinder and cushion cylinder are so designed that the total energy of expansion from these two sources is very nearly constant, with the result that the total energy of compression in the working cylinder must also be nearly constant. The compression rises rapidly as the pistons approach the inner dead point, and therefore a small change in compression energy will cause only a very small displacement of the inner dead point. Since the compression characteristic is substantially constant, the expansion characteristic is determined by the quantity of fuel injected, and therefore the position of the outer dead point varies with the quantity of fuel injected, that is with the load, the stroke being substantially shorter at light loads than at full load. Figs. 1, 6 and 7 show the pistons at their outer dead point under maximum load, while Figs. 8 and 9 show the position occupied by the piston 11 at the outer dead point under partial load.

This compressor, as so far described, is of known type and operates in known manner. In accordance with this invention, the scavenge ports 33 are shaped, as shown in Figs. 2, 3, 8 and 9, in such a manner that their angle to the radial varies progressively along their depth, so as to compensate for the change in air velocity hereinbefore described. At the opening edge 35 of the ports (Figs. 2 and 3) their angle A to the radial direction is a minimum, and this angle increases progressively until at the opposite edge 36 of the ports the port angle B approaches a tangential direction in relation to the cylinder, as shown by the dotted lines in Fig. 3, which show the section of the ports at the plane A—A in Fig. 2. Consequently, when the stroke is short and only a small fraction of the total depth of the ports is uncovered by the piston near the edge 35, the average direction of that part of the port which is uncovered makes a relatively small angle to the radial, as will be clear from Fig. 9, where the full lines at the ports 33 show their section at the plane of the head of the piston 11 when this piston is at the outer dead point of such a reduced stroke, the angle of the ports to the radial at this plane being denoted by C, and the mean angle of the part of the port which is uncovered being denoted by M_S. As the stroke increases according to the load on the engine, so a larger fraction of the depth of the ports is exposed by the piston and the average angle of the ports is increased up to the maximum angle M_L (Fig. 3), and their mean directional effect on the air flowing through them varies accordingly. However, as the stroke is increased, the effective port area and the time available for the entry of the scavenge air are increased, and the velocity of the air through the ports is reduced as described. This would reduce the rate of swirl if it were not for the fact that the average port angle is increased at the same time. By a proper choice of the angles of the ports along their depth the rate of swirl can be made practically constant, whatever the stroke of the piston.

The direction of the air entering the cylinder through a partly open port agrees more closely with the direction of the port as the ratio of depth to width of the port opening increases. Consequently the width of the port may be varied, instead of or in addition to its angle over the depth of the port, in order to compensate for the change in area of port opening due to variation in piston stroke.

Fig. 4 shows scavenge ports 33A having a uniform angle to the radial throughout their depth, but decreasing progressively in width from the opening edge 35A to the opposite edge 36A.

In this connection another factor to be considered in the design is the effect of a lip on the opening edge of the port, sometimes used to prevent this edge of the port building up with carbonised oil. Such a lip is shown in Fig. 5 at 37 in a scavenge port 33B which is otherwise arranged as hereinbefore described with reference to Figs. 2 and 3 or to Fig. 4. With such a lip a smaller rate of swirl is obtained for a given width of port at a given small fractional opening below the opening edge 35B, because the area of the port over the main part of its length is then larger in relation to the area of the opening into the cylinder, so that the air velocity along the port is relatively low, and therefore the direction of the port has less influence on the direction of the air entering the cylinder.

The amount by which the angle or the width of the port is varied is dependent on the characteristics of each particular engine. It will be dependent on the variation of the stroke and also on the variation in the amount of scavenge air pumped through the ports with change of stroke.

While the invention has been described as applied to a free-piston engine, namely one in which the movement of a piston assembly or of an interlinked number of piston assemblies is controlled solely by the gas pressures acting on the several pistons, the invention is also applicable to other kinds of engines, such for example as a semi-free piston engine in which a piston assembly is controlled by an oscillating crank or the like which determines the inner dead centre of a working piston but wherein the position of the outer dead point is variable with variation in the operating conditions.

I claim:

1. A two-stroke compression-ignition internal-combustion engine of the kind having a working cylinder, opposed pistons in said cylinder which is provided with scavenge ports and exhaust ports controlled respectively by said pistons, said scavenge ports being shaped to give a swirl to the charge admitted by them, synchronizing linkage connecting said pistons together for constraining them to move equally and oppositely, a reciprocating compressor having a compressor piston connected to one of said working pistons for reciprocation in unison therewith through strokes which vary with variation in working conditions so as to vary the outer dead points of said working pistons and in consequence the maximum depth of port opening of said scavenge ports, wherein said scavenge ports are so shaped that the mean direction of the charge fluid flowing through any one of said scavenge ports is nearer to the radial direction from the axis of said working cylinder at small port openings than at large port openings.

2. An engine as claimed in claim 1, wherein the angle of said scavenge ports to the radial direction is increased along their depth in the direction of the "out" stroke.

3. An engine as claimed in claim 1, wherein the width of said scavenge ports is decreased along their depth in the direction of the "out" stroke.

4. An engine as claimed in claim 1, wherein the angle of said scavenge ports to the radial direction is increased along their depth in the direction of the "out" stroke, and the width of said scavenge ports is decreased along their depth in the same direction.

5. A two-stroke internal-combustion engine of the kind including a working cylinder having a tangentially directed scavenge port, a piston in said cylinder controlling said port, and means operable while said engine is running for varying the position occupied by said piston at the end of its "out" stroke and in consequence the maximum depth of opening of said port by said piston, characterized in that the angle of said port to the radial direction increases along its depth in the direction of the "out" stroke, so that its mean directional effect on the charge flowing through it into said working cylinder varies with the depth of port opening in such a manner as to compensate, at least in part, for variation in the rate of swirl resulting from the change in the area of port opening due to variation in the position of said piston at the end of its "out" stroke.

6. A two-stroke internal-combustion engine of the kind including a working cylinder having a tangentially directed scavenge port, a piston in said cylinder controlling said port, and means operable while said engine is running for varying the position occupied by said piston at the end of its "out" stroke and in consequence the maximum depth of opening of said port by said piston, characterized in that the width of said port decreases along its depth in the direction of the "out" stroke, so that its mean directional effect on the charge flowing through it into said working cylinder varies with the depth of port opening in such a manner as to compensate, at least in part, for variation in the rate of swirl resulting from the change in the area of port opening due to variation in the position of said piston at the end of its "out" stroke.

7. A two-stroke internal combustion engine of the kind including a working cylinder having a tangentially directed scavenge port, a piston in said cylinder controlling said port, and means operable while said engine is running for varying the position occupied by said piston at the end of its "out" stroke and in consequence the maximum depth of opening of said port by said piston, characterized in that the angle of said port to the radial direction increases along its depth in the direction of the "out" stroke, and the width of said port decreases along its depth in the said direction, so that its mean directional effect on the charge flowing through it into said working cylinder varies with the depth of port opening in such a manner as to compensate, at least in part, for variation in the rate of swirl resulting from the change in the area of port opening due to variation in the position of said piston at the end of its "out" stroke.

8. A two-stroke internal-combustion engine of the kind including a working cylinder having a tangentially directed scavenge port, a piston in said cylinder controlling said port, and means operable while said engine is running for varying the position occupied by said piston at the end of its "out" stroke and in consequence the maximum depth of opening of said port by said piston, characterized in that the shape of said port varies along its depth in such a manner that its mean directional effect on a charge flowing through it into said working cylinder varies by departing from the radial direction as the depth of port opening increases, so as to compensate, at least in part, for variation in the rate of swirl resulting from the change in the area of port opening due to variation in the position of said piston at the end of its "out" stroke.

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