Jan. 14, 1964 H. J. VENEDIGER 3,117,566

PORT-CONTROLLED, OPPOSED-PISTON, TWO-CYCLE INTERNAL-COMBUSTION ENGINE

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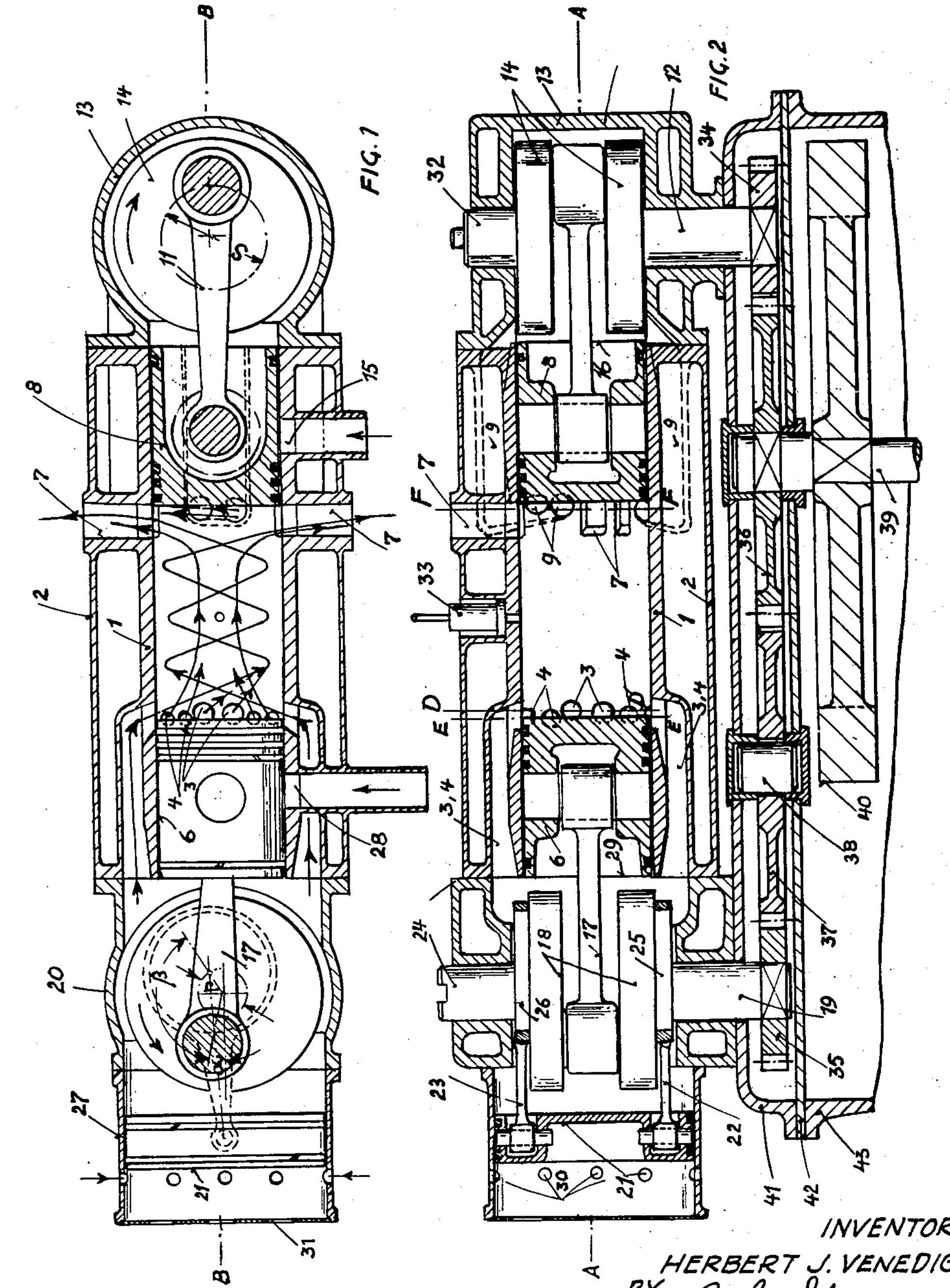

INVENTOR
HERBERT J. VENEDIGER
BY C. G. Stratton
ATTORNEY

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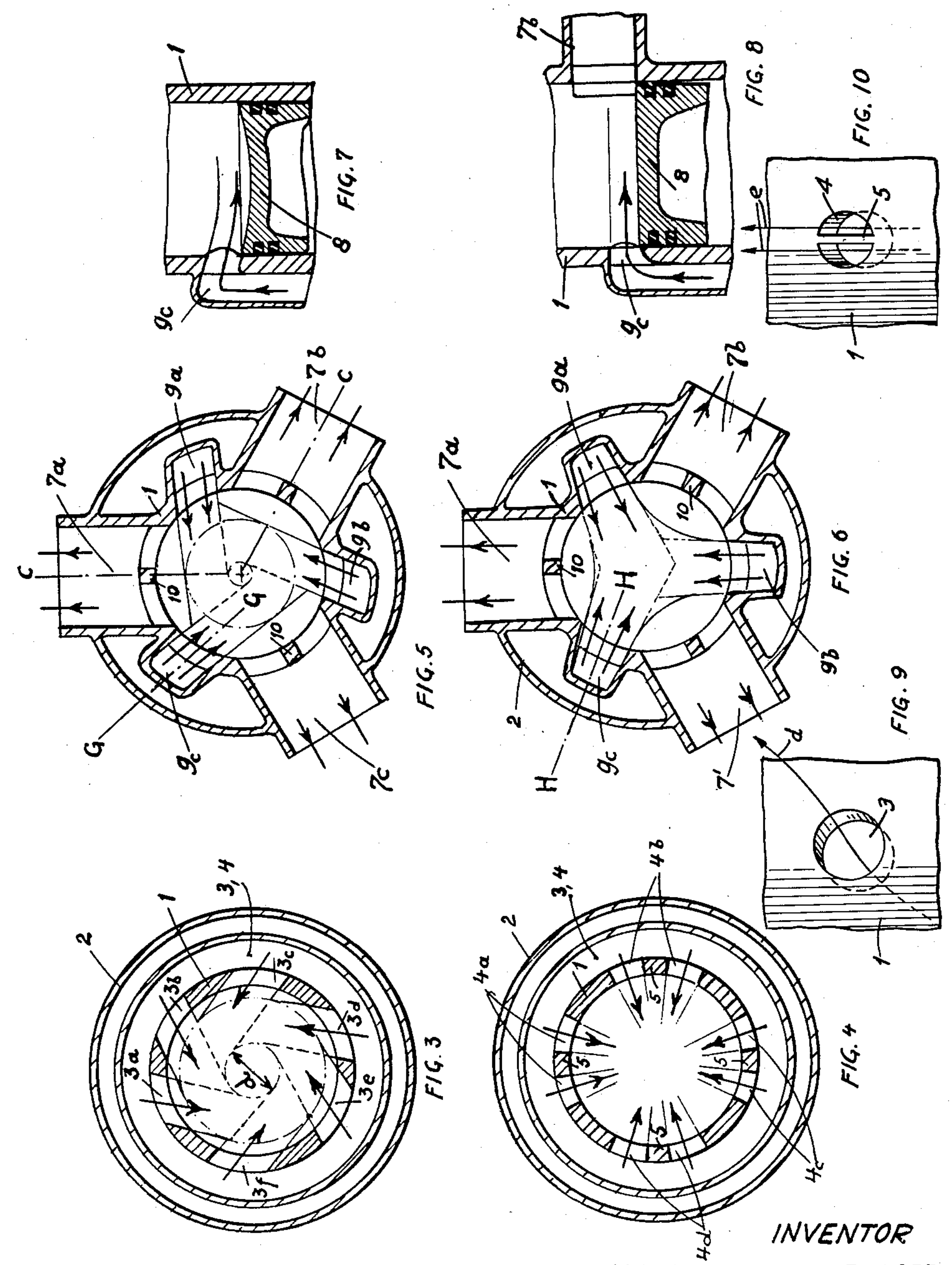

INVENTOR
HERBERT J. VENEDIGER
BY C. G. Stratton
ATTORNEY

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PORT-CONTROLLED, OPPOSED-PISTON, TWO-CYCLE INTERNAL-COMBUSTION ENGINE

Herbert J. Venediger, Heinrich Heinestrasse, Furth, Bavaria, Germany, assignor of one-half to Thomas B. Danckwortt, Beaumont, Calif.

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This invention relates to a port-controlled, opposed-piston, two-cycle internal-combustion engine with two working pistons moving toward and away from each other in a common working cylinder, of which one piston controls the intake ports in its end of the cylinder, and the other controls the exhaust ports in the opposite end of the cylinder, whereby the two pistons with their piston tops form the combustion chamber or a part thereof when the pistons are in their inner positions. In this type of engine, all intake ports are located in one cylinder end and all exhaust ports are located in the other cylinder end, and the scavenging and charging air flows through the cylinder according to the inclination or position of the intake ports distributed around the cylinder circumference, either in paths approximately parallel to the cylinder axis or in the form of helical lines; or the port arrangement is such that part of the scavenging and charging air enters the cylinder tangentially and moves in helical formation motion toward the exhaust ports, while the balance of the scavenging and charging air enters by other ports radially arranged to the cylinder and progressing in helical line motion in approximate parallel axial paths from inlet to exhaust.

This known construction, commonly known as an opposed-piston, two-cycle internal-combustion engine with uniflow scavenging, has a serious disadvantage in that the working piston controlling the exhaust ports (also called the exhaust piston) is constantly exposed to the exhaust gases and is insufficiently cooled during the fuel-to-power exchange process. In order to obtain a good quantitative scavenging degree, i.e., retention of the largest possible amount of scavenging and charging air introduced in the cylinder, the front of the air mass should not have proceeded further than to the surface of the exhaust piston when the same is closing the exhaust ports. It is evident that, in this case, there is no cooling effect of the scavenging and charging air upon the surface of the exhaust piston, as the air does little or no moving relative to and over the surface of the working piston that controls the exhaust. Since such an operation is not possible on account of the excessive thermal loading of the exhaust piston, the two-cycle uniflow opposed-piston engine is forced to operate with an excessively high amount of scavenging air.

The scavenging and charging air effort is the quotient of the space volume of the air introduced into the working cylinder per working cycle and the piston displacement of the working cylinder. The value of this quotient for opposed-piston, two-cycle engines of known construction must be 1.50 or higher, to assure a reliable, continuous operation with acceptable output. A scavenging air amount of 1.50 or 1.60 makes necessary the use of a special larger charger or supercharger. Coupling of such a device to the two-cycle engine requires considerable power. About 25% of the scavenging and charging air introduced into the cylinder at a 1.50 ratio leaves the cylinder through the exhaust during the exhaust and scavenging process. Therefore, the surface of the exhaust piston is being inefficiently cooled by scavenging air that is escaping at high speed. A reliable operation with good horsepower output requires that known uniflow opposed-piston engines be supplied with a large air volume. Air manifolding and a separate supercharger, requiring considerable horsepower for this air production, are needed for this purpose. Therefore, all attempts to operate opposed-piston two-cycle diesel engines with crankcase-air scavenging has failed, because crankcase pumps produce an air volume of only 0.75% to 0.80%. Even a low horsepower output of such a two-cycle engine could not prevent the thermal difficulties of the exhaust piston, due to the above reasons.

This invention eliminates the thermal difficulties of the exhaust pistons of uniflow, scavenged, opposed-piston, two-cycle engines by a special arrangement of the cylinder end with the exhaust ports, and makes possible the creation of a two-stroke diesel engine with good specific output, working with a crankcase air pump and making unnecessary large-volume scavenging air receivers.

According to the present invention, piston-cooling channels are arranged between adjacent exhaust ports for the purpose of intensifying cooling of the exhaust piston that controls the exhaust ports. Additional scavenging air is directed against or over the piston crown, distributed over the cylinder circumference, and tangentially or radially directed. These piston-cooling channels are opened by the exhaust piston considerably later than the exhaust ports and are preferably directed against the flat or concave curved end of the exhaust piston, as the case may be.

Instead of the above, the piston-cooling channels may enter the cylinder perpendicularly to the cylinder axis, or, in the case of a slightly convex exhaust piston top, slightly inclined toward the top. The piston-cooling air channels are connected to the scavenging-air receiver.

In continuation of the inventive concepts, the crankcase housing the running gear of the exhaust piston is preferably constructed as a crankcase-scavenging pump. This pum sucks on the necessary scavenging or cooling air volume for the exhaust piston and presses it into the piston-scavenging channels. If the crankcase pump of the exhaust piston is properly designed, a cooling-air volume of .35% to .45% is made available for the temperature reduction of the exhaust piston, sufficient to eliminate the thermal difficulties above mentioned.

According to the invention, the crankcase chamber which incorporates the running gear of the working piston that controls the intake ports may also be built as a crankcase-scavenging air pump. The running gear of this intake piston may also drive, in a known manner, an auxiliary opposed charging piston of proportionally large diameter and short stroke. This charging piston operates back and forth in a special charging cylinder and increases the air-delivery efficiency of the crankcase-scavenging pump of the intake piston.

It is quite feasible to increase the air volume delivered by the crankcase-scavenging pump plus an auxiliary charging piston by suitably proportioning the influential factors, such that the air volume entering the working cylinder through the air intake ports will be over 1.0. The intake of the fresh air into the crank chamber of the exhaust piston and that of the intake piston takes place by means of fresh air intake ports in the wall of the working cylinder controlled by the lower edge of the respective working piston.

According to the invention, the scavenging pressure and charging pressure may be regulated on an opposed-piston, two-cycle, internal-combustion engine with crankcase pumps arranged on opposite ends of the working cylinder by connecting both crankcase pumps and the respective scavenging-air receivers, by a common fresh air suction manifold. Furthermore, the arrangement may be such that the piston-scavenging channels may be connected to the scavenging-air receiver and the respective crankcase-scavenging pump of the intake side instead of to the crank chamber of the exhaust side; or they may be connected to both crankcase-scavenging pumps.

In case of a two-cycle, opposed-piston engine according to this invention, which operates as an Otto-cycle engine or as a gas engine, the piston-scavenging channels introduce fresh air mixed with some lubricating oil into the working cylinder while the inlet channels on the other cylinder end blow a mixture of fuel-air for combustion, specifically gas-air mixture, into the working cylinder where the mixture is ignited by an electric spark.

A port-controlled, opposed-piston, two-cycle, internal-combustion engine has been known wherein both cylinder ends have crankcase-scavenging pumps. However, according to this prior engine, the exhaust ports and intake-scavenging ports have about the same relationship to the end of the working piston and are both located on the same cylinder end, while the cylinder-scavenging follows the reverse loop principle and the intake ports on the other cylinder end deliver merely after-charging air.

In contradistinction to this prior engine, the present invention is more nearly related to the uniflow scavenging-type of engine or a combination of pure uniflow scavenging and helical-line scavenging plus exhaust-piston cooling improvements.

This invention also has for its objects to provide such means that are positive in operation, convenient in use, easily installed in a working position and easily disconnected therefrom, economical of manufacture, relatively simple, and of general superiority and serviceability.

The invention also comprises novel details of construction and novel combinations and arrangements of parts, which will more fully appear in the course of the following description, and which is based on the accompanying drawings. However, said drawings merely show, and the following description merely describes, preferred embodiments of the present invention, which are given by way of illustration or example only.

In the drawings, like reference characters designate similar parts in the several views.

FIG. 1 is a longitudinal sectional view of an engine according to the present invention, as taken on the line A—A of FIG. 2.

FIG. 2 is a similar view as taken on the line B—B of FIG. 1, with the portion at the right following the line C—C of FIG. 5.

FIGS. 3, 4 and 5 are cross-sectional views taken respectively on the lines D—D, E—E and F—F of FIG. 2.

FIG. 6 is a cross-sectional view similar to FIG. 5, showing a modification.

FIG. 7 is a fragmentary longitudinal sectional view as taken on the line G—G of FIG. 5.

FIG. 8 is a similar view as taken on the line H—H of FIG. 6.

FIGS. 9 and 10 are enlarged fragmentary detail views of intake ports, as used in the present invention.

In the drawings, the working cylinder **1**, shown in FIG. 1, is water-cooled and, therefore, has a water jacket **2**, but the same could as well be built for air cooling and provided with cooling fins. At one end, the cylinder **1** is provided with inlet port groups **3** and **4** which, as shown in FIGS. 3 and 4, show, for example, six air-intake ports entering the working cylinder **1** at an angle, and four inlet ports **4*a***, **4*b***, **4*c*** and **4*d*** entering the cylinder radially. The intake ports **4*a*** to **4*d*** are each divided by a rib **5**. The intake-port group **3** is located nearer the middle portion of the cylinder than is the intake-port group **4**. Such difference in port-group positions causes the intake piston **6** to open the ports **3** earlier and close them later than the intake ports **4**. In FIG. 1 the intake piston **6** has not yet reached its inner dead-center position, and is short of dead-center by the crank angle α (FIG. 1).

As shown in FIGS. 1, 2 and 3, the intake ports **3*a*** to **3*f*** of port group **3** enter through the wall of cylinder **1** at an angle that is inclined toward the axis of cylinder **1**. Therefore, the scavenging streams introduced into the cylinder through said ports proceed in a helical path toward the exhaust end of the cylinder, as suggested by the helical lines in FIG. 1. FIG. 3 shows the favorable inclination of the ports **3*a*** to **3*f***, which has the effect of producing a hollow cylinder in the direction of the cylinder axis by the helical form of the scavenging streams. Such a cylinder is shown with a diameter *d*, which is about one-third of the internal diameter of the working cylinder.

The other intake ports **4*a*** to **4*d*** enter the working cylinder **1** in a radial direction, as in FIG. 4, but at an angle inclined toward the cylinder axis, like the ports **3*a*** to **3*f***, at a greater angle. Therefore, the streams from ports **3*a*** to **3*f*** enter the cylinder at a steeper angle. The intake streams from ports **4*a*** to **4*d*** meet the cylinder axis under an entry angle of about 45°, while the angle of the inlet stream of ports **3*a*** to **3*f*** is so selected relative to a plane perpendicular to the cylinder axis that a scavenging stream coming out of the ports **3*a*** to **3*f*** proceeds one and one-half to two turns or convolutions of the helical line before entering the exhaust port.

FIG. 9 shows a port **3** and its compound angle, as above described, to produce an intake flow that follows the helical path, here designated *d*. Several turns or convolutions of such flow path are shown by the helices within the cylinder in FIG. 1. FIG. 10 shows a port **4**, with its dividing rib **5**, and shown at the single angle inclined toward the cylinder axis to produce the longitudinal path suggested by the arrows *e*.

The scavenging air streams from ports **4*a*** to **4*d*** flow within the hollow *d* of the cylinder, shown in dotted lines in FIG. 3, to exhaust along the axis of the cylinder **1**. Therefore, the streams of scavenging air from the ports **3** move substantially longitudinally along the axis of cylinder **1**, and the streams from ports **4** move in a swirling path around the first streams. FIG 3 illustrates these two different paths of air from ports **3** and **4**.

The outlet port group **7** of cylinder **1** is located at the end of said cylinder opposite to where the port groups **3** and **4** are located. Said port group **7** is controlled by an exhaust piston **8** at its inner dead center position, when the intake piston **6** is still away from its inner dead center by the crank angle α. FIGS. 1 and 2 show this condition.

The exhaust-port group **7** is shown as a row of exhaust ports distributed around the circumference of the cylinder **1**. FIGS. 5 and 6 show the port arrangement, in which the exhaust port group **7** consists, for example, of three exhaust ports **7*a***, **7*b*** and **7*c***, spaced one hundred twenty degrees apart and each having one dividing rib **10** in the center. The extension of the exhaust ports **7** beyond the end of exhaust piston **8** at retracted dead center position of said piston is greater than the extension of the intake ports **3** beyond the end of piston **6** at its α angle position, and the extension of the intake ports **4**, in turn, by said end of piston **6** at said position, is greater than the extension of ports **3**. Thus, the intake port group **3** will be opened by the intake piston **6** when the exhaust piston **8** has opened the exhaust port group **7** such that the crank of the exhaust piston **8** is ten to fifteen degrees closer to its inner dead center position than the crank of the intake piston **6**. When the intake piston **6** opens the intake port group **4**, its crank has already passed about ten degrees in the direction of the inner dead center after opening of the intake group.

An essential characteristic of the invention is the arrangement of a group of piston-cooling air channels **9**, shown specifically as channels **9*a***, **9*b*** and **9*c***, as disposed alternately with the exhaust ports **7*a***, **7*b*** and **7*c***. The flow from said channels cools the exhaust piston **8**, intensively, by blowing over across its end surface at high speed when said piston is at or near inner dead center.

FIGS. 5 and 6 show only two preferred forms of channels **9** although other possibilities may be used. The piston-cooling channels **9*a***, **9*b*** and **9*c*** enter cylinder **1** so that the same are opened later by the exhaust piston **8** and closed correspondingly earlier than the exhaust port group 7. It is quite favorable to have the exhaust piston 8 open the piston-cooling ports *9a* to *9c* when the crank of the exhaust piston 8 is about fifty degrees from its inner dead center. The piston-cooling channels *9a* to *9c* enter the cylinder at a tangential angle, as shown in FIG. 5, so that the cooling scavenging streams follow a turbulent stream line over the surface of the end of exhaust piston 8, as shown in FIG. 5; or they may be directed radially toward the cylinder axis, as shown in FIG. 6. It is desired that the cooling air streams cover as great a portion of the surface of the piston 8 as possible. For this purpose, it is favorable to direct these streams against the end (or crown) of piston 8, which may be made flat or concave, as desired. In case the end is concavely shaped, as shown in FIG. 7, the bottom of the end of the opposed piston 6 may be convexly shaped so as to get sufficiently high compression for diesel operation. This design also favors the initial guidance of the scavenging streams from the intake air ports 3 and 4.

With a flat end on the exhaust piston 8, the piston-cooling channels *9a* to *9c* may also enter the cylinder 1 radially transverse, as shown in FIG. 8. With a convex or concave end of the exhaust piston, the air channels may enter the cylinder slightly inclined in one direction or the other, as the case may be.

In case the opposed-piston two-cycle internal-combustion engine is operated with a supercharger and its necessary large air-receiver manifold, the piston-cooling channels *9a* to *9c* may be attached to the air receiver where they get the amount of air for the cooling action of the exhaust piston 8. It does not matter how much of this air, used for cooling the exhaust piston 8, remains in the cylinder or escapes through the still open exhaust ports *7a* to *7c*, since the primary task of this air is the intensive cooling of the exhaust piston.

It is also possible to reverse the turbulent fresh air motion of the cooling air from the channels *9a* to *9c*, as in FIG. 5, and let the air rotate or swirl in a direction opposite to that of the helical flow of the scavenging air coming from the air inlet group 3. In such case, a heavy turbulence occurs at the end of the exhaust piston 8, the same being beneficial for heat extraction from the piston end.

Similar effects are produced by the meeting of air streams from *9a* to *9c* in the piston center according to FIG. 6 where the temperature of the exhaust piston is at its greatest.

The above-described arrangements of inlet port groups, exhaust port groups and piston-cooling channels will bring about effective cylinder-scavenging even with a scavenging air quantity of less than 1.5 and will otherwise safeguard the piston and insure an intensive cooling of the same to overcome the most formidable thermal and operational difficulties of known opposed-piston engines.

The cooling air for the direct cooling of the exhaust piston 8 may be produced simply by designing the crankcase as a crankcase-scavenging pump. The crank housing 13, forming the crankcase, houses the running gear, such as the connecting rod 11 and the crank shaft 12. The crank cheeks 14 may be best formed as circular members. The piston-cooling channels *9a*, *9b* and *9c* may be arranged parallel to the cylinder axis from where they enter the cylinder at the crankcase air pump and extend the shortest distance toward their entry into the cylinder.

The inlet of fresh air is sucked by the exhaust piston 8 into the crankcase housing 13 through air ports 15, of which only one port is shown, the same entering the cylinder between the exhaust ports 7 and the crank housing 13, as shown in FIG. 1. The timing of the air inlet ports is accomplished in a known manner by the end edge 16 of the exhaust piston 8 when the latter piston approaches its outer dead center. It is not difficult to get an air delivery ratio of eighty percent even with a wide range of revolution speeds with a crankcase air pump built according to the above construction, considering that the piston 8 of the crankcase air pump 11, 12, 13, 14 opens, for each working stroke, fifty percent of the stroke volume of the opposed-piston two-cycle engine. The amount of air volume available for piston cooling will be 0.80 times 0.50 which equals a 0.40 volume—a small amount remaining in the working cylinder, determined by advantageous arrangement, direction and position of the piston-cooling channels *9a* to *9c*, as shown, for example, in FIGS. 5 and 6.

Another significant feature of the invention is the creation of an especially simply constructed opposed-piston two-cycle engine with uniflow scavenging and port control which produces the necessary scavenging and charging air for the cylinder 1 by building the intake air side also as crankcase pump similar to pump on the exhaust side.

The running gear—crank rod 17, crank cheeks 18 and crankshaft 19—of intake piston 6 in crank housing 20 are quite similar to gear in the crank housing 13 of exhaust piston 8.

In the event it is desired to introduce an amount of fresh air into the working cylinder 1 corresponding to an air delivery ratio of 0.90 to 1.10 or more, it is proposed to use an auxiliary charging piston 21, which is, for example, driven by two connecting rods 22, 23 from the crankshaft 19, 24 of the intake piston 6, by means of eccentric discs 25 and 26, on the crankshaft 19, 24. This structure effects the stroke motion of the auxiliary piston 21.

The auxiliary charging piston operates in its own cylinder 27, which has an axis that is preferably in line with the axis of cylinder 1. The arrangement is such that the auxiliary charging piston 21 comes very close to the circular cheek discs 18 at its inner dead center. The auxiliary charging piston 21 reaches its inner dead center together with the intake piston 6, or, more advantageously, a little later, as shown in FIG. 1. Here, the angle between cranks of the intake piston 6 and the auxiliary charging piston 21 is not 180°, but an angle β that is at 150° to 180°. This lesser angle decreases the maximum scavenging air pressure somewhat and delivers scavenging air through the auxiliary charging piston into the intake port groups 3 and 4 until the intake piston 6 closes the intake port groups 3 and 4, again.

The fresh air sucked into the crankcase housing 20 by the crankcase scavenging pump above described, enters, through intake ports 28, into the working cylinder 1. Said ports are opened and closed by the end edge 29 of the intake piston 6 when the latter piston is closed to outer dead center position.

Instead of the above, the intake ports may also be arranged in the cylinder 27 of the auxiliary charging piston 21, as shown by ports 30 in FIG. 2. A third solution is the provision of air intake ports 28 in cylinder 1, as well as ports 30 in the auxiliary charging piston cylinder 27. In this case, the port groups 28 and 30 may be connected, although such connection is not shown in the drawing. The end of cylinder 27, open to the outside, is protected by a plate 31 provided with openings.

The weights of the pistons 6, 8 and 21 may be balanced so that the opposed-piston two-cycle engine, in operation, has perfect balance of the two oscillating masses.

The crankshaft ends 24 and 32 may drive auxiliary equipment, for instance, fuel pumps, magnetos, generators, etc. A fuel injector is shown at 33, but the same represents a spark plug, as well, the required carburetor is connected to intake ports 28, or both the port pumps 28 and 30, while the piston-cooling channels *9a* to *9c* are connected to introduce pure air or lubrication oil mixed with air, into the cylinder 1. The same applies for gas engines.

The crankshaft ends 12 and 19 deliver the horsepower output of the opposed-piston engine and, therefore, are connected by a gear drive. Crankshaft 12 carries a driving gear 34 and the crankshaft 19 carries a driving gear 35 of the same size. Both gear wheels are fastened securely to their respective crankshafts. Gear 34 meshes with a gear 36 and gear 35 with a gear 37. The gears 36 and 37 are the same size, and are in mesh with each other. Gear 37 is an idler, while gear 36 is connected solid to an output shaft 39. Shaft 39 is the power take-off shaft and carries a flywheel 40.

The driving gears 34, 35, 36 and 37 are mounted in an oil-tight housing 41 which is covered with the plate 42. Plate 42, in turn, is mounted upon a housing 43 that carries the entire engine, if, as shown, the crankshafts of the opposed-piston engine stand vertically. Of course, every other position of the crankshafts in space is possible.

The working cylinders need not be aligned as shown in the drawings, but may be arranged in multiple relationship alongside each other, or one above the other, as well as vertically or horizontally.

While the foregoing has illustrated and described what is now contemplated to be the best mode of carrying out the invention, the constructions are, of course, subject to modification without departing from the spirit and scope of the invention. Therefore, it is not desired to restrict the invention to the particular forms of construction illustrated and described, but to cover all modifications that may fall within the scope of the appended claims.

Having thus described this invention, what is claimed and desired to be secured by Letters Patent is:

1. In an opposed piston, two-cycle internal combustion engine including cylinder means receiving the opposed working pistons and forming therewith a combustion space between the pistons, the cylinder means having an axis along which said working pistons are in longitudinal alignment, rotary crankshafts coupled to the working pistons, the improvement that comprises crank housings for said crankshafts at the opposite ends of the cylinder means, each said crank housing comprising a scavenging pump, a plurality of scavenging ports provided in one end of the cylinder means, exhaust ports provided in the opposite end thereof so that scavenging air, during scavenging, moves in the cylinder means from the former to the latter ports, one scavenging pump being connected to the opposed piston controlling the scavenging air ports and being provided with a reciprocative auxiliary piston that has counter-reciprocative movement to the latter piston, and a plurality of piston-cooling channels extending from the crank housing of the other scavenging pump to the interior of the cylinder means and directed across the operative end of the opposed exhaust piston to cool the same, the cylinder means having an air intake, controlled by the latter piston, and connected to the latter crank housing for supplying said cooling air.

2. An opposed-piston, two-cycle internal combustion engine according to claim 1 in which the piston-cooling channels enter the cylinder means outward of the exhaust ports, the exhaust piston opening the same later than opening of the exhaust ports, and closing the same earlier than closing of the exhaust ports.

3. An opposed-piston, two-cycle internal combustion engine according to claim 1 in which the piston-cooling channels, where the same enter the cylinder means, are disposed at an angle to direct the cooling air streams against the operating end of the exhaust piston.

4. An opposed-piston, two-cycle internal combustion engine according to claim 1 in which the piston-cooling channels are directed at an angle that is tangent to an imaginary smaller cylinder on the axis of the cylinder means.

5. An opposed-piston, two-cycle internal combustion engine according to claim 1 in which an air inlet is provided for the piston-cooling channels, and an independent inlet for carbureted fuel is provided in the cylinder means and controlled by the first-mentioned piston for direction toward the crank housing thereof and displacement by the pump in the latter housing through the scavenging ports into the cylinder.

6. An opposed-piston, two-cycle internal combustion engine according to claim 1 in which all of the scavenging ports are directed at an angle to the cylinder means axis to scavenge in a longitudinal direction toward the exhaust ports.

7. An opposed-piston, two-cycle internal combustion engine according to claim 1 in which all of the scavenging ports are directed along a chord of the cylinder means to form a tangent to an imaginary cylinder on the axis of the cylinder means, whereby said latter ports are disposed at a compound angle that directs scavenging streams of air that proceed in helical convolutions toward the exhaust ports.

8. An opposed-piston, two-cycle internal combustion engine according to claim 1 with at least two longitudinally offset rows of scavenging ports of which the ports farther from the exhaust ports are directed at an angle to the cylinder means axis to scavenge in a longitudinal direction toward the exhaust ports, and the other of said scavenging ports are directed along a chord of the cylinder means to form a tangent to an imaginary cylinder on the axis of the cylinder means, whereby said latter ports are disposed at a compound angle that directs scavenging streams of air that proceed in helical convolutions toward the exhaust ports, the remaining scavenging ports, because of their angle direction, creating scavenging air streams that move longitudinally within the convolutions above mentioned.

9. An opposed-piston, two-cycle internal combustion engine according to claim 1 in which the piston-cooling channels are directed at an angle that is tangent to an imaginary smaller cylinder on the axis of the cylinder means, the scavenging ports that produce scavenging streams that proceed in helical convolutions being disposed nearer to the exhaust ports than are the other scavenging ports and thereby being opened earlier than said other ports and closed later by the piston controlling the same.

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