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TWO-CYCLE COMBUSTION ENGINE

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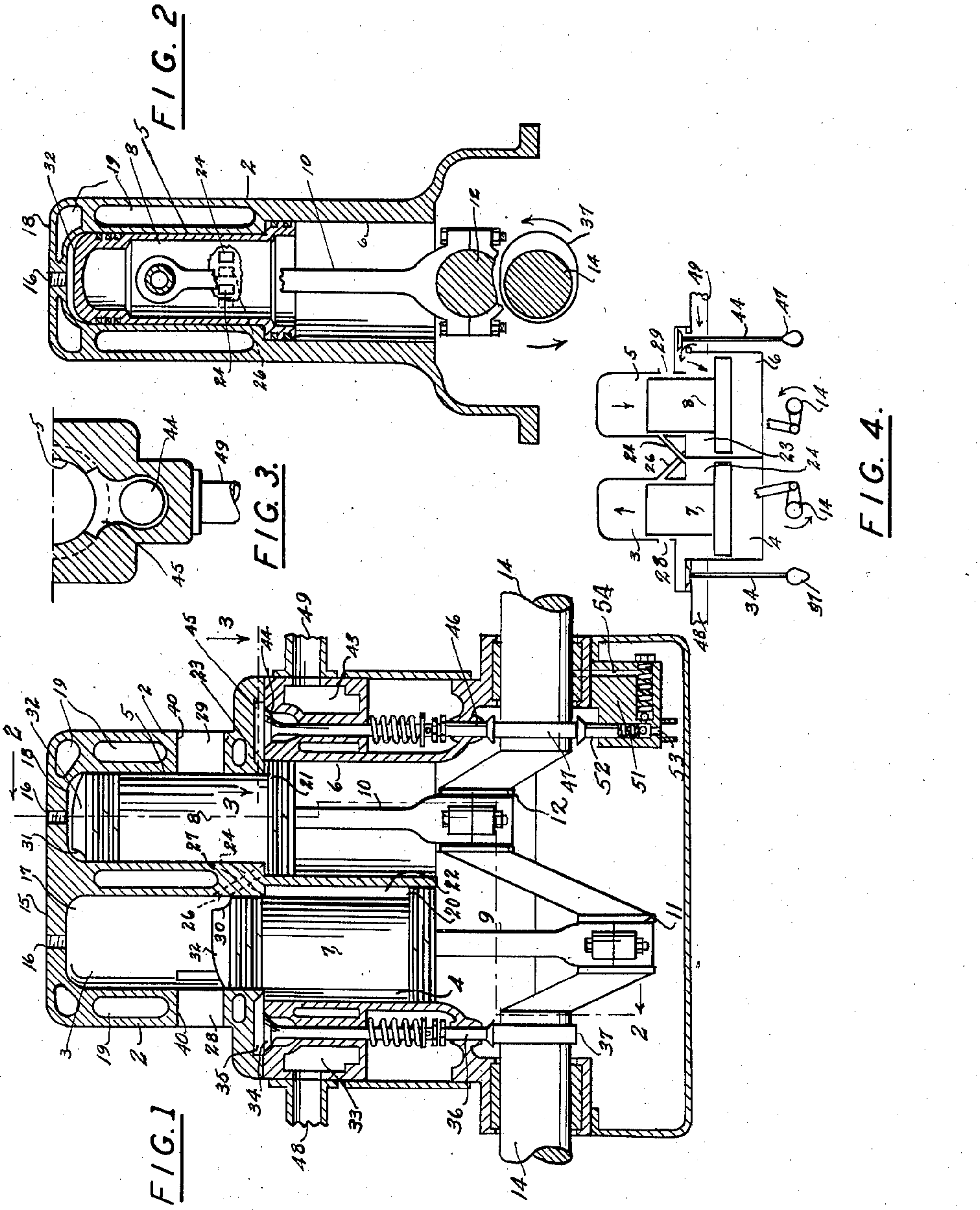

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TWO-CYCLE COMBUSTION ENGINE

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This invention relates generally to two cycle engines and more particularly to that type of engine having twin or companion cylinders wherein double pistons operating in parallel positioned cylinders are supported at opposite throws of a common crank shaft. The objects of my invention are: First, to provide an internal combustion engine of the type described which will operate in a more efficient manner than motors heretofore made; Second, to provide a motor of the type described having greater overall fuel efficiency and greater economy of operation than motors of said type heretofore constructed; Third, to provide a means for introducing a measured volume of combustible mixture into the combustion chamber of the firing cylinder under a definite pre-determined pressure; Fourth, to provide an engine of the type described wherein the volume of the explosive fuel mixture introduced is not dependent on the cubical displacement of the piston in the explosion chamber during the entire stroke, but wherein the charge introduced is measured to equal only the volume of the ignition cylinder above the opening of the exhaust ports; Fifth, to provide an engine of the type described wherein the design and construction permits the use of exhaust port openings higher than is commonly used in conventional engines of this type, to secure better scavenging of exhaust gases; Sixth, to provide an engine of the type described wherein the ignition of said measured charge, operating through the entire stroke, transmits it power to a crank-throw corresponding to the said entire stroke whereby greater mechanical advantage, resulting in greater torque, is secured than in conventional types of engines; Seventh, to provide a construction and arrangement of parts in an engine of the type described, whereby a minimum amount of power is required to compress the explosive mixture prior to its introduction into the ignition chambers. Other objects will apepar hereinafter.

I attain these objects by means of the construction and design of the engine and its component parts illustrated in the accompanying drawing, herewith made a part of this application, and in which—

Fig. 1 is a vertical side section of an engine built according to my design; Fig. 2, a transverse section thereof, taken substantially on lines 2—2, Fig. 1; Fig. 3, a partial horizontal section taken on the plane indicated by line 3—3, Fig. 1; and Fig. 4, a diagrammatic view of the engine illustrating the position of the valves and pistons at the middle of a stroke, being 90° advanced from the position shown in Figs. 1 and 2; the cylinders being represented as in Fig. 1, whereas the crank in the lower portion of the diagram is represented as in Fig. 2, or at right angles to the upper portion of the diagram to better show the relative position of the cranks and cams. Similar numerals refer to similar parts in all views.

Before describing in detail the several parts of my engine, as herewith illustrated, and in order to better explain the reason for its particular design and construction, I will make the following explanation as to its general construction and operation. Heretofore, the most successful engines of the two cycle type have been designed with a larger bore than stroke. It has been found, so far as the construction generally of efficient high speed engines is concerned, that greater overall economy is attained by opening the exhaust port from 40 to 57 degrees in advance of lower dead center.

It has been found, however, that when two cycle engines had a longer bore than stroke—since the cylinder was charged from the crank case—the volume of gas drawn into the crank case was too great for the volume needed to fill the working portion of the combustion cylinder above the upper edge of the exhaust ports. That is, since the volume displaced by the conventional type piston below the piston within the crank case is always equal to the total area displaced above the piston, the volume of the charge drawn into the crank case will always be equal to the total displaced volume above the piston in the firing chamber. Now, if exhaust ports open any considerable distance above the lower travel of the piston, which would be true if they uncovered when the crank pin is approximately 57° above lower dead center, the volume of the gases transferred from the crank case to the firing chamber would exceed the amount necessary to efficiently charge the combustion chamber. Therefore, part of the charge would be mixed with exhaust gases and, if premature ignition did not take place, would be diluted by and blown out with the exhaust. By reason of the fact that the area of the compression chambers may be varied and their ratio of compression increased to any reasonable limit, a higher compression than is ordinarily possible in the crank case type engine may be produced. Thus the charge, prior to entrance into the ignition chamber, may be put under a pressure as high as fifty pounds per square inch. This increased pressure causes the gases to expand from the compression chamber head to the ignition chamber, through the cross ports, with greater velocity than in the conventional type of engine and, consequently, when my engine is run at high speeds there is a more efficient transfer of the entire compressed charge.

By the design and construction hereinafter explained, the volume of gases drawn into the charging cylinder is measured to equal the volume of the combustion chamber into which they are introduced above the top opening of the exhaust ports. This measured volume is determined by the design of the engine.

Since the metered volume of gases compressed in the compression chamber is less than in the conventional crank case compression motor, if other factors are the same with my type of motor, less energy need be expended in compressing the charge prior to introducing it into the ignition chamber.

From the construction and design hereinafter described it will be apparent that a two cycle engine may be constructed with a bore considerably longer than the stroke; that the exhaust port may be provided with its upper opening edge a considerable distance above the lower limit of travel of the piston to the effect that the exhaust will open 60° above lower dead center; that a charge may be introduced into the combustion chamber which will equal only the volume of the combustion chamber above the upper edges of the exhaust port opening, and as a result, this measured charge of fuel and air operates on a crank-throw proportioned to the entire stroke, and therefore, greater torque is produced. The two cycle engine, thus, has the advantage of a four cycle engine with a longer stroke than bore, together with the advantage of greater compactness and simplicity.

Within the casting **2**, dual or twin parallel double cylinders are cast and machined out in the usual manner. Numeral **3** indicates the left hand upper or ignition cylinder, and **4** the left lower cylinder, while **5** indicates the right hand upper cylinder and **6** the right hand lower cylinder. Pistons **7** and **8** respectively are fitted in these cylinders and attached by connecting rods **9** and **10** to the two throws **11** and **12** of the crank shaft **14**. The crank journals are positioned 180 degrees apart. Thus, when piston **7** is at the lower end of its travel, piston **8** will be at the upper end of its travel. The upper cylinders are closed at the top by a head **15**, provided with spark plug openings **16**, and all parts adjacent the firing chambers **17** and **18** are provided with water jackets **19**, as is the usual practice with internal combustion engines.

The lower cylinders **4** and **6**, which I term compression measuring cylinders, are of a considerably larger diameter than the upper or ignition cylinders, **3** and **5**. The lower portion or skirt of each of the pistons **7** and **8** is enlarged at **20** and **21** and provided with rings to form a working gas tight closure for the lower portions of said cylinders. In enlarged cylinder **4**, this closure forms an annular chamber **22**, between the upper end of this lower cylinder and the enlarged skirt **20** of the piston therein. A further similar chamber **23** is similarly formed within lower cylinder **6**. These annular chambers constitute and may be termed compression cylinders, and by design their cubical contents may be changed by altering their outer diameter relative to the diameter of the upper portion of the piston operating therein. That is, their cubical contents is governed by the difference between the diameter of the ignition cylinder and the diameter of the skirt at the bottom of each piston, and may be increased by enlarging the latter and/or by decreasing the former.

Compression cylinder or chamber **22** is connected by cross ports **24**, to communicate with the interior of ignition cylinder **5**, and compression cylinder **23** is connected with ignition cylinder **3** by cross ports **26**. These ports are cast in that portion of block **2** indicated by numeral **27**, and being between the two upper ignition cylinders. This construction makes it possible for these cross or intake ports to be made straight and lead directly from the upper end of one compression cylinder to the lower portion of the opposite ignition cylinder, so that the incoming charge travels in a straight line directly toward the deflector and the top on the piston within the ignition cylinder.

Exhaust ports **28** and **29** respectively are provided in the ignition cylinders **3** and **5**. Each of the pistons **7** and **8** have deflector recesses **30** and **31** cast in their heads **32**. These are positioned adjacent their respective inlet or cross ports **24** and **26**, and shaped to force the incoming explosive charge up toward the top of the ignition cylinder.

So far as concerns the left hand cylinder, a charge of explosive mixture (air and hydrocarbon fuel) is admitted through passageway **33**, poppet valve **34**, and passage **35** to chamber **22**. Valve **34** is operated by push rod **36**, actuated by cam **37** on the main or crank shaft **14**. The same arrangement pertains to the right hand companion cylinder; the corresponding parts being numbered **43**, **44**, **45**, **46** and **47**. Chambers **33** and **43** are connected to a carbureter by manifolds **48** and **49**.

Referring to Fig. 4, which shows the position of the pistons on the crank when advanced 90° from the position shown in Figs. 1 and 2, it will be noted that poppet valve **34** is closed since a charge is being compressed in compression cylinder **22**, while poppet valve **44** is opened to permit the entering of a charge into compression cylinder **23** on the down stroke of piston **8**, while ignition takes place in cylinder **5**. Cams **37** and **47** are positioned on shaft **14** relative to the position of crank pins **11** and **12** to effect this operation.

It is to be understood that, as the crank cycle progresses, intake poppet valve **44** closes when piston **8** has reached the bottom of its stroke to permit the compression of the charge drawn into cylinder **23**. As the stroke cycle further proceeds, piston **8** starts upward and piston **7** downward at which time cam **37** has advanced to open valve **34**. Meanwhile the charge compressed in cylinder **23** has been released into ignition cylinder **3**, by the uncovering of port **26** by the top of piston **7**.

It will be noted that the top edge **40** of both exhaust ports is higher than is the usual practice in engines of this type. This is designated to permit exhaust gases to be released as soon as the piston has traveled from the top of its stroke downward to a position where the crank pin has advanced throughout 120° from its position at the top of the firing stroke. The intake port is uncovered later, and when the crank pin has traveled approximately 160° from the top of the stroke. The difference between these two positions permits the gases of combustion to start exhausting through approximately 40° of the crank cycle before the intake port opens. After the piston uncovers the exhaust, the pressure of the ignited charge is rapidly reduced and the piston is driven through the lower portion of its crank cycle—60°—largely by the expansion of the ignition charge and its own inertia. Power transmitted in this portion of the stroke is, therefore, proportionately less than practically any other portion of the ignition stroke, but this is compensated for by the fact that the ignition gases have an adequate time to escape whereby back pressure is minimized and the charge more efficiently scavenged before a fresh charge enters the ignition cylinder. This charge is measured or metered by the design and size of the compression cylinder from which it is forced, so that it is sufficient only to effectively fill the upper portion of the ignition cylinder substantially above the horizontal plane at the top or upper edge **40** of the exhaust port. In other words, this ignition charge is metered so that it is entirely ignited and principally expanded by the time the piston uncovers the exhaust port.

Now in all two cycle engines of the conventional type wherein the crank case or any portion thereof is used to draw in an explosive charge prior to forcing it into the firing chamber, the volume induced by the up stroke of the piston is always equal to the total volume on the upper side of the piston. That is to say, the total volume of the ignition cylinder. Thus, the charge entering the ignition cylinder at ordinary atmospheric pressures on exhaust and carbureter is equal to the entire volume of the ignition chamber. A portion of this firing charge is mixed with the outgoing exhaust and wasted. This objection is overcome by the construction of the engine I have hereinabove described. Furthermore, it is obvious from the above description, that I have produced an engine wherein the metered fuel charge is ignited in an ignition cylinder so that the charge is ignited and expanded throughout a stroke cycle wherein the length of the stroke may be easily made to greatly exceed the size of the bore. Therefore, since the throw of the crank is in proportion to the length of the stroke, this explosive charge operates to greater mechanical advantage on this somewhat larger crank throw and in this way produces a greater torque as a result of firing of each metered charge.

The construction of the engine with the cam shaft carried direct on the crank shaft **14** provides easy adaptability to mechanism for both fuel and oil pumps. Thus, as shown in Fig. 1, an oil pump **51** is attached to the lower portion of the case below the right hand main journal. A push rod plunger **52**, held in contact with cam **47** by a spring, forms a pump cylinder. Small ball bearings may be used to form the valves, as is the usual practice, and oil may be drawn in from the crank case through duct **53** and led to the bearing through duct **54**. Obviously, a similar pump may be attached to the left hand journal, and used either for oil or fuel pump purposes.

I claim:

1. An engine of the two-cycle type having double pistons operative in double parallel positioned cylinders whereby ignition cylinders are formed in the upper smaller bore in each of said cylinders and compression chambers are formed by an expanded skirt portion of the piston operating in larger bores in the lower portion of said cylinders, said pistons having deflector heads and being operatively connected to a common crank shaft on crank throws positioned 180° apart, cross intake ports directly connecting the top portion of each lower compression cylinder with each opposite companion ignition cylinder, exhaust ports opening into said ignition cylinders positioned to be opened and operated by the piston therein and having a vertical depth not less than twice that of said intake port, passageways connecting the upper end of said compression cylinders to a source of carburation, said passageways being provided with poppet valves, mechanically operated from cams positioned on said engine crank shaft timed to open on the down stroke of the piston operating in its respective cylinder, said compression cylinders being proportioned so that their piston displacement equals the volume of the upper portion of the ignition cylinder into which they discharge above the plane of the upper edge of its exhaust port.

2. An engine of the type described in claim 1, wherein the upper edge of said exhaust ports is uncovered by the pistons therein when the crank throw has traveled through approximately 120° from top dead center and wherein the intake port is opened approximately 20° thereafter.

3. An engine of the type described having parallel double pistons with a smaller bore in the upper portion of each to form an ignition cylinder and a larger bore in the lower portion to form a compression cylinder, double pistons operative in each of said cylinders having rings whereby the upper portion of each piston forms a closure for each ignition cylinder and an expanded skirt having rings forming a closure for each compression cylinder, said pistons being operatively connected by means of a connecting rod with a crank shaft having crank pins 180° apart, said pistons having heads provided with deflectors positioned opposite intake ports in said cylinders to assist in directing incoming gases toward the top of said cylinders, exhaust ports in the walls of each of said ignition cylinders having the top of its upper edge substantially above the upper edge of said intake port, intake ports cut in the walls of said ignition cylinders, positioned substantially diametrically opposite to said exhaust ports, cross ports angularly positioned to direct the gases toward the deflectors on the tops of said pistons, and directly connecting the intake ports of each ignition cylinder with the top of the opposite companion compression cylinder, means for inducing a charge of combustible gases into said compression cylinders including passageways connected to the top of each of said compression cylinders and a poppet valve operative therein driven by mechanical mechanism associated with said crank shaft whereby said poppet valve opens on the down stroke of each piston respectively and closes on the corresponding up stroke.

4. An engine of the type described in claim 3, wherein the length of the stroke exceeds the diameter of the bore of the ignition cylinders by the depth of the exhaust port.

5. An engine of the type described in claim 3, wherein the length of the stroke exceeds the diameter of the ignition cylinders, and wherein the exhaust ports, cut in said ignition cylinders, is positioned to uncover and open by the downward travel when the crank pins are within approximately 60° of lower dead center, and said intake port opens when said crank pins are within approximately 20° of lower dead center and wherein the piston displacement of said compression cylinders equals the volume of said ignition cylinders above the opening of the exhaust ports.

6. An internal combustion engine of the two cycle type having a cylinder casting comprising bore twin cylinders, the upper smaller bore forming an ignition cylinder, and the lower larger bore forming a pump cylinder by cooperating with an expanding skirt on the piston operating therein, pistons operative in said cylinders provided with rings at the top for working closure of the ignition cylinder and rings on an expanding skirt forming a working closure of said pump cylinder; said pistons being connected to a common crank shaft having its crank pins 180 degrees apart, passage-ways connecting the tops of said pump cylinders directly with the lower portion of the opposite ignition cylinder terminating in an intake port positioned to be uncovered by the lower terminal movement of the upper part of the ignition piston; said casting also having longitudinal extensions at each end provided with a hollow portion forming a passage-way returning from said pump cylinders to a valve seat provided in the lower portion of said passage-way and connecting through a recess therebelow to a carburetor manifold, poppet valves operative in each of said valve seats actuated by cams on said crank shaft and set to open and close at pre-determined positions in order to induct a charge of combustible gases into said pump cylinders, exhaust ports opening in the walls of said ignition cylinders opposite said intake ports and extending from the lower limit of the piston travel upward to the extent that their upper edges permit uncovering and opening when the cranks of each cylinder after igniting have proceeded not more than 120 degrees from top dead center and the said intake ports being positioned so that they do not uncover until the crank has proceeded not less than 20 degrees therefrom; said ignition cylinders being proportioned so that the length of the stroke exceeds the diameter of the bore by not less than the vertical distance of the exhaust port opening and said pump cylinders having a piston displacement equal only to the volume of the ignition cylinders above the plane of the exhaust port opening.

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