ABSTRACT: Two-cycle, crankcase compression engines, spark or compression ignition, carburetion or fuel injection types, with or without supercharging, having improved balancing of restrictions to flow of air or air-fuel fluids through the engine; having at least one crankcase inlet port(s) location in the cylinder wall on the same side of the cylinder as the exhaust port(s) and below the level of the exhaust ports, having a plurality of transfer ports, at least some of which are located on the opposite side of the cylinder with the flow therefrom directed toward the cylinder head plus other transfer port(s) located and shaped so as to direct flow in a plane more nearly at right angles to the cylinder axis and thence merging with flow from the oppositely located transfer port(s), along the cylinder wall in a direction away from the exhaust port(s); the height of the exhaust port(s) for optimum operational characteristics being more than approximately 35 percent of stroke; in a modified form, in addition to the inlet below the exhaust port, including another subsidiary inlet(s) into the crankcase located below the oppositely situated transfer port(s); in engine propelled vehicles such as snowmobiles, where operator position is such that the engine is between the legs and near his crotch, the improvement of having inlet (carburetor) and exhaust on the same side of engine and locating such slide away from crotch of operator, the side of the engine towards the operator being devoid of hot or vaporous engine protuberances. Improvements that, in some forms, the engine is combined with a resonant exhaust system; improvements in attachment of inlet and exhaust manifolds are provided to minimize and/or regulate heat transfer between exhaust manifold, cylinder and intake manifold, to minimize backflow of heat from the exhaust manifold to the cylinder and inlet manifold and to regulate temperature of various engine parts.
TWO CYCLE REAR COMPRESSION ENGINE PORTING
AND TRANSFER PASSAGE ARRANGEMENT

This invention relates to two-cycle internal combustion engines of the crankcase compression type having exhaust and induction ports in that end portion of the cylinder wall which is adjacent the crankcase. The invention may be utilized in both spark ignition and compression ignition engines. Supercharging of the fuel/air charge may be utilized before or during introduction of the charge into the crankcase.

The flow path of the gaseous fluids through two-cycle crankcase compression engines may be likened to flow through a conduit having certain impediments and constrictions. The first is through an air cleaner, where used, and thence optionally through a precompressor (supercharger), where this is used, and thence through a carburetor or carburetors, or a manifold, with or without fuel injection, and thence into the crankcase. The flow continues, on an interrupted basis, from the crankcase via a transfer passage or passages to a transfer port or ports in the cylinder wall and thence into the cylinder. After the combustion cycle is completed, flow on an interrupted basis is continued out of the cylinder via an exhaust port (or ports) to an exhaust system which may include pipes, mufflers, and various devices for utilization of pressure waves in varying combinations. Where the exhaust system is "tuned" there will be provided, in a selected speed range or ranges, resonant, reflective compression (and also expansion in most instances) of unburned charge which has passed through the cylinder and out through the exhaust port(s). This unburned charge, or some of it, is compressed back through the (still open) exhaust port, and into the cylinder.

Each element in the flow path, e.g. air cleaner, when used, carburetor, when used, inlet or fuel injection manifold, when used, devices such as valves or ports for controlling inflow into the crankcase, crankcase, transfer passage(s) and port(s), flow path through the cylinder, exhaust port(s), and exhaust system exerts an individual and combined effect on the overall rate of flow of fluids through the engine, and hence an influence upon power output of the engine, and even though one constriction may be lessened, such as by enlargement, shaping, or duplication, bettered performance may not always be fully achieved, since another constriction may then become the restricting factor. Hence an attempt at improvement in performance occasioned by some variation in the total flow path may not come to full fruition.

One example of the foregoing is found in respect to the utilization of resonant exhaust systems, which I have found produce useful improvements in engine performance in many engines. However, theoretical maximum improvement, for example increased power output in the resonant speed range(s), may not be achieved solely by use of the resonant exhaust system(s). I have found that in engines having resonant exhaust systems improvement toward the maximum is gained by use in combination of the present invention. The present invention is not, however, limited to use in only those engines having resonant exhaust systems.

A most demanding criteria in design concerns inlet, transfer and exhaust port(s) particularly in respect to the number, location, area and height (measured as a percentage of piston travel) of such ports. This is due to the fact that there is only a limited amount of cylinder wall surface that is available for placement of ports.

In the situs of two-cycle, crankcase compression engines the present invention provides, and it is an object of the invention to provide an improved flow of gaseous fluids through such engines and improvements in performance and reliability of such engines; to provide improved combinations of exhaust, inlet and transfer passages and porting in such engines; to provide in such engines having improved combinations of exhaust, inlet and transfer ports and passageways, exhausts having a height of more than 35 percent of stroke, and for such engines having exhaust ports of substantially greater height to provide in the combination of such engines resonant exhaust system(s); to provide improvements in the location and number of inlet openings for flow of air or fuel-air mixture from carburetor or air inlet, into the crankcase; to provide improvements in the connections of inlet and exhaust manifolds to the engine; optionally to provide for multiple oppositely located crankcase inlets; to provide reduction in port height of crankcase inlet ports for improved power output; to provide for improved cooling of the exhaust side of piston and cylinder; to provide more uniform temperatures as between opposite sides of piston and cylinder and improved reliability and endurance particularly during operation under conditions of high power output; to provide an engine configuration wherein carburetor location is such as not to be an inconvenience to the user or interfere with location of other components, more specifically in an engine such as snowmobiles where the operator position is such that the engine is between the legs of the operator and near his crotch, to provide a vehicle and engine configuration wherein the engine exhaust port(s) and inlet port(s) are situated on the same side of the engine and that side of the engine is away from the operator's crotch, the inlet port(s) being situated below the exhaust port(s).

The invention is illustrated in the drawings wherein:

FIG. 1 is a plan (top) view of one exemplary form of engine of the invention;
FIG. 2 is a vertical sectional view taken along the line in the direction of arrows 2-2 of FIGS. 1 and 5;
FIG. 3 is a vertical, side elevational view, partly in section, taken along the line and in the direction of arrows 3-3 of FIG. 2;
FIG. 4 is a vertical partial sectional view taken along the line in the direction of arrows 4-4 of FIGS. 1, 2 and 5;
FIG. 5 is a horizontal sectional view taken along the line and in the direction of arrows 5-5 of FIG. 4;
FIGS. 6A, 6B and 6C are similar fragmentary vertical sectional views, similar to that portion of FIG. 2 between the levels 6-6 of FIG. 2, showing modified forms of the transfer passageways and related portions of the cylinder and piston, the same being opposed the exhaust port-inlet port side of the cylinder;
FIGS. 7A, 7B, 7C and 7D are similar fragmentary vertical sectional views of that portion of the cylinder including the exhaust and crankcase inlet ports, wherein, in these figures there are illustrated various modified forms of attachment of the exhaust and inlet manifolds to the cylinder, FIG. 7D corresponding to FIG. 2;
FIG. 8 is a side elevational view, partly broken away showing an exemplary vehicle-engine (snowmobile) configuration of the invention; and
FIG. 9 is a vertical sectional view, similar to FIG. 2 showing a modified form of the invention wherein provision is made for two generally oppositely located crankcase inlets through the cylinder, under the piston skirt and into the crankcase.

Referring to the drawings,
FIGS. 1-5 and 7D show one exemplary embodiment of the engine of the invention having a crankcase generally designated 10, composed of two halves 10A and 10B that are bolted together by bolts 11 on a center plane normal to the crankshaft at the middle of the crankpin 17. Bearings 12 are suitably retained in the crankcase halves and are provided with external seals, as at 14, the latter being retained by plate 14A and bolts 14B. The bearings 12 rotatably support crankshaft 15 having crank discs 15A connected by a crank pin 17.
To the crankcase there is attached by capscrews, not shown, a cylinder generally designated 16, here illustrated as air-cooled and hence provided with cooling fins 16A. Liquid cooling may be used, if desired. A cylinder head, generally designated 18, also provided with cooling fins 18A and 18B when desired, is attached to the cylinder by capscrews 19. In the illustrated embodiment a spark plug 20 is shown and this may be supplemented or replaced by a fuel injector nozzle where desired or necessary.
The air or fuel-air mixture for the engine is introduced via inlet manifold 21, which may be suitable flanged at its outer end 21A to connect to a carburetor, or other suitable device such as fuel injector, not shown, in the spark ignition version, or to an air intake manifold in the diesel compression ignition version. That end of manifold 21 proximate cylinder 16, is provided with a flange 21B, see FIG. 3 and the flange is held against the cylinder by studs and nuts 21C. A gasket 22 separates flange 21B from the cylinder. Gasket 22 may, if desired, be made of a material having a low thermal conductivity (low heat conductivity) so as to minimize flow of heat between the cylinder and the intake manifold flange 21B and manifold 21. It is a feature of the invention that the length of the hot portion 24B of the passage through the cylinder wall to port 24 is minimal and accordingly does not permit much time or provide much area by which heat may be transferred to heat the (cooler) incoming air or fuel-air, which is therefore kept desirably cool and dense. Such heat as is absorbed by the intake or fuel-air charge at this portion desirably cools the hot bridge 16B between the intake port 24 and the exhaust port 25 and also cools that side of piston 40 which is adjacent. This is a feature of the invention since the cooling of the piston in the area of exhaust port 25, contributes to longer piston and ring life in high performance engines. Also, the temperature of the entire exhaust port side of the cylinder is reduced. The port 24 is wide with rounded square corners, as shown in FIG. 3 and here is shown as provided with a port bar 24A which may be desirable, though optional. The height 24H of the intake port may be widely varied but is generally in the range of approximately 30 percent to 50 percent of stroke, and is here shown as 40 percent of stroke. The port 24 is uncovered as the lower edge 40B of the piston 40 rises toward top center position of the piston during the compression stroke, and is covered as the piston descends in the cylinder during the power stroke. The piston 40 is connected by connecting rod 42 to the crankpin 17.

Note: In this specification the expression port(s) is used. By this is meant that the port (inlet, exhaust or transfer) may be one or plural. Sometimes there may be only one port, or separate side-by-side ports may be provided. Sometimes a wide port is provided with one or more port bars thereby (technically) providing multiple ports as viewed from the cylinder. The expression "port(s)" is intended to include these variants.

In FIGS. 2 and 4, piston, generally designated 40, is at its lowermost position (lower dead center position of crankpin 17) and at such position has fully uncovered exhaust port 25, which is illustrated as having a port height 25H.

According to my invention, in order to obtain the most benefit from the invention, the height of the exhaust port(s), expressed as a percentage of stroke, will normally be 35 percent or more of stroke and at this and even greater exhaust port heights, performance and horsepower are acceptable and improved over what is presently available. The exhaust port height may be increased considerably above the 35 percent level to 50 percent or even more of stroke by use, in the invention, of a resonant exhaust system, and this is intended to be within the purview of the invention. In the drawings the exhaust port is illustrated as being 40 percent of stroke, and this and even higher exhaust port levels will operate advantageously without resort to resonant exhaust systems. Reducing the height of the exhaust port to less than 35 percent unnecessarily reduces performance and will normally not be done except where lower performance and horsepower are acceptable. In general, as the exhaust port height is made to approach 50 percent or more, performance and horsepower will not be improved except that by utilizing resonant exhaust system or systems, exceptional performance will be obtained at resonant speed(s). This is a feature of the invention.

The exhaust port 25 preferably also has only a short length indicated at dimension 25B, substantially only the cylinder wall thickness as measured to the flat surface 25F upon which gasket 26 is seated. Flange 28B of exhaust tube 28 seals on gasket 26, being held in place by studs and nuts 28C. This is Fig. 5. Gasket 26 may, if desired, be made of low heat conductivity so as to minimize flow of heat between the exhaust tube 28 and the exhaust port area of cylinder 16. According to this invention the heat of the hot exhaust gases is primarily dissipated in the exhaust tube 28 (and exhaust system connected thereto) rather than by the cylinder cooling facilities. It is a feature of the invention that the dimension 25B of port 25 to the exhaust gasket 26 is kept minimal, thus limiting the surface area and time during which heat of the hot exhaust gases are permitted to conduct heat to the exhaust port area of the cylinder per se. This adds to the life of the cylinder, piston and piston rings, which are often subject to deterioration in the region of the exhaust port area.

In some instances, it is desirable to make one or the other of gaskets 22 or 26 of heat conductive material, the other being of heat-flow barrier. Thus if gasket 22 is made heat conductive, the heat of the cylinder is to some extent drawn off into the inlet manifold, thus helping to cool the cylinder, which of course heats the incoming charge to some extent.

It is a feature of the invention that at least one of the inlet port into the engine 22 is located below the exhaust port 25. This not only brings about desirable utilization of the heat absorbing capabilities of the incoming cool gaseous fluids to cool piston rings, piston and cylinder in the region of the exhaust port, but also, by locating the inlet port below the exhaust port, space of the cylinder wall on the opposite side of the cylinder is made available for other uses, now to be described. There are also other advantages, i.e. with the exhaust and inlet on one side of the engine, the other side is left free, and this is a distinct advantage in some vehicles, such as snowmobiles, where the operator sits astride the engine, with the cylinder adjacent the operator's crotch. By orienting the engine so that the exhaust and inlet ports face away from the operator, the operator's position against the other side of the cylinder (free of exhaust and inlet) is made much more comfortable, a real advantage in case of, for instance, a flooded carburetor.

The air of fuel-air mixture introduced via inlet port 24 into the crankcase is compressed by the movement of the piston toward the crankcase and in two-cycle, crankcase compression engines this charge so compressed is conveyed to the cylinder. As shown in FIG. 2, the flow of exhaust gases will be, generally in the direction of arrow 29. It will be understood that as the upper edge of piston 40 descends and begins to uncover the upper edge of exhaust port 25 flow immediately occurs and due to the initially high pressure within the cylinder such flow will be at very high velocity. As more of the exhaust port 25 is uncovered, flow continues and as the pressure drops the rate of flow decreases. Meanwhile, the upper edge of the downwardly descending piston 40 will reach and begin to uncover the upper edges of the transfer ports, which will now be referred to.

It is a feature of the present invention that the transfer ports (and their related passageways) are of two kinds, via (a) transfer port(s) which are on generally adjacent in respect to the exhaust port, i.e. alongside the exhaust port in areas 33A and 33B, see FIG. 5, and (b) transfer port(s) that are in an area generally indicated as zone 32, substantially opposite the exhaust ports zone 34. For convenience of nomenclature these will hereinafter be referred to as "side transfer ports" (zones 33A and 33B) and "opposite transfer ports," (zone 32).

Referring to FIGS. 2, 4 and 5 especially, please notice that in the plane of section 4—4 which is vertical through the crankshaft, there are provided two transfer passageways generally designated 30—30 joining the crankcase at 30A—30A. These rise generally parallel to the cylinder axis and outside the portions 16B—16B of the cylinder 16, and then at their upper ends form ports 30B—30B, see FIG. 2, at zones 33A and 33B (see FIG. 5). The passages 30—30 adjacent ports 30B—30B are shaped by walls 39C and 30D so as to direct the flow of incoming gases (1) substantially generally.
horizontally or generally horizontally and somewhat upward across the cylinder and (2) towards a target zone 31 (see FIGS. 2 and 5) which is near the zone 32 of the cylinder which is opposite the exhaust and intake port area 34 (see FIGS. 2 and 5). These flows from transfer passages 30—30 and out of ports 30B—30B are illustrated by arrows 37—37. It is a characteristic of the invention that these flows, thus emanating from the “side” transfer ports 30B—30B will generally meet at zone 31, and since the top of the piston 40 is adjacent (and the flows hence cannot go downward), the flows will merge and turn upward, this upward turning being assisted by the transfer flow simultaneously entering the cylinder by way of the “opposite” transfer ports, generally designated 35A and 35B at zone 32, FIG. 5. One or more of such “opposite” transfer passages and ports may be used; here, two are illustrated.

It is a characteristic of the invention that the flow 36 from the passage or passages 35A and 35B should enter the cylinder in a direction approaching a condition of parallelism with the cylinder axis (and should be toward the distal end of the cylinder (i.e. most remote from the crankcase). In the illustration of FIG. 2, the surface 35A—A, angle K, is approximately 30°, which is to say approximately 60° to a plane normal to the cylinder axis. This angle may be varied somewhat with the objective that the flow denoted by arrow 36, through passages 35A and 35B should merge with the meeting and upward moving flows from ports 30B—30B (denoted by arrows 35—35) and all of the flows from all transfer passages 30B—30B (denoted by arrows 35—35) and all of the flows from all transfer passages 30B—30B and 35A and 35B will then continue upward and sweep the distal end of the cylinder at cylinder head 18. This total flow of incoming (transfer) charge, denoted by arrows 38, thus scavenges the cylinder head end of the cylinder, and ideally does not excessively come into contact with the spent gases meanwhile exiting via exhaust port 25.

It is another feature of the invention that there are provided a plurality of transfer ports and passageways, greater in number and greater in effective area than has heretofore been available in piston ported engines, all without increasing the height H of the upper edge of the transfer ports, (as measured by the cylinder position of stroke). Dimension “H” is the distance from the top edge of the piston to the top of the transfer port when the piston is at its bottom dead center position. The “height” of the transfer ports generally is in the range of 18 percent to 25 percent of stroke, measured at “H” above the top edge of the piston when it is at the bottom dead center piston position, and is here illustrated as approximately 22 percent.

In the use of scavenged engines it is desirable to keep the exhaust port height as low as consonant with good engine performance and since the transfer port height is usually kept even lower than the exhaust port height and this plus the requirement that the transfer ports must be aimed in a propitious direction so as to promote scavenging of the spent gases from the cylinder, it has, in effect, placed a design limitation on the size and area of the transfer ports, with the ultimate result that the transfer ports become, in effect, a limiting constriction on the total flow of fluid through the engine, and is hence determinative of power output.

I have discovered that by placing the intake port below the exhaust port and in such combination utilizing the both the “side” and “opposite” forms of transfer ports (and their related passageways) as above described, that the effective transfer port area is increased. The crankcase inlet port(s) is then no longer crowded and may be desirably widened in proportion, and this also desirably affects performance. Thus, according to this invention, utilization of opposite and side transfer ports in combination with placement of the inlet port below the exhaust port (optionally with a second inlet port, as will be described with reference to FIG. 7) gives more effective use of these ports and passageways with consequent reduction in resistance to flow of the air-fuel-air fluid through the engine, all of which are desirable for improving performance.

Then, with this combination, I have discovered that in order to obtain fullest benefit and advantages of the invention the exhaust port height may be increased to 35 percent of stroke or more, and may be increased even to 50 percent or more as hereinafter explained. For engines having an exhaust port height of 35 percent of stroke and even more, it is not essential in the invention to use a resonant exhaust system, but as the exhaust port height is increased toward and even beyond 50 percent of stroke, the use of a resonant exhaust system is increasingly desirable and beneficial in the combination of the invention. The invention will therefore be considered as optionally utilizing such resonant exhaust system as illustrated in FIG. 2, where the exhaust system is shown diagrammatically. A form of resonant exhaust system suitable for use in the present invention is disclosed in my U.S. Pat. No. 3,367,311. It is noted however, that this patent shows a multiple-tube resonant exhaust system. For the purposes of the present invention single or multiple-tube resonant exhaust systems may be used.

The placement of the inlet 24 below the exhaust 28 frees the opposite side of the cylinder from protuberances, and this is a distinct advantage in some vehicles, for example some types of snowmobiles, where the user sits in a position such that the engine is essentially between the legs with one side of the engine near the crotch. In such vehicles, the exhaust side of the engine is generally away from the position of the user, because of the heat evolved, and in vehicles utilizing engines of known configurations this has resulted in having the inlet (carburetor) quite close to the user, in fact close to the crotch, which is a disadvantage. This undesirable configuration is avoided by utilizing engine configurations of the present invention, since the exhaust and inlet (carburetor) can be on one side only, of the engine and in the vehicle (such as a snowmobile) this side (having exhaust and inlet) can be faced away from the user’s crotch.

Thus, referring to FIG. 8, there is illustrated an exemplary vehicle-engine configuration of the invention wherein the user sits astride the vehicle with the crotch of the user closely adjacent the engine cylinder area. In this particular illustration the vehicle is a snowmobile, but it will be understood this configuration also occurs in other vehicles such as scooters, minibikes, some watercraft, recreational and racing vehicles and the like.

In the illustration of FIG. 8, the vehicle generally designated V, has an engine E made as herein described with reference to FIGS. 1—5, 6A, 6B and 6C and FIGS. 7A—7D, having cylinder 16, cylinder head 18 and crankcase 10. The engine is oriented on the vehicle V so that the exhaust manifold 28 and inlet manifold 21, are on the “front side” of the engine E, as it is positioned in the vehicle. The inlet manifold 21 connects to carburetor C and through it to air cleaner AC at such “front side” position. The rear side ER of engine E has no exhaust port, no exhaust manifolding and has no inlet port and no carburetor and air cleaner and hence the user is not subjected to annoyances such as bumping into these protuberances proximate the crotch of the user to the engine.

Referring to FIGS. 2 and 6A, 6B and 6C, there are in these figures illustrated several ways in which the port area (of transfer ports 35A and 35B) can be fed from the crankcase. In all of these figures the actual port areas 35A and 35B, (only port 35A is illustrated) are similar, but the passageways leading from the crankcase vary.

In FIG. 2, the passageway 35A—B is merely a groove. The groove 35B—B is identical, see FIG. 4. The piston itself covers these grooves and in effect makes them passageways. This form, shown in FIG. 2, is easy to make but does not allow cylinder wall surface available for supporting the piston; this can be a disadvantage.

In FIG. 6A, the groove 35A—B of FIG. 2 has been converted into a passageway 35A—C, by providing a portion of cylinder 16C at 16C. This gives more cylinder wall area for piston support but may be somewhat more expensive to fabricate.

In FIGS. 6B and 6C the hollow inside of piston 40 is utilized as a passageway for compressed crankcase charge and a port
3,612,014

40A is provided in piston 40 opening into the lower end of a groove 35A-D (of FIG. 6B) or a passegway 35A-E (of FIG. 6C) the latter being formed through the cylinder body thus providing cylinder wall area 16D for better piston support. These forms, FIGS. 6B and 6C are especially useful in the embodiment of the invention illustrated in FIG. 9, hereinafter described. Combinations of transfer ports and channels as shown in FIG. 2, 6A, 6B and 6C may be utilized in the same cylinder for handling transfer of the charge. The various forms of passageways illustrated in FIGS. 2, 6A, 6B and 6C, may if desired, also be utilized for feeding the "side" transfer ports 39-39B.

FIGS. 7A through 7D illustrate variations in mounting of the flanges of the exhaust and inlet systems manifolding against the exhaust and inlet ports bolting areas. FIG. 7D corresponds to that shown in FIG. 2. In these figures, dimensions 24B and 25B are kept to a minimum, and there are provided heat insulating gaskets at 22 and 26, thus resulting in minimal heat flow between the exhaust manifold and cylinder 16 and from cylinder 16 to the inlet manifold 21, with the results already explained.

In FIG. 7A the dimensions 24B and 25B (of FIGS. 2 and 7D) are increased respectively to 24B-7B and 25B-7B, which increases the heat picked up from the exhaust gasses by the cylinder and in part utilizes the inflow of cool air to absorb such heat.

In FIG. 7B the dimension 25B remains the same as in FIGS. 2 and 7D, but the dimension 24B is increased to 24B-7C, with increased utilization in fresh charge inflow for cooling the cylinder adjacent the exhaust port area, which in some instances is an advantage.

In FIG. 7C, where cooling of the cylinder will otherwise suffice, dimension 25B (of FIGS. 2 and 7B) is increased, which increases the heat inflow to the cylinder at the exhaust port region.

The forms shown in FIGS. 7A through 7D are design alternatives, each presenting advantages under selected conditions.

An embodiment of the engine of the invention, having advantages in some installations, utilizes the transfer passages shown in FIGS. 6B and 6C and is shown in FIG. 9, wherein two inlets on opposite sides of the crankcase are provided rather than only one inlet location below the exhaust port. For a given area of inlet port, the greater "width" of inlet port(s) so provided by the FIG. 9 exemplification, permits a reduction in intake port height as measured at a percentage of stroke. The engine of FIG. 9 is generally like that shown in FIGS. 1-5 except that inlets at two locations, viz. 24 and 44, are provided, inlet 24 being as previously described relative FIGS. 1-5. Inlet 44 is similar but is located in that zone where the cylinder wall opposite the exhaust and intake, viz zone 32, FIG. 5, and below transfer passage 35A-D (compare FIGS. 6B and 6C with FIG. 8). I have discovered that by utilizing the FIGS. 6B-6C configuration, space (below the transfer passage 35A-B or 35A-C) is made available and may be utilized as further crankcase inlet space, thus reducing an impediment (or restriction) to total flow, previously mentioned. In some engines the crankcase is brought up around the lower port of the cylinder, to a greater extent than here illustrated and, in such designs might cover the inlet port 24. In such designs the crankcase would then, simply be provided with a port matching port 24; similarly re FIG. 9 for port 44.

The vertical location of the ports 24 and 44 may be the same or the same. They are valved by being uncovered by the lower edges 24E and 44E, respectively, of the skirt of the piston. The skirt of the piston may be lengthened or shortened within available limits and as needed, so as with the location of ports 24 and 44, to achieve the desired port timing.

Where carburetors are used, one at each inlet and their throttles properley linked at their throttle valves, one carburetor alone can function as an idling or slow speed carburetor and, as more speed or power is needed the second carburetor is opened and the two will then function in unison at moderate and high outputs and speeds.

While the invention has been illustrated with reference to single cylinder engines, it may be utilized with good results and without diminution of advantages in multiple cylinder engines. While the drawings illustrate the exhaust manifold 28 as being directly above inlet manifold 24, it will be understood that these may be placed at an angle to each other or curved so as to permit noninterfering placement and connection to carburetor(s) exhaust system, etc.

What I claim is:

1. In a two-cycle piston-valved, internal combustion engine wherein at least some of the gases comprising the combustion engine wherein at least some of the gases comprising the combustion charge are received into the crankcase through an inlet port valve by the piston skirt and compressed in the crankcase for transfer to the cylinder, and in which the combustion gases are released from the cylinder through an exhaust port valve by the top edge of the piston, the improvement comprising,
locating said inlet port and exhaust port in the same side of the cylinder spaced in axial direction of said cylinder and at least partially overlapping circumferentially in said cylinder with the inlet port spaced between the crankcase and the exhaust port, providing transfer passageways from the crankcase and emanating as ports in the cylinder wall so as to be valved by said piston, at least one of said transfer ports, called the opposite transfer port, being in that portion of the cylinder wall, called the opposite side of the cylinder, opposite from the portion having the exhaust port, and other transfer ports, called side transfer ports, in the cylinder wall and spaced so as circumferentially to be between that side of the cylinder having the exhaust port therein and the opposite side of the cylinder, said side and opposite transfer ports and passages adjacent thereto being shaped so that flow therefrom meets and flows generally away from the crankcase and along the opposite side of the cylinder.

2. The combination of claim 1 further characterized in that the height of the exhaust port measured from the top edge of the piston at lower dead center position to the top of the exhaust port is more than 35 percent of stroke of the piston.

3. The combination specified in claim 1 further characterized in that at least one of said transfer passages is in the form of a groove in the cylinder wall, said groove extending from the crankcase to the corresponding transfer port.

4. The combination specified in claim 1 further characterized in that at least one of said transfer passageways communicates with a port in the piston wall so as to be open to the interior of the piston and receive compressed crankcase charge therethrough when the port served by said transfer passage is uncovered by the top edge of the piston.

5. The combination of claim 1 further characterized in that the exhaust port communicates through an exhaust passage in the cylinder wall to an exhaust exit port in an exhaust flange seat and the inlet port communicates through an inlet passage in the cylinder wall to an inlet entrance port in an inlet flange seat and that flanged exhaust and inlet manifolds are bolted respectively to the exhaust flange seat and intake flange seat, gaskets being provided between each flange and its seat, at least one of said gaskets being of a material having low heat conductivity.

6. The combination of claim 5 further characterized in that the length of the exhaust and intake passages are of minimal length.

7. The combination of claim 5 further characterized in that the exhaust and intake passages are not of equal length, one being long and the other short.

8. The combination of claim 4 further characterized in that a supplemental inlet port is located in the cylinder below the level of that transfer port which communicates with a port in the piston.

9. The combination of claim 1 further characterized in that said exhaust exit port is connected to a resonant exhaust system.
10. The combination of claim 2 further characterized in that said exhaust port is connected to a resonant exhaust system.

11. The two-cycle piston-valved internal combustion engine of claim 1 further characterized in that a major portion of said inlet port and exhaust port overlap circumferentially.

12. The combination as specified in claim 11 wherein said transfer ports are arranged to open through said cylinder wall in all sectors of the circumference of said cylinder wall except in the region of said exhaust port.

13. The combination as specified in claim 12 wherein the transfer port opposite from said exhaust port opens to a transfer passageway defined by surfaces which direct entering fresh charge gases generally toward the distal end of the cylinder.
UNITED STATES PATENT OFFICE
CERTIFICATE OF CORRECTION

Patent No. 3,612,014 Dated October 12, 1971

Inventor(s) William L. Tenney

It is certified that error appears in the above-identified patent and that said Letters Patent are hereby corrected as shown below:

Column 8, lines 11 and 12, (Claim 1, lines 2 and 3) delete "wherein at least some of the gases comprising the combustion engine".

Signed and sealed this 4th day of July 1972.

(SEAL)
Attest:

EDWARD M. FLETCHER, JR. ROBERT GOTTSCHALK
Attesting Officer Commissioner of Patents