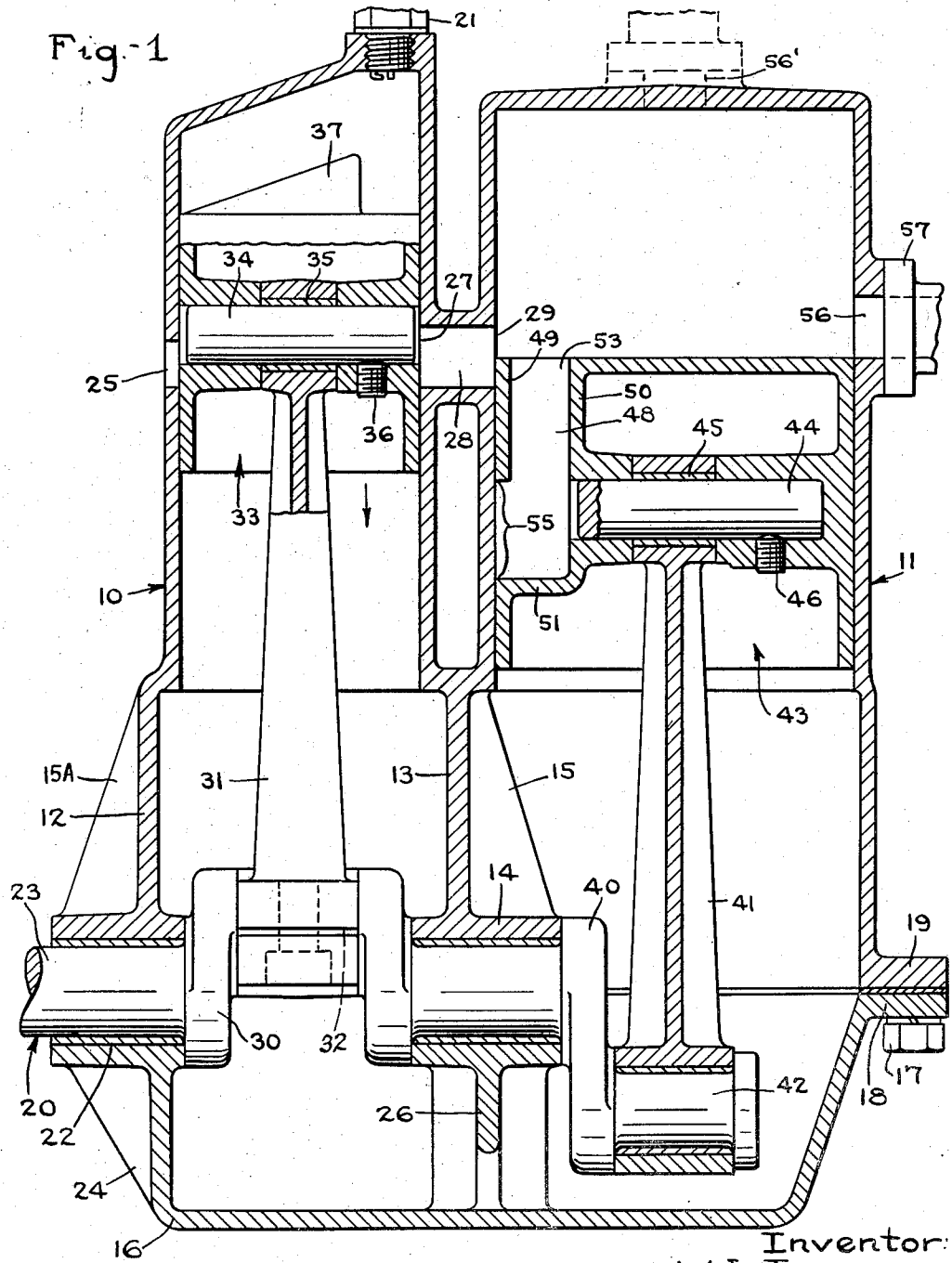
Sept. 19, 1950 W. L. TENNEY 2,522,649

TWO-STROKE CYCLE ENGINE CYLINDER AND PUMP

Filed Oct. 6, 1945 6 Sheets-Sheet 1

Fig. 1



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TWO-STROKE CYCLE ENGINE CYLINDER AND PUMP

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Fig-2

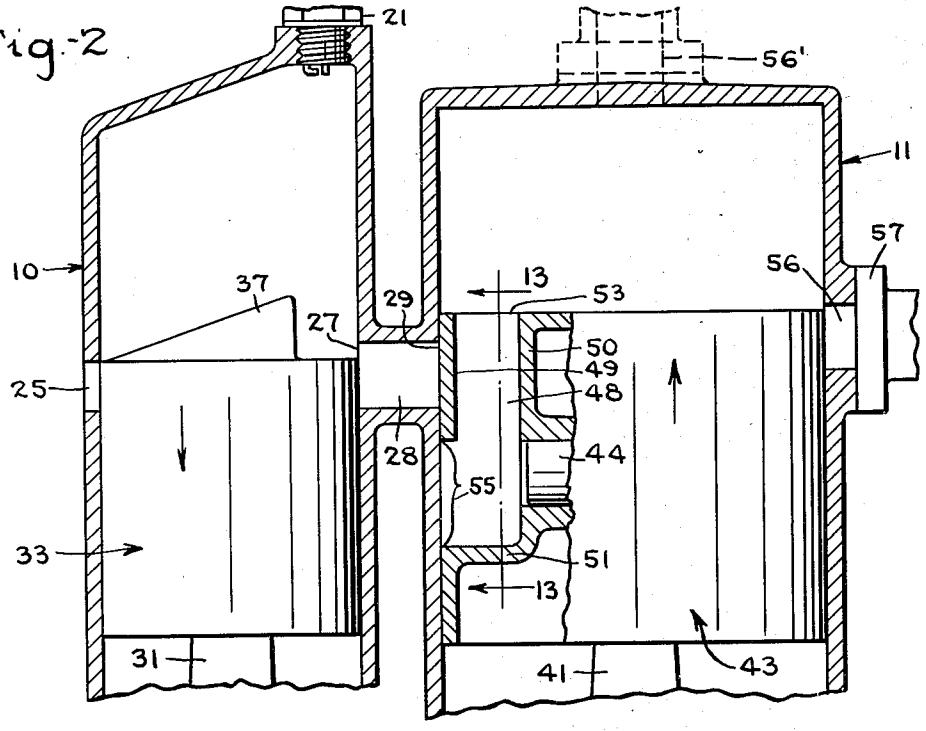
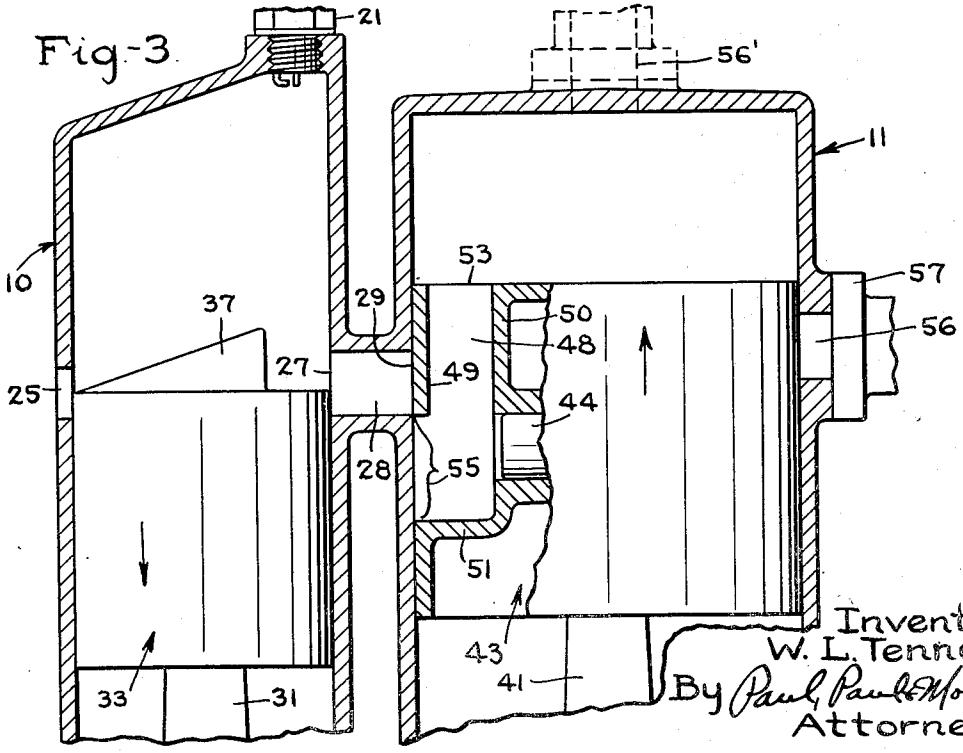


Fig-3



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TWO-STROKE CYCLE ENGINE CYLINDER AND PUMP

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Fig.-4

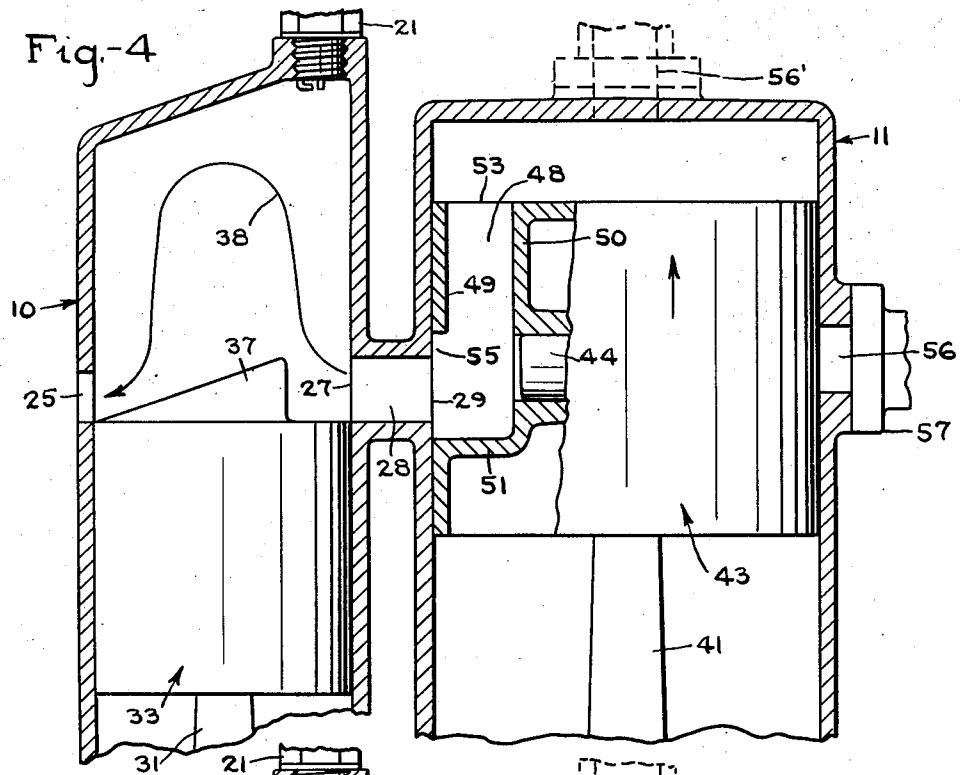
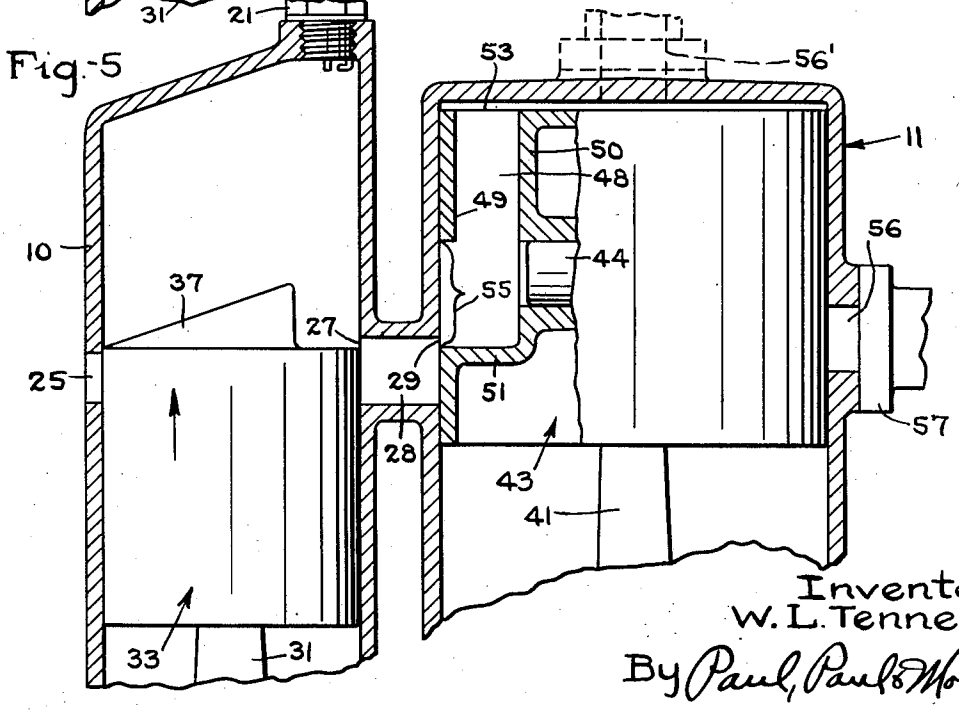


Fig.-5



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TWO-STROKE CYCLE ENGINE CYLINDER AND PUMP

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Fig. 13

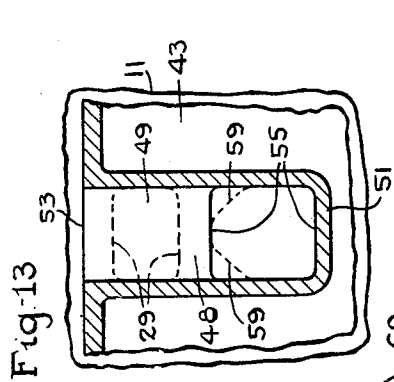

Fig. 11

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TWO-STROKE CYCLE ENGINE CYLINDER AND PUMP

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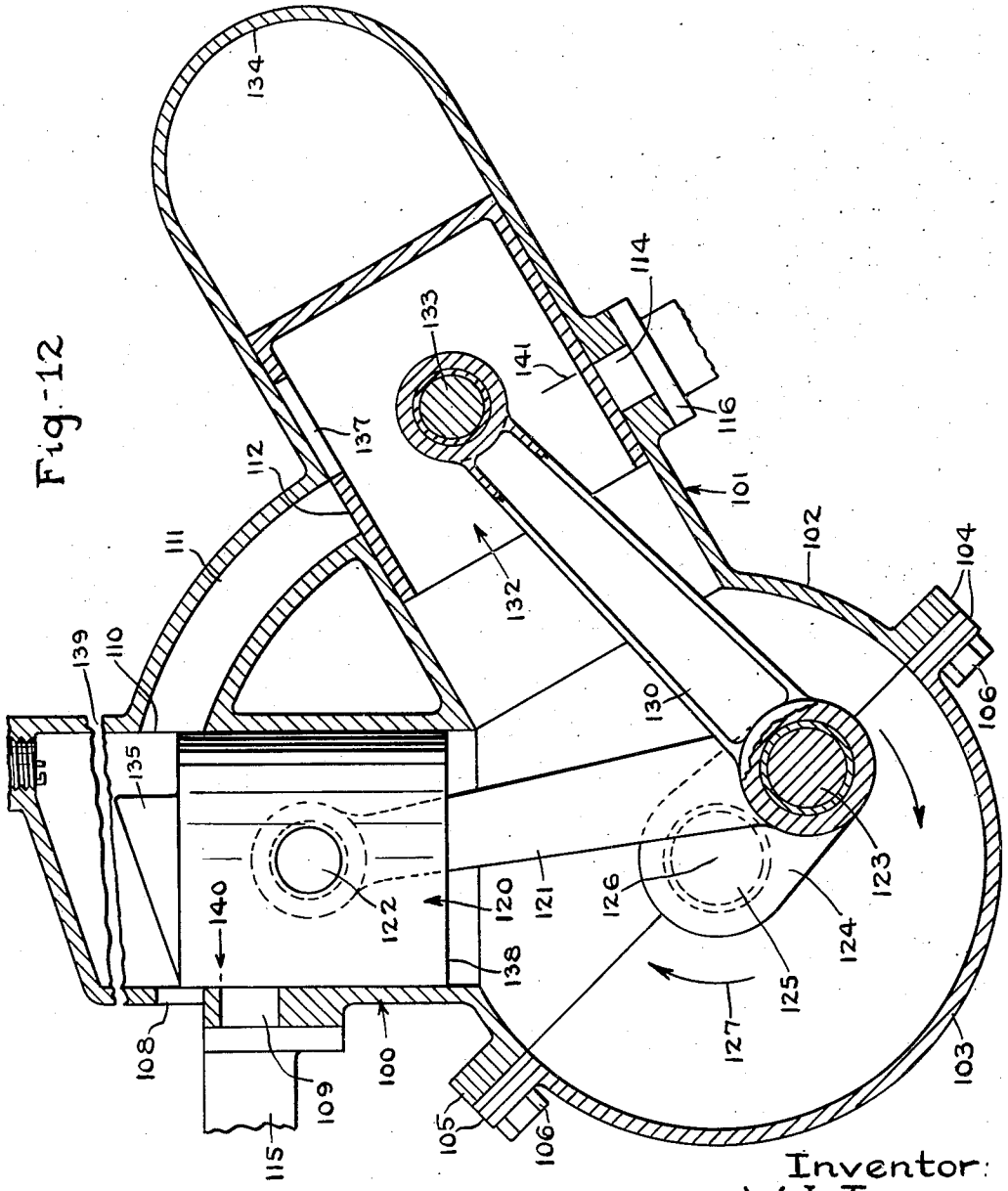
Fig. 12



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UNITED STATES PATENT OFFICE

2,522,649

TWO-STROKE CYCLE ENGINE CYLINDER AND PUMP

William L. Tenney, Crystal Bay, Minn.

Application October 6, 1945, Serial No. 620,707

6 Claims. (Cl. 123—70)

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This invention relates to internal combustion engines of the two-stroke cycle type wherein a piston reciprocating within a cylinder is employed as a pump for purposes of or for aiding in scavenging, charging or supercharging or all three, or any combination thereof, of the combustion cylinder. The primary object of the invention is to provide a most simple, positive, inexpensive means of controlling the connection of the pump air outlet with the engine combustion cylinder inlet ports, over and above the control provided by the piston of the engine combustion cylinder in acting as a valve means for opening and closing the engine combustion cylinder inlet ports which are connected to the pump outlet. The desirability for this additional control has long been recognized and various means of accomplishing the control have been proposed and to some extent employed.

One of the primary purposes of this additional control has been to insure the introduction and retention in the engine combustion cylinder of the greatest possible weight of fresh air or fresh fuel-air mixture. Thus, in two-stroke cycle engines of the type employing both inlet and exhaust ports located about the periphery of the lower end of the engine combustion cylinder with no control of the inlet air or fuel-air mixture other than that provided by the motion of the engine combustion cylinder piston, it is generally necessary to provide exhaust ports, the upper edges of which project higher in the plane of the piston top than the upper edges of the inlet ports. This is done in order that exhaust port discharge may lower the combustion cylinder pressure to or below the pressure of the inlet supply of fresh air or fuel-air mixture before the inlet ports are connected to the supply, so that flow of combustion cylinder contents at high temperature and pressure into the inlet supply is prevented. This imposes a definite disadvantage on the return or compression stroke for, in such a design, the combustion cylinder inlet ports necessarily close prior to the exhaust ports on the return stroke. As a result any substantial positive pressure which may have existed in the combustion cylinder prior to the closing of the combustion cylinder inlet ports is allowed to escape or be substantially reduced during the period in which the combustion cylinder inlet ports are closed and exhaust ports still open.

Under such conditions retention of fresh air or fuel-air mixture in the combustion cylinder is very considerably reduced in comparison with the results possible when the exhaust ports close

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first and the inlet ports close later. When the inlet ports close later, it is possible to establish a substantial positive pressure within the combustion cylinder at the time of final port closure, and this permits the retention of a maximum weight of fresh air or fuel-air mixture in the combustion cylinder by means of avoiding the loss that would otherwise occur with the exhaust ports still open when the inlet ports close. Additionally, a higher thermal efficiency is made possible due to a lower percentage loss of air out the exhaust ports which results in lower pumping losses and, in the case of carburetor engines, in less loss of unburned fuel out the exhaust port. Further, a somewhat longer expansion stroke is attained tending to result also in higher thermal efficiency.

It has been found that in engines wherein the inlet and exhaust ports are positioned at the same end of the cylinder and the upper edges of the inlet ports project higher in the plane of the piston top than the exhaust port, control means must be provided between the fresh air or fuel-air mixture supply and the combustion cylinder inlet ports in addition to that provided by motion of the combustion piston in acting as a valve. When this is done, blow-back of products of combustion at high pressure and temperature into the inlet air or fuel-air mixture supply does not occur during the period that the combustion cylinder inlet ports are open and the pressure in the combustion cylinder is substantially above that of the inlet air or fuel-air mixture supply.

In addition to the purpose set forth above, additional control means of the inlet ports has been provided in order to divide the inlet period into two or more stages for purposes of providing improved scavenging (Bokemuller Patent 2,113,979); so as to permit division of the inlet air or fuel-air mixture into separately timed scavenging and supercharging flows; and for various other purposes. Separation of the scavenging air from the supercharging air is desirable for two reasons. First, the scavenging air is generally required in larger quantities and at lower pressures, whereas the supercharging air is generally required in smaller quantities and at higher pressures. Additional control of inlet ports permits the scavenging and loading air to be supplied at low pressure, and supercharging air to be supplied from a separate compressor at higher pressures, resulting in the type of compressor most suitable for each function being utilized (Garve et al. 2,216,074), and hence in

lower pumping losses. Also, in carburetor engines air alone may be used for scavenging and charging and an over rich fuel-air mixture supplied for supercharging, resulting in lower specific fuel consumption due to less unburned fuel or no unburned fuel being lost out the exhaust port.

The most widely employed means heretofore available for exercising additional control has been the use of automatic valve controlled ports, and these have been of limited application in slow rotating, compression ignition engines of the oil-burning type. The automatic valves have been of the spring or gravity controlled types which are held closed by the pressure within the engine combustion cylinder until the exhaust port discharge or "blow-down" has lowered the combustion cylinder pressure to or below the pressure of the inlet supply of fresh air or fuel-air mixture, at which time the inlet supply pressure opens the spring or weight controlled valves, allowing discharge from the supply through the valves and in turn through the combustion cylinder inlet ports. The use of such automatic valves is limited to engines of relatively slow rotative speeds since the inertia of the valves themselves, in addition to the spring or weight loading, causes a lag in operation rendering them unsuitable at relatively high rotative speeds such as are generally required in portable, automotive or aircraft engines.

For engines of relatively higher rotative speeds various types of mechanically controlled valve ports have been employed, perhaps the most widely used having been the rotary valve (Dusenberg racing car engine, Indianapolis races 1926, the Violet engines made in France, the Straub Patent 2,265,677 and Bokemuller Patent 2,113,979). In addition, the well known opposed piston uniflow-type of Diesel engine (Junkers aircraft and truck engines) as well as the U-cylinder type engines (the pre-war Austrian Puch motorcycle engine, the present day General Motors aircraft engine, etc.) afford a similar type of operation by a different method, in that they keep the inlet port open after the closing of the exhaust port, but at the cost of even greater mechanical complication and a great deal more expensive engine structure. It may thus be seen that the addition of this control means has always entailed additional moving parts with resultant greater possibility of wear and mechanical trouble, as well as additional initial and maintenance expense and complexity in the engine constructon.

The primary object of this invention is to provide control of the inlet air or fuel-air mixture in addition to that provided by the movement of the piston in the combustion cylinder, for any of the purposes outlined above or for any combination of such purposes, wherever a reciprocating piston type pump is used as an air source for any inlet ports of the engine combustion cylinder and where the pump piston does not operate in phase with the combustion cylinder piston. In accordance with the present invention, additional combustion cylinder inlet control is accomplished by employing a section of the pump piston wall or means moving with it, which acts in cooperation with a port in the pump cylinder wall to provide the desired additional control. By this means it becomes possible to exercise additional control over the engine combustion cylinder inlet ports supplied by the pump without any additional mechanism or complexity such as has previously been required in the form of rotary valves, plate valves, spring loaded valves, etc.

Another object of the present invention is to provide means for controlling the air or fuel-air mixture flow from a reciprocating piston type charging pump through the combustion cylinder inlet ports of a two-stroke cycle engine so that pressure and volume of flow suited to the functions of scavenging, charging and supercharging are provided at the appropriate periods in the engine cycle when each of these functions takes place.

The following description relates to illustrative applications of the method and apparatus of the present invention to simple two-stroke engines of the type wherein the inlet and exhaust ports lie in the same end of the cylinder, and where a single pump is used for scavenging, charging and supercharging. There are a very wide variety of more complicated engine designs in which the present invention is applicable, and space precludes mentioning very many of them. Simple illustrations have been chosen, but it is to be distinctly understood that the illustrations do not limit the scope of the invention to the particular arrangements and purposes illustrated. The methods of the present invention are equally applicable for any of the purposes wherein additional inlet port control is desirable. They are likewise applicable to any physical arrangement of the charging pump in relation to the engine combustion cylinder, for example in V at other angles than as shown, in radial engines wherein the pumps lie in the same plane or in front of or behind the engine combustion cylinders, for W and X type combinations, for different combinations of in-line, multiple pumps and combustion cylinders, for arrangements wherein the pumps may lie parallel to the crankshaft or for any other pump to cylinder arrangement or engine cylinder arrangement or pump cylinder arrangement desired. It is also applicable to double acting pumps and double acting combustion cylinders in any suitable arrangements or combinations.

The invention is illustrated by the drawings in which Figures 1 through 5 are a set of related views of an engine having a combustion cylinder and pumping cylinder in line. Figure 1 is a side elevational view, partly in section, showing the piston of the combustion cylinder in the course of its power stroke. Figure 2 is a fragmentary side elevational view, partly in section and partly broken away, corresponding to Figure 1 and showing the piston of the combustion cylinder when the exhaust port is about to be opened. Figure 3 is a view corresponding to that shown in Figures 1 and 2 and showing the position of the combustion cylinder piston when the exhaust port is partly open and with the additional inlet port control valve about to open. Figure 4 is a view corresponding to Figures 1, 2 and 3 showing the piston in the combustion cylinder at the bottom of its stroke with the exhaust and inlet ports full open and the additional port control valve of the pumping piston likewise fully open. Figure 5 is a view corresponding to Figures 1–4 showing the piston of the combustion cylinder just after it has closed off its exhaust port and with the combustion cylinder inlet port and additional port control valve still open.

Figures 6–10 are diagrammatic views of the engine crank and connecting rods from a position facing the combustioin cylinder from end **23** of crankshaft **20**. Figures 6–10 correspond to Figures 1–5, respectively, and show the angular re-

lationship of the cranks and connecting rods and the positions of the piston pins of the combustion cylinder and pump cylinder. The notations on Figures 6 through 10 indicate the timing of the ports, the functioning of the additional inlet port control and various other operational factors.

Figure 11 is a cross sectional view partly broken away illustrating an application of the principles of the invention to a 90° V-type arrangement of combustion cylinder and pump cylinder.

Figure 12 is a cross sectional view showing an application of the principles of the invention to another illustrative type of engine wherein an auxiliary pump cylinder and a combustion cylinder are arranged in V at 60° and the air or fuel-air mixture is compressed in the crankcase.

Figure 13 is a fragmentary elevational view, partly in section, taken along the lines 13—13 of Figure 2, and illustrates a modified form of port shape used under certain conditions.

Throughout the drawings corresponding numerals refer to corresponding parts.

Referring to Figures 1–10 there is illustrated an engine of the type having the combustion cylinder generally designated 10 arranged in line with the pumping cylinder, generally designated 11. In the illustrated engine the combustion cylinder 10 and pumping cylinder 11 are cast integral with the upper half 12 of the crankcase. The crankcase is provided with an open or closed internal web 13 between the combustion cylinder and pumping cylinder which serves as a support for the upper half of bearing 14. The upper half of the crankcase is adequately braced at 15 and 15A. The lower half of the crankcase is illustrated at 16 and is fastened by suitable retaining cap screws 17 arranged at the connecting flanges 18 and 19. The lower half of the crankcase is adquately braced at 24 and has an internal open web 26 which supports the lower half of bearing 14.

A crankshaft generally designated 20 is rotatably supported in bearings 22 and 14, the power take-off of the crankshaft being at end 23 which is shown broken off. It will be understood that suitable ignition means are provided where a spark ignition type engine is indicated and that a suitable fuel injection pump and injection means together with spark plugs, if desired, may be used in the Diesel, semi-Diesel, or gasoline injection types of engines. A spark plug is indicated at 21, but it will be understood that this may be replaced by a suitable injection nozzle or combined spark plug and fuel injection nozzle. Where fuel injection is utilized this may be done at the inlet port 27 or elsewhere, if desired.

In the drawings, the cooling apparatus, necessary with any type of engine, has been eliminated for purposes of clarity, it being understood that the cylinders will be provided with suitable water jackets, or fins for air cooling, for the combustion cylinder, and also for the pump cylinder if desired. Likewise, for purposes of clarity, the lubrication system of the engine has been omitted, but it will be understood that adequate pressure or splash lubrication will be provided. In addition, piston rings and counterweights on the crankshaft have been omitted but it will be understood that they will be used where desirable.

The combustion cylinder is provided with an exhaust port 25 which, it is understood, is indicative of one or more exhaust ports arranged around a segment of the cylinder wall. At the opposite side of the combustion cylinder there are provided one or more inlet ports 27 which are connected through the passageway 28 to corresponding port or ports 29 in the pump cylinder 11. It will be understood that the position of the inlet port 27 and the exhaust port 25 may be at any desired position around the periphery of the cylinder to accomplish any system of scavenging flow feasible and desired.

The crankshaft 20 is provided with a crank 30 and crank pin 32 on which there is mounted a connecting rod 31 having the usual removable large-end bearing cap. The connecting rod supports a piston generally designated 33. The piston pin 34 is journaled in the connecting rod as indicated at 35, the piston pin itself being locked in place by a retaining screw 36. The piston pin is either solid or is plugged so as to prevent any direct passage of gases through from the inlet to the exhaust ports where those ports are oppositely located. In Figures 1–5 the piston 33 is indicated as being provided with a deflector 37 so as to direct the flow of the inlet gases through a scavenging flow generally indicated by the arrow 38 of Figure 4, but a flat or shaped-top piston may be used according to the system of inlet and exhaust port design and scavenging flow utilized. The flow of scavenging air or fuel-air mixture can, to some extent, be controlled by the direction and position of the inlet and exhaust ports and these are correlated with the piston top shape to provide the scavenging flow desired, which, per se, forms no part of the present invention.

The crankshaft 20 passes through the bearing 14 and is provided with a second crank 40 and crank pin 42 on which there is mounted a connecting rod 41. Connecting rod 41 supports the pump piston generally designated 43. The piston 43 is provided with a solid or plugged piston pin 44 which is journaled in the connecting rod as shown at 45, the piston pin being attached solidly to the piston by means of retaining screw 46.

It is a purpose of the invention to provide a pump piston having a control port therein. An example of one form in which the piston may be made is shown at 43 in Figures 1–5, although many other forms are possible. The piston 43 is provided with a passageway 48 which is formed by the piston side wall 49 or an extension thereof (as shown in the modified form, Figure 11) and an internal web 50 which is closed off at the bottom as shown at 51. The passageway 48 is ported through the top surface of the piston as indicated at 53 and is ported through the side wall of the piston as indicated at 55. The piston 43 is illustrated as having one such passageway 48 but it will be understood that where desired a number of passageways 48 may be utilized. The location, shape and direction of the passageway or passageways may be varied to fit the requirements of the design. Likewise there may be provided a number of ports 29 to cooperate with the passageway or passageways 48 and ports 55 in the piston 43.

The displaced volume of cylinder 11 is preferably made substantially greater than the volume of cylinder 10 in order to provide supercharging as hereinafter described. The inlet to the pumping cylinder 11 is through port 56 and flange 57, the latter being connected to a carburetor or to an air supply, depending upon the system of engine operation utilized. Thus, in the case of a carburetor type engine the flange 57 represents the flange of the carburetor. In the case of fuel injection engines the flange 57 represents a suitable connection to atmospheric air or to an air

supply under pressure, or to a fuel injection means, or to an air cleaner, where utilized.

As illustrated, port 56 represents the inlet port for the air or fuel-air mixture utilized in the engine. If desired, port 56 may be located in the upper part of the cylinder as illustrated by the dotted lines 56' and where control is provided by any suitable valve means.

Referring to Figures 1–5 and also to Figures 6–10, which correspond to Figures 1–5, it will be observed that the connecting crank 30 is arranged at an angle to crank 40. Crank 40 is arranged at an angle with respect to crank 30 so that at the time of final combustion cylinder inlet port closure, the piston 43 of the pump cylinder will be at or near the end of its pumping stroke. The angular relationship between cranks 30 and 40 is of importance in that it controls the travel of pump piston 43 on its compression stroke in timed relation to the functions of scavenging and supercharging in the combustion cylinder 10, in addition to its effect on engine balance. The importance of control of the travel of pump piston 43 on its compression stroke in timed relation to the functions of scavenging and supercharging will become apparent as description of the engine cycle progresses.

The direction of rotation in Figures 6–10 is counterclockwise. In the particular engine design illustrated the pumping cylinder crank 40 leads the engine cylinder crank 30 by 120°, although it will be understood that the exact angularity may be varied within limits so long as the phase relationship of the pump piston and engine piston is such that the pump piston is at or near the end of its compression stroke at the time of final closure of the combustion cylinder inlet port, and reasonable engine balance is retained. The angularity, of course, depends upon whether the cylinders are in line or displaced with respect to each other as will be more apparent with reference to the modifications shown in Figures 11 and 12. The 120° displacement of Figures 1–10 represents a reasonable setting for the engine illustrated.

In Figures 1 and 6 the combustion piston power stroke which began at A has continued through approximately 60 degrees to point B. At this time the pumping piston is at its lowermost position F, as represented by crank 40. In this condition of operation the exhaust port 25 and inlet port 27 of the combustion cylinder are closed. The pump cylinder outlet port 29 is uncovered by the wall 49 but no flow takes place due to the blocking of this port by the combustion cylinder piston 33. At this time the inlet air or fuel-air mixture continues to enter through the then open inlet port 56 of the pumping cylinder.

In Figures 2 and 7 the cycle has progressed to the point at which the combustion cylinder inlet port 27 has already opened slightly, the opening point of which is indicated at C on the phase diagram, Figure 7, and the exhaust port 25 is just about to be opened, as indicated at point D on the phase diagram. At this time the products of combustion are under pressure in cylinder 10, but flow does not occur back through the pump cylinder outlet port 29 due to the fact that the pump piston 43 has in the meantime moved to point I on the phase diagram, at which time the piston side wall 49 has reached a position closing off the outlet port 29 of the pumping cylinder. The inlet port 56 of the pump cylinder is about to be closed off in Figures 2 and 7.

As the engine rotates to the condition shown in Figures 3 and 8, the engine piston 33 has moved to point E and has partly uncovered the exhaust port 25, and the inlet port 27 of the combustion cylinder is additionally uncovered as compared to Figure 2. At this time the pump piston 43 has moved upwardly to the point J on the phase diagram, Figure 8, at which time the piston side wall 49 is about to bring port 55 of the piston into registry with port 29. During the period between the conditions shown in Figures 2 and 3, exhaust gases under pressure in the combustion cylinder 10 have discharged through the exhaust port 25, the desideratum being that the exhaust gases be permitted in this interval to discharge down to approximately the pressure developed above the piston in the pumping cylinder 11.

In Figures 4 and 9 the combustion cylinder piston 33 has reached point F on the phase diagram, Figure 9, and the pumping piston 43 has reached point K. Under these conditions the combustion cylinder piston 33 has completely uncovered the exhaust port 25 and inlet port 27 and the pump cylinder side wall 49 has moved to a position such that port 55 registers with the outlet port 29 of the pump cylinder. Air or fuel-air mixture which has been partly compressed by the action of the pump piston 43 therefore flows freely through the then open passageway 28 and strikes the piston deflector 37 of the combustion cylinder piston 33, deflects upwardly through the cylinder 10 and drives the exhaust gases through the exhaust port 25. This action continues through the period representing the change from the position shown in Figure 4 until shortly before the condition shown in Figure 5.

Figure 5 corresponds to the phase diagram shown in Figure 10. In this figure the combustion cylinder piston has moved to slightly beyond point of exhaust port closure, point G on phase diagram Figure 10, to point H and pump piston has moved to point A, i. e. the top of its stroke in the illustrated modification. Under these conditions the piston 33 has moved to a position so as to close off exhaust port 25, but inlet port 27 is still open as is also the passageway between the port 55 in piston 43 and the outlet port 29 in the pump cylinder wall.

It is intended that this engine be supercharged, and it is to this end that the pumping cylinder 11 is designated as of substantially larger displaced volume than the combustion cylinder 10. Supercharging in this type of engine may be described as the attainment of a pressure substantially above atmospheric pressure in the combustion cylinder at the time of final port closure. Supercharging in this type of engine does not occur solely in the period when piston 33 has closed exhaust port 25 while inlet port 27 still remains open to an air or fuel-air mixture supply under pressure, as has been commonly believed. A combination supercharge and scavenge occurs from the moment of inlet port flow because of the fact that the instantaneous open area of the inlet port is larger than that of the exhaust port. When air or fuel-air mixture is supplied under pressure through the combustion cylinder inlet ports and allowed to escape out the exhaust ports, the pressure drop will be greater across the smaller orifice, resulting in a pressure above atmospheric in the combustion cylinder.

The dominance of the supercharge effect over the scavenge effect tends to increase as combustion piston 33 rises from its lower center position

because of the fact that as it rises the instantaneous open area of exhaust port **25** becomes an increasingly smaller percentage of the instantaneous open area of inlet port **27**, resulting in increasing pressure within combustion cylinder **10** even before the exhaust port **25** is finally closed. At the point of closure of exhaust port **25**, the scavenge effect is ended and the supercharge effect continues alone until closure of inlet port **27**, provided the pressure from the inlet supply is greater than that existing in the combustion cylinder during this period. It will be noted that upward motion of combustion piston **33** after closure of exhaust port **25** in itself tends to produce an increased pressure in cylinder **10** through its compressing motion. If the pressure of air or fuel-air mixture inlet supply is insufficient during this period, back-flow through inlet port **27** will occur with consequent loss in combustion cylinder pressure and supercharge. The pump piston **43** is timed to be at or near the end of its compression stroke at the time of final port closure in order to insure a sufficiently high pressure to prevent back-flow at this point in the cycle.

It may thus be seen that a combination scavenge and supercharger occurs from the moment of inlet port **27** opening and that from the lower center position of combustion piston **33** until closing of exhaust port **25** the supercharge effect gradually becomes dominant. Scavenging ceases with closure of exhaust port **25** and supercharging alone continues until final inlet port closure.

In an engine of the type illustrated in Figure 1–10 wherein both the scavenging and supercharging air or fuel-air mixture are supplied from a single source, the piston type charging pump may be employed for both functions to great advantage when the travel of the pump piston **43** on its compression stroke is correctly controlled in timed relation to the occurrence of these functions in the combustion cylinder **10** with the result that the supply of air or fuel-air mixture is delivered through inlet port **27** at the most appropriate pressures and volumes.

It is well known that the scavenging air or fuel-air mixture is best supplied at lower pressures corresponding to the pressures required to pass the desired amount of the scavenging medium through the inlet port **27**. It is likewise known that the supercharging supply is required at higher pressures. These two requirements may be met with the same pump when a reciprocating piston type of pump is used, as in illustrations Figures 1–10, by so timing the travel of pump piston **43** that little compression has occurred before the opening of pump outlet port **29** and commencement of the scavenging process. Accordingly, the scavenging air or fuel-air mixture is supplied at the start at a low pressure, as required. An appropriate part of the travel of pump piston **43** is then reserved for the wide open port condition of combustion cylinder **10**, so as to continue the scavenge at low pressure.

As has already been pointed out, a slight supercharge effect occurs from the moment of commencement of flow through the inlet port, with the supercharge gaining dominance as the combustion cylinder piston **33** rises from its lower center position. Accordingly, the air or fuel-air mixture should be supplied at increasing pressure through inlet port **27** from the lower center position of combustion cylinder piston **33** until inlet port **27** has been closed. This is accomplished by timing the travel of pump piston **43** so that it reaches substantially the end of its compression stroke at the time of closing of combustion cylinder inlet port **27**. The end of the compression stroke of pump piston **43** corresponds to the least clearance volume of the pump and hence the highest pressure. Inlet port **27** being gradually reduced in area by the motion of piston **33** gradually increasing pressure occurs as a result of travel of pump piston **43**, in accordance with the pressure requirements previously defined. Stated briefly, the pump pressure rises as the supercharge effect gains dominance over the scavenge, and is at its highest pressure at the termination of the supercharge period. Thus, the pump pressure corresponds to the known pressure requirements of scavenging and supercharging when its piston travel occurs in correctly timed relation to the scavenging and supercharging functions of combustion cylinder **10**.

Control over the volumes of air or fuel-air mixture supplied through combustion cylinder inlet port **27** during the scavenge and supercharge periods, respectively, is also of great importance. Such volume control is exercised when a piston type charging pump is employed and the travel of the pump piston **43** is controlled in correctly timed relation to the occurrence of the scavenging and supercharging functions in combustion cylinder **10**. The volume requirements are met by control of timed relation of travel of pump piston **43** to combustion piston **33** which corresponds well with the timed relation dictated by pressure requirements as described above. Hence, a pump having optimum timing of piston **43** will fulfill both the pressure and volume requirements.

In general, it is required to keep the volume supplied for scavenging at a desirable minimum. This requirement is dictated by the fact that the larger the volume of the scavenging medium supplied, the larger the percentage of that volume is lost out exhaust port **25** without taking part in the combustion. This results in high fuel consumption in carburetor engines, and in any case in pumping losses which reduce the brake thermal efficiency of the engine. The entire amount of air or fuel-air mixture supplied as a supercharge, however, is retained in combustion cylinder **10** and takes part in the combustion. It is therefore desirable to keep the supercharge volume supplied as large as practical.

In order to keep the scavenging delivery at a desirable minimum, it is necessary to control the motion of pump piston **43** so that its travel is limited during the period of flow through combustion cylinder inlet port **27** when combustion cylinder piston **33** is approaching, reaching and passing the lower center position, for it is in this period that the scavenge effect is dominant. Thus, a large portion of the compressing travel of pump piston **43** is reserved for the remainder of the inlet port **27** open period, and a large volume is likewise reserved for the period in which the supercharge effect becomes dominant and finally replaces the scavenge effect.

From the foregoing it becomes evident that by arranging the travel of pump piston **43** on its compression stroke to occur in correctly timed relation to the occurrence of the scavenging and supercharging functions in combustion cylinder **10**, the pressure and volume requirements appropriate for each function, (i. e. scavenging and supercharging), can be met by one pump. It is an object of this invention to provide en-

gines in which such results are accomplished.

Fortunately, the appropriate timed relationship requisite to correct pressures and volumes for both scavenging and supercharging can be accomplished by linking pump piston 43 by means of connecting rod 41 to crank 40 which is displaced in phase with respect to crank 30 so that pump piston 43 is brought substantially to the end of its compression stroke at time of closure of combustion cylinder inlet port 27, with little compressing motion of pump piston 43 occurring before scavenge flow is allowed to begin through combustion cylinder inlet port 27. Likewise, a large portion of the compression stroke of pump piston 43 is reserved for supercharging, and thus the requirements are met.

So far in the discussion it has been assumed for purposes of clarity that gas inertia, R. P. M. and fluid friction effects are non-existent. Actually, such effects must be taken into account and will, of course, affect the correct angular displacement of cranks 30 and 40. It will be noted that with the angular displacement embodied in illustrations 1–10, the control of pump piston 43 is such that at any given moment piston 43 is somewhat further along in its travel than would be required by the reasons previously given, and this is purposely so in order to compensate for effects of gas inertia, R. P. M. and fluid friction.

While angular displacement of cranks 30 and 40 is the principal means of controlling the travel of pump piston 43 in accord with the requirements of supercharging and scavenging pressures and volumes, it is not intended that this be the sole means. In addition, advantage may also be taken of relationships of angularity gained by displacing either combustion cylinder 10, or pump cylinder 11 or both, so that while their center lines remain parallel, they no longer intersect the center line of the crankshaft. Ports 55 or 29, or both, may also be shaped or timed, or both, so as to restrict or promote flow from pump cylinder 11 at the desired times. Figure 13 shows a sectional view through passageway 48 in Figure 2 illustrating how port 55 may be shaped to accomplish some restriction in the initial flow of the charge. The full lines 55 would allow a maximum flow. By narrowing the port to the shape shown by dotted lines 59, only a small initial width of port is opened and this increases as piston 43 moves upwardly. Any of these facilities may likewise be used in the engines illustrated in Figures 11 and 12.

Referring to Figure 11 there is illustrated an engine having a combustion cylinder generally designated 60 arranged at right angles to a pump cylinder generally designated 61, the two cylinders being cast integral with the upper half of the crankcase 62. The lower half of the crankcase is illustrated at 63 and is held in place by suitable attachment studs or cap screws 64. The pump cylinder 61 has a larger diameter and displaced volume than the combustion cylinder 60. In Figure 11 the cooling arrangement, lubrication facilities, piston rings, crankshaft counterweight, fuel injection or ignition system or both, and other auxiliaries have been omitted for purposes of clarity of the drawings.

The combustion cylinder is provided with an exhaust port 66 and an inlet port 67. The upper edge of inlet port 67 is at a higher level than the upper edge of exhaust port 66. The inlet port 67 is connected through passageway 68 to the outlet port 69 of the pump cylinder. As in the engine illustrated in Figures 1–5, the inlet port 67 and exhaust port 66 may be extended around a segment of the cylinder or plurality of inlet and exhaust ports may be provided around segments of the cylinder wall, and they may be shaped, positioned or directed so as, in cooperation with the shape of the piston top, to provide any desired system of scavenging flow.

Within the combustion cylinder there is positioned a piston 72 having deflector 73 thereon, the piston being mounted upon a connecting rod 75 which is journaled on crank pin 76, the latter being carried upon crank 77 pivoted about the crankshaft axis 78. The direction of rotation is as indicated by arrow 79. Alongside connecting rod 75 there is positioned a connecting rod 81 which serves to mount the pump cylinder piston 82 which is designed along the lines shown for the engine of Figures 1–5 except that the pump cylinder side wall which does the valving of the pump outlet port 69 is made partially as an extension 83 which is adequately braced to the pump cylinder piston by webs 85. The head of the pump cylinder piston is shown at 87 and connects to internal walls 88 and 89 leading to the skirt at 90. A passageway 92 thus extends from above the pump piston 82 to the port 94. The pump cylinder 61 is provided with an outlet port 69 which may be extended around a segment of the cylinder, if desired, or a number of ports may be used. Port bars may be supplied as needed. The upper part of the pump cylinder 61 is formed as indicated at 95 to conform closely to the shape of the extension 83 and bracing webs 85, where a high compression ratio is desired. When a lesser compression ratio is desired the upper part 95 of cylinder 61 may be shaped off to provide greater clearance and hence less compression.

Figures 1–5 and Figure 11 show two of the many possible variations in piston design that may be used in carrying out the present invention. The port of the pumping piston, for example port 94, may be shaped and proportioned, divided by bars or may be replaced by a multiplicity of smaller ports so as to control the scavenging and supercharging flow of the charge and provide adequate mechanical strength. Two or more passageways 92, ports 94 and side wall segments 83 may be used, if desired.

The inlet of air or fuel-air mixture to the pump cylinder 61 is introduced through any suitable port, here shown as the piston valved pump inlet port 70 which is illustrated as connected to flange 71. Any other suitable inlet port such as a port 70′ may be used, and, of course, a suitable valve is then used over the inlet port to hold the charge during the compression stroke and to allow introduction of the charge during the suction stroke. A plurality of inlet ports may be used, if desired.

The engine illustrated in Figure 11 is of the fuel injection type having a fuel nozzle 96 mounted at the top of cylinder 60, although it will be understood that any desirable position for nozzle 96 may be utilized. Thus, for example, in some instances it is desirable to locate the fuel injection nozzle in passageway 68 so as to supply the fuel into the combustion cylinder through the port 67. In the case of a fuel injection type engine flange 71 is merely connected to an air supply or to atmosphere. An air cleaner is, of course, used where desired. For carburetor type engines the nozzle 96 is replaced by a spark plug,

and flange 71 is then connected to a suitable carburetor.

In Figure 11 the engine is illustrated with the combustion cylinder piston 72 near the end of its stroke with the piston 72 in a position so as partly to uncover the exhaust port 66. Blow-back of exhaust gases through port 69 is prevented, however, due to the fact that the partially extended piston side wall 83 is in a position so as to cover the outlet port 69 of the pump cylinder. During this period when exhaust port 66 is open and port 69 is still closed, the exhaust gases above the piston 72 in the combustion cylinder 60 blow down through the exhaust ports 66 to a pressure which is desirably approximately equal to the pressure that exists above the piston 82 in the pump cylinder 61.

During the ensuing operations, the pump piston 82, in moving to the right in Figure 11, uncovers the port 69 and when the inlet port 70 is closed, the air above the piston 82 is forced through passageway 92 and thence through port 94 (which is then in registry with pump outlet port 69) and through the passageway 68, inlet port 67 against deflector 73 on the combustion cylinder piston 72.

It will be noted that there is a very short period in the cycle of valve operation shown in Figure 11 when inlet port 70 is open, ports 69 and 94 are in registry and inlet and exhaust ports 67 and 66, respectively, of the combustion cylinder are also open, or stated in another way, there is a through passage from inlet port 70 to exhaust port 66. This open-port condition occurs immediately after the exhaust gas blow-down period and the "carry out" effect of the exhaust gases may thus be utilized to draw in fresh charge gases. If it is not desired to utilize this effect it may be eliminated by timing the closure of port 70 to occur at approximately the opening of port 69.

The incoming gases scavenge the exhaust gases through the exhaust port 66. Then, as explained with reference to Figures 1–5, as the motion continues the exhaust port 66 begins to close and since its upper edge is at a lower position than the upper edge of the inlet port 67, the reduction in area of exhaust port 66 occurs and reaches the closed port position more quickly than the inlet port 67. In the meantime compression is occurring due to the action of pump piston 82 and the air above the piston is forced through the course described and into the combustion cylinder. The motion continues, and scavenging and supercharging occur, with the scavenging function predominating at first. The supercharging effect then becomes dominant and scavenging ceases as exhaust port 66 is closed, the supercharging being continued thereafter until the inlet port 67 is closed. When both the inlet port 66 and exhaust port 67 have been closed, the air above the piston 72 is compressed until the piston 72 reaches nearly the uppermost part of its stroke and fuel which has been introduced or is introduced at that instant is then ignited and the ensuing power stroke causes the piston 72 to be driven down through the successive operations described.

In Figure 11 the 90° displacement of the combustion cylinder 60 with respect to the pump cylinder 61 permits the use of a single crank 77, rather than displaced cranks, as illustrated in Figures 1–5. As will appear later in the discussion, the 90° angle has certain advantages but it is no wise intended that the angle of V, when the V arrangement of pump cylinder to combustion cylinder is used, shall be limited to 90° for purposes of the present invention.

The connecting rods 75 and 81 are shown as placed side by side on crank pin 76, but it will be understood that they may also be placed in the known fork and blade arrangement or that only one rod may be attached to crank pin 76, and the other rod articulated. Articulation may be specifically employed in order to bring about more favorable relationships between the pump piston and the combustion cylinder piston as to scavenging and supercharging pressure and volume requirements by means of taking advantage of the changed angularity relations brought about by the articulation. In addition for the same reasons, pump cylinder 61 or combustion cylinder 60, or both, may be offset so that the cylinder center line no longer intersects the crankshaft center line in order to alter the kinematic relationship of the pistons and ports, and shaping, proportioning and timing of ports 94 and 69 may also be employed, as explained with reference to Figures 1–10.

It will be noted that the section of pump piston wall 83 is formed in part as an extension of the pump piston wall proper. The port or ports 94 and attendant passageway or passageways 92 in pump piston 82 (which control piston pump outlet port 69) may be formed entirely in the piston proper, partially in the piston proper and partially in an extension, or entirely in an extension.

Varying the angle of V in the V-type pump to cylinder arrangement with a single crank pin has, of course, the same effect as varying the displacement of angle of cranks 30 and 40 in Figures 1 to 10. This effect has been quite thoroughly described with reference to Figures 1–10 and is applicable equally well to the arrangement of Figure 11. The 90° V arrangement has been chosen here, however, since it combines the possibility of good engine balance with a favorable control of travel of pump piston 87 in timed relation to the scavenge and supercharge periods. In addition, Figure 11 with its 90° arrangement, in comparison with the 120° angle of Figures 1–10, illustrates the variation in angularity that may be used without departing from the teaching of the invention, and these are by no means the limits. The timed relation of Figure 11 is suited to engines of slower revolution speed than in the case of Figures 1–10, since little allowance is made for effects of gas inertia, fluid friction etc.

If a crankshaft is employed having two cranks angularly displaced, one for the pumping piston and one for the combustion piston, it will be understood that correct timed relationship of pump piston and combustion piston may be maintained regardless of the angle of V so long as the combined angular displacement of cranks and cylinders is correct.

Little or no pressure is developed in the pump cylinder 61 prior to the opening of port 69, resulting in very little negative work being done in pre-compressing the air charge to higher pressures than necessary. In addition the scavenging flow and pressure is built up gradually, tending toward improved scavenging. Rapid travel of piston 87 occurs during the period that inlet port 67 and exhaust port 66 are fully open, or nearly so, thus timing the principal scavenge flow so that it takes place during the period when it is most effective. When the inlet port 67 is correctly proportioned the most effective scavenge pattern occurs with wide open ports,

Velocity of piston 87 on the compression stroke is maintained at a high level up to the time of final port closure, after a slow start, thus assuring high pressures and adequate volumes for supercharging purposes as have previously been described to be desirable.

From the above it may again be seen that a piston type of charging pump is highly desirable in this type of two-stroke engine since its pressure and discharge characteristics may be timed to meet the needs of the scavenging and supercharging processes. With rotary blowers either a constant, or worse, a falling pressure results during the inlet port open period, resulting in less efficient operation.

In Figure 11 as crank 77 revolves, piston 72 closes off inlet port 67, and at this time piston 82 is not yet at the end of its stroke, but has reached a point at which the piston velocity rapidly decreases, the piston stops and then begins to accelerate in the opposite direction. The charge is compressed to some degree but all or most of the energy is recovered by the immediately following expansion. In the 90° V design the advantageous gradually increasing and rapid motion portions of travel of piston 82 are reserved for the scavenging and supercharging functions. During the end of the period in which the piston slows down, dwells and reverses its direction, the piston merely performs an idle function.

Referring to Figure 12 there is illustrated another type of engine utilizing the present invention wherein the motion of the combustion piston and of an auxiliary pumping piston serves to introduce and compress the air or fuel-air mixture in the engine crankcase prior to transfer of the charge to the combustion cylinder. In Figure 12 combustion cylinder generally designated 100 and the auxiliary pumping cylinder generally designated 101 are cast integrally with the upper half 102 of the crankcase. The lower half of the crankcase is illustrated at 103 and is removably attached at flanges 104 and 105 by a plurality of cap screws 106.

As explained with reference to Figures 1–5 and 11, various constructional details, accessories and auxiliaries have been omitted from the drawings for purposes of simplicity. Thus, Figure 12 does not include a cooling arrangement, piston rings, carburetors or fuel injection devices, ignition devices, lubrication system, air cleaners, etc., but it will be understood that these and any other engine auxiliaries and accessories are supplied as needed.

The combustion cylinder is provided with an exhaust port 108, a combustion cylinder inlet port 110, and, if desired, a piston valved admission port 109 feeding the crankcase. Port 110 is connected through passageway 111 to the outlet port 112 of the auxiliary cylinder 101. The inlet of the auxiliary pumping cylinder into the crankcase is shown at 114. Each of the inlet ports, 109 of cylinder 100 and 114 of cylinder 101, is connected either to a carburetor, to a source of pre-compressed air or to atmospheric air through an air cleaner. The connection at 109 is made by way of flange 115 and the connection at 114 is made by way of flange 116.

In cylinder 100 there is a combustion cylinder piston generally designated 120 mounted upon a connecting rod 121 by means of wrist pin 122. The connecting rod 121 is mounted upon the crank pin 123 on crank 124 which pivots about the crankshaft 125, the axis of which is shown at 126. The direction of rotation is as shown by the arrow 127. Alongside connecting rod 121, or articulated thereto, is another connecting rod 130 which is connected to the piston 132 of the auxiliary pumping cylinder by means of wrist pin 133. The two connecting rods 130 and 121 may also be mounted by the known fork and blade arrangement.

The angle of V between cylinders 100 and 101 may be varied as hereinafter explained, even with the connecting rods mounted on the same crank pin. By using a crankshaft having two crank pins angularly displaced in the plane of rotation of the crankshaft the angle of V between the cylinders may be made anything desired.

The upper portion of the auxiliary pumping cylinder is provided with a closed dome 134 which, as illustrated, has neither inlet nor outlet ports, although if desired it may be provided with suitable inlet and outlet valves and used as an air compressor absorbing some or all of the power produced in the engine.

In cylinder 100 is a piston 120 having on its upper surface a deflector 135 which serves to produce satisfactory scavenging of the upper portion of the cylinder 100. The arrangement and number of inlet ports 110 may be varied widely, as explained with reference to Figures 1–5, so as to produce the desired scavenging pattern and flow in cooperation with the exhaust port or ports 108 and the piston top shape. The pump piston 132 is provided with a port 137 which is positioned so as to register with the port 112 and thus open and close port 112 during appropriate portions of the reciprocatory motion of piston 132 in its cylinder. In this way blow-back of combustion gases into the fresh air or fuel-air supply is prevented and the introduction of the fresh air or fuel-air is appropriately timed. As in Figures 1–10 and 11 a plurality of ports 137 and 112 may be used, and they may be shaped as suggested in Figure 13 so as to assist in timing the flow of charge.

The inlet of air or fuel-air mixture through port or ports 109 occurs when the piston within cylinder 100 has progressed upwardly in the cylinder far enough so that the under edge 138 of the skirt on piston 120 uncovers the port 109. It will be noted that the cylinder 100 is broken off at 139 indicating additional volume in the upper part of the cylinder 100 permitting the upward movement of the piston 120. The upper limit of the travel of piston 120 is such that the skirt 138 never goes above the lower edge of the exhaust port 108 and in the engine shown, the upper limit is at approximately level 140. The pump piston 132 likewise travels upwardly in its cylinder to line 141 so as completely to uncover the inlet port or ports 114.

It may be noted that the ports 109 and 114 are thus valved by their pistons 120 and 132, respectively, and that this represents a simple means of valving the inlet air or fuel-air mixture into the crankcase. However, the inlet into the crankcase may likewise, if desired, be effected through any suitable valve means.

In Figure 12 the piston 132 is about to bring port 137 into registry with the outlet port 112 of the cylinder 101. Prior to this time the partial opening of the exhaust port 108 has permitted the combustion gases to blow down to a pressure approximately equal to that existing in the crankcase at the time the port 137 is about to open port 112. It will be noted that with the

connecting rods 121 and 130 both mounted upon the crank pin 123, both pistons are operated in synchronism and in displaced phase such that pistons 120 and 132 are drawn downwardly in their cylinders at the same time through the major portion of their stroke. Thus, both contribute to the compression of the air or fuel-air mixture in the crankcase. Then, as port 137 is registered with port 112, the thus compressed air or fuel-air mixture is driven through ports 137 and 112 through the passageway 111 and through the inlet port 110 against the deflector 135 and the incoming stream of air or fuel-air mixture scavenges and charges the cylinder. As port 137 registers with port 112, the motion of piston 132 is about maximum for a given angular velocity of the crank and thus the compression of the gas in the crankcase is desirably large during the open-port condition. As piston 120 reaches its lower center position and begins then to travel upwardly in cylinder 100, it tends to decrease the compression in the crankcase, but phase displacement is such that piston 132 is, under those conditions, still moving downwardly at a sufficient rate to maintain the compression and produce supercharging. As piston 120 moves upwardly it restricts the opening through exhaust port 108 at a rate more rapidly than a restriction is produced in the inlet port 110 and consequently some supercharging occurs above piston 120 even before the exhaust port 108 is completely closed. Then after exhaust port 108 is completely closed, the final supercharging is applied through the then still open inlet port 110. As piston 120 moves up and closes inlet port 110, compression begins and near the top of the compression stroke ignition or fuel injection occurs in the usual manner.

In an engine of the type shown in Figure 12 where the air or fuel-air mixture is compressed in the crankcase, the lubrication may be provided by mixing the lubricating oil with the fuel or may be provided by a separate lubrication system metering oil directly to the cylinder walls and to the various bearings involved.

In operation the air in the closed space above piston 132 is simply compressed and decompressed as the engine revolves. The amount of work recovered by expansion of the air within the space is approximately equal to the amount of work involved in compressing the air during the preceding compression portion of the cycle. Accordingly, less loss is involved by closing cylinder 101 by means of dome 134, than by permitting air to rush into and out of the cylinder through a restricted opening.

Engines of the type illustrated in Figure 12 tend to eliminate the chief objections to the simple crank chamber type, as employed in most American outboard motors, model airplane engines, etc. at the expense of little additional mechanical complication. The chief attraction of the crank chamber type is its extreme mechanical simplicity, resulting in very low first cost. Lubrication is generally by means of oil mixed with the gasoline in the case of carburetor engines, eliminating the need for costly lubrication systems. The same type of lubrication may be employed in the type of engine illustrated in Figure 12.

The pressure and volume relationships of the usual crank chamber pump arrangement lead to inefficient operation, however. Scavenging takes place under conditions of decreasing pressure, and there is no possibility of a supercharge since

the displaced volume of the pump is equal to the displaced volume of the combustion cylinder. Inefficient consumption of the scavenging medium results, with low brake mean effective pressure, and, in the case of carburetor engines, high fuel consumption. In the type illustrated in Figure 12 the decreasing pressure in the crank chamber during the combustion cylinder inlet port open period may be largely eliminated, and in combination with the application of the present invention and greater pumping volume than combustion cylinder volume, supercharging may be obtained. Thus, the faults of inefficient consumption of the scavenging medium and low brake mean effective pressure may be largely cancelled out. In addition, starting is greatly improved since purity of charge in the combustion cylinder will approach that of four-cycle engines, whereas in the usual crank chamber type the charge is at least 50% diluted with products of combustion and at starting will only ignite under optimum or near optimum conditions.

It will be understood that the general efficiency of the type of engine illustrated in Figure 12 does not have the same possibilities as the types illustrated in Figures 1 to 11, nor will the starting be as easy with mixture of oil and gasoline and attendant oil-fouling possibilities. However, with careful attention to detail design, the general efficiency may be brought to nearly as high a level as that of the engines shown in Figures 1 to 11 and the starting qualities will be good.

In the illustrative example, Figure 12, the auxiliary pumping cylinder is shown at an angle of 60 degrees to the combustion cylinder. With this setting the falling pressure characteristic of the usual crank chamber two-cycle engine which occurs during the combustion cylinder inlet port open period, may be largely obviated. It must be remembered, however, that the effective volume of the crank chamber as a pump will be reduced as the differential motion of the pistons at upper and lower centers is increased, due to the increased period on the crank chamber suction stroke when the pistons are traveling in opposite directions. Thus, more favorable crankcase pressure-volume relations for scavenging and supercharging will under some conditions be obtained with a 90° cylinder setting but the effective suction volume of the crank chamber will be reduced. The reduction in effective suction volume can, however, be offset by the employment of an auxiliary suction cylinder of larger displaced volume.

It may be seen that the angle of V in the illustrative engine, Figure 12, and also in the illustrative engine, Figure 11, and as the angle of crank displacement is in the in-line engine, Figures 1–10, may be varied to achieve the requisite pressure-volume relationship for adequate scavenging and desired supercharging.

In V-type arrangements of the type shown in Figures 11 and 12, it will be understood that two crank pins, displaced at an angle around the axis of the crankshaft, may be employed with corresponding change in cylinder angularity where desired. Linking of the pumping piston to an eccentric attached to the crankshaft, rather than to a crank arm and crank pin, may also be employed in any of the types of engines illustrated herein.

With careful attention to details of design such as keeping the crank chamber clearance air space to a minimum, choosing favorable angles of auxiliary pump piston and combustion piston

crank and cylinder relationship, in combination with proper volume relationship, and by taking advantage of the angularity relationships made possible by means of offsetting cylinders and/or articulating one of the connecting rods, it is possible in an engine of the type illustrated in Figure 12 to achieve a pressure and volume relationship of the crankcase pump favorable both to the scavenging and supercharging functions. It should particularly be noted that construction of pump piston 132 may be varied by placing the piston head at or near the skirt end of the piston and allowing the wall of the piston and the attendant valving means to project above the head in order that the crank chamber clearance air space may be reduced. As in the case of engines illustrated in Figures 1 to 11, the timing and proportions of the port outlet and passageway in the pump piston may also be altered as suggested in Figure 13, for the purposes previously explained.

While the foregoing description has related purely to the application of the present invention to simple types of two-stroke cycle engines wherein all the combustion cylinder inlet ports are connected to the pumping cylinder outlet control port, it will be understood that the methods of the present invention are equally applicable when a separate set of combustion cylinder inlet ports is connected to a source of air or fuel-air mixture supply other than the pumping cylinder or when a separate set of combustion cylinder inlet ports is connected to a separate pump cylinder outlet control port, or in other combinations desirable to improve the engine efficiency in more complex types.

It will also be understood that it is sometimes desirable, especially in large engines, to position pump outlet control ports on opposite sides of the pump cylinder in order that back pressure from the combustion cylinder be balanced with regard to its effect on the pump piston and that this arrangement falls within the scope of the present invention.

The simple illustrative engine types have been employed for purposes of clarity and to demonstrate that the low brake mean effective pressure, hard starting and high fuel consumption generally associated with the simplest type of two-stroke cycle engine may be overcome by the application of the principles of the present invention and that this may be accomplished at very little additional expense in engine structure or mechanical complication. It will, of course, be understood that the illustrative engine types may be employed using a plurality of cylinders.

In accordance with the principles of this invention the timing of the flow of air or fuel-air mixture into the combustion cylinder is time-related to the volume-pressure characteristics of the pumping cylinder. As illustrated by the examples given, this time relationship is accomplished by operating the combustion cylinder piston and pumping piston in synchronism and in displaced phase, the displacement in phase being produced by an angular arrangement of the cranks which operate the pistons, by an angular arrangement of the cylinders, or both. Further variations in phase are available to the designer by utilizing articulation, offsetting the cylinders, etc., as previously explained.

In each of the illustrated examples of engines utilizing the invention additional control valving of the flow from the pumping cylinder is accomplished by simple porting of the pumping cylinder in cooperation with an appropriately constructed pumping piston, but any other valve which is timed to the operation of the pumping piston may be used in utilizing the volume-pressure characteristic of the pumping cylinder in the relationship herein outlined. In the modification shown in Figures 1–5 and in Figure 11 the entire pumping effect is produced by the pumping cylinder, whereas in Figure 12 cylinders 100 and 101 cooperate to produce the pumping effect. In each instance, however, the timing of the flow of the air or fuel-air mixture into the inlet port of the combustion cylinder is controlled not only by the inlet porting arrangement of the combustion cylinder but also by the outlet porting arrangement in a pumping cylinder wherein the piston operates in timed relation to the piston in the combustion cylinder, and the supercharging and scavenging flows are further controlled by the pumping characteristics of the pumping cylinder.

In each of the illustrated examples the combustion cylinder inlet port has been shown as opening prior to the combustion cylinder exhaust port and closing later than the combustion cylinder exhaust port. However, if only a mild supercharge is desired, the inlet and exhaust ports may be made to open and close simultaneously. Then if the exhaust outlet is restricted, for example, by shaping one or both of the ports so that the exhaust port momentary open area is smaller than the intake port momentary open area at each position of the piston, the desired pressure rise in the cylinder will be effected and a mild supercharge will result.

In any of the engines made in accordance with the present invention, backfire screens and/or throttles may be provided in the communicating passageway between the combustion cylinder inlet port and the pumping or auxiliary cylinder outlet port.

It is to be understood distinctly that the invention is applicable to reciprocating installations wherein the reciprocating action of the combustion cylinder piston is utilized to operate a reciprocating load, as for example in a direct connected air compressor or in an engine operated hammer. In such installations the connecting rod and crank arrangement is replaced by appropriate connection between the combustion cylinder piston and the load, and the combustion cylinder piston is connected to the pumping or auxiliary piston by any suitable reciprocating mechanism such as a bell crank.

As many apparently widely different embodiments of this invention may be made without departing from the spirit and scope thereof, it is to be understood that I do not limit myself to the specific embodiments herein except as defined by the appended claims.

What I claim is:

1. In an internal combustion engine of the two-stroke cycle type, a combustion cylinder and a pump cylinder, said combustion cylinder having inlet and exhaust ports and said pump cylinder having inlet and outlet ports, at least the inlet port of the combustion cylinder and the outlet port of the pump cylinder being connected together, said combustion cylinder exhaust port being closed prior to the closure of the inlet port, means forming a part of the pump piston cooperating with the outlet port of the pump cylinder for opening and closing the connection between the pump cylinder outlet and the combustion cylinder inlet, means for moving said pistons in synchronism and in displaced phase,

said pump piston being actuated during its compression stroke for initiating the movement of a controlled volume of charge from the pump cylinder to the combustion cylinder prior to the time the piston therein reaches lower dead center and then continuing the movement of said charge, said pump outlet port being positioned and the shapes of said outlet port and means on said pump piston being correlated for restricting the earlier part of the flow from the pump cylinder to the combustion cylinder.

2. A two-stroke cycle engine comprising a combustion cylinder having a combustion cylinder piston therein, a crankshaft-connecting rod mechanism connected to the combustion cylinder piston for reciprocating it in the combustion cylinder, inlet and exhaust ports located in the combustion cylinder wall so as to be opened in the order named as the piston moves through its power stroke and to be closed in reverse order during the ensuing reverse stroke of said piston, a pump cylinder having a piston therein connected to the crankshaft-connecting rod mechanism so as to be reciprocated in constant angular synchronism with the combustion cylinder piston, inlet and outlet ports in said pump cylinder, said outlet port being connected to the combustion cylinder inlet port, and means forming a part of the pump cylinder piston for maintaining the pump cylinder outlet port closed until after the exhaust port of the combustion cylinder has been opened, and for opening the outlet port of said pump cylinder prior to the time the combustion cylinder piston reaches the end of its power stroke at lower dead center.

3. An internal combustion engine of the two-stroke cycle type comprising a combustion cylinder and a pump cylinder each of which has a piston therein, an exhaust port and an inlet port in the combustion cylinder side wall arranged so that the inlet port will be opened before the exhaust port is opened when the piston of the combustion cylinder moves through its power stroke and closed in reverse order when the piston moves in its compression stroke, an inlet port and an outlet port in the pump cylinder at least the latter being in the pump cylinder side wall and connected to the combustion cylinder inlet port, means for reciprocating the pistons synchronously with the pump cylinder piston leading by a constant displacement of crank angle whereby the pumping stroke approaches completion during the beginning of the compression stroke in the combustion cylinder, and means forming a part of the pump cylinder piston cooperating with the pump cylinder outlet port for timing the beginning of flow between the pump cylinder to the combustion cylinder after the combustion cylinder piston has opened the inlet and exhaust ports thereof and before said piston has reached its lower dead center position.

4. The apparatus of claim 3 further characterized in that the cylinders are in line and each piston is provided with a connecting rod which is connected to a crank pin on a crankshaft, the phase displacement being accomplished by providing an angular displacement between the cranks of the crankshaft.

5. The apparatus of claim 3 further characterized in that the cylinders are arranged in V and each piston is connected by a connecting rod to a crank pin on a crankshaft, the phase displacement being provided by a displacement in the cylinders with respect to each other.

6. An internal combustion engine of the supercharged two-stroke cycle type comprising a combustion cylinder and a pump cylinder each of which has a piston therein, said pump cylinder having a substantially larger volume than the combustion cylinder, an exhaust port and an inlet port in the combustion cylinder side wall arranged so that the inlet port will be opened before the exhaust port is opened when the piston of the combustion cylinder moves through its power stroke and closed in reverse order when the piston moves in its compression stroke, an inlet port and an outlet port in the pump cylinder at least the latter being in the pump cylinder side wall and connected to the combustion cylinder inlet port, means for reciprocating the pistons synchronously with the pump cylinder piston leading by a constant displacement of crank angle whereby the pumping stroke approaches completion as the combustion cylinder inlet port is closed by the piston in said cylinder, and means forming a part of the pump cylinder piston cooperating with the pump cylinder outlet port for timing the beginning of flow between the pump cylinder to the combustion cylinder after the combustion cylinder piston has opened the inlet and exhaust ports thereof and before said piston has reached its lower dead center position.

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