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(54) **A two-stroke cycle petrol engine**

(57) A two-stroke cycle power cylinder-piston assembly 100 has two horizontally opposed pistons and incorporates uniflow scavenging, a reciprocating type scavenging pump means including at least one pump cylinder-piston assembly driven by the engine in synchronization therewith, and has a two-stage carburetor 500 which incorporates, in addition to normal throttle valves 510, 516, a control valve 520 which automatically increases its opening as the rotational speed of the engine increases. Valve 520 is actuated electromagnetically by a voltage which increases with increasing engine speed so that the valve is wide open when the engine is rotating at its maximum speed. There is also a valve 518 which is unbalance mounted, or operated by a diaphragm subject to the depression at venturis 542, 544 so that it opens as the air flow increases.

FIG. 2

FIG. 7

FIG. 1

ENGINE EQUIPPED WITH A CONVENTIONAL TWO-STAGE
TWO-BARREL TYPE CARBURETOR

ENGINE EQUIPPED WITH
A SINGLE BARREL TYPE
CARBURETOR

ENGINE OF THE PRESENT
INVENTION

INTAKE VOLUMETRIC EFFICIENCY OF PUMP %

90
80
70

0   1000 2000 3000 4000

ENGINE ROTATIONAL SPEED rpm ⟶

FIG. 5

FIG. 6

FIG. 7

FIG. 2

FIG. 3

FIG. 4

FIG. 8

FIG. 9

POWER PISTON TDC

PUMP BDC

COMPRESSION STROKE OF CRANKCASE

COMPRESSION STROKE OF PUMP OF
THE PRESENT EMBODIMENT

EXHAUST PORTS OPEN Eo

SCAVENGING PORTS OPEN So.

PHASE RANGE
OF PUMP TDC

POWER PISTON
BDC

SCAVENGING PERIOD

15°

15°

15°

SCAVENGING PERIOD

EXHAUST PORTS CLOSE Ec

SCAVENGING PORTS CLOSE Sc

PUMP TDC
(THE PRESENT
EMBODIMENT)

FIG. 10

FIG. 11

FIG. 12

FIG. 13

FIG. 14

TDC POWER PISTON

EXHAUST PORTS
CLOSE Ec

Eo EXHAUST PORTS
OPEN

SCAVENGING PORTS
CLOSE Sc

So SCAVENGING
PORTS OPEN

BDC

SCAVENGING
PERIOD

FIG. 15

PRESSURE OF
CRANKCASE atm

So

BDC

0

Sc

TDC (POWER PISTON)

POSITION OF POWER PISTON

FIG. 16

SPECIFICATION

A two-stroke cycle gasoline engine

5 The present invention relates to a two-stroke
cycle gasoline engine, and, more particularly,
to a two-stroke cycle gasoline engine adapted
for use with automobiles.
    A two-stroke cycle engine has theoretically
10 the advantage that an engine of a certain size
can generate a greater power than a four-
stroke cycle engine of a bigger size because
the two-stroke cycle engine has twice as many
work cycles per revolution as the four stroke
15 engine. In fact, however, a conventional two-
stroke cycle gasoline engine employing a car-
buretor has drawbacks, such as: that it has
high fuel consumption as compared with a
four stroke cycle engine due to the loss of
20 fuel-air mixture caused by the direct escape,
i.e. blow-out, of scavenging mixture to the
exhaust manifold during scavenging; and that
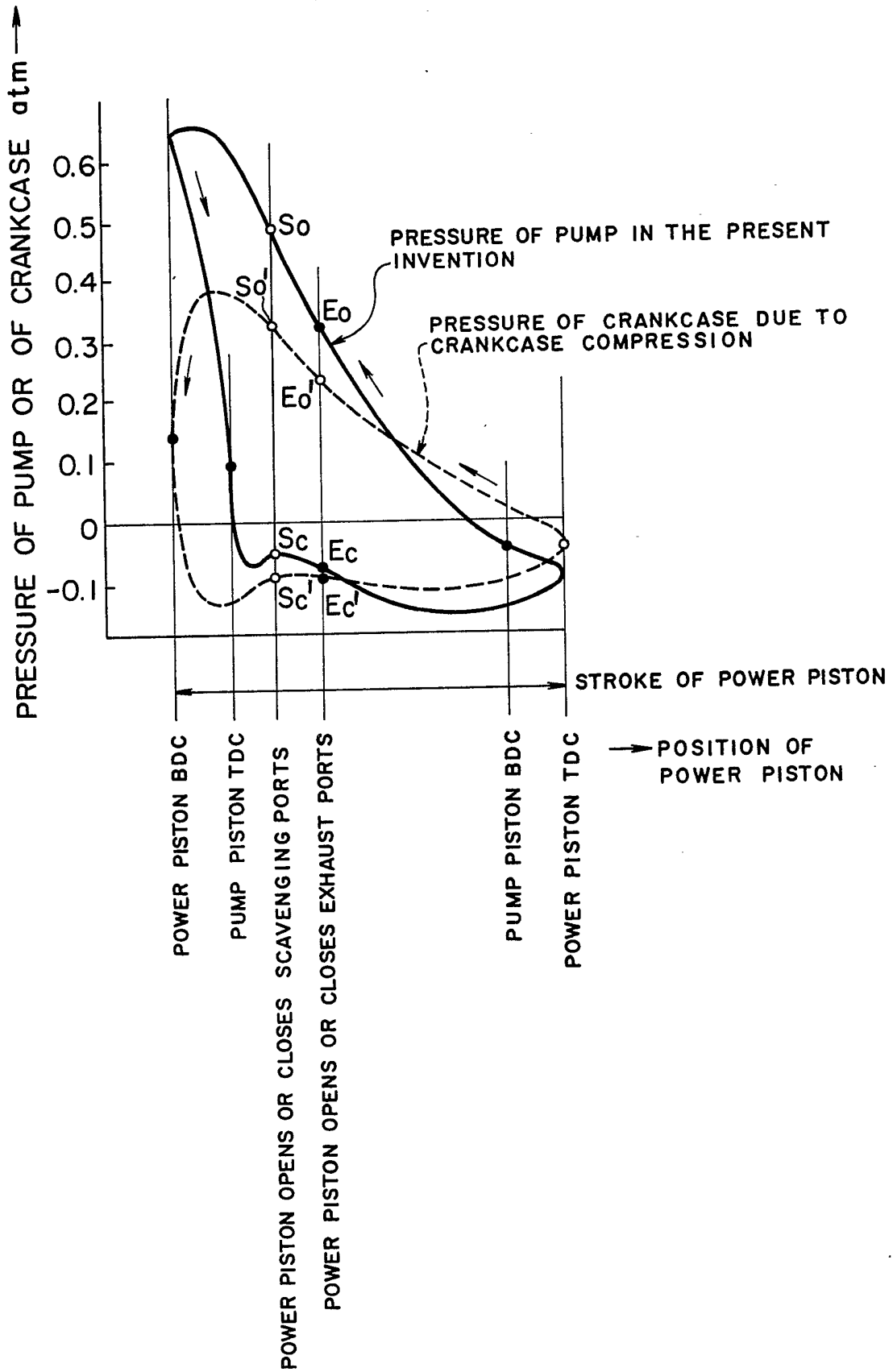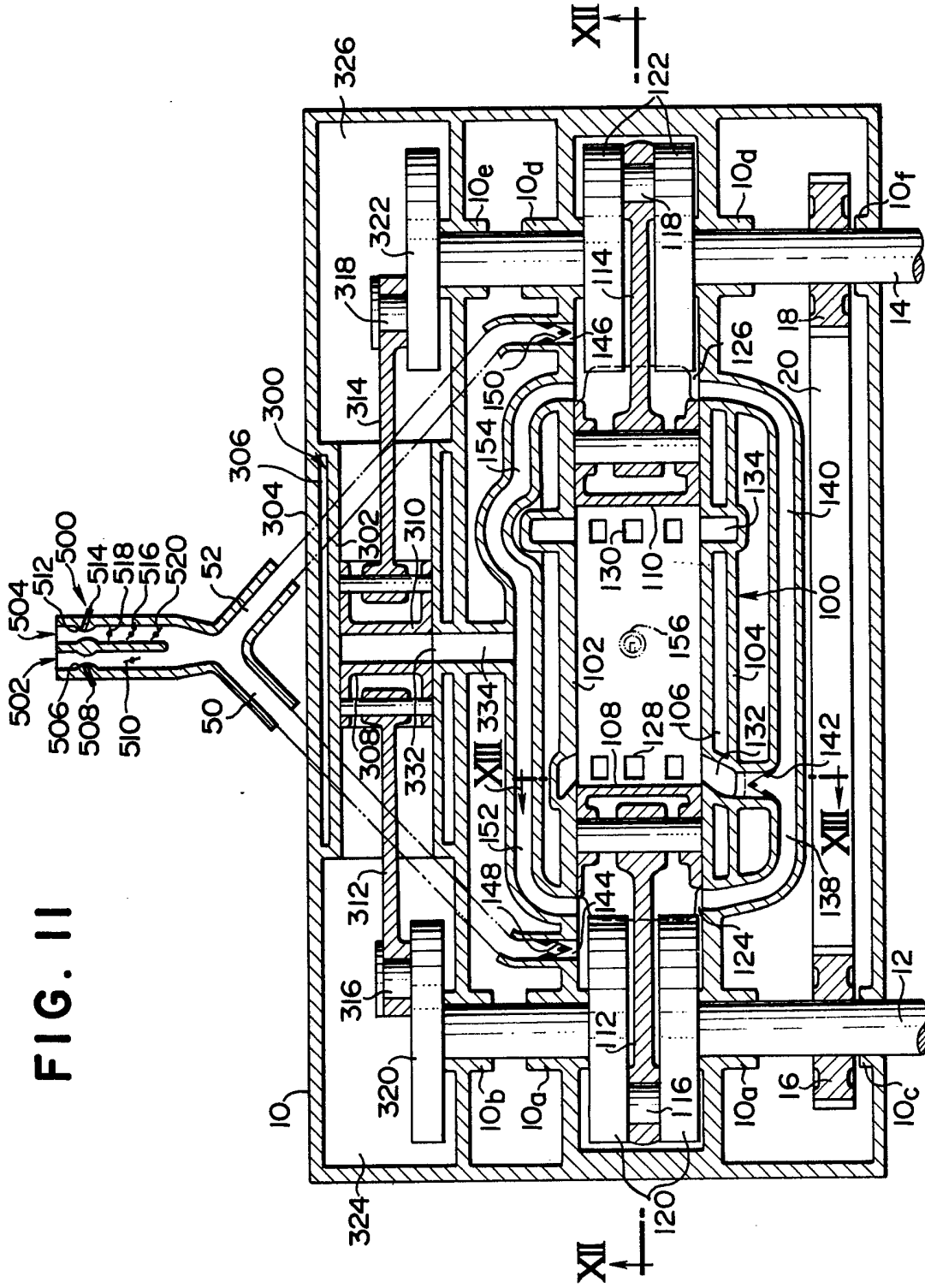it cannot generate such a high power as
expected from the fact that it has twice as
25 many work strokes as the corresponding four-
stroke cycle engine, due to the fact that the
scavenging is still insufficient.
    As methods of scavenging in two-stroke
cycle engines are conventionally known cross
30 scavenging, loop scavenging, and uniflow sca-
venging. In this connection, if the amount of
scavenging mixture is increased so as to im-
prove scavenging efficiently, uniflow scaveng-
ing is considered to be most desirable, in
35 order to obtain the highest scavenging effici-
ency without causing direct escape of the
scavenging mixture to the exhaust manifold.
    In view of this, and in view of the aforemen-
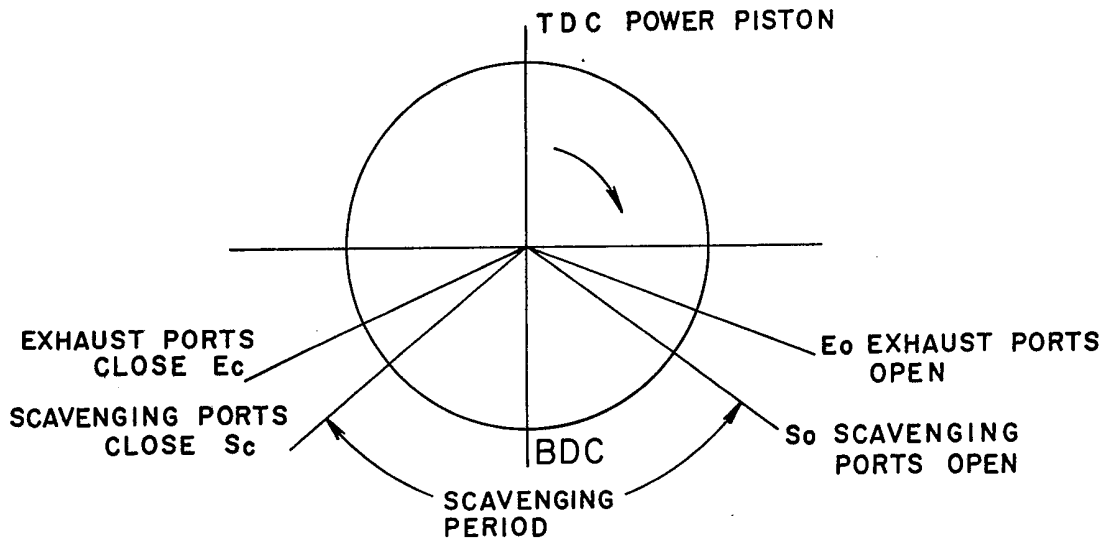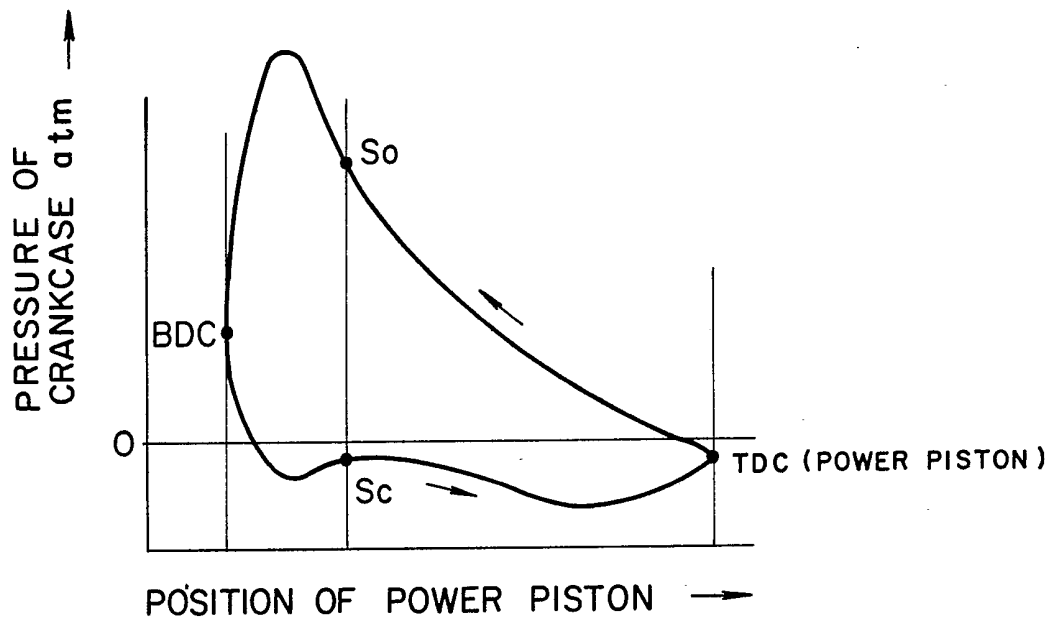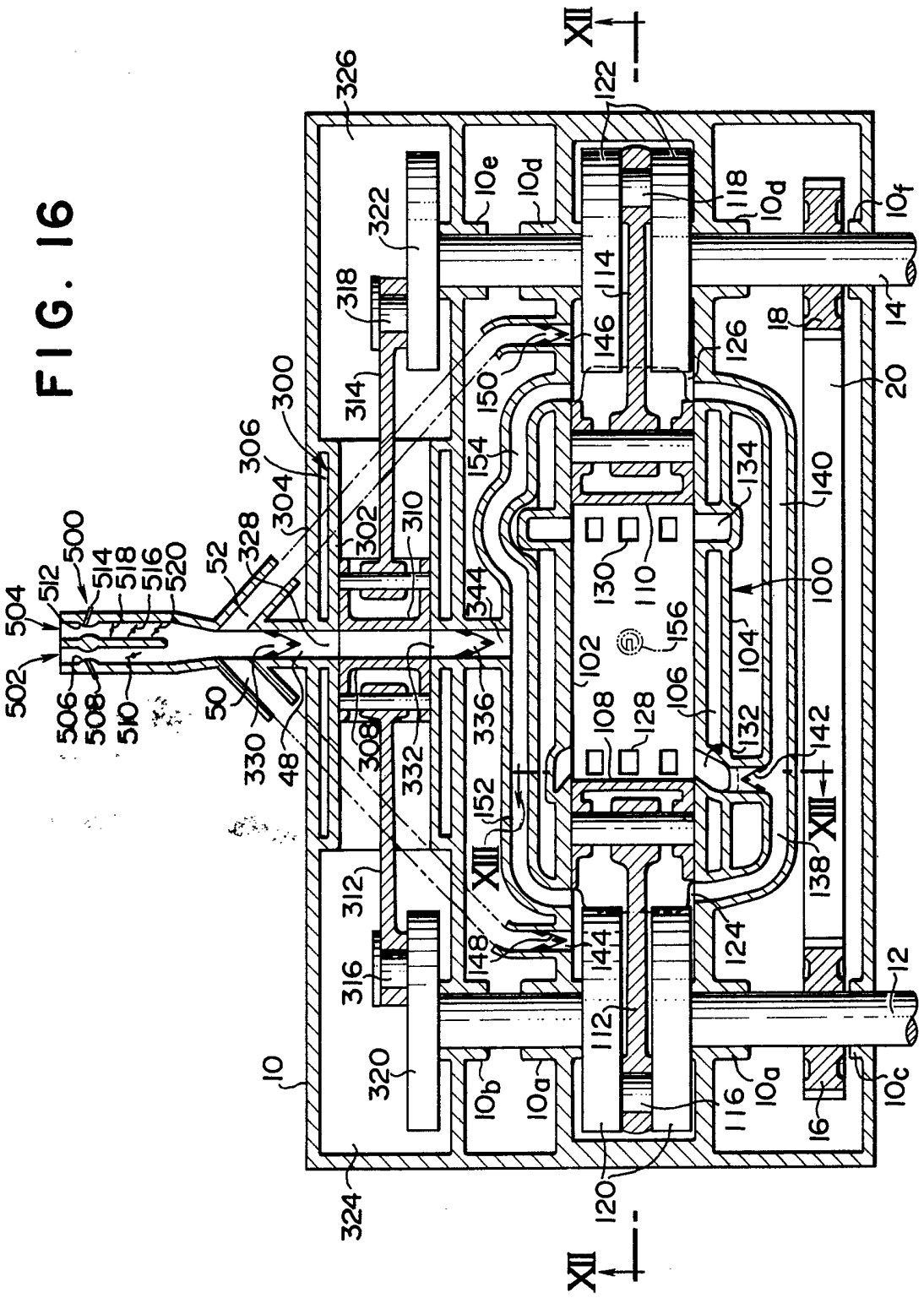tioned drawbacks, the actual application of
40 two-stroke cycle gasoline engines has been
conventionally limited to the field of small-size
engines in which simplicity of structure and
low manufacturing cost are essential condi-
tions. Therefore, conventional two-stroke cycle
45 gasoline engines presently used generally em-
ploy crankcase compression for scavenging.
However, scavenging by crankcase compres-
sion cannot deliver a sufficient amount of
scavenging mixture, thereby causing incom-
50 plete scavanging, which leads to a relatively
low volumetric efficiency.
    In view of the fact that such a low volumet-
ric efficiency is the principal cause of the poor
output power of conventional two-stroke cycle
55 gasoline engines, in a preceding patent appli-
cation no. 7909719 we have proposed a two-
stroke cycle gasoline engine particularly suit-
able for use as an automobile engine, which
comprises at least one two-stroke cycle power
60 cylinder-piston assembly incorporating uniflow
scavenging and two horizontally opposed pis-
tons, and at least one scavenging pump cylin-
der-piston assembly of the reciprocating type,
with or without incorporating crankcase com-
65 pression, wherein the total stroke volume of

the scavenging pump means is 1.35 to 1.85
times as large as that of the power cylinder-
piston assembly, so that the volumetric effici-
ency is substantially increased so as to gener-
70 ate high power output when compared with
conventional two-stroke cycle gasoline en-
gines.
    Furthermore, in view of the fact that, even
when a separate pump cylinder-piston assem-
75 bly is employed as proposed in the abovemen-
tioned former application, if the conventional
crankcase compression is also incorporated,
the operational phase relation between the
power cylinder-piston assembly and the sca-
80 venging pump means is substantially re-
stricted, we have proposed, in our former
application no. 7909720, not to utilize at all
crankcase compression, and to provide a two-
stroke cycle gasoline engine which comprises
85 at least one two-stroke cycle power cylinder-
piston assembly of the reciprocating type and
driven by said power cylinder-piston assembly
in synchronization therewith with a phase dif-
ference therebetween, wherein the total stroke
90 volume of said pump cylinder-piston assembly
is between 1.15 and 1.65 times as large as
that of said power cylinder-piston assembly,
and said phase difference between said power
and said pump cylinder-piston assemblies is
95 so determined that the top dead center of a
pump cylinder-piston assembly is, as viewed
in the crank angle diagram, in a range be-
tween 15° in advance of and 15° behind the
midpoint between the bottom dead center and
100 the scavenging port closing phase point of the
power cylinder-piston assembly to which it
supplies scavenging mixture, thereby substan-
tially improving its scavenging efficiency when
compared with conventional two-stroke cycle
105 gasoline engines, so that the engines can
generate high output power and is suitable for
use as an automobile engine.
    In either of the abovementioned formerly
proposed two-stroke cycle gasoline engines, in
110 order to improve scavenging efficiency, the
amount of scavenging mixture is increased by
employing an additional or separate pump
when compared with the conventional scaven-
ging dependent only upon crankcase com-
115 pression, so that the volumetric efficiency of
the power cylinder becomes as high as
75%–100%. In this connection, in the case
of a power cylinder-piston assembly having
two horizontally opposed pistons and incorpo-
120 rating uniflow scavenging, when the path of
the flow of scavenging mixture from the sca-
venging port to the exhaust port, and its
length, which are determined by the geometry
of the power cylinder and the scavenging and
125 exhaust ports provided at its scavenging and
exhaust sides, respectively, and the manner of
flow of the scavenging mixture in the power
cylinder which is ejected into the power cylin-
der through the scavenging ports and gener-
130 ally flows along a helical path, are given, the

speed of scavenging, i.e. the speed at which the scavenging mixture moves from the scavenging port to the exhaust port while urging the exhaust gases remaining in the power
5 cylinder, is determined by the difference between the pressure of the scavenging mixture and that of the remaining exhaust gases, and by the amount of scavenging mixture which backs up the scavenging mixture introduced
10 into the power cylinder, i.e. by the total amount of scavenging mixture, and has no direct relation to the rotational speed of the engine. On the other hand, however, the time which lapses from the time point when the
15 scavenging port is opened to the time point when the exhaust port is closed becomes shorter as the rotational speed of the engine becomes higher. Therefore, when so-called engine matching is so performed that, when
20 the engine is operating at that load and rotational speed which are most frequently employed, the exhaust port is closed when the scavenging mixture has just pushed the exhaust gases out of the exhaust port (such a
25 rotational speed is called "matching rotational speed"), assuming that the volumetric efficiency of the scavenging pump is constant regardless of the rotational speed of the engine, blowing-through of scavenging mixture
30 will occur when the engine is operating at a rotational speed lower than the matching rotational speed, while exhaust gases will remain in the power cylinder when the engine is operating at a rotational speed higher than the
35 matching rotational speed.
   The volumetric efficiency of a reciprocating piston type scavenging pump means such as crankcase compression and a reciprocating type pump cylinder-piston assembly is gener-
40 ally higher as its rotational speed is lower; that is, when the pump means is directly driven by an engine, as the rotational speed of the engine is lower. Actually, however, an intake system which includes a carburetor
45 connected to a scavenging pump means, an intake passage, and other flow resistant means such as reed valves incorporated in the intake passage, is subject to a pulsating effect; and, since such a pulsating effect
50 changes in accordance with engine rotational speed, the volumetric efficiency of the scavenging pump means does not necessarily increase in accordance with decrease in engine rotational speed. Generally, in fact, the
55 volumetric efficiency of a scavenging pump means is better when engine rotational speed is not very low but is relatively high. Further, when a larger number of pump cylinder-piston assemblies are operated with phase differ-
60 ences therebetween, the maximum volumetric efficiency is obtained at a higher rotational speed of the engine.
   Generally, when an intake system operates with a single barrel type carburetor, the intake
65 volumetric efficiency of a scavenging pump

served by this carburetor changes in accordance with an upwardly convex curve as shown in Fig. 1, wherein the efficiency first increases in accordance with increase of en-
70 gine rotational speed so as to reach the maximum value at a certain engine rotational speed, and then lowers when engine rotational speed further increases, provided that the venturi diameter is constant and the throt-
75 tle opening is constant, as, for example, when the throttle is set at full throttle. In this case, the point of maximum volumetric efficiency shifts in the direction of high rotational speed as the venturi diameter is increased. However,
80 increase of the venturi diameter means reduction of the speed of air flowing through the venturi, and causes poor atomization of gasoline in low load operation of the engine, thereby reducing engine torque, increasing
85 fuel consumption, and reducing accelerating performance. Further, when the scavenging pump includes a plurality of chambers, the poor atomization of gasoline also causes poor distribution of fuel between these pump
90 chambers and the power cylinders separately connected to these pump chambers. Therefore, when an engine is equipped with an intake system having a single barrel type carburetor and a venturi of a certain diameter
95 so that the maximum intake volumetric efficiency of a scavenging pump driven by the engine is obtained at a medium point in the range of engine rotational speed, a difficulty is encountered with regard to matching of the
100 power cylinder for scavenging. That is, if the power cylinder is matched for scavenging at an engine rotational speed which provides the maximum intake volumetric efficiency of the pump, as the rotational speed of the engine
105 increases beyond this matching rotational speed, the scavenging becomes more and more incomplete. However, if the power cylinder is matched for scavenging at a high engine rotational speed, as the rotational
110 speed of the engine lowers, more blowing through of scavenging mixture will occur.
   On the other hand, in order to obtain good atomization of fuel over a wide range of engine rotational speed, it is already known to
115 employ a two stage two barrel type carburetor which is opened or closed in two stages in accordance with engine load in such a manner that, when engine load is relatively low, intake air is drawn only through its first stage barrel,
120 and when engine load increases its second stage barrel is also opened so that air is drawn through both the first and second stage barrels. In these conventional two stage two barrel type carburetors the first and second
125 stage barrels generally incorporate first and second throttle valves, respectively, of which the first throttle valve which controls opening of the first stage barrel is directly operated by an accelerating mechanism including an accel-
130 erator pedal, while the second throttle valve

which controls opening of the second stage barrel is interconnected with the first throttle valve by a link mechanism so that the second throttle valve is opened when the first throttle
5 valve is opened beyond a predetermined opening, or alternatively the second throttle valve is opened by a diaphragm means which is operated by the venturi vacuum in the first stage barrel so that the second throttle valve
10 is opened when the venturi vacuum in the first stage barrel increases beyond a predetermined value due to increase of air flow speed in the first stage barrel over a predetermined value. When scavenging mixture is supplied
15 to a scavenging pump by employing this conventional two stage two barrel type carburetor, it is possible to set the engine rotational speed at which the intake volumetric efficiency of the pump becomes maximum at a
20 point of high engine rotational speed, as shown by a broken line in Fig. 1, without deteriorating atomization of fuel in low speed operation of the engine, so that the matching of the power cylinder for scavenging is ac-
25 complished more desirably. However, the change of pump intake volumetric efficiency relative to engine rotational speed obtained by the conventional two stage two barrel type carburetor is still relatively moderate as seen
30 in Fig. 1, and in this case, if the power cylinder is matched for scavenging at its maximum rotational speed, in low load operation considerable blowing-through of scavenging mixture still occurs.
35 A low intake volumetric efficiency of a scavenging pump means that the ratio of the delivery amount to the stroke volume of the pump is low, and that therefore a pump having a large stroke volume is required to
40 obtain a predetermined delivery amount. However, from the viewpoint of energy economy, the loss of energy which increases in accordance with lowering of of the intake volumetric efficiency of the pump corresponds to the
45 increase of the loss of power in driving the pump, and this is smaller than the loss of energy caused by the fuel being blown out from the power cylinder without being combusted due to poor matching of the power
50 cylinder for scavenging. In the conventional two-stroke cycle gasoline engines which depend upon only crankcase compresion for the compression of scavenging mixture, a reduction in the intake volumetric efficiency of a
55 scavenging pump directly causes a reduction in the output power of the engine, and therefore the intake volumetric efficiency of a scavenging pump seriously affects the performance of the engine. However, the engine in
60 which the present invention is incorporated is of the type which incorporates a reciprocating type pump cylinder-piston assembly driven by a power cylinder-piston assembly in synchronization therewith, regardless of whether or not
65 the crankcase compression of the power cylin-

der is utilized for compressing scavenging mixture, so as to increase the total stroke volume of the scavenging pump to be larger than the total stroke volume of the power
70 cylinder-piston assembly, and in this case, therefore, a reduction in the intake volumetric efficiency of the scavenging pump can be easily compensated for by slightly increasing the total stroke volume of the separate or
75 independent pump cylinder-piston assembly. In this case, by changing the intake volumetric efficiency of the scavenging pump more greatly relative to engine rotational speed than in the case of employing a conventional two
80 stage two barrel type carburetor, it is possible to operate an engine which was matched for scavenging at its maximum rotational speed in its low rotational speed operational region in well matched condition, wherein no blowing
85 out of scavenging mixture occurs.
It is therefore the object of the present invention to improve the engines proposed in the aforementioned prior applications, which are of the type comprising at least one two-
90 stroke cycle power cylinder-piston assembly having two horizontally opposed pistons and incorporating uniflow scavenging, and a reciprocating type scavenging pump means including at least one pump cylinder-piston as-
95 sembly of the reciprocating type driven by said power cylinder-piston assembly in synchronization therewith, so as to be able to maintain the most desirable matching of the engine for scavenging over the entire rota-
100 tional speed of the engine.
In accordance with the present invention, the above-mentioned object is accomplished by incorporating, in a two-stroke cycle gasoline engine of the aforementioned type, a
105 carburetor which includes a throttle valve and a control valve which is controlled to increase its opening as the rotational speed of the engine increases.
By a carburetor being equipped with a
110 control valve adapted to increase its opening as the rotational speed of the engine increases, in addition to a normal throttle valve adapted to be operated by the driver, the flow resistance of the carburetor can be more
115 greatly changed in accordance with change of engine rotational speed, so that the intake volumetric efficiency of a scavenging pump which receives supply of fuel-air mixture from the carburetor can be more greatly changed in
120 accordance with change of engine rotational speed, i.e. the ratio of the reduction of the intake volumetric efficiency of a scavenging pump relative to the lowering of engine rotational speed from its maximum value can be
125 determined to be greater than in the case . where the conventional two stage two barrel type carburetor is employed, and therefore it is possible to operate an engine which is matched for scavenging at its maximum rota-
130 tional speed in its low speed operational re-

gion while maintaining the desirable matching condition for scavenging, thereby avoiding blowing-out of scavenging mixture in low speed operation of the engine, thus maintain-
5 ing high fuel consumption over the entire rotational speed range of the engine.

In this connection, the maximum rotational speed of the engine according to the present invention is about 3800 rpm at the highest.
10 This limitation is imposed in order to utilize the feature that the two-stroke cycle engine has twice as many working strokes as the four cycle engine to its extremity by increasing the volumetric efficiency of the power cylinder,
15 thereby making it possible to operate the engine at lower rotational speed, so as to reduce frictional loss caused by high speed rotation, and to increase effective output power per unit stroke volume of the power
20 cylinder. These matters have already been explained in the specifications of the afore-mentioned prior applications nos. 7909719 and 7909720.

In accordance with a particular feature of
25 the present invention, the abovementioned carburetor may be constructed as a two stage two barrel carburetor having first and second barrels and adapted to be opened and closed in two stages in accordance with engine load,
30 wherein the aforementioned control valve is incorporated in the second stage barrel which is opened in high load operation of the en-gine.

The present invention will become more
35 fully understood from the detailed description given hereinbelow and the accompanying drawings, which are given by way of illustra-tion only, and are thus not limitative of the present invention, and wherein:
40   Figure 1 is a diagram which shows changes of the intake volumetric efficiency of a sca-venging pump relative to changes of the rota-tional speed of the engine which drives the scavenging pump, available from the present
45 invention and from conventional engines equipped with conventional carburetors;

Figure 2 is a diagrammatical plan sectional view showing a first embodiment of the two-stroke cycle gasoline engine of the present
50 invention;

Figures 3 and 4 are sectional views along lines III–III and IV–IV in Fig. 2;

Figure 5 is a diagrammatical longitudinal sectional view showing an embodiment of a
55 carburetor which is incorporated in the two-stroke cycle gasoline engine of the present invention;

Figure 6 is a rear side part-sectional view of a part of the carburetor shown in Fig. 5,
60 showing a link mechanism which connects throttling valves provided in the first and second stage barrels of the carburetor;

Figure 7 is a diagrammatical longitudinal sectional view showing another embodiment
65 of the carburetor to be incorporated in the two-stroke cycle gasoline engine of the pre-sent invention;

Figure 8 is a diagrammatical view showing an embodiment of the operating mechanism
70 for the control valve incorporated in the carbu-retor in the two-stroke cycle gasoline engine of the present invention;

Figure 9 is a crank angle diagram showing the operational phases of the engine shown in
75 Figs. 2–4;

Figure 10 is an indicator diagram showing changes of the pump pressure relative to the position of the power piston in the engine shown in Figs. 2–4, also showing, for the
80 purpose of comparison, the same relation in an engine incorporating conventional crankcase compression;

Figure 11 is a diagrammatical plan sec-tional view showing a second embodiment of
85 the two-stroke cycle gasoline engine of the present invention;

Figures 12 and 13 are sectional views along lines XII–XII and XIII–XIII in Fig. 11;

Figure 14 is a crank angle diagram showing
90 the operational phases of the engine shown in Figs. 11 and 13;

Figure 15 is an indicator diagram showing changes of the crankcase pressure relative to the position of the power piston in the engine
95 shown in Figs. 11–13; and

Figure 16 is a diagrammatical plan sec-tional view showing a third embodiment of the two-stroke cycle gasoline engine of the present invention.
100   Referring to Figs. 2–4, showing the first embodiment of the present invention, the two-stroke gasoline engine herein shown com-prises a cylinder block 10, the overall shape of which is like a relatively flat block rectangu-
105 lar in a plan view, and which is adapted to be installed substantially with the two largest of its six faces arranged horizontally. In the cylin-der block there are provided a pair of crank-shafts 12 and 14 which are arranged along
110 opposite edges of the cylinder block and are rotatably supported by bearings 10a–10c and 10d–10f, respectively. In this embodiment, for example, the crankshaft 12 may be con-nected to auxiliaries of the engine while the
115 crankshaft 14 may serve as the power output shaft of the engine. In the cylinder block 10 there are incorporated a power cylinder-piston assembly 100 and an independent reciprocat-ing type scavenging pump 300, which is in
120 this embodiment a pump cylinder-piston as-sembly having horizontally opposed pistons.

First, the power cylinder-piston assembly 100 will be described. The assembly includes a power cylinder 102 supported by the cylin-
125 der block 10. The power cylinder is sur-rounded by a cooling jacket 106 defined by a jacket wall 104. In the cylinder 102 are arranged two power pistons 108 and 110, one being located on the scavenging side or
130 the left side in the figure, while the other is

located on the exhaust side or the right side of the figure, respectively. The pistons 108 and 110 are individually connected with connecting rods 112 and 114, which in turn are
5 individually connected with crankpins 116 and 118. The crankpins 116 and 118 are individually supported by crank arms 120 and 122. The two crank mechanisms each including a crank arm and a crank pin are
10 individually housed in crankcases 124 and 126. Since in this case no crankcase compression is involved, the crankcase may have any clearance volume.

The cylinder 102 has a plurality of scaveng-
15 ing ports 128 and a plurality of exhaust ports 130. These scavenging ports and exhaust ports are connected with a scavenging plenum 132 and an exhaust plenum 134, respectively. The exhaust plenum 134 is con-
20 nected with exhaust pipes 136. As shown in Fig. 4, the scavenging ports 128 include a pair of scavenging ports 128a which open towards the central axis of the power cylinder 102, and six ports 128b which open along
25 axes tangential to a phantom cylinder C coaxial with the cylinder 102. Further, all these scavenging ports 128a and 128b are inclined toward the exhaust side of the cylinder so that the flows of scavenging mixture discharged
30 from these scavenging ports have a velocity component towards the exhaust ports 130. Thus, the flows of scavenging mixture discharged from the scavenging ports 128b are formed into swirl flows which proceed heli-
35 cally in the cylinder 120 towards the exhaust ports, while on the other hand the flows of scavenging mixture discharged from the scavenging ports 128a collide with each other at the center of the cylinder 102 and then
40 proceed along the central axis of the cylinder towards its exhaust side. At a longitudinal central portion of the cylinder 102 is provided a spark plug 156.

Next, the pump 300 will be described. The
45 pump 300 includes a pump cylinder 302 supported by the cylinder block 10. The pump cylinder 302 is surrounded by a cooling jacket 306 defined by a jacket wall 304. This cooling jacket 306 serves to remove the com-
50 pression heat of mixture generated in the pump 300, so as to increase the volumetric efficiency of the pump, while further when the engine is operated in cold weather it serves to warm the pump cylinder so as to expedite
55 atomization of the gasoline. For this purpose, the cooling jacket 306 is connected with the cooling jacket 106 of the power cylinder by a passage means not shown in the figure. In the pump cylinder 302 are provided a pair of
60 pump pistons 308 and 310 as opposed to each other. The pistons 308 and 310 are individually connected with connecting rods 312 and 314 which in turn are individually connected with crank pins 316 and 318. The
65 crank pins 316 and 318 are individually sup-

ported by crank arms 320 and 322. The crank mechanisms, composed of the connecting rods, crank pins, and crank arms, are individually housed in crankcases 324 and
70 326. These crankcases of the pump assembly are connected with the crankcases 124 and 126 so as to balance the pulsations of crankcase pressure caused by the pistons 308 and 310. Furthermore, these crankcases are
75 connected with the internal space of an air cleaner (not shown in the figure) by a positive crankcase ventilation valve (also not shown in the figure).

The carburetor generally designated by 500
80 is a two stage two barrel type carburetor including a first stage barrel or carburetor 502 and a second stage barrel or carburetor 504. The first stage carburetor 502 includes a venturi portion 506, a first stage main fuel
85 nozzle 508 which opens at the throat of the venturi portion, and a first stage throttle valve 510. Further, the second stage carburetor 504 includes a venturi portion 512, a second stage main fuel nozzle 514 which opens at
90 the throat of the venturi portion 512, a second stage throttle valve 516, an auxiliary valve 518, and a control valve 520 provided particularly in accordance with the present invention. The detailed structures and opera-
95 tion of this two stage two barrel type carburetor will be explained afterwards.

The mixture outlet port of the carburetor 500 is connected with an inlet port 328 of the pump 300 by way of a passage 48. In
100 the passage 48 or in the port 328 is provided a reed valve 330 which allows fluid to flow only toward the pump chamber. The outlet port 332 of the pump 300 is connected with the scavenging plenum 132 of the power
105 assembly 100 by way of the passage 153. In this passage and in proximity to the scavenging plenum 132 is provided a reed valve 142 which prevents blowback of combustion gases from the cylinder 102. If there is no danger of
110 such blowback, this reed valve may be omitted. Furthermore, the feed valve 142 serves to interrupt the reverse flow of mixture from the scavenging plenum to the pump cylinder when high vacuum is generated in the pump
115 300 due to its performing its suction stroke after the pump pistons have passed their top dead center (TDC). However, when the two-stroke cycle gasoline engine of the present invention is constructed so as to supply sca-
120 venging mixture by an independent scavenging pump, the TDC of which is positioned substantially behind the BDC of the power assembly 100, as proposed in the aforementioned application serial no. 79,9720, so that
125 the TDC of the pump is, as viewed in the crank angle diagram, in a range between 15° in advance of and 15° behind the midpoint between the bottom dead center and the scavenging port closing phase point of the
130 power cylinder-piston assembly, the period

between pump piston TDC and the exhaust port closing phase point is short, and therefore there exists positive pressure around the scavenging ports due to inertia effects even when the pump 300 has entered its suction stroke. Therefore the drawing action of the pump 300 does not substantially affect scavenging of the pump cylinder 102. Therefore, unless the aforementioned blowback of combustion gases should occur, there would exist no reverse flow which would need to be interrupted by the reed valve 142. However, when the pump TDC and power piston BDC are relatively close to each other, it is desirable that the reed valve 142 should be provided. This is due to the fact that, although the drawing action of the pump is relatively weak in such a region extending between 15° in advance of and 15° behind the TDC of the pump piston because of the small rate of movement of the piston relative to the rotational angle of the crank arms, when the crank phase proceeds beyond this phase region, the independent pump cylinder-piston assembly generates a stronger drawing action than does conventional crankcase compression.

The crankshafts 12 and 14 are drivingly connected with each other by sprocket wheels 16 and 18 individually mounted on the crankshafts and an endless chain 20 engaged around these sprocket wheels so as to co-rotate in the same direction at the same rotational speed. The phase relation between the crankshafts 12 and 14 is so determined that the crankpins 116 and 118 related with the power pistons 108 and 110 are shifted from one another by a phase difference of 180°. Depending upon this phase relation between the crankshafts 12 and 14, the crankpins 316 and 318 related to the pump pistons 308 and 310 are also shifted from each other by the same phase difference of 180°.

In Fig. 5, the carburetor 500 diagrammatically shown in Fig. 2 is shown in more detail in the form of a cross section of a more concrete embodiment. In this case the venturi portion 506 of the first stage carburetor 502 includes a large venturi 506a and a small venturi 506b. The first stage main fuel nozzle 508 opens at the throat of the small venturi 506b. In the first stage carburetor 502 and in proximity to the inlet portion of the venturi is provided a choke valve 522. The venturi portion 512 of the second stage carburetor 504 is constructed as a triple venturi which includes a large venturi 512a, a first stage small venturi 512b, and a second stage small venturi 512c.

As shown in Fig. 6, which is a partial sectional view of a part of the carburetor shown in Fig. 5 as seen from the rear side of Fig. 5, the second stage throttle valve 516 is operationally connected with the first stage

throttle valve 510 by a link mechanism which includes link elements 524, 526, 528, etc. The link element 524 is fixed to the throttle shaft 510a of the first stage throttle valve 510 so as to rotate together with the throttle valve 510. The link element 524 has an arcuate groove 524a, in which is slidably engaged an end 526a of the link element 526. The other end 526b of the link element 526 is pivotally connected with the free end portion of the link element 528 which is fixed to the throttle shaft 516a of the second stage throttle valve 516. This link mechanism itself is well known in the art, and operates so as to open the second stage throttle valve 516 when the first stage throttle valve 510 is opened beyond a predetermined opening (normally about 45°–55°).

The auxiliary valve 518 itself is also well known in the art, and, in the embodiment shown in Fig. 5, it is flexibly maintained at its closed position by a weight 530. As the rotational speed of the engine increases so as thereby to increase intake vacuum, pressure difference which acts across the valve increases, and, when the pressure difference exceeds a predetermined value, the valve is opened against the closing force applied by the weight 530. For this purpose, as apparent from Figs. 5 and 6, the shaft 518a of the auxiliary valve 518 is shifted from the center of the valve to one side thereof, so that a pressure difference acting across the valve produces a moment which drives the valve 518 anticlockwise as seen in Fig. 5. When this moment overcomes the moment produced by the weight 530 and acting in the reverse direction, the auxiliary valve 518 is rotated towards the valve opening direction, i.e. anticlockwise as seen in Fig. 5. In the engine of the present invention, the auxiliary valve 518 is set to open when the rotational speed of the engine exceeds about 1500 rpm.

Fig. 7 is a view similar to Fig. 5, showing a modification of the two stage two barrel type carburetor 500. In this modification, the second stage throttle valve 516 is operated by vacuum. In more detail, the throttle valve 516 is operated by a diaphragm means 532 which has a diaphragm chamber 534, a diaphragm 536 which defines the diaphragm chamber on one side thereof, and a rod 538 connected with the diaphragm, wherein the rod 538 is connected with a free end portion of an arm 540 fixed to the shaft of the throttle valve 516. The diaphragm chamber 534 is connected with a vacuum port 542 which opens at the throat of the large venturi 506a in the first stage carburetor as well as with a vacuum port 544 which opens at the throat of the large venturi 512a of the second stage carburetor by a passage 546, so that the diaphragm chamber 534 is supplied with a vacuum which is of the mean level of the vacuums at the vacuum ports 542 and 544. Such

an operating mechanism depending upon vacuum for the second barrel throttle valve itself is well known in the art.

In this case, as the first stage throttle valve
5 510 gradually opens from its closed position, the venturi vacuum existing in the port 542 gradually increases. However, when the second stage throttle valve 516 is closed, no substantial vacuum appears in the port 544.
10 Therefore, the vacuum taken out from the first stage port 542 is attenuated by the air flowing in through the second stage port 544, so as to provide an attenuated vacuum to the diaphragm chamber 534 through the passage
15 546. When the first stage throttle valve 510 is more opened, so that the attenuated vacuum increases beyond a predetermined vacuum level, the diaphragm 536 is shifted rightwards in the figure so as to open the
20 second stage throttle valve 516. When the second stage throttle valve 516 is opened, venturi vacuum appears in the second stage port 544, whereby the vacuum supplied to the diaphragm chamber 534 through the pas-
25 sage 546 rapidly increases, so that the second stage throttle valve 516 is rapidly opened thereafter. In this carburetor, in which the second stage throttle valve 516 is opened by venturi vacuum so as to be opened in accor-
30 dance with air flow through the carburetor, the auxiliary valve 518 in the embodiment shown in Figs. 5 and 6 is not required. The further portions in Fig. 7 corresponding to those shown in Fig. 5 are designated by the
35 same reference numerals as in Fig. 5.

The control valve 520 shown as incorporated in the carburetor in Figs. 2 and 5–7 is a valve provided in accordance with the present invention so as to be more opened as the
40 rotational speed of the engine increases. Fig. 8 is a diagrammatical view showing an embodiment of the control means for operating the control valve 520. In Fig. 8. 10 designates an engine such as shown in Figs. 2–4,
45 the crankshaft 12 of which, for example, drives an AC generator 602 forming a part of the controller generally designated by 600 for operating the control valve 520. The alternating current generated by the AC generator is
50 converted by a rectifying circuit 604 to a direct current having voltage V which changes in accordance with the rotational speed of the engine. 606 designates a DC motor adapted to be driven by the DC current generated by
55 the rectifying circuit 604. In this case, however, the armature 608 is not adapted to be continuously rotated in one direction but is adapted to generate varying torque in accordance with the voltage V of the direct current
60 supplied to it. In the DC motor 606: 610 is a commutating pole; 612 is a magnetic field current adjusting resistance; 614 is a magnetic field coil; and 616 is a constant voltage external source for energizing the magnetic
65 field coil. The DC motor 606 including the

armature 608, the commutating pole 610, the magnetic field current adjusting resistance 612, and the magnetic field coil 614 itself is well known in the art.
70 The armature 608 of the DC motor 606 is connected with a pulley 620 by way of a shaft 618, which is shown in Fig. 8 in a diagrammatical perspective form, and in fact is coaxial with the armature 608 and the
75 pulley 620. A wire 622 which is connected at its one end to an expansion coil spring 624 which is supported at its one end by a fixed housing 626 is connected at its other end 622a to a portion of the periphery of the
80 pulley 620, whereby the pulley is exerted with a torque which drives the pulley clockwise, against which the pulley 620 is driven anticlockwise in the figure by the armature 608. The control valve 520 is adapted to
85 be driven for opening and closing by an arm 628 which is firmly connected to its valve shaft 520a. The arm 628 has a slot 630 in its free end, in which is slidably engaged a pin 632 which is supported by the wire 622. The
90 revolution of the arm 628 in the clockwise and anticlockwise directions as seen in the figure is limited by stops 634 and 636.

The armature current I in the DC motor 606 is given by the following formula:
95

$$I = (V - E)/(ra + rc + R) \qquad (1)$$

wherein:
V is the input voltage,
100 E is the back electromotive force,
ra is the resistance of the armature 608,
rc is the resistance of the commutating pole 610, and
R is the resistance of the adjusting resistance
105 612.

Since the armature 608 does not move when the torque generated by the motor 606 is balanced by the force of the tension coil spring 624, at this time E equals zero. Torque
110 T of the motor 606 is given by the following formula:

$$T = k \times M \times I \qquad (2)$$

115 wherein:
k is a proportional constant, and
M is the magnetic flux.

Therefore, the torque of the motor 606 when the armature 608 is not moving is
120 given by the following formula:

$$T = k \times M \times V/(ra + rc + R) \qquad (3)$$

Thus, since M is constant, torque T of the
125 motor is proportional to voltage V, and therefore is proportional to the rotational speed of the engine. When the arm 628 is in abutting contact with the stop 636, the control valve 520 is fully closed, while the expansion coil
130 spring 624 is so mounted that in this condi-

tion it is just not extended. When the arm
628 is turned clockwise in the figure from this
position, the distance of movement of the pin
632, i.e. the distance for which the expansion
5 coil spring 624 is expanded, is proportional to
the torque T of the motor 606 at that mo-
ment, by Hooke's law. When the arm 628
abuts against the stop 634, the control valve
520 is fully opened. The rotational angle of
10 the control valve 520 between its fully closed
position where the arm 628 is in contact with
the stop 636 and its fully opened position
where the arm 628 is in contact with the stop
634 is about 80°. When the relative arrange-
15 ment between the control valve 520 and the
expansion coil spring 624 is so determined
that the bisector of the full rotational angle of
the arm 628 crosses the expansion coil spring
approximately at a right angle, the shifting of
20 the pin 632 in the aforementioned manner is
approximately proportional to the opening of
the control valve 520. By this arrangement,
therefore, the opening of the control valve
520 becomes approximately proportional to
25 the rotational speed of the engine.
    Therefore, when the arrangement is so de-
termined that, for example, the pin 632 is
shifted as much as the maximum distance d
at an engine rotational speed of 3500 rpm, so
30 as fully to open the control valve 520, the
opening of the control valve at engine rota-
tional speed lower than 3500 rpm becomes
approximately proportional to engine rota-
tional speed. By the opening of the control
35 valve 520 incorporated in the second stage
carburetor 504 being set so as to be approxi-
mately proportional to engine rotational
speed, the intake volumetric efficiency of a
scavenging pump such as the pump cylinder-
40 piston assembly 300 in the embodiment
shown in Figs. 2–4, if it is so adjusted that its
intake volumetric efficiency is the maximum at
an engine rotational speed of 3500 rpm,
lowers more rapidly than in the case of em-
45 ploying a conventional two stage two barrel
type carburetor having no control valve 520,
as engine rotational speed lowers from 3500
rpm, as shown in Fig. 1, whereby the delivery
amount of scavenging mixture per one stroke
50 of the power piston assembly is much reduced
as engine rotational speed lowers, so that
matching condition of the engine for scaveng-
ing in low to medium speed operation is
maintained in the most desirable condition, as
55 in high speed operation of the engine.
    Fig. 9 is a crank angle diagram showing the
operational phases of the engine shown in
Figs. 2–4. The operation of this engine is,
with regard to its operational phases, similar
60 to that of the engine proposed in the afore-
mentioned prior application no. 7919720.
That is, in this engine, the phase relation
between the power cylinder-piston assembly
and the pump cylinder-piston assembly is so
65 determined, as viewed in the crank angle

diagram, that the TDC of the pump cylinder-
piston assembly is in a range between 15° in
advance of and 15° behind the midpoint
between the BDC and the scavenging port
70 closing phase point of the power cylinder-
piston assembly which receives supply of fuel-
air mixture from the pump cylinder-piston
assembly. Particularly, in the embodiment
whose performance is shown in Fig. 9, the
75 TDC of the pump cylinder-piston assembly is
set exactly at said midpoint. In this connec-
tion, in the engine shown in Figs. 2–4, as in
the engine proposed in the aforementioned
prior patent application no. 7919720, the
80 total stroke volume of the pump cylinder-
piston assembly 300 is determined to be
1.15–1.65 times as large as that of the
power cylinder-piston assembly 100.
    Fig. 10 is an indicator diagram showing
85 changes of the pump pressure relative to the
position of the power piston in the engine
shown in Figs. 2–4. In this figure, for the
purpose of comparison, the crankcase pres-
sure due to conventional crankcase compres-
90 sion is also shown.
    The operation of the engine shown in Figs.
2–4 will now be explained, with reference to
Figs. 9 and 10. When the pump pistons 308
and 310 move from their TDC towards their
95 BDC, the reed valve 330 opens and fuel-air
mixture is drawn into the pump cylinder 302
from the carburetor 500. When the power
pistons 108 and 110 have moved 30° from
their TDC, the pump pistons traverses their
100 BDC and enter into their compression stroke,
and the reed valve 330 is closed. When the
compression stroke of the pump has pro-
ceeded so far that its delivery pressure over-
comes the spring force of the reed valve 142,
105 the reed valve is opened, and scavenging
mixture is charged into the scavenging
plenum 132. Then, first the exhaust ports
130 are opened, so as to discharge exhaust
gases through the exhaust plenum 134 and
110 the exhaust pipes 136. When the pressure of
the exhaust gases remaining in the power
cylinder has substantially lowered, the sca-
venging ports 128 are uncovered by the pis-
ton 108, whereupon scavenging mixture is
115 ejected into the power cylinder. At this time
the scavenging pressure is at the level of So
in Fig. 10, while the scavenging pressure is
changing in accordance with change of the
pump pressure as shown in Fig. 10.
120   However, the actual scavenging pressure is
affected by various factors. For example, if the
reed valve 142 is almost closed after the
scavenging plenum 132 has been filled with
scavenging mixture, after the initiation of dis-
125 charge of scavenging mixture from the sca-
venging ports 128, the scavenging pressure
slightly lowers for a moment because it takes
a time for the reed valve 142 to be again
opened. However, when the engine is operat-
130 ing at such a high rotational speed that the

scavenging ports 128 are opened immediately
after the scavenging plenum 132 has been
filled with scavenging mixture, the reed valve
142 is maintained in its open condition, and
5 therefore no temporary lowering of the sca-
venging pressure occurs. Although the pump
pistons reach their TDC at a phase point
which is 30° in advance of the end of the
scavenging period, since the pump pistons do
10 not move much in a phase region such as 15°
by crank angle behind their TDC, thereby
generating only a small pressure, if the stroke
volume of the pump is large so as to deliver a
large amount of scavenging mixture, the flow
15 of scavenging mixture into the power cylinder
102 is maintained over a substantial phase
region after the pump TDC by the inertia
effect of mass flow, thereby increasing volu-
metric efficiency of the power cylinder.
20   The speed of the mixture flow in the pas-
sage extending from the pump outlet port
332 to the scavenging ports 128 and that of
the mixture flow through the scavenging ports
128 are determined by the delivery pressure
25 and the delivery amount of the pump, and are
not directly related with the rotational speed
of the engine. On the other hand, the time
lapse between pump TDC and the scavenging
port closing phase point Sc varies in accor-
30 dance with engine rotational speed, so that it
becomes shorter as engine rotational speed
increases. In this connection, since in the
present invention the intake volumetric effici-
ency of the pump is changed as shown in Fig.
35 1 in accordance with rotational speed of the
engine, so that the speed of the mixture flow
from the pump delivery port 332 to the
scavenging ports 128 and the speed of sca-
venging mixture discharged through the sca-
40 venging ports 128 are changed in accordance
with engine rotational speed, if the adjustment
of these is properly done, the flowing in of
scavenging mixture into the power cylinder
102 is maintained over the entire scavenging
45 period over a wide range of engine rotational
speed without causing blow-out of scavenging
mixture to the exhaust plenum.
   Here it is to be noted that, although in the
embodiments shown in Figs. 2 and 5–7 the
50 control valve 520 is incorporated in the sec-
ond stage barrel, which is shut down in low
load operation of the engine, in this low load
region the intake volumetric efficiency of the
pump automatically decreases without the
55 control valve, as the rotational speed of the
engine lowers.
   When the pistons 108 and 110 further
move, the scavenging ports 128 are closed by
the piston 108, and then the exhaust ports
60 130 are closed by the piston 110 (Ec), where-
upon the engine enters into compression
stroke, so that mixture is compressed, and the
compressed mixture is ignited by the spark
plug 156 at a phase point which is somewhat
65 in advance of power piston TDC, wherefrom

the power cylinder enters into combustion
stroke.
   Fig. 11 is a diagrammatical plan sectional
view showing a second embodiment of the
70 two-stroke cycle gasoline engine of the pre-
sent invention, and Figs. 12 and 13 are
sectional views along lines XII–XII and XIII–
XIII in Fig. 11. In these figures, the portions
corresponding to those shown in Figs. 2–4
75 are designated by the same reference
numerals as in Figs. 2–4. This second embod-
iment employs crankcase compression in addi-
tion to a supplementary scavenging pump for
compressing scavenging mixture. In this em-
80 bodiment the crank arms 120 and 122 of the
power assembly 100 are disk-shaped, and
crank assemblies, each composed of a pair of
these crank arms and a crank pin connecting
the pair of crank arms, are received in the
85 crankcases 124 and 126, respectively, so that
a substantial part of the space in each
crankcase is occupied by the crank assembly,
regardless of the rotational angle of the crank
arms, so as to minimize the clearance volume
90 of the crankcase and to increase the effective-
ness of the crankcase compression. In this
embodiment the outlet of the carburetor 500
is connected with the crankcases 124 and
126 of the power assembly 100 by way of
95 passages 50 and 52, so that the mixture
generated by the carburetor 500 is directly
supplied to the crankcases 124 and 126. In
ports 144 and 146 at which the passages 50
and 52 open to the crankcases 124 and 126,
100 respectively, are provided reed valves 148
and 150, respectively, so as to prevent re-
verse flow of scavenging mixture. The
crankcase 124 and 126 are connected with
an outlet port 332 of the supplementary sca-
105 venging pump 300 by way of passages 152
and 154 respectively and a common passage
334, on the one hand, while on the other
hand the crankcases 124 and 126 are con-
nected with the scavenging plenum 132 by
110 way of passages 138 and 140 respectively.
   Also in this embodiment the carburetor 500
is constructed as a two stage two barrel type
carburetor having a first stage barrel or carbu-
retor 502, and a second stage barrel or carbu-
115 retor 504 which further includes a control
valve 520 adapted to be controlled in accor-
dance with the rotational speed of the engine,
as in the embodiment shown in Figs. 2–4.
Further, the total stroke volume of the sca-
120 venging pump means including the
crankcases 124 and 126 and the pump 300
is determined to be 1.35–1.85 times as large
as that of the power cylinder-piston assembly,
as in the engine proposed by the aforemen-
125 tioned prior patent application no. 7919719.
Therefore, the total stroke volume of the
pump 300 in this embodiment is to be
0.35–0.85 times as large as that of the
power cylinder-piston assembly 100.
130   In this second embodiment, which employs

crankcases and a supplementary scavenging
pump for the compression of scavenging mix-
ture, in the suction stroke the crankcases 124
and 126 take in mixture through the passages
5 50 and 52 and a carburetor 500, while the
pump 300 which operates in roughly the
same phase as the crankcases 124 and 126
takes in mixture from the crankcases 124 and
126 through the passages 152, 154, and
10 334, and therefore, as a whole, further
through the passages 50 and 52 and the
carburetor 500. When the crankcases 124
and 126 and the pump 300 begin to perform
their compression strokes, the mixture in the
15 crankcases 124 and 126 is compressed in
these crankcases while remaining therein,
while on the other hand the mixture drawn
into the pump cylinder 302 is again pushed
back to the crankcases 124 and 126 through
20 the passages 334, 152, and 154 as it is
compressed between the pump pistons 308
and 310. The mixture thus compressed in
either of the two crankcases or in both the
pump cylinder and either of the two
25 crankcases is supplied towards the scavenging
plenum 132 through the passage 138 or
140.
   Fig. 14 is a crank angle diagram showing
the opening and closing phases of the sca-
30 venging and exhaust ports in the engine
shown in Figs. 11–13. Fig. 15 is an indicator
diagram showing changes of the crankcase
pressure relative to the position of the power
piston in the engine shown in Figs. 11–13, in
35 full throttle operation of the engine. Since the
crankcase pressure abruptly lowers when the
power pistons traverse their BDC, as shown in
Fig. 15, in this case it is also desirable that
the operational phase angle of the pump
40 pistons should be further delayed relative to
that of the power pistons by an angle not
larger than 15°, so that the operational phase
of the pump pistons should be delayed rela-
tive to the power pistons by an angle of
45 180°–195°, as proposed in the aforemen-
tioned former patent application no.
7919719, whereby scavenging in the last
scavenging period after power piston BDC is
further improved.
50   The valve 520 in this embodiment func-
tions in the same way as in the first embodi-
ment, and the same advantages accrue there-
from.
   Fig. 16 is a diagrammatical plan sectional
55 view showing a third embodiment of the two-
stroke cycle gasoline engine of the present
invention, which resembles the second em-
bodiment shown in Figs. 11–13. In Fig. 16,
the portions corresponding to those shown in
60 Figs. 2–4 and 11–13 are designated by the
same reference numerals as in these figures.
The sections along lines XII–XII and XIII–XIII
are the same as Figs. 12 and 13. In this third
embodiment, the mixture outlet port of the
65 carburetor 500 is connected with the inlet

port 328 of the pump 300 by way of the
passage 48 as well as with the ports 144 and
146 opening towards the crankcases 125 and
126 of the power cylinder-piston assembly
70 100 by way of passages 50 and 52, respec-
tively. In the passage 48 and in proximity to
the port 328 is provided the reed valve 330
which allows mixture to flow only towards the
port 328. In this embodiment, a part of the
75 fuel-air mixture generated in the carburetor
500 is directly supplied to the crankcases
124 and 126 of the power cylinder-piston
assembly 100 so as to be compressed in the
crankcases, while the other part of the fuel-air
80 mixture is supplied to the pump 300, and
after having been compressed in the pump is
supplied to the crankcases 124 and 126 by
way of the passages 344, 152 and 154, so
that the fuel-air mixture generated by the
85 carburetor is as a whole compressed by the
crankcases 124 and 126, thereby increasing
the pressure and the amount of scavenging
mixture supplied to the power cylinder 102.
Also in this case, as in the second embodi-
90 ment, it is desirable that the total stroke
volume of the scavenging pump means com-
posed of the crankcases 124 and 126 and
the pump 300 should be 1.35–1.85 times as
large as that of the power cylinder 100; that
95 is, the total stroke volume of the pump 300
should be 0.35–0.85 times as large as that of
the power cylinder 100. Comparing this third
embodiment with the second embodiment
shown in Figs. 11–13, the third embodiment
100 has an advantage that it draws in fuel-air
mixture through a relatively short passage
such as 48 in its intake stroke, while the
second embodiment must draw fuel-air mix-
ture through a relatively long passage includ-
105 ing the passages 50 and 52, the crankcases
124 and 136, and the passages 152, 154,
and 334, which provides a relatively large
flow resistance. On the other hand, however,
in the delivery stroke of the pump 300 the
110 third embodiment must supply fuel-air mixture
through the reed valve 336 which causes a
pressure loss of the scavenging mixture, while
the second embodiment which has no such
reed valve is free from such pressure loss.
115 Further, a reed valve, in its actual structure,
occupies a relatively large space, and there-
fore, if the reed valve 336 is provided in the
passage 334, the passage cannot be so short
as diagrammatically shown in Fig. 16. There-
120 fore, the clearance volume involved in the
passage 334 will be considerably different
with respect to the two embodiments shown
in Figs. 11 and 16—i.e. of course the clear-
ance volume will be substantially smaller in
125 the engine shown in Fig. 11. Further, with
respect to the pipe arrangement at the outlet
of the carburetor the second embodiment has
an advantage over the third embodiment in
that it is more simple. However, regardless of
130 these advantages and disadvantages, the sec-

ond embodiment shown in Figs. 11–13 and
the third embodiment shown in Fig. 16 have
substantially the same performance.

The valve 520 in this embodiment, again,
5 functions in the same way as in the first
embodiment, and the same advantages accrue
therefrom.

Although the invention has been shown and
described with respect to some preferred em-
10 bodiments thereof, it should be understood
that various changes and omissions of the
form and detail thereof may be made therein
by one skilled in the art without departing
from the scope of the invention.
15

CLAIMS
1. A two-stroke cycle gasoline engine
comprising at least one two-stroke cycle
power cylinder-piston assembly having two
20 horizontally opposed pistons and incorporating
uniflow scavenging, a reciprocating type sca-
venging pump means including at least one
pump cylinder-piston assembly of the recipro-
cating type driven by said power cylinder-
25 piston assembly in synchronization therewith,
and a carburetor incorporating a throttle valve
and a control valve which is controlled to
increase its opening as the rotational speed of
the engine increases.
30   2. An engine as claimed in claim 1,
wherein said carburetor is a two stage two
barrel type carburetor which includes first and
second stage barrels containing respectively
first and second throttle valves which are
35 opened and closed in two stages, said control
valve being incorporated in said second stage
barrel in series with said second throttle valve.
3. An engine as claimed in claim 1 or 2,
comprising an electrical drive means for oper-
40 ating said control valve, which includes an AC
generator driven by the output shaft of the
engine, a rectifying circuit which rectifies the
alternating current generated by said genera-
tor, and a torque generating means which is
45 energized by the direct current obtained by
said rectifying circuit from said alternating
current.
4. An engine as claimed in any of claims
1 to 3 wherein the total stroke volume of said
50 pump cylinder-piston assembly is between
1.15 and 1.65 times as large as that of said
power cylinder-piston assembly, and the
phase difference between said power and
pump cylinder-piston assemblies is so deter-
55 mined that the top dead center of a pump
cylinder-piston assembly is, as viewed in the
crank angle diagram, in a range between 15°
in advance of and 15° behind the midpoint
between the bottom dead center and the
60 scavenging port closing phase point of the
power cylinder-piston assembly to which it
supplies scavenging mixture.
5. An engine as claimed in any of claims
1 to 3 wherein the total stroke volume of said
65 scavenging pump means is between 1.35 and

1.85 times as large as that of said power
cylinder-piston assembly, and the operational
phase of a power cylinder-piston assembly is
so shifted relative to that of the power cylin-
70 der-piston assembly to which it supplies sca-
venging mixture that, when the power cylin-
der-piston assembly is at its bottom dead
center, the pump cylinder-piston assembly is
at or slightly before its top dead center.
75   6. A two-stroke cycle gasoline engine sub-
stantially as hereinbefore described with refer-
ence to, and as illustrated in, Figs. 2 to 4,
and 7 or Figs. 2 to 6, and 8, or Figs. 2 to 4,
7 and 8, or Figs. 11 to 13, or Fig. 16 of the
80 accompanying drawings.