ABSTRACT: An internal combustion engine which is dynamically balanced and which achieves a highly efficient derivation of power from the combustible fuel-air mixture. The engine has all of the components which react with the pistons during reciprocation of the latter arranged so as to form groups of components which move in synchronism while being opposed to each other so as to achieve a dynamic balance of the moving structure of the engine. In addition, the engine has various passages, spaces, and the like through which the fuel-air mixture flows during combustion, compression, and exhausting thereof, and through various valves as well as through the opposed ends of the pistons themselves the fluid is acted upon so as to achieve such features as scavenging, supercharging, and the like.
Fig. 14

Fig. 15

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The object of the invention is an internal combustion engine, having in its design a new mechanism which converts the movement of translation in a movement of rotation, by means of which one can achieve a permanent dynamic balance of normal and inertia forces, provided with a new scavenging system of the flue gases which avoids losses of fresh gases together with the flue gases in the scavenging process of the two-stroke carburator engine, having the possibility of supercharging when used with the four-stroke engine, the adjusting of supercharging degree, that is the load with maximum combustion pressure, the achieving a method of controlled combustion in the case of carburator engine as well as the injection engine and a new sealing method of the pistons in high pressure conditions.

In the field of two-and four-stroke internal combustion engine, there are known two ways of taking over the normal force which occurs due to connecting rod inclination and oscillation with regard to cylinder axis. The piston itself delivers the normal force to the cylinder first through the inner part of its liner and then by means of the crosshead with the engines having shank and crosshead sliding between two sliding bars.

Both types of engines have the disadvantage of taking over the normal force by friction of linear and alternative sliding which due to low specific allowable pressures between the contact surfaces is limiting the maximum combustion pressure and consequently the energy capacity of the unit.

The single cylinder as a functional unit is not possible to be balanced.

The scavenging of combustion gases in two-stroke engines with carburator is obtained with mixed gas (petrol-air mixture), supplied by the scavenging pump, evacuating during this process at the same time with the combustion gases a part of the mixed gas, which represents a loss of fuel proportional with the engine capacity.

The four-stroke engine supercharging needs special set apart from the engine.

The maximum combustion pressure varies with the supercharging degree (variation of inlet and compression pressure at constant compression ratio), a fact which lowers the economy of the engine at partial loads; the compression ratio of supercharged engines is chosen to be lower as the supercharging degree is higher, a thing which determines that at low loads (low inlet pressures) the compression and the combustion pressure may diminish substantially in relation with pressures corresponding to maximum loads, consequently lowers also the economy.

The compression ratio is limited by the petrol detonation. The air-petrol mixture is limited by the ignition possibility of the sparking plug. Thus the petrol octane index requires an absolute limit to the maximum value of the compression ratio, and the petrol-air mixtures cannot be ignited when they are either too high or too low.

In the case of the diesel cycle it was found that the short time which assures the fuel on efficient combustion in the area around the upper dead point, is greatly influenced by the delay period of the firing and the kinematics of the combustion process which at the increase of rotation lengths the expansion, thus lowering the performance of the cycle.

The normal sealing method with free ring in the piston groove, permits infiltration the combustion gases in the rear of the compression ring and the increase of the friction pressure between the ring and cylinder; the axial clearance of the rings causes the oil to be pumped towards the combustion chamber promoting the coaking of the rings and increase in oil consumption.

The new internal combustion engine in a permanent balance, according to the invention, avoids the above disadvantages. It eliminates the normal force which occurs with the known engines between the piston and cylinder, that is between the crosshead and the sliding bar, by the hinged coupling of the upper ends of the connecting rods by means of a strap guided by rollers or bumpers, tangential with two guiding ways, which form together with the pistons rods a jointed mechanism which actuates two synchronised crosshead mechanisms coupled oppositely.

The new scavenging system avoids the losses of fresh gas mixture together with the combustion gases, during the exhaust process, by the fact that the scavenging is obtained in two flows, the first one consisting of clear air which cleans the flue gases and the second one with fresh mixture, the latter being prevented to come into contact with the combustion gases. This is obtained by means of the two compression spaces intended to discharge from one of them clear air and the second one high petrol-air mixture—which exist at the bottom of the pistons.

The new air supercharging system in three stages, corresponding to the four-stroke engine operation, permits the obtaining of the natural inlet, double and treble charging—the two last ones forming the supercharging systems, due to the possibility of independent operation of the compressing spaces (behind the pistons) from the power space in front of the pistons.

The variation in steps or continuous of the compression chamber volume, depending on the supercharging degree (inlet pressure) and load permits the compression and combustion pressure to be constant (a constant thermodynamic ratio) in all regimes, improves the outputs also at partial loads.

The new combustion method available with carburator engines avoids the detonation and increases the field of using the low mixtures (α = 1.6 2).

At the same time it permits the increase of the compression ratio for the following reasons:

At maximum power rate pure air is admitted in one of the adjacent cylinders and to the second cylinder a rich mixture below the flash point (α = 0.6), which at the end of compression is mixed in the common combustion chamber, controlled by the tubular slide valve, which is shut during the inlet, compression and exhaust phases, opening only at the end of compression phase and beginning of expansion phase.

In the common combustion chamber limited by the tubular slide valve, there are residual hot gases from the preceding cycle at a pressure equal to the compression or somewhat below it, but a temperature much higher than that at the end of compression; this causes in the moment of space untight, the fresh air and the rich mixture to be injected from the respective cylinders into the common chamber—following two tangential directions given by two corresponding slots provided on the cylinder head bottom surface.

In this process of double mixing—pure air with rich mixture and with hot residual gases a progressive combustion takes place controlled by the process of feeding the common chamber with air and mixture—produced by the pistons traveling towards the upper dead point.

As the compression of the pure air and rich mixture are separately one—(below the flash point)—the compression ratio which can be obtained are much higher than those obtained today with the carburator engine.

At partial loads (economical) in both cylinders or only in a single one a low mixture allowed to enter which gets heated due to compression but cannot ignite with a sparking plug.

The ignition is obtained at the end of the compression cycle when the spaces become united, lean gas mixture is mixed with the residual gas of high temperature producing the self-ignition of the mixture and the progressive burning which takes place as the content of the cylinders is injected into the united combustion chamber.

The starting is assured by an incandescent sparking plug located excentrically in the united combustion chamber which stops operating as the combustion cycles are obtained, which will store the hot residual gases in the united combustion chamber.
In the case that in addition to the incandescence plug a centric injector is installed, then it is possible to obtain the fuel combustion any time or even continuously in the combustion chamber which is separated from the rest of the working space—which assures an equivalent of 700° crankshaft revolution at the disposal of the fuel for vaporization and preparation.

The fuel injection takes place in a space full of flue gases, and the vaporization and superheating time is approximately equal to the four-stroke of the diesel cycle.

In view of this fact, the idea of delay in ignition—that is the idea of fuel grade makes no sense, the use of any fuel oil being possible. When the combustion chamber is joining the two cylinders, the quantity of pure air which is delivered to the combustion chamber provides a progressive combustion, accurately controlled by its discharging process of the pistons which are approaching the top dead joint.

The tangential penetration of the air in the chamber assures an intense turbulence which provides within the chamber a good mixture of gases and pure air.

The new type of sealing for the high-pressure cylinders eliminates the possibility of the gases to infiltrate behind the rings, maintaining a permanent contact with the top of the ring's grooves in the piston. The group of the three rings have the slits staggered, there is no possibility of flow for gases or oil.

We give below one example of the present application of the invention, according to the illustrated FIGS.

FIG. 2 illustrates schematically a driving mechanism in permanent dynamic balance, actuated by a double cylinder;

FIG. 3 and 4 are schematic side- and front-elevation views of a driving mechanism in permanent dynamic balance with opposite pistons in a monocylinder (two views);

FIG. 5 is a partial schematic elevation of an internal combustion, two-stroke engine in permanent dynamic balance with carburator, having a double-flux scavenging through outlets;

FIG. 6 is a partial schematic elevation of an internal combustion two-stroke engine in permanent dynamic balance with carburator, having a double flux scavenging through the inlets and exhaust valves;

FIG. 7 is a schematic partial elevation of an internal combustion four-stroke engine in permanent dynamic balance, diesel cycle, supercharged in three steps (longitudinal section);

FIG. 8 is a cross-sectional view along the line shown by arrows in FIG. 7;

FIG. 9 is a sectional view through the common and supplementary combustion chamber of the four-stroke engine illustrated in FIGS. 7 and 8;

FIG. 10 is a distribution diagram of the treble supercharging phase;

FIG. 11 shows external characteristics of the three feeding rates;

FIG. 12 is a schematic partial elevation view of the internal combustion four-stroke engine in permanent dynamic balance with carburator or injection and with controlled ignition and combustion (longitudinal section);

FIG. 13 is a plan sectional view of separable common combustion chamber of the engine illustrated in FIG. 12;

FIG. 14 is a cross-sectional view of the valve actuator and combustion chamber slide valve;

FIG. 15 is an enlarged detail view of separable united combustion chamber;

FIG. 16 is a schematic side-elevation view of the sealing system for high pressures in a piston cylinder.

According to the invention the driving mechanism in permanent dynamic balance in accordance with this invention, actuated by a double cylinder as shown in FIG. 1, consists of a double cylindrical block 1 having pistons 2, actuated from the common combustion chamber 3 which transmits the force P to the rods 4, connecting rods 5, their top heads being connected by strap 6, the crankshafts 7 are synchronized by synchronizing wheels 9 having a common sump 9.

According to the invention the driving mechanism in permanent dynamic balance is actuated by a monocylinder with different piston, as shown in FIG. 2, has the same components.

According to the invention, the driving mechanism in permanent dynamic balance with opposite pistons in a monocylinder, as shown in FIGS. 3 and 4, consists of the same components as in FIGS. 1 and 2, in addition having a synchronizing mechanism of the opposite pistons consisting of auxiliary connecting rods 10 and rockers 11 with a fixed scavenging point 12, the rods being sealed and guided in the stuffing boxes 13.

According to the invention the internal combustion two-stroke engine in permanent dynamic balance with carburetor having a double-flux scavenging through outlets, as shown in FIG. 5, consists of the driving mechanism described in FIG. 1, where the double cylindrical block of the cylinders are provided with exhaust openings 14 scavenging openings 15 connected by ducts 16 to the pure-air compressor 17, to the scavenging openings of the rich-mixture compressor 19; the inlet is controlled by the sliding valves 20 and the rich mixture is delivered by the carburetor 21.

According to the invention the two-stroke internal combustion engine in permanent dynamic balance with carburetor, having a double-flux scavenging through the inlets and exhaust valves, consists of the same driving mechanism as shown in FIG. 1, with the specification that both cylinders from the double-cylinder block are provided with orifices for scavenging the pure air 15 and the mixture 18, the respective compressors 17 and 19 (for air and rich mixture) controlled by the rotating valves 20 which act also as camshaft bearing by means of which the exhaust valves are controlled 22.

According to the invention the internal combustion four-stroke engine in permanent dynamic balance, diesel cycle, supercharged in three stages according to FIGS. 7, 8 and 9 consists of the driving mechanism described in FIG. 1 having in addition the distribution through the exhaust valves 22, inlet valves 23, rotating valve-camshaft bearing 25, buffer tubes 26, pressure discharge valves 27 and the connecting grooves 28; the supplementary chamber 29 with a variable volume due to the piston 30 adjusted from the load governor of the injection pump by the lever 31.

FIG. 10 shows the distribution diagram for operating with treble supercharging regime, consisting of the natural inlet phase (α + β) and blast phase of two charges (γ) delivered by the compressor behind the pistons.

FIG. 11 shows the three outside features corresponding to the three feeding phases.

According to the invention the internal combustion four-stroke engine in permanent dynamic balance with carburetor and controlled ignition and combustion (longitudinal section) as in FIGS. 12, 13, 14 and 15, consists of the cylindrical block 1 which contains the pistons 2 which actuates the same driving mechanism described in FIG. 1, having in addition the turbulent combustion chamber 3 for both the cylinders, provided with the carburetor 21 and intake valves 22 and 23 driven by the camshaft 25 which transmits the motion to the sliding block 23 which by means of rods 33 and 34 drive the rocker lever 34 of the combustion chamber sliding valve 36 as well as the rocker lever 35; the combustion chamber sliding valve 36 is loaded by the spring 37 which is resting on the combustion chamber flange plate 38 which in turn is resting on the cylinder head cover 39; the combustion chamber slide valve 36 is resting on the heat resisting steel seat; the tangential grooves 41 extend along the lower surface of the engine cylinder head.

The tubular slide valve of the combustion chamber is sealed from outside and inside by rings 42 and, the original ignition being provided by the incandescent sparking plug 44.

According to the invention, the piston sealing system at high pressure as shown in FIG. 16, consists of identical compres-
sion rings 45 having the slits staggered at 180°, the radial ex-
pander 46 with the slit not to coincide with the other two on
the compression rings, the axial expander 47 which has a
sinusoidal shape; the piston interior is provided with oil-purging
ports 48.

The operation of the internal combustion engine in per-
manent dynamic balance shown in FIG. 1, is assured by the
fact that the identical pressures caused by the uniting of the
working spaces of the two cylinders which are traveling
through the same cycle produce the identical force P on the
pistons 2 which drive through rods 4 the connecting rods 5.
The decompressing valve 7 in two directions one along the
connecting rod axis 5 and the second along the connecting
strap axis 6 simultaneously in both crank-cross head
mechanism, the shafts 7 of which are coupled by end gearings
8 in 1:1 ratio of phase permanent opposition, results that the
energy distribution in all points of the system be identical and
in permanent reciprocal opposition, which produces a per-
manent dynamic balance of the two mechanisms.

The normal forces N which occur in the connecting strap 6
are permanently equal and of opposite sense, a fact which
stresses the strap in tension, or in compression respectively.
This arrangement avoids the normal forces that occur during
the working stroke of the connecting rod be transmitted out-
side.

The inertial horizontal force components are naturally
balanced due to symmetrical oscillation and in permanent
phase opposition of the two mechanisms.

The vertical components of the inertia force of the first
order are also balanced, by the driving mechanism which,
being provided with counterweights on the counterrotating
shafts, become automatically dynamic balancers of the first
order.

In the case of FIG. 2 the situation is the same, adding the
remark that the differential pistons system acts also as connect-
ing strap for the two driving mechanisms.

The operation of the internal combustion engine with per-
manent dynamic balance with opposed pistons shown in FIGS.
3 and 4, is assured by the identical forces which press the two
opposed pistons, thus being synchronized by rods 4 auxiliary
connecting rods 10 and rockers 11 centered in the oscillation
point 12, which in this way become a mechanical system in
permanent dynamic balance with accurate identity of the
forces which occur in all the points of the system.

The useful work of the resultant forces act on the two
mechanisms' crank-and crosshead which are also identical
and hence in permanent dynamic balance.
The engine unit shown in FIGS. 3 and 4 form a system in
perfect and permanent dynamic balance without forces and
free from inertia moments.

The two-stroke internal combustion engine with permanent
dynamic balance with carburator and a double-flux scaveng-
ing, according to FIG. 9 (outlets with interrupted line) respect-
ively FIG. 5 and 6, operates as follows:
The engine spaces are connected by the common com-
bustion chamber 3, the spaces from behind the pistons are
used to deliver to one of them fresh air and the other rich mix-
ture (air + petrol) absorption from carburator 21.

In the expansion phase, the piston which controls the ex-
hauot opens the exhaust outlets for flue gases, further the piston
which controls the scavenging opens first the fresh-air
inlets 15 supplied through the groove 16 by the compressor 17
then are opened the inlets 18, for rich mixture (air+petrol)
supplied by compressor 19 thus obtaining the evacuation of
flue gases by fresh air, avoiding the contact between the old
flue gases and the new carburated gases (air + petrol mixture).
The first part of the scavenging process (fresh-air delivery
under pressure) reproduces the identical scavenging conditions
as with the two-stroke diesel engine, consequently
without fuel losses involved with the exhaustion of flue gases.
In the case of the designs shown in FIGS. 3, 4 and 5 the
pistons themselves control the exhaust as well as the scaveng-
ing, the distribution diagram being symmetrical.

In the case of the version shown in FIG. 6 the evacuation
is governed by valves 22 by which one achieves asymmetrical
distribution diagrams.

The control slide valves of the inlet permit the inlet process
to take place along the whole distance of 180° due to this
phase in the compressors from the lower surface of the
pistons.
The internal combustion four-stroke engine in permanent
dynamic balance, diesel cycle supercharged in three steps, ac-
cording to FIGS. 7, 8, 9, 10 and 11 is operating as follows:

1. Normal feeding condition.
The slide valve 24 is fastened in the position drawn in FIG. 7
which assures the connection between the driving space and
atmosphere simultaneously with the isolation of compressor
delivery; the decompressing valve 24 is lifted from its seat
eliminating thus the compressor behind the pistons.
The four-stroke diesel cycle is achieved in conditions of a
direct air suction from the atmosphere which can be expressed
by the feature N (FIG. 11).

2. Double feeding condition.
The slide valve 24 is rotating by 180° against the former
position—thus providing the connection between the com-
pressors' delivery and driving spaces’ inlet, the valve 24 is
returned on its seat, cutting thus the connection with at-
mosphere, and providing in this way the working of the com-
pressors.

By the fact that the driving cycle has four strokes, hence a
single inlet stroke, at the same time the compressor cycle hav-
ing two strokes, producing in the same period two complete
compressor cycles hence two deliveries under pressure, one
can achieve the supercharging of the driving space with ap-
proximately two equivalent charges of natural suction—
consequently supercharging.

As to this increased quantity of air is corresponding an in-
creased quantity of fuel, the position of the injection pump
rack will be different in agreement with the new regime—a
fact which will influence the position of lever 31 which will
shift the piston 30 which will cause the increase of the sup-
plementary chamber volume. This will provide the same
compression pressure in supercharging and normal admission
conditions. The combustion will then result at the same maximum
pressure in both conditions.

The result is that the engine's power increases having the
external feature N 3  without causing the maximum stress in the
parts—however the average stress increases according to the
new conditions.

3. Treble feeding condition.
The slide valve is connected kinetically with the camshaft 25 so that during the time of normal inlet (which lasts \(\alpha + \beta\) according to FIG. 10) the driving space is con-
ected with the atmosphere. At the beginning of the period
(\(\gamma\)) the connection with the atmosphere is changed over to
the buffer ducts 26, where the two charges delivered by the
compressors behind the pistons are stored, blasting this quan-
tity of air to that taken normally from the atmosphere. In this
way, the equivalent of three volumes admitted from the at-
mosphere is provided.

The new increase in the air quantity will correspond to
another position for the injection pump rack, that is with an
increase in the quantity of the injected fuel and with another
increasing variation of the supplementary chamber volume,
which maintains constant the compression and combustion
pressure. The result is a new increase in power (the external
feature \(N_3\)), without the equivalent increase of maximum
stress in the motor parts.

The combustion process is ensured by the common com-
bustion chamber 3 of toroidal shape provided with connecting
grooves inclined on the direction of V- and T-arrows,
establishing thus a double vortex motion of the air which
comes from the cylinder; the resultant R producing a twisting
motion of a spiral on the torus—which assures one intimate
mixture between the injected fuel and the blast air. The com-
bustion done, the pressure in the chamber increases and re-
jects the gases incompletely burnt in the chambers formed by the pistons' recesses according to a direction tangential to the cylinder, causing thus a new plane vortex motion which provides a new mixture of the incomplete burnt gases and the air from the cylinder.

This treble vortex motion allows the use of a minimum excess of combustion air and an excellent insensitiveness to the grade of the used fuel.

The combustion engine in permanent dynamic balance, with carburation or injection having a controlled ignition and combustion, as shown in FIGS. 12, 13, 14 and 15, operates as follows:

In the case it operates with carburation, one admits in a cylinder fresh air or very lean mixture and in the second cylinder very rich mixture. The lean mixture as well as the rich mixture are placed out of the possible combustion interval of the respective mixture.

In the two cylinders takes place the compression separately—towards the end of the compression the combustion chamber slide valve 36 is opening allowing the blasting of the fresh air through the tangential grooves 41 of the fresh air in a cylinder and of rich mixture in the second cylinder in the common combustion chamber which contains fresh gases at a pressure equivalent or somewhat lower than the compressed gases.

It results a double mixing of the air with the rich mixture and the high temperature fresh gases—a fact which leads to the establishing of an air + fuel ratio which can ignite and burn properly.

The presence of hot residual gases and at pressure near to that of the compression, provides a superheating of the new mixture and a safe ignition.

As the mixing process is determined by the discharge of the air and mixture by the stroke of the pistons towards the upper dead point, the combustion process is performed at the rate and dynamics required by the discharging conditions.

It results that the combustion rate in permanent relation with the engine revolution and the process controlled without having the possibility to operate independently.

In the case of operation with injection—in both cylinders 40 admitted fresh air. In the combustion chamber separated by the two cylinders the fuel is injected in the atmosphere of residual fresh gases of high temperature with a lead which can reach of 650—700° crankshaft revolution.

In this period which is equivalent with the period of the four-stroke cycle occurs the process of heating, vaporization at preparation of superheated fuel vapors and residual gases.

When the combustion chamber slide valve opens, the compressed and heated air is blast in the common combustion chamber and as it mixes with the fresh air the burning proceeds, the rate of which is determined only by the fresh-air blasting process.

Since the time of fuel preparation and the thermal conditions under which this process is fulfilled are good the delay in ignition does not exist as phenomenon and idea.

Under these conditions the engine can run with liquid or gaseous fuel, especially those which have a good resistance to ignition (heavy oil, low cetane number etc.).

The slide valve adjustment 36 is performed by the engine camshaft 25 which transmits the lifting motion to the rockering lever 34 against the spring (37) which is resting on the combustion chamber ceiling plate 38.

By the rings 42 and 43 the combustion chamber slide valve sealing is secured, and the heat-resisting steel plate 40 has the role of warm wall on the combustion chamber, on its surface being injected the sprayed fuel.

The performance of the sealing system at high pressure reduces itself to the condition that the axial force of expander 47 may assure a permanent contact of the package rings 45 and 47 with the upper part of the grooves of the piston ring. Under these conditions all the flow interstices became annihilated, hence a perfect tightness is obtained, for gases as well as for oil and at the same time the impossibility to form gas pressure behind the ring—proportional with the chamber pressure which is the major source of losses through friction and of excessive wear. The alternative movement of the rings in the corresponding grooves being eliminated, it is impossible for the hot gases and oil to come into contact and consequently an impossibility of coking regardless of the engine thermal working conditions. Moreover there is an improvement of heat transfer conditions from the pistons to the rings by maintaining a permanent contact between the ring and piston which takes place on clean surfaces without coke. The oil gathered by the rings return to the sump through the ports 48.

The advantages of the new engine are:

The complete canceling of the normal force which is not taken up by specific pressure of the contact between the piston and cylinder allowing the increasing of the combustion maximum pressures (up to over 300 atms.) by increasing the compression ratio and degree of supercharging.

The same effect is obtained by eliminating the gas pressures from behind the rings.

Eliminating the evaluation of the cylinder and piston ovalization, hence increasing of service life.

Increase in mechanical efficiency, due to the absence of friction between piston and cylinder, also decrease of that between the rings and cylinder.

Excellent dynamic balance in the case of the variants shown in FIGS. 1 and 2 where all the horizontal inertia forces are eliminated and also the vertical forces of the first degree are equally balanced and consequently an in the case of the variant shown in FIGS. 3 and 4, where all the inertia forces of all degrees from 1 to ** are perfectly balanced.

The obtaining of the double-flux scavenging (fresh air and mixture) without being necessary for this purpose, two special compressors—eliminates the losses of mixtures at the same time with the flue gases—simultaneously with the engine becoming simpler by the elimination of compressors apart from the engine.

The performance of four-stroke diesel engine with the load adjustment, respectively the supercharging degree, with maximum combustion pressure of cycle is highly profitable in all working beginning with the reduced speed to the maximum load.

The possibility of operating in three supercharging conditions, together with that of high admissible pressures for the thermal cycle give very high output per litre of the unit.

The toroidal combustion chamber with double vortex makes possible the operation with minimum air + fuel ratio.

The three external features of power revolution for each double cylinder gives the unit the flexibility of the first degree to the requirements of the driving clutch similar to that rendered by the electromagnetical converters and as a matter of fact eliminates the gear box.

In the case of thermal cycle with adjusted combustion (FIG. 12) with carburation, the detonation is eliminated as the compression ratio increases much above the present limits at the same time with the possibility of operation with partial regimes with finally very lean mixtures of air-fuel.

In the case of thermal cycle with adjusted combustion and the injection of fuel in the separable combustion chamber—permits the preparation of any fuel—no matter how heavy, having at its disposal a time equal with the four-stroke thermal cycle itself (700—720°) and the obtaining of a combustion process accurately controlled and governed by the compressed air injection speed in the separable chamber, at the same time with the possibility of operation with a minimum excess of air + fuel in maximum load condition or with very low mixtures at low loads—hence a high rentability.

The proper combustion begins when the separable chambers join the cylinder.

The beginning of the combustion concurs with the beginning of compressed air injection and lasts all along the injection time, that is until the piston reaches the upper dead point. Due to this fact, in the expansion period, the retarded combustion does not any longer take place.
The reason of this is that the whole quantity of air is forced to mix with the prepared fuel (vaporized and supercharged) in optimum thermodynamical conditions, namely at maximum temperatures and pressures.

The above elements lead to the following advantages:

Increasing of mechanical efficiency by eliminating the normal force and decreasing of ring friction.

Increase of indicated efficiency by the possibility to increase to maximum the cycle pressure and the compression and supercharging ratios.

The use of toroidal chamber with double vortex or of the separable chamber with adjusted combustion result in effective efficiencies and output per litre which up to now were not reached by any thermodynamical unit—which have as direct effect a very low specific fuel consumption.

The permanent agreement between the load and the maximum pressure, by varying the chamber volume and constant maintenance of the maximum pressure, that is of that of the compression, assures economical operation conditions also in the period when it works at reduced load.

Its ability of operating with any grade of fuel without detonation or retarded ignition, beginning with petrol and finishing with residual heavy oil in conditions of controlled combustion, gives to the new motor a versatile use and maximum economy.

We claim:

1. In an internal combustion engine, piston means including a pair of piston rods symmetrically arranged with respect to a given axis in a pair of piston cylinders disposed in said internal combustion engine and being at all times equidistant from said given axis, a pair of two cross-interconnecting said rods, guide means coacting with said strap for guiding the latter for movement in the direction of movement of said piston rods, a pair of identical and opposed synchronously operating crank means geared to each other pivotally connected to said strap and adapted to reciprocate said piston means, and passage-defining means defining passages for a combustible mixture in communication with said pair of piston cylinders, said passages coacting with opposed ends of said piston means for achieving fluid flow which includes compression of the fluid in both directions of reciprocal action of said piston means.

2. The combination of claim 1 and wherein said engine is a two-stroke engine having a carburator, and said passage-defining means coacting with both ends of said piston means for providing scavenging by way of a pair of fluid-flow paths one of which accommodates fresh air and the other of which accommodates a fuel-air mixture.

3. The combination of claim 1 and wherein said engine is a four-stroke engine while said passage-defining means coacts with one end of said piston means to provide for compression, combustion, and exhaust of a fuel-air mixture, and said passage-defining means coacting with the other end of said piston means to provide for supercharging.

4. The combination of claim 1 and wherein said piston means is a differential piston and said rods are connected by said crank means in opposition.

5. The combination of claim 1 and wherein said piston means includes a differential piston composed of a pair of piston portions situated in a common cylinder and movable in opposition toward and away from each other, said passage-defining means coacting with the space between said piston portions.

6. The combination of claim 2 and wherein said piston means coacts with ports forming part of said passage-defining means, and all of the compression and fluid flow within the engine being derived solely from movement of said piston means.

7. The combination of claim 6 and wherein said piston means includes a pair of pistons situated in side-by-side relation, a pair of cylinders respectively accommodating said pistons and communicating with said passage-defining means, the latter including a combustion chamber coacting with ends of both of said pistons and valve means coacting with said pistons and one group of said ports of said passage-defining means for providing scavenging, said passage-defining means directing air to be compressed by ends of said pistons which are opposed to and directed away from said combustion chamber.

8. The combination of claim 3 and wherein said piston means includes a pair of pistons, a pair of cylinders in which said pistons are accommodated, said passage-defining means providing combustion chambers coacting with said pair of pistons at one of the ends thereof, and said passage-defining means providing passages communicating with the opposed ends of said pistons for achieving a supercharging compression by way of the movement of the latter opposed ends.

9. The combination of claim 8 and wherein an inlet valve means coacts with said combustion chambers for admitting air into the latter, and control valve means for providing in predetermined positions of the latter communication between the combustion chambers and the spaces at that side of said piston opposed to said combustion chambers, to increase the supply of air to said combustion chambers, thus achieving the supercharging action.

10. The combination of claim 9 and wherein an additional suction inlet forms part of said passage-defining means for further increasing the air suction directly from the atmosphere so as to further increase the air by approximately three times the quantity consumed when said control valve means is closed and said additional suction inlet is closed while said additional suction inlet is closed and said control valve means is open and connecting strap supplied through said inlet valve means is approximately doubled.

11. The combination of claim 10 and wherein a volume-control means coacts with said passage-defining means to adjust the combustion chamber volume to a value which will maintain compression at a constant level and which will achieve combustion at any load and at any degree of supercharging.

12. The combination of claim 11 and wherein said combustion chambers are combined and consist of at least two spaces one of which has a constant volume and torus-shaped configuration for providing turbulence, said passage-defining means providing for admission of air along tangents to said torus-shaped turbulence chamber, and the other of said spaces having a variable volume and configuration and acting as an air accumulator, said volume-control means including a movable piston for varying the volume, an injection pump for supplying fuel, and a mechanical transmission between said injection pump and volume-controlling piston for positioning the latter in accordance with the operation of said injection pump, maintaining turbulence in said torus-shaped turbulence chamber during combustion and expansion through flow of fluid in the variable volume chamber to provide approximately stoichiometric combustion conditions.

13. The combination of claim 1 and wherein said passage-defining means includes a volume-regulating means coacting with said piston means for selectively reducing the volume of a combustion chamber thus enabling said piston means to act on a small quantity of combustible mixture at high temperatures, said volume regulating means acting during the expansion stroke of said piston to provide a predetermined lead of admission of fuel-air mixture.

14. The combination of claim 10 and wherein said passage-defining means includes an air inlet for independent admission of fresh air and of a fuel-air mixture enriched above the admission limit, said passage-defining means coacting with said volume-regulating means for providing separate compression of said fresh air and of said enriched mixture and then interconnecting 70 spaces in which the fresh air and enriched mixtures are separately compressed, achieving a combustion dependent upon displacement of said piston means for automatically adjusting the combustion to the engine speed.

15. The combination of claim 14 and wherein a valve means coacts with the spaces in which said fresh air and enriched
mixture are separately compressed while maintaining for both pistons in the cylinders in which they reciprocate identical compression ratios with identical combustion and expansion simultaneously in both cylinders up to the moment when said valve means closes.

16. The combination of claim 1 and wherein said piston means has an exterior groove receiving a pair of piston rings which are angularly displaced one with respect to the other, axial expanding means situated between said rings and urging the latter respectively against opposed sides of said groove, and radial-expanding means also acting on said rings for radially expanding the latter assuring compression and lubrication without sacrificing gas tightness.