ENERGY CONVERSION PROCESS FOR INTERNAL COMBUSTION ENGINE AND PISTON ENGINE FOR CARRYING OUT SUCH PROCESS.

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Description

This invention relates broadly to what conventionally is known as a combustion process of the type used in piston-type internal combustion engines as well as combustion chamber designs for such engines and arrangements for distributing fuel and air in the combustion chambers during engine operating cycles.

In accordance with the prior art, it is generally known that combustion can be improved by supplying air into the combustion zone from an air chamber in which part of the intake air has been previously compressed by a moving piston of the engine. The expanding compressed air is believed to impart high turbulence to the mixture in the combustion space and to improve the combustion process in accordance with various theories. For example, U.S. Patent No. 1,344,352 discloses an apparatus for carrying out such a process in a diesel engine environment. However, this patent does not disclose the use of a variable width connecting passageway between the air chamber and the combustion zone nor is there any specific provision for preventing contamination of the air chamber with fuel. Moreover, due to the design of the air reservoir chamber (annular chamber communicating with the combustion zone through a series of nozzles), the necessary molecular action utilized in accordance with the present invention will not likely occur.

It is also known to provide a supplemental air chamber in the top part of a moving piston of an internal combustion engine which receives both fuel and air which react in the reservoir during the combustion process to augment the reaction being carried out in the main combustion zone. Such an illustration is shown in U.S. Patent No. 3,923,032. In accordance with the present invention, only air (perhaps with slight contamination of fuel, but not to the extent that it is combustible) is maintained in the air chamber so that combustion does not occur in the chamber itself. It is quite important to prevent combustion in the chamber of the present invention, for reasons that will be evident in the specification set forth below. In accordance with this invention, substantially fuel-free air always remains available in the reservoir chamber to provide a source of highly energized oxygen molecules that are controllably admitted into the combustion zone during the entire combustion process.

In accordance with U.S. Patent No. 2,119,219, it is known to provide a variable area passageway between a combustion zone and an adjacent chamber. However, the present invention is readily distinguishable over the prior art exemplified by this patent in that, in accordance with this invention, only air is provided in the auxiliary chamber and substantially all fuel is maintained in the main com-

bustion chamber. Thus, in accordance with the present invention, auxiliary air supplied to the combustion zone from an auxiliary, combustion sustaining air chamber is maintained substantially free from contamination with hydrocarbon fuel.

According to the present invention there is provided a combustion type energy conversion process for use in a work producing moveable piston internal combustion engine wherein charges of fuel and air are cyclically supplied to and combusted in a combustion chamber and wherein a portion of the air part of each charge is disposed in a combustion sustaining air chamber of substantially fixed volume connected with and closely adjacent the combustion chamber at the beginning of each expansion cycle, the connecting passageway between the air and combustion chambers being delimited by a gap including an open width and length facing the combustion chamber and a depth directly rearward of the open area and extending to the bottom of the air chamber, characterized by; formulating the charges in response to power demand of the engine in a manner such that the total air to fuel ratio thereof is stoichiometric at full power demand and on the lean side of stoichiometric at all other times; and causing the combustion at each charge to occur at least above idling conditions with the fuel to air proportion in the combustion chamber on the rich side of stoichiometric throughout the combustion event by controlling the amount of oxygen molecules supplied to the combustion area from the air chamber through the connecting passageway, such control over the quantity of oxygen molecules being achieved solely by the geometry of the connecting passageway, and including the step of varying the size of the opening of the passageway from a smaller to a larger size between initiation and completion of each combustion event.

This process, in addition, may include the step of varying the size of the opening of the passageway by using the position of the piston or pistons to selectively restrict and open the passageway.

In one embodiment of the process, the entire fuel quantity is aspirated into the combustion chamber during each charge intake event and the fuel to air proportion is varied by varying the quantity of the fuel in each charge without varying the total quantity of air; and furthermore, fuel is supplied to the combustion chamber beginning not earlier than 30 to 50° with respect to output shaft rotation after the start of each respective charge intake event.

In another embodiment of the process, at least part of the fuel of each charge is injected directly into the combustion chamber during the compression of the charge, but not later that 30 to 40° before initiation of the combustion, and the fuel to air proportion is varied by varying the
quantity of fuel in each charge without varying the total quantity of air.

In another embodiment of the process, the entire fuel quantity of each charge is injected under pressure into the combustion chamber during the central period of the intake and compression events; and the fuel to air proportion is varied by varying the quantity of fuel in each charge without varying the total quantity of air.

Also in accordance with the invention there is provided a work producing engine for carrying out the process referred to above, such engine including a moveable piston or pistons for compressing each charge and reacting to the expansion forces resulting from the combustion of same, a variable volume combustion chamber at least in part defined by the moveable piston or pistons; a fuel and air charge supplying and distribution means, an air chamber located adjacent to and in communication with the combustion chamber through a passageway delimited by a gap including an open width and length facing the combustion chamber and a depth directly rearward of the open area extending to the bottom of the air chamber, characterized by; said open area of said passageway being disposed relative to said piston or pistons in a manner whereby movement of the piston causes the size of said open area of said passageway to vary between a minimum open area when the combustion chamber volume is at minimum and a larger open area when the combustion chamber volume is at greater than minimum; said fuel and air charge supply and distribution means is arranged to cause distribution of all the fuel portion of each charge to the combustion chamber and a part of the air portion only of each charge to the air chamber for each operative cycle of the engine; and a fuel to air proportion control system for varying the ratio of fuel to air of the charges supplied to the combustion chamber.

In one embodiment of such engine, the solid surfaces delimiting the combustion chamber, air chamber and passageway are geometrically configured to avoid straight line of sight paths between the principal combustion and air chamber volumes and to provide such paths between the combustion chamber and the passageway volume.

The ratio of the air chamber volume to the minimum combustion chamber volume is preferably between 0.2 and 1.8; and the width of the passageway when it is at its minimum area is not less than 0.050 inch (1.27 mm) and at its maximum area not greater than 0.20 inch (5.08 mm).

The piston or pistons may reciprocate within a cylindrical working chamber and the maximum passageway open area is between 0.05 and 0.15 times the square of the diameter of the combustion chamber. Further preferably, the passageway volume (gap area times depth) is between 0.10 and 0.35 times the total air chamber volume when the passageway area is at its maximum size.

In an embodiment of the engine, said moveable piston or pistons is a single piston reciprocally mounted within a cylindrical bore with the combustion chamber disposed between a closed head end of the bore and a closed top end of the piston, the piston includes an upper compression seal and a generally concave arcuate top end, characterized in that the bore head end includes a centrally converging side wall area and a generally concave arcuate end area, the air chamber is disposed in the peripheral area of the piston below its top end between said top end and said compression seal; a radial clearance gap is provided between the piston top end and the cylinder wall, said gap defining said passageway, and said passageway is varied in size as said piston approaches and recedes from said converging side wall area.

The peripheral area of the piston end adjacent the passageway is preferably shaped so that, with the adjacent cylinder bore sidewall the passageway converges in a direction away from the combustion chamber, and the surfaces on the piston leading to the passageway from the combustion chamber are preferably convexly curved.

In a further embodiment of the engine, said piston or pistons is a single piston reciprocally mounted within a cylindrical bore, the combustion chamber is located between a closed head end of the bore and a closed top end of the piston, the bore head end includes an inwardly converging side wall area and a convex arcuate end area, the piston includes an upper compression seal and a concave arcuate top end area, characterized in that the air chamber is disposed in the peripheral area of the bore head end; the length of the passageway area extends along the periphery of the bore head end; a peripheral area of the piston top end lies adjacent the passageway area in partially blocking relationship when the combustion chamber is at minimum volume and is clear of the passageway when the combustion chamber is larger than minimum volume.

In another embodiment of the engine, said piston or pistons are a pair of closed ended pistons with top compression sealing means and reciprocally mounted within a single cylindrical bore, the pistons, together with the closed top end and the bore, defining said combustion chamber, characterized in that the piston top ends each have concave arcuate central portions and respective peripheral top edge portions that are radially spaced from the bore wall to define a maximum passageway width; said air chamber lies in the peripheral area of the top end of each piston between said peripheral edge portion and said compression sealing means of each piston; and the distance between said piston peripheral top edge
portions when the combustion chamber is at minimum volume defines a minimum passage-
way width.
Prefered embodiments of the process and apparatus are described in detail below and in
the accompanying drawings.

Detailed description of the invention
Description of drawings
With reference to the drawings appended to this specification, which illustrate preferred
embodiments of the invention:
Figure 1 is a pressure-volume diagram showing
the theoretical events of a power cycle for an
internal combustion engine operating in
accordance with the present invention and a
schematic outline of the thermodynamic system
depicting a theoretical engine capable of carry-
ning out the cycle:
Figure 2 is similar to Figure 1, showing the
contribution of Region I to the events of the
power cycle of the engine depicted in Figure 1;
Figure 3 is similar to Figure 1, showing the
contributions of Region II to the events of the
power cycle of the engine depicted in Figure 1;
Figure 4 shows a side elevational sectional
view of a working chamber layout of an actual
engine for carrying out the process in accord-
ance with one embodiment of the invention;
Figure 5 is a top sectional view of the work-
ing chamber taken along line V—V in Figure 4;
Figure 6 is an enlarged detail view of the
area generally bounded by circular line VI in
Figure 4;
Figure 7 is similar to Figure 6 but shows the
piston of the engine lower than top dead center;
Figure 8 schematically shows the fuel and air
supply system for the engine illustrated in
Figure 4;
Figure 9 schematically shows an alternate
embodiment of the fuel and air supply system
for the engine illustrated in Figure 4;
Figure 10 shows still another alternate
embodiment of the fuel and air supply system
for the engine illustrated in Figure 4;
Figures 11 and 12 graphically illustrate prior
art Otto and Diesel four stroke power cycle
events in timed relationship with a rotary power
output shaft;
Figures 13 and 14 are similar to Figures 11
and 12, only showing the power cycle events
according to the present invention in connec-
tion with spark or glow plug ignited and spon-
taneously ignited cycles, respectively;
Figure 15 shows an elevational sectional
view of an alternate working chamber layout
of an engine constructed in accordance with the
present invention;
Figure 16 shows an enlarged detailed view
of the area at XVI in Figure 15;
Figure 17 is similar to Figure 16, showing the
piston below top dead center;
Figure 18 is a sectional view of an alternate
working chamber layout constructed in accor-
dance with this invention;

Figures 19 and 20 are enlarged views of the
area IX in Figure 18 with the pistons in two
positions;
Figure 21 is a sectional view of an alternate
working chamber layout constructed in accor-
dance with this invention; and
Figures 22 and 23 are enlarged detailed views
of the area XXII in Figure 21, with the pistons in
two positions.

Description
With reference to Figure 1 of the drawings,
an ideal gas power cycle for an ideal internal
combustion engine operating according to the
present invention is represented by a plotting of
pressure versus volume of the gas in the
variable volume working chamber of the engine
during cycle events call beading to activation
(compression), addition of heat, expansion, and
rejection of heat. The diagram is similar to a
conventional pressure-volume diagram for
conventional air standard heat engines nor-
manly used to illustrate various predictable
equilibrium states reached by the gas mass
during the cycle based upon certain classical
assumptions that are well recognized and
explained in various thermodynamic text books.
In classical diagrams of this sort, for example, it
is assumed that the equilibrium states are
reached through a series of reversible pro-
cesses over a closed cycle. As will be seen, this
is not the case in the cycle of the present inven-
tion, and therefore, the lines between the points
on the diagram are intended to graphically
depict this fact.
The ideal cycle diagram shown in Figure 1
must be considered in connection with a theo-
retical internal combustion engine 10 that is
assumed to have variable volume working
chambers divided into Regions I and II that are
separated by a theoretical ideal partition 12
that is assumed to be capable of controlling
interaction between the regions to the extent
that the pressures within the regions are always
equalized during the power cycle, while the
temperatures, for example, in each region may
be different at any point in time. Stated dif-
nently, and from a thermodynamic viewpoint,
the internal energy status or entropy of the
molecules in each region can be different, but
the manifestation of average pressure within
each region is always equal. As will be
discussed in more detail below, this requires the
assumption that the partition 12 can be variable
in volume to enable the pressure-volume-
temperature relationships in each region to
satisfy classical equilibrium requirements when
heat is added to or rejected from a mass of gas
within the regions under adiabatic conditions.
The volume of Regions I and II is assumed to be
varied by pistons 14 that move in unison.
An ideal gas cycle for a heat engine, as is
well understood, is one capable of carrying out
the events of a closed power cycle over a series
of reversible or irreversible processes between
equilibrium states, which processes eventually return the system to its original state, and having as a fundamental function the transformation of heat, derived from conversion of chemical energy into thermal potential, into usable work. In ideal so-called internal combustion engines, the heat is cyclically drawn into the system from external reservoirs while ignoring the actual rapid dissociative chemical reactions between activated fuel components and oxygen which periodically sustain expansion of the heated reaction products to drive a moveable piston element that forms one wall of the variable volume working chamber of the engine for transforming the heat into usable work.

From a theoretical standpoint, the expected work that is available can be projected by using a pressure-volume diagram to compare the effects of varying compression ratio of different systems, that is the ratio between the total volume of the combustion or working chamber at its maximum volume and the volume of the combustion chamber at its minimum volume. However, it is believed that theoretical expectations in large part are not fulfilled in actual internal combustion piston engines due to the fact that such engines have been designed without a recognition on the part of the designers of the importance of the time factor that must be accommodated in a real engine during the addition of heat event.

Based upon a consideration of the natural progression of minute discrete events that must occur during a dissociative reaction process, such process sometimes simply being referred to as "combustion", the present invention first proposes that the combustion reaction event be described in the molecular or "micror" domain, since this is required for the evaluation of the contribution of the time bounded requirements of the addition of heat event of the power cycle of the internal combustion engine, while recognizing that heat is supplied by converting chemical energy to thermal potential to drive a continuously moving piston element. The present invention then proposes a process and apparatus for accommodating the time requirement needed to accomplish complete reaction of the fuel component by controlling in a sustained manner the availability of activated oxygen within the working chamber during the time span corresponding to the addition of heat event in the power cycle, while controlling the proportions of reactants in the working chamber during the reaction.

In summary, the power cycle according to the present invention is based on a refinement of classical assumptions surrounding the addition of heat events during the cycle, using suitable microcoordinates (as distinguished from macrocoordinates) for describing the probability distribution function of the molecular interactions during the cycle. Such microcoordinates describe the system in the domain of modern quantum theory, statistical mechanics, statistical thermodynamics, etc., and permit the analyst to accommodate the theoretical evaluation of the time bounded minute discrete contributions of energy flux through which the supply of heat is generated.

Referring again to Figure 1, the total system is described as a volume isolated adiabatically from the surrounding environment and filled to its maximum capacity at its stable point 1 with a mass of ideal gas. The gas is composed of a charge of ideal molecules capable of reacting in a manner resulting in the conversion of the chemical potential of the charge into a pre-established quantity or value Q of thermal potential. The points 1 and 2 shown as solid dots are assumed to be equilibrium states in the power cycle, the same as a conventional pressure-volume diagram, where points 1 and 2 represent the pressure-volume relationship of the gas in the combined working chamber Regions I and II, assuming no heat flow from or into the system, and assuming that the partition 12 permits pressure equalization but not necessarily temperature or other internal energy equalization between the regions. Between points 1 and 2, the gas in Regions I and II is adiabatically compressed by reducing the working chamber volume by means of piston or pistons 14 and the pressure and temperature of the gas increases to reflect the increased internal energy of the gas molecules. The density, or population, of the gas is also increased, naturally, and point 2 therefore represents an equilibrium state of the gases in Regions I and II that has been reached through a series of irreversible minute processes or events of conversion of externally supplied mechanical energy into internal energy of the system.

A quantity of heat Q is assumed to be added to the system according to the present invention starting at point 2 and it is assumed also that a portion of the total heat Q is added at constant volume and the remainder portion of the total heat is added at constant pressure. States 3 and 4 of the system are shown as a circle within a cross to indicate that these are undefined states and they are connected by cross lines to indicate that the thermodynamic process is undefined and irreversible, since neither state in the process can be defined using macro thermodynamic coordinates of the system. The net effect, however, can be plotted on the pressure-volume scales in the manner illustrated to allow a non-equilibrium thermodynamic representation of the cycle.

A careful analysis of points 3 and 4 requires one to consider that the first of the total heat portion is assumed to be added to Region I as a continuous, yet quasi-instantaneous series of minute energy releasing events assumed to occur along an irreversible constant volume path. The partition 12 between regions allows pressure in Region II to follow pressure in Region I, but the temperature in Region II does not follow the temperature in Region I.
system therefore cannot be defined in equilibrium terms at point 3, and furthermore the assumed capabilities of partition 12 must satisfy the natural classical pressure-volume-temperature relationships of the gases that must exist at point 3. In actuality, the partition 12, as point 3 is approached, must accommodate the change in volume of Region II that must exist at point 3. This will be explained in somewhat more detail below in connection with the description of Figure 3.

The remaining heat portion is assumed to be added to the system at point 3 also in a continuous, time bounded irreversible series of minute energy conversion events. Since the partition 12 permits interchange to the extent that pressures in each region remain equalized, the states of Region I at points 3 and 4 are identical since the temperatures and pressures in Region I are unchanged. However, Region II reaches a new state at point 4 as a result of the additional, secondary heat contribution. Now, the temperatures between Regions I and II are different, so the system is not at equilibrium at point 4, requiring that the partition 12 once again must accommodate the gas pressure-volume-temperature relationships then existent in the Regions I and II.

From point 4 to point 5, the gas in the working chamber is permitted to expand adiabatically until the chamber is at its original volume, and thereafter heat Qp is rejected from the system to return the latter to its original starting point 1. The state of the system at point 5 again is undefined because the temperature of the mass in Regions I and II are different. At point 1 again, the pressures and temperatures are equalized and the state of both regions returns to starting conditions.

The pressure-volume diagram of Figure 1 resembles the theoretical pressure-volume of a classical combined Otto-Diesel cycle wherein a portion of fuel is combusted at constant volume and a portion combusted at constant pressure. However, it will be seen that, in actuality, the process according to the present invention prolongs the heat input event by supplying a suitable, fast reacting charge of reactants in the working chamber before the reaction is initiated, and thereafter accommodates the natural time requirements of the reaction by continuously supplying activated oxygen before, during and after initiation of the reaction, without regard to constant volumes or constant pressure conditions.

In Figures 2 and 3, a closer analysis of what transpires in each of Regions I and II during the cycle is shown, and helps one to further understand the theoretical basis for the process of the invention. In Figure 2, Region I is shown as it changes its states during the cycle, while in Figure 3, Region II is depicted to show it various states along the same cycle, assuming the existence of the theoretical partition 12.

In Figures 2 and 3, both Regions I and II reach state 2 along the same process as is depicted in Figure 1. Heat portions ZQ and (1-Z)Q of the total heat added Q are then respectively assumed to be supplied to the system in Regions I and II. In Figure 2, the states of Region I at points 3 and 4 as a result of the assumed supply of heat ZQ in Region I are the same because the pressure, volume, and temperature of the gas in Region I are unchanged as the total system approaches states 3 and 4. In Figure 3, on the other hand, it is evident that the Region II adiabatically proceeds from its state at point 2 to its state at point 3, since it shares the pressure increase occurring in Region I as a result of the addition of heat event in Region I. Point 3 for Region II therefore must lie to the left of point 2 on the pressure-volume diagram. The adiabatic pressure increase in Region II must also be assumed to be accompanied by a decrease in volume of the gas in Region II, with the partition 12 accommodating the volume differential, since the temperatures in Regions I and II at points 3 and 4 are different, and the sum of the volumes in Regions I and II at state 3 will no longer be theoretically equal to the total volumes of Regions I and II at point 1. When the heat portion (1-Z)Q is assumed to be supplied to Region II at constant pressure, Figure 3 illustrates that the state of Region II approaches point 4 along an irreversible path as shown. The state of the system in Region I is unchanged because the pressure and volume in Region I is not affected by the assumed addition of heat in Region II. Adiabatic expansion of Region II to point 5 then follows, and rejection of heat between points 5 and 1 returns the system to starting conditions.

The ideal equilibrium and quasi-equilibrium assumptions in the macro domain of a classic thermodynamic heat balanced cycle proposed for heat engines, when compared with Air Standard Otto and Diesel cycles, reveals that for equal gas mass, compression ratio and total heat input, the heat balanced cycle can theoretically produce lower peak pressure and higher mean effective pressure than the Otto cycle and is capable of operating with a higher thermal efficiency than the comparable Otto cycle. The above comparisons, of course, are based on theoretical thermodynamic calculations and assume that the system follows reversible processes between equilibrium states and that heat inputs can be separately maintained. The latter, of course, represents an accomplishment that is difficult to achieve in a real engine.

In the power cycle of the present invention, the theoretical limit of the "heat balanced" cycle is fully scientifically and technologically refined in a manner that accommodates real internal combustion engine environments. The use of a micro point of view for describing the irreversible processes constituting the events of the cycle recognizes the micro domain which is required for evaluating the contributions of the
minute, discrete, enormous number of time bounded events through which energy conversion occurs. Accordingly, fuel and oxygen quantity and distribution controls are used in conjunction with a working chamber and piston geometry that enables better separation of the heat input contributions of what can roughly be considered the functional equivalents of the theoretical Regions I and II.

In the present invention, the separate regions in the real cycle are embodied in a variable volume working chamber that is in constant communication with an auxiliary fixed volume air chamber called a "sustaining chamber," through an open passageway or partition area and/or volume between the chambers. The partition area or volume is carefully controlled in size throughout the cycle and varies from a minimum at initiation of the reaction to a maximum during the expansion state of the cycle. At all times, the communicating area and volume enable pressure equalization between chambers, but travel of molecules of gases between chambers promoted by molecular internal energy differential between the chambers is controlled by careful design of the solid surfaces surrounding the communicating area and volume. By distributing the fuel and oxygen components so that all the fuel is in the working chamber and the oxygen (supplied in air) is divided between the working and sustaining chambers in a predetermined ratio, and thereafter activating the components both by mechanical compression and the reaction process itself, and finally controlling sustained participation of the highly activated oxygen in the reaction during the cycle, the following occurs. The reaction is initiated and proceeds to useful completion with the proportion of partially unreacted fuel to uncombined oxygen always on the fuel rich side of stoichiometric; the probabilities of successful collision between fuel and oxygen molecules (including molecules of various intermediate fuel species and oxygen) in the working chamber is maximized; and the time requirements for completion of the reaction are accommodated during the cycle. As a result, a more natural progression of energy releasing bond breaking events occurs during the reaction, beginning with dissociation of unstable "easy" bonds, followed by dissociation of the more stable "hard" bonds characteristic of various species of partially reacted fuel.

The foregoing advantages must be compared with and evaluated against prior art combustion processes, in which the quasi-instantaneous combustion of a normally rich, near stoichiometric homogeneous mixture of fuel and air resulted in virtually total utilization of available high energy oxygen to rapidly break the unstable bonds between fuel molecules to release energy at a high rate which resulted in complete depletion of oxygen required to react with the more stable intermediate fuel mole-

cular species. The prior art combustion reaction classically went from its initial, usually "rich" condition, to a richer and richer oxygen deprived condition, with resultant production of undesirable products of incomplete combustion under high pressure and temperature conditions which promoted dissociation of products of combustion and molecular species of nitrogen and carbon. Attempts in the prior art to provide a "sustained" supply of oxygen by various lean combustion techniques has required, on the part of engine designers, a precarious balance between stable, power producing combustion on one side, and quenching of combustion before completion on the other side because the proportion of remaining fuel to remaining oxygen normally proceeds to the lean side in such processes until the mixture is outside the flammability limits. High manufacturing costs and the danger of poor engine performance, of course, are endemic to lean burn engine configurations.

The process according to the present invention can best be understood in the context of specific engine configurations arranged to carry out the process. The exemplary engine embodiments to be considered are specifically configured as variations of existing reciprocating type internal combustion engines operating in Otto or Diesel modes, 2 or 4 stroke power cycles, with spark, glow or spontaneous ignition systems. The principles to be discussed are equally applicable to rotary piston engines or engines with reciprocating cylinders and "fixed" pistons, and it is not intended to imply that the process of the present invention can only be carried out using such "conventional" engine configurations.

In Figures 4 through 7, an exemplary combustion chamber geometry suitable for carrying out the process is illustrated. Typical possible fuel and air supply and distribution systems for such an engine are illustrated in Figures 8 through 10, while Figures 13 and 14 graphically depict timing of various intake, activation, reaction and exhaust events in the context of a piston driving a rotary output shaft over a four stroke Otto and Diesel cycle, respectively.

With reference to Figure 4, a working (combustion) chamber arrangement is illustrated as an example of one form of apparatus that could be used to carry out the process of the present invention. The working chamber 20 is a variable volume chamber defined by concave arcuate sidewall portions or areas 22 at the head end of a cylindrical bore 24, and a piston 26 reciprocally mounted in the bore 24 for varying the volume of the chamber 20 between a minimum value when the piston is at top dead center position and a maximum when the piston is at the bottom of its stroke (assuming here that the piston 26 is connected to the usual rotary crankshaft). The usual inlet and outlet valves 28, 30 for cyclically admitting fuel
and air and exhausting products of reaction into and out of the chamber 20 are shown, along with a spark plug or igniter 32. This type of engine is an air breathing engine, whereby the oxygen portion of the reactant is supplied by the air.

Fuel and air are cyclically supplied to working chamber 20, activated by compression, ignited and reacted to generate thermal potential which is converted to work via the piston 26, and the products of reaction are exhausted from the chamber. The spark plug initiates the reaction, which is essentially a dissociative, energy releasing chain reaction between some type of fuel molecules (usually hydrocarbon, hydrocarbon derivative, alcohol and/or ketone) and oxygen. The released heat increases the pressure of the gas in the working chamber 20 to drive the piston 26 to produce work in the usual manner. The energy releasing event can occur every other stroke of the piston or every fourth stroke of the piston to achieve a two cycle or four cycle operation, as is well understood. Ignition can be by spark, glow plug, or by spontaneous ignition. Fuel and air can be aspirated or injected under pressure, or a combination of aspiration and injection.

The head end of the bore 24 is at the top as seen in Figure 4, and it can be seen that the closed top end or working surface 34 of piston 26 lies above an upper compression seal 36 on the periphery of the piston 26, as shown. The central portion 38 of the top working surface of piston 26 is concave arcuate in form and cooperates with the concave arcuate end area 40 of the bore 24 to form a generally spherical working chamber 20 when the latter is at its minimum volume. At the outer peripheral area of the top of piston 26 a chamber 42 of a substantially fixed volume called a "sustaining" or "air" chamber is provided, this chamber being located in this instance between the top sealing means 36 and the extreme top end area 44 of piston 26.

A radial clearance gap 46 (see Figure 6) defines a minimum width of an open area or volume that provides continuously open passageway communication between chambers 20 and 42. The passageway gap 46 has a peripheral length so that a passageway partition area defined as the width of the gap at its minimum transverse dimension times the length of the gap is provided between the chambers. In this example, the gap extends entirely around the piston, but this need not always be the case. Likewise, the partition passageway length need to be continuous, nor does the gap width need to be constant.

As will be evident by comparing Figures 4, 6 and 7, the transverse gap 46 varies from a minimum when the working chamber 20 is at minimum volume (Figure 6) to a maximum (Figure 7) when the piston 26 is away from its top dead center position, due to the inward convergence of the sidewall area 22 of the head end of the cylinder bore 24. That is, when the piston 26 is at the top of its stroke as shown in Figure 6, the peripheral area of the top end of the piston more closely approaches the cylinder bore sidewall 22 than when the piston is away from its top dead center position as shown in Figure 7.

In addition, the solid boundary surfaces leading towards and extending away from the gap 46 are specifically contoured to favor rebound of molecular motion in a particular fashion. Specifically, the boundary surfaces are configured to favor to a maximum extent the rebound motion of molecules in such a manner that most of the molecules within the working chamber 20 tend to remain within the working chamber and only a relatively small quantity tend to diffuse into the sustaining chamber 42. Moreover, due to the curvature of the sidewalls of chamber 20, the highly energized molecules in chamber 20 striking its sidewalls tend to rebound ultimately back towards the central region of the chamber as a focal point or area. Thus, looking at Figure 6, the arcuate convex curvature of the top end surface 44 of piston 26, the concave curvature of surface 38, and the concave curvature of surfaces 22 and 40, all combine to favor rebound of molecular motion originating in chamber 20 back into the chamber 20 and to discourage migration of molecules into the sustaining chamber. It is desirable to avoid straight line-of-sight paths leading from chamber 20 into sustaining chamber 42 in order to favor maximum rebound motion and minimum diffusion of fuel molecules into the sustaining chamber 42. Preferably, rebound motion is directed towards the central region of chamber 20.

It will be seen that the gap 46 itself extends between a sharp lip 47 at the top peripheral area of the piston 26 and the sidewall of the bore. On the other side of the gap 42, that is within the sustaining chamber region 42, the interior sidewalls of the central region of the chamber are spherical in cross section as shown in Figure 6. The spherical contour here tends to cause the molecules moving within chamber 42 to rebound back towards the region 42, preferably its central region, and not to diffuse out through the partition gap 46 into the working chamber 20. Molecules moving from chamber 42 through the gap 46 towards chamber 20, tend to rebound towards chamber 20 and back into chamber 42. However, the shaded area shown in Figure 7 represents a volume 48 defined by the partition passageway area (gap width times gap length) projected over the straight line distance 50 extending between the gap 46 and the back wall 52 of the sustaining chamber 42. The distance 50 represents a depth dimension of the volume 48 that always extends normal to the gap dimension 46, whether the latter is at its minimum or maximum dimension (Figure 6 or Figure 7, respectively). The gap 46, of course, shall always be the minimum transverse dimension across
the open area between chambers 20 and 42, regardless of piston size, shape or position, or gap configuration. The volume 48 shall extend along a straight line-of-sight vector between the gap 46 and the back wall 50 in chamber 42, since this volume has a very important function in the present invention, as does the gap area 46.

It will also be seen in Figures 4—7 that the sustaining chamber 42 is located within the top peripheral area of the piston 26 and is defined in this embodiment by a chamber 42 within the piston 26 that is partially closed along one side by the cylinder sidewall, except for the gap 46. The chamber 42, of course, will always be above the top sealing means 36 so that molecular interaction across the gap 46 between chambers 20 and 42 can occur in a controlled manner.

The theory of the invention is based upon the sustained control of oxygen availability within chamber 20 during very critical time periods in the energy conversion cycle. Such time periods are at initiation of the reaction, during the period when the working chamber is at minimum volume, and during the expansion of the working chamber volume. The theory also requires that fuel and oxygen reactants be cyclically distributed within the engine working chamber in such a manner that the fuel proportion of each charge is entirely within chamber 20 upon completion of each compression stroke of the piston (in the micro domain, upon completion of the activation event) and a proportion of the oxygen bearing air be disposed in the sustaining chamber 42, where it becomes compressed and highly activated both by mechanical compression and the reaction process itself.

It should be mentioned at this time that during the power cycles there is no or very little macroscopic flow of gases between chambers 20 and 42, particularly during the reaction event. The molecular interchange that occurs across the gap is such that the pressures in the chambers 20 and 42 always tend to equalize throughout the cycle with minimum macro flow between the chambers.

The curvature of the surfaces defining the working chamber are such that highly activated fuel molecules, including species of partially reacted fuel molecules tend not to travel into the sustaining chamber through the gap 46. The probabilities of successful collision of those isolated fuel molecules migrating below the gap are highly diminished because with the large population of oxygen molecules in the partition volume 48 causes the proportion of fuel to oxygen to proceed towards the lean side to the extent that the rate of reaction is virtually zero. Thus, another key factor in the process is the absence of any substantial energy releasing reaction between fuel and oxygen in the sustaining chamber area 42.

The quantity of oxygen made available for participation in any reaction occurring within chamber 20 is, according to the invention, dependent primarily upon the size of the volume 48, since it is only along this straight line path that activated molecules of oxygen can readily migrate into the reaction chamber 20. Since the volume 48 can be controlled by controlling the gap width and length, as well as the height or depth 50, the oxygen participation in the working chamber 20 can be controlled in any desired manner once the proper distribution of fuel and air has been established at the start of the reaction corresponding to point 2 of the diagram in Figure 1.

Since the reaction charge in the working chamber 20 at the beginning of the reaction will be controlled by the fuel and air distribution system (to be explained below in connection with Figures 8—10) so that a charge that is on the "rich", or excess fuel side of stoichiometric proportions is contained within the working chamber when the reaction is initiated, it is very important for the process that additional oxygen availability in chamber 20 be strictly limited at the time of initiation of the reaction (i.e. ignition of the charge) corresponding roughly to the constant volume portion of the pressure-volume diagram in Figure 1 (from points 2 to 3). However, when the reaction proceeds, since initial oxygen is depleted in the reaction, additional oxygen must be supplied to maintain the proportion of fuel to oxygen near stoichiometric to insure maintenance of the reaction at the maximum expected rate.

Thus, the volume 48 is varied from a minimum value when the working chamber is at minimum volume as shown in Figure 6, to a maximum value when the chamber size is greater than minimum volume, as shown in Figure 7. Migration of molecules of oxygen from chamber 42 into working chamber 20 is therefore minimal as the working chamber approaches and is at minimum volume, but increases and sustains the required charge proportions as the reaction proceeds during the expansion of the working chamber.

It must be remembered that oxygen availability in the working chamber 20 does not result from a substantial macroscopic flow of air out of chamber 42 into chamber 20 during the reaction, as will be explained below. Rather, diffusion-like motion sustains the migration of molecules from chamber 42 into chamber 20 despite pressure equalization between the chambers, due to the fact that random energy distribution of the molecules with the chambers are different during the reaction. Since the temperatures are different while the pressures are the same, the population of molecules in the chamber 42 is greater than the population of molecules in chamber 20. Accordingly, movement of high energy oxygen molecules from the sustaining chamber 43 through the partition volume 48 and through the partition area or gap 48 into chamber 20 will occur at a rate that is
dependent upon the parameters of the partition volume 48, and will be favored in order to maintain pressure equilibrium.

The next matter considered to be important in the process is the quantitative amount of oxygen participation that can be permitted during each reaction. This requires, as a prerequisite, a consideration of the objectives sought in controlling oxygen availability in the working chamber during the reaction. Essentially, there is sufficient oxygen availability in the chamber 20 at initiation of the reaction to sustain the major part of the chain reaction between fuel and oxygen and to carry out a large portion of the heat releasing events of the cycle by converting chemical energy into thermal potential through the molecular dissociation of the fuel molecules. Yet, only the relatively instable bonds are broken during the initial part of the reaction, and this leaves the more stable bonds between fuel and fuel species to be dissociated. In prior art combustion processes, as has been stated, means were not available to supply oxygen molecules into the reaction zone with sufficient energy levels to cause further dissociation of the molecules having strongly affinitive bonds, nor was there a way available to control the rate at which such oxygen could be supplied on a sustained level to enable the reaction to proceed along its natural combustive course at a maximum rate without the usual restrictive constraints imposed by engine designs based upon virtually instantaneous combustion of a homogenous fuel mixture.

Accordingly, the process here requires that, first, the charge of reactants initially in the working chamber 20 be on the fuel "rich" side of stoichiometric, and, secondly, that the fuel proportion of reactants in chamber 20 be maintained on the rich side until the reaction has proceeded to useful completion (i.e., until the rate of heat release is insufficient to perform more useful work). The partition volume and partition area, therefore, must be configured and dimensioned so that for any working chamber, oxygen replenishment into the working chamber during the useful part of the reaction will sustain the proportions of the remaining reactants in the chamber 20 (including, oxygen, fuel, and partially reacted fuel species) on the rich side of stoichiometric for maintaining the proper maximum reaction rate.

For different required performances, fuels and geometries, the following limits, or ranges of values, will insure such control over oxygen availability in a combustion chamber configured generally in accordance with Figures 4—7 and 15—20 in the drawings. The ratio of the sustaining chamber volume to the working chamber volume should lie between .2 and 1.8 for most engines, since a higher ratio tends to promote excess availability of activated oxygen into the working chamber and the oxygen participation in the reaction is difficult to control. The gap width 46 at its smallest dimension should not be much less than .050 in. (1.27 mm), because a smaller gap is not conducive to the desired interchange between working and sustaining chambers during the cycle. For related reasons, the gap 46 should never be much greater than .20 in. (5.08 mm) to prevent substantial mass flow of air from the sustaining chamber during the expansion phase of the reaction and to prevent fuel contamination in the sustaining chamber during the reaction.

The maximum total partition area along the gap 46 should be between approximately .05 and .15 times the square of the diameter of the working chamber or cylindrical bore to permit one to limit total oxygen availability in the working chamber. The partition volume 48 should be approximately between .10 and .35 times the total sustaining chamber volume when the partition gap 46 is at its maximum opening to achieve proper diffusion control over the activated oxygen.

There remains to be explained, for a fuller understanding of the process, how the fuel and air constituents are distributed in the working and sustaining chambers to achieve the separation of a portion of air of each charge between the working and sustaining chambers, with minimum contamination of fuel molecules in the sustaining chamber, and while assuring the presence of a desired fuel and air mixture in the reaction chamber. The distribution of the fuel and air must occur during the compression activation event and be maintained to a large extent during the reaction itself. The working and sustaining chamber geometries, the fuel and air supply systems, and appropriate controls over the timed relationship between the supply of fuel and air during each cycle must all function in a harmonious manner to achieve the desired distribution of reactants in the working and sustaining chambers by the time the reaction is initiated during each power cycle.

First considering broad approaches to the supply system for fuel and air, and more particularly, the system for cyclically forming fuel and air charges, Figures 8 through 10 schematically illustrate how typical systems can be arranged to achieve the desired fuel distribution in various engine configurations. In the embodiment shown in Figure 8, an engine 60 having a working chamber generally constructed in accordance with Figure 4 is illustrated, including a spark ignition plug 62, inlet and exhaust valves 64, 66, a source of fuel 68, and an air line 70. Fuel and air in this embodiment are independently aspirated into the working chamber during each intake event and, after activation, the fuel and air charge is ignited to initiate the combustive reaction between the fuel and oxygen reactants to drive the piston 72 to produce work. The fuel and air supply are independently regulated by a suitable control system schematically shown at 74. A central control unit 74 associated with a throttle or
power regulator for the engine adjusts the position of an air valve 76 so that the valve remains essentially open except in the idle regime. Air is thus aspirated into the working chamber without additional constraints in response to power demand of the engine. Fuel flow, on the other hand, is controlled by a valve 78 so that the amount of fuel aspirated into the working chamber through the intake valve 64 varies as a function of the power demand of the engine. Fuel flow could be induced by means of a conventional venturi 80 or by other suitable means.

The important aspect of the fuel and air supply control system is that the relationship of total fuel and air proportions in each charge only varies from stoichiometric at full power demand to an excess of air at less than maximum power operating conditions. That is to say, except at full power, when the charge proportions approach stoichiometric, the charge proportions will always be on the "lean" or excess air side of stoichiometric and never on the rich side, with only the fuel proportion being varied in accordance with power demand.

Other fuel supply and control systems could be based upon pressurized injection of fuel into the intake manifold, as shown in Figure 9. In Figure 9, an intake pipe or manifold 84 includes an idle control air valve 86 regulated by controller 88, a fuel injector nozzle 90 and a timed injection controller 92 connected to a fuel supply. A plug 94 could be spark or glow type and the engine 98 includes the usual intake and exhaust valves 98, 100, respectively, and piston 102. Fuel would be supplied under moderate pressure in timed relation to each intake event during the intake stroke of the piston 102. Only the fuel quantity would be controlled in response to the power demand of the engine, and the proportion of fuel to air in each charge would always be in accordance with the principles set forth in connection with the discussion relating to Figure 8.

In Figure 10, still another embodiment of the fuel supply is illustrated, including a pressurized fuel injector nozzle 106 used to inject fuel directly into the working chamber of the engine 108. Air is aspirated normally through air inlet valve 110, and the fuel is supplied to the nozzle 106 under pressure through an injector system 112 of the general type well known in the art. The fuel injector system supplies a controlled quantity of liquid fuel into the reaction chamber in timed relationship with the piston strokes to achieve the desired fuel and air distribution in the working chamber. A glow plug 114 may be utilized for starting the engine, but as a general rule the reaction will be initiated in this embodiment by compression ignition. The fuel to air ratio of the charge in this embodiment varies as that for engines described in Figures 8 and 9.

It will thus become apparent that the timed nature of the fuel supply is of great importance in the process of this invention, as is the charge control. For a better understanding of the timed or phased relationship between fuel admission into the working chamber and the engine operation, again it is important to understand that the objective that is sought is twofold: First, to distribute the fuel and air in the working and sustaining chambers so that the air in the sustaining chamber is completely separate from the charge in the working chamber; and second, to distribute the fuel and air so that the proportions of the reactant charge in the working chamber is always on the fuel rich side of stoichiometric at initiation of the reaction (i.e., ignition). The rich proportions of the charge in the working chamber favor an optimum rate of reaction for the available fuel, considering the chemical composition of the fuel, the particular engine configuration and the working chamber geometry. To achieve such a distribution, the fuel to air proportions of the supplied charge are varied as explained previously, but, just as importantly, the fuel must be admitted into the working chamber in a specific timed relationship with the variation of volume of the working chamber during the intake and compression events of the cycle, such timed relationship being selected from a broad range compatible with the process.

In the Figures 11 and 12, generally conventional timing of fuel and air supply to a combustion chamber of an internal combustion engine is graphically depicted by a standard circular graph or chart comprising multiple concentric circles that illustrate the cycle beginning with the innermost circular segment and ending with the outermost segment. The angular coordinates of the chart coincide with the angular positions of the power output shaft of an engine capable of carrying out the cycle illustrated in the chart. Thus, in Figure 11, intake of fuel and air in a conventional Otto cycle engine occurs as an aspiration of a homogenous mixture of fuel and air over the period of shaft rotation shown on the innermost circular segment 116, that is, from about 0° up to about 190° of shaft rotation during an intake stroke, depending upon the timing of intake valve closure. Compression then occurs up to a few degrees before piston top dead center (0°) as shown by segment 118, at which point ignition occurs followed by quasi-instantaneous combustion and then expansion over the rotation period covered by the next outer segment 120. The exhaust phase is shown at segment 122. The cycle of Figure 11 is a four stroke cycle and it can readily be seen how the fuel and air mixture according to this cycle must be completely placed within the combustion chamber as a readily and quickly combustible mixture during each intake event, and that very little control is available over the supply of the fuel component other than as part of the total aspirated mixture charge. It is well known, of course, that timed fuel injectors can be used to more carefully control the mixture in the combustion chamber.
In accordance with engine power demands, but such systems are limited to optimizing the homogeneity of the mixture in the chamber at the moment of ignition to achieve a rapid, virtually instantaneous, knock-free combustion of the fuel. This has the disadvantage that complete vaporization of fuel is difficult to achieve in the short time available during that portion of the intake event that the fuel is being injected.

In Figure 12, an exemplary diesel cycle is illustrated, showing how air and fuel can be separately injected into the combustion chamber to achieve the standard or high speed diesel cycle. However, it is well known that high compression ratios are required to provide sufficient activation of the fuel and air, and that elaborate precautions must be taken to ensure ignition. Separation of the fuel before and during combustion. Diesel engines traditionally require careful regulation and control over the starting of the supply of injected fuel to limit the amount of reaction before top dead center in order to control maximum peak pressure in the combustion chamber to avoid engine damage. On the other hand, because the charge always includes an excess of air, as the combustion progresses the proportion of oxygen with respect to unreacted fuel proceeds towards a leaner and leaner condition until, when the charge is outside the flammability limits, quenching of combustion occurs. Also, species of fuel produced by the fracturing of the fuel drops during their surface combustion cannot totally react, and characteristic smoke is emitted in the Diesel exhaust.

In Figures 13 and 14, a typical cycle according to the present invention is depicted to show the range of timing of fuel admission that is compatible with carrying out the cycle in view of the required distribution of fuel and air in the working chamber and the required control of oxygen availability during the reaction, as explained previously. A four stroke power cycle is depicted in Figure 13 and a spark or glow plug ignition of the reactants is assumed, rather than compression ignition. Intake of air by aspiration occurs over the period of output shaft rotation corresponding to the innermost segment 126. However, fuel is admitted (segment 128) anytime during the intake and compression events between 30° to 50° after beginning of the intake and up to 30° to 40° before ignition. Supply of fuel under moderate pressure during the central period of the intake and compression events, that is from 140° after beginning the intake and up to 120° before ignition, is considered to be advantageous in some cycles, depending upon working chamber geometry and engine configuration. Activation of the reactants by compression is denoted by segment 130 and can overlap the fuel supply period, as shown. Ignition, followed by sustained reaction then follows over segment 132, followed finally by the exhaust, depicted by segment 134. Significantly, due to the distribution of fuel and air in the working chamber and the charge of reactants that is available, the peak pressure in the working chamber is controllable because of the sustained nature of the reaction that occurs in accordance with the process of this invention.

In Figure 14, a process according to the invention is illustrated where fuel is injected into the working chamber under pressure, and ignition occurs by spontaneous initiation of the reaction by activation alone. The cycle differs essentially from that shown in Figure 13 in that the supply of fuel, shown by segment 136, should be started at any point about midway between intake and compression, and should terminate not later than 35° to 40° before the ignition point, the latter being necessary to provide sufficient time for the required activation of the molecules of the injected fuel. The distribution of the fuel and oxygen reactants in the working and sustaining chambers, and due to the sustained reaction achieved by the process, satisfactory work producing cycles using compression ignition can be achieved with compression ratios between 5:1 and 12:1, with the required control of peak pressures, without restrictions over composition of fuel and critical timing of the injection of the fuel. The compression ratio, of course, means the ratio of the sum of the maximum working and sustaining chamber volumes to the sum of the minimum working and sustaining chamber volumes.

The process of the invention enables one to carry out a chemical to thermal energy conversion in a manner that produces substantially lower proportions of noxious products of partial reaction than conventional Otto and Diesel cycles. The sustained nature of the reaction is carried out in accordance with the invention and results in the progress of the reaction towards more complete conversion of the fuel and oxygen reactants into stable final products of reaction with lower peak cylinder pressures than known combustion processes.

Comparing exhaust products of Otto, Diesel and the cycle of this invention using present day evaluation techniques, and assuming basically orthodox piston engine layouts used to carry out each cycle, the proportions of oxygen (O₂), carbon dioxide (CO₂), carbon monoxide (CO) and partially or unreacted hydrocarbon (UHC) contained in the exhaust stream of each engine can be used as an indication of the ability of each cycle to convert its respective fuel charge into thermal potential assuming that each engine is running in its natural mode without emission controls.

At full power conditions, an Otto cycle, with its virtually instantaneous combustion of a homogeneous fuel mixture, will naturally tend to produce 6 to 11% CO in the exhaust stream, 1000 to 5000 parts per million (ppm) UHC, and no O₂ will be present, since it has been depleted early
in the combustion cycle. The exhaust of a conventional Diesel engine, also at full power demand, will normally contain .5 to .8% CO, .5 to 1% O₂ and smoke, consisting of carbon and UHC in various proportions.

At cruise power conditions, the exhaust of an Otto cycle engine will normally contain .5 to 1% CO, 200 to 1000 ppm UHC, and .5 to .8% O₂, and the exhaust of a Diesel engine will contain .3 to .5% CO; 500 to 1500 ppm UHC and 2 to 4% O₂.

At idle, the exhaust of an Otto engine will contain roughly 6 to 9% CO; 100 to 3000 ppm UHC, and no O₂, while a Diesel exhaust at idle normally will include .2 to .5% CO, 5 to 8% O₂, and various amounts of carbon and UHC displayed in smoke.

The process of this invention is carried out to produce, in the exhaust of a reciprocating piston engine carrying out the process and operating at full power conditions, a maximum of .2 to 3% CO, 100 to 1800 ppm UHC, and 0 to .2% O₂. At cruise power conditions, the reaction will produce a maximum of .1 to 1% CO, 50 to 1500 ppm UHC and .2 to 3.0% O₂ in the exhaust. At idle, the exhaust will include a maximum of .2 to 1.0% CO, 100 to 1000 ppm UHC and 2 to 4% O₂.

It is to be noted that the supply and distribution of reactants in the engine carrying out the process, and the controlled participation of the excess oxygen during the reaction, results in a supply of excess activated oxygen adjacent the reaction region when the exhaust valve of the working chamber opens. At this moment, the pressure in the working chamber rapidly drops as the reaction products expand through the valve. The excess activated oxygen at this moment flows out of the sustaining chamber and expands into the working chamber to extend the continuation of the reaction in the working chamber and in the exhaust stream. Thus, the proportion of unreacted products in the exhaust stream is diminished by this process as compared with standard internal combustion processes.

One embodiment of suitable engine apparatus for carrying out the process of the invention has been described in Figure 4 and the related Figures 5—7, Figures 15—23 show alternate engine arrangements contemplated for carrying out the process.

In Figure 15, the working chamber is shown to include a cylinder bore 140 having a head end 142 including a concave arcuate surface area 144. A piston 146 having a closed top end 148 having a concave arcuate surface 149 reciprocates within bore 140 to vary the volume of the working chamber 150 between the closed top end 148 of the piston and the head end of the bore. Intake and exhaust valves 152, 154 are included, as is a spark plug igniter 156. The valves and igniter could be differently arranged, of course, if it is intended to operate the engine in a two cycle mode using a glow plug, for example.

The sustaining chamber 158 is located adjacent and below the surface of the head and of the cylinder bore in this embodiment, and the partition gap 160 (see Figures 16 and 17) is located virtually in the same plane as or closely adjacent to the piston extreme top end portion 162 when the working chamber 150 is at minimum volume. The extreme top end portion 162 of the piston 146, accordingly, partially obstructs the gap 160 and effectively reduces its size as shown in Figure 16 when the working chamber is at its minimum volume. However, when the piston moves away from the minimum volume position (Figure 17) the full gap area 160 is open to furnish a communicating area between chambers 150 and 158. Thus, the partition volume 164 varies from a minimum value when the working chamber is at a minimum volume to a maximum value when the piston moves away from its top dead center position. The partition volume 164 shown in Figure 17 is shaded to show how it is enlarged compared to the partition volume when the piston is at its top dead center position.

It will be observed that the surfaces adjacent the gap 160 and defining the chambers 150 and 158 are all contoured to favor rebound of molecular motion in the manner described in connection with the engine configuration shown in Figure 4, including the area adjacent the top end of the piston 146.

In the engine embodiment of Figures 18—20, two closed-ended opposed pistons 170, 172 are reciprocally mounted in a single bore 174 having reactant inlet/outlet ports 176. The working chamber 178, at minimum volume, is defined essentially by the volume between the concave arcuate end wall areas 180, 182 of the pistons 170, 172. The sustaining chamber volume 184 is divided between two volumes located in the outer peripheral area of pistons 170, 172 between the outer extreme end wall areas or top edges 186 of the pistons 170, 172 and the top sealing rings 188 of each piston. In this embodiment, the minimum gap 190 as best seen in the detail view of Figure 19, extends between the working and sustaining chamber volumes between the top end areas 186 of pistons 170, 172 and the maximum partition gap 192 is defined as the radial gap between the top end areas of the pistons and the cylinder sidewall, as shown in Figure 20. The respective maximum and minimum partition volumes are illustrated by the shaded areas in Figures 19 and 20. Thus, the steps of compressing and distributing reactants in the working and sustaining chambers includes reciprocating the pistons 172 towards and away from each other so that the gap between their extreme end wall areas varies between a minimum and maximum value.

In the engine embodiment of Figures 21 through 23, opposing pistons 196 having a
working chamber 198 between their closed top ends are reciprocally mounted in cylindrical bore 200. The extreme top end areas 202 of pistons 196 approach and recede from each other as pistons 196 reciprocate, with the distance 206 representing the minimum clearance between the piston extreme top ends. The sustaining chamber 204 in this embodiment is disposed in the cylinder sidewall and communicates with the working chamber through an aperture 208. The minimum partition gap in this embodiment is the clearance 206 between the piston extreme top end areas 202, and the maximum partition gap is represented by the width of the aperture 208 as shown in Figure 23.

In the embodiment of engine configurations shown in Figures 4—7 and 15—23, various preferred approaches are illustrated by way of examples as to the manner in which the partition volume can be varied during the reaction to achieve control over the availability of activated oxygen within the respective reaction chambers. These are not intended to represent the only ways in which the partition volume can be controlled in various engine configurations, but are intended to illustrate here the range of choices available to designers of current, state-of-the-art engines as to how existing engine systems can be readily modified to carry out the process of the present invention. Clearly, the choice of mechanical expedience is as broad as the choice of engine designs themselves.

Claims

1. A combustion type energy conversion process for use in a work producing moveable piston internal combustion engine wherein charges of fuel and air are cyclically supplied to and combusted in a combustion chamber (20) (150) (174) (198) and wherein a portion of the air part of each charge is disposed in a combustion sustaining air chamber (42) (158) (184) (204) of substantially fixed volume connected with and closely adjacent the combustion chamber at the beginning of each expansion cycle, the connecting passageway between the air and combustion chambers being delimited by a gap including an open width (46) (160) (190, 192) (206, 208) and length facing the combustion chamber and a depth (50) directly rearward of the open area and extending to the bottom (52) of the air chamber, characterized by: formulating the charges in response to power demand of the engine in a manner such that the total air to fuel ratio thereof is stoichiometric at full power demand and on the lean side of stoichiometric at all other times; and causing the combustion at each charge to occur at least above idle conditions with the fuel to air proportion in the combustion chamber on the rich side of stoichiometric throughout the combustion event by controlling the amount of oxygen molecules supplied to the combustion area from the air chamber through the connecting passageway, such control over the quantity of oxygen molecules being achieved solely by the geometry of the connecting passageway, and including the step of varying the size of the opening of the passageway from a smaller to a larger size between initiation and completion of each combustion event.

2. The process according to claim 1, including the step of varying the size of the opening of the passageway by using the position of the piston or pistons (26) (146) (170, 172) (196) to selectively restrict and open the passageway.

3. The process according to claim 1, characterized by aspirating the entire fuel quantity into the combustion chamber during each charge intake event and varying the fuel to air proportion by varying the quantity of the fuel in each charge without varying the total quantity of air; and furthermore supplying fuel to the combustion chamber beginning not earlier than 30 to 50° with respect of output shaft rotation after start of each respective charge intake event.

4. The process according to claim 1, characterized by injecting at least part of the fuel of each charge directly into the combustion chamber during the compression of the charge, but not later than 30 to 40° before initiation of the combustion, and varying the fuel to air proportion by varying the quantity of fuel in each charge without varying the total quantity of air.

5. The process according to claim 1, characterized by injecting the entire fuel quantity of each charge under pressure into the combustion chamber during the central period of the intake and compression events; and varying the fuel to air proportion by varying the quantity of fuel in each charge without varying the total quantity of air.

6. A work producing engine for carrying out the process claimed in any of the previous claims, such engine including a moveable piston or pistons (26) (146) (170, 172) (196) for compressing each charge and reacting to the expansion forces resulting from the combustion of same, a variable volume combustion chamber (20) (150) (174) (198) at least in part defined by the moveable piston or pistons; a fuel and air charge supplying and distribution means (74) (92, 88) (112), an air chamber (42) (158) (184) (204) located adjacent to and in communication with the combustion chamber through a passageway delimited by a gap including an open width (46) (160) (190, 192) (206, 208) and length facing the combustion chamber and a depth (50) directly rearward of the open area and extending to the bottom (52) of the air chamber, characterized by: said open area of said passageway being disposed relative to said piston or pistons in a manner whereby movement of the piston causes the size of said open area of said passageway to vary between a minimum open area when the combustion chamber volume is at minimum and a larger open area when the
combustion chamber is at greater than minimum; said fuel and air charge supply and distribution means is arranged to cause distribution of all the fuel portion of each charge to the combustion chamber and a part of the air portion only of each charge to the air chamber for each operative cycle of the engine; and a fuel to air proportion control system for varying the ratio of fuel to air of the charges supplied to the combustion chamber.

7. The engine according to claim 6, characterized in that the solid surfaces delimiting the combustion chamber, air chamber and passageway are geometrically configured to avoid straight line of sight paths between the principal combustion and air chamber volumes and to provide such paths between the combustion chamber and the passageway volume.

8. The engine according to claim 6 or 7, characterized in that the ratio of the air chamber volume to the minimum combustion chamber volume is between .2 and 1.8 and the width of the passageway when it is at its minimum area is not less than .050 inch (1.27 mm) and at its maximum area not greater than .20 inch (5.08 mm).

9. The engine according to claim 6 or 7, characterized in that the piston or pistons reciprocate within a cylindrical working chamber and the maximum passageway open area is between .05 and .15 times the square of the diameter of the combustion chamber.

10. The engine according to claim 9, characterized in that the passageway volume (gap area times depth) is between .10 and .35 times the total air chamber volume when the passageway area is at its maximum size.

11. The engine according to any one of claims 6 through 10 wherein said moveable piston or pistons is a single piston (26) reciprocally mounted within a cylindrical bore (24) with the combustion chamber (20) disposed between a closed head end (40) of the bore and a closed top end (38) of the piston, the piston includes an upper compression seal (36) and a generally concave arcuate top end (38), characterized in that the bore head end includes a centrally converging side wall area (22) and a generally concave arcuate end area (40), the air chamber (42) is disposed in the peripheral area of the piston below its top end between said top end and said compression seal; a radial clearance gap (46) is provided between the piston top end and the cylinder wall, said gap defining said passageway, and said passageway is varied in size as said piston approaches and recedes from said converging side wall area.

12. The engine according to claim 11, characterized in that the peripheral area (44, 47) of the piston end adjacent the passageway is shaped so that, with the adjacent cylinder bore sidewall (24) the passageway converges in a direction away from the combustion chamber, and the surfaces (44) on the piston leading to the passageway from the combustion chamber are convexly curved.

13. The engine according to claim 6, wherein said piston or pistons is a single piston (146) reciprocally mounted within a cylindrical bore (140), the combustion chamber (150) is located between a closed head end (144) of the bore and a closed top end (149) of the piston, the bore head end includes an inwardly converging sidewall area (144) and a convex arcuate end area, the piston includes an upper compression seal and a concave arcuate top end area (149) characterized in that the air chamber (158) is disposed in the peripheral area of the bore head end; the length of the passageway area extends along the periphery of the bore head end; a peripheral area (162) of the piston top end lies adjacent the passageway area in partially blocking relationship when the combustion chamber is at minimum volume and is clear of the passageway when the combustion chamber is larger than minimum volume.

14. The engine according to claim 6, wherein said piston or pistons are a pair of closed-ended pistons (170, 172) with top compression sealing means (188) and reciprocally mounted within a single cylindrical bore (174), the pistons, together with their closed top end and the bore, defining said combustion chamber (174), characterized in that the piston top ends each have concave arcuate central portions (180, 182) and respective peripheral top edge portions (186) that are radially spaced from the bore wall to define a maximum passageway width (192); said air chamber (184) lies in the peripheral area of the top end of each piston between said peripheral edge portion and said compression sealing means of each piston and the distance between said piston peripheral top edge portions when the combustion chamber is at minimum volume defines a minimum passageway width (190).

Revendications

1. Procédé de conversion d'énergie du type à combustion pour utilisation dans un moteur à combustion interne à piston mobile produisant du travail, selon lequel des charges de carburant et d'air sont délivrées cycliquement à une chambre de combustion (20, 150, 174, 198) et brûlées dans cette chambre et dans lequel une partie de la portion d'air de chaque charge se trouve dans une chambre d'air d'entretien de combustion (42, 158, 184, 204) de volume substantiellement fixe reliée avec la chambre de combustion et immédiatement adjacente à cette dernière au début de chaque cycle d'expansion, le passage de liaison entre les chambres d'air et de combustion étant délimité par un jeu comprenant une largeur (46; 160; 190, 192; 206, 208) et une longueur ouverte en face de la chambre de combustion et une profondeur (50) située directement à l'arrière de la zone ouverte et s'étendant jus-
qu’au fond de la chambre d’air, caractérisé par : formation de charges en réponse à la demande de puissance du moteur de manière que le rapport d’air total au carburant de ces charges est stochiométrique en cas de demande de pleine puissance et plus pauvre que stochiométrique dans tous les autres cas ; et en déterminant pour chaque charge, au moins en dessous des conditions de marche à vide, que la combustion se produise avec une proportion du carburant à l’air dans la chambre de combustion stochiométriquement riche pendant toute la phase de combustion par le contrôle de la quantité des molécules d’oxygène délivrées dans la zone de combustion par la chambre d’air à travers le passage de liaison, un tel contrôle de la quantité des molécules d’oxygène étant réalisé uniquement par la géométrie du passage de liaison, et comprenant l’étape de variation de la grandeur de l’ouverture du passage d’une valeur faible à une valeur forte entre le début et la fin de la phase de combustion.

2. Le procédé selon la revendication 1, comprenant l’étape de variation de la grandeur de l’ouverture du passage en utilisant la position du ou des pistons (26 ; 146 ; 170, 172 ; 196) pour restreindre et élargir sélectivement le passage.

3. Le procédé selon la revendication 1, caractérisé par une aspiration de la quantité totale de carburant dans la chambre de combustion pendant chaque phase d’admission de charge et par une variation de la proportion de carburant à l’air en variant la quantité de carburant dans chaque charge sans changer la quantité totale d’air ; et en outre par le fait que la délivrance de carburant à la chambre de combustion de commence pas plus tôt que 30 à 50° par rapport à la rotation de l’arbre de sortie après le début de chaque phase respective d’admission de charge.

4. Le procédé selon la revendication 1, caractérisé par l’injection d’au moins une partie du carburant de chaque charge directement dans la chambre de combustion pendant la pression de la charge mais pas plus tard que 30 à 40° avant le début de la combustion et en variant la proportion de carburant à l’air par variation de la quantité de carburant dans chaque charge sans changer la quantité totale d’air.

5. Le procédé selon la revendication 1, caractérisé par l’injection sous pression de la quantité totale de carburant de chaque charge dans la chambre de combustion pendant la période centrale des phases d’admission et de compression ; et par la variation de la proportion de carburant à l’air en variant la quantité de carburant dans chaque charge sans changer la quantité totale d’air.

6. Un moteur producteur de travail pour la mise en œuvre du procédé revendiqué dans n’importe laquelle des revendications précédentes, ce moteur comprenant un ou des pistons mobiles (26 ; 146 ; 170, 172 ; 196) pour comprimer chaque charge et réagissant aux forces d’expansion résultant de la combustion desdites charges, une chambre de combustion à volume variable (20, 150, 174, 198) définie au moins en partie par le ou les pistons mobiles ; des moyens pour délivrer et distribuer une charge de carburant et d’air (74 ; 92, 98 ; 112), une chambre d’air (42, 158, 184, 204) adjacente à la chambre de combustion et en communication avec cette chambre par un passage délimité par un jeu comprenant une largeur (48 ; 160 ; 190, 192 ; 206, 209) et une longueur ouverte en face de la chambre de combustion et une profondeur (50) directement à l’arrière de la zone ouverte s’étendant jusqu’au fond (52) de la chambre d’air, caractérisé en ce que la zone ouverte dudit passage est disposée relativement au ou aux pistons de manière que le mouvement du piston produise une variation de la zone ouverte entre une largeur lorsque le volume de la chambre de combustion est minimum et une grandeur supérieure lorsque la chambre de combustion est plus grande que le minimum ; en ce que les moyens pour délivrer et distribuer la charge de carburant et d’air sont disposés de manière à produire la distribution de toute la portion de carburant de chaque charge à la chambre de combustion et une partie seulement de la portion d’air de chaque charge à la chambre d’air pour chaque cycle de fonctionnement du moteur ; et par un système de contrôle de la proportion de carburant à l’air pour varier le rapport du carburant à l’air des charges délivrées à la chambre de combustion.

7. Le moteur selon la revendication 6, caractérisé en ce que les surfaces solides délimitant la chambre de combustion, la chambre d’air et le passage ont une configuration géométrique évitant les chemins rectilignes des lignes de visée entre les volumes de la chambre de combustion principale et de la chambre d’air et produisant de tels chemins entre la chambre de combustion et le volume de passage.

8. Le moteur selon la revendication 6 ou 7, caractérisé en ce que le rapport du volume de la chambre d’air au volume minimum de la chambre de combustion est entre 0,2 et 1,8 et en ce que la largeur du passage dans sa surface minimum n’est pas inférieure à 0,050 pouce (1,27 mm) et pas supérieure à 0,50 pouce (5,08 mm).

9. Le moteur selon la revendication 6 ou 7, caractérisé en ce que le ou les pistons ont un mouvement alternatif dans une chambre de travail cylindrique et en ce que la surface ouverte maximum du passage est entre 0,05 et 0,15 fois le carré du diamètre de la chambre de combustion.

10. Le moteur selon la revendication 9, caractérisé en ce que le volume du passage (surface du jeu fois profondeur) est entre 0,10 et 0,35 fois le volume total de la chambre d’air lorsque la surface du passage est de grandeur maximum.
11. Le moteur selon l’une quelconque des revendications 6 à 10, dans lequel le ou les pistons mobiles est un seul piston (26) monté en interaction dans une ouverture cylindrique (24) avec la chambre de combustion (20) disposée entre une extrémité frontale fermée (40) de l’ouverture et une extrémité supérieure fermée (38) du piston, le piston comprenant un joint de compression supérieur (38) et une extrémité supérieure courbée de forme généralement concave (38), caractérisé en ce que l’extrémité frontale de l’ouverture comprend une zone de paroi latérale convergente vers le centre (22) et une zone d’extrémité courbée de forme généralement concave (40), en ce que la chambre d’air (42) est disposée dans la zone périphérique du piston en dessous de zone extrémité supérieure entre ladite extrémité supérieure et ledit joint de compression; en ce qu’un jeu radial (46) est prévu entre l’extrémité supérieure du piston et la paroi du cylindre, ledit jeu définissant ledit passage, et en ce que ledit passage varie en grandeur lorsque le piston s’approche et s’éloigne de ladite zone de paroi latérale convergente.

12. Le moteur selon la revendication 11, caractérisé en ce que la zone périphérique (44, 47) de l’extrémité du piston adjacente au passage est formée de manière qu’avec la paroi latérale adjacente de l’ouverture du cylindre (24), le passage converge dans une direction s’éloignant de la chambre de combustion et en ce que les surfaces (44) du piston conduisant au passage depuis la chambre de conduction sont courbées en forme convexe.

13. Le moteur selon la revendication 6, dans lequel le ou les pistons est un seul piston (146) monté en interaction dans une ouverture cylindrique (140), la chambre de combustion (150) est située entre une extrémité frontale fermée (144) de l’ouverture et une extrémité supérieure fermée (149) du piston, l’extrémité frontale de l’ouverture comprend une zone de paroi latérale convergente vers l’intérieur (144) et une zone d’extrémité courbée de forme convexe, le piston comprenant un joint de compression supérieur et une zone d’extrémité supérieure courbée de forme concave (149), caractérisé en ce que la chambre d’air (158) est disposée dans la zone périphérique de l’extrémité frontale de l’ouverture, en ce que la longueur de la zone de passage s’étend le long de la périphérie de l’extrémité frontale de l’ouverture, en ce qu’une zone périphérique (162) de l’extrémité supérieure du piston est adjacente et bloque partiellement la zone de passage lorsque la chambre de combustion est à son volume minimum et dégage le passage lorsque le volume de la chambre de combustion excède le volume minimum.

14. Le moteur selon la revendication 6, dans lequel le ou les pistons sont une paire de pistons à extrémités fermées (170, 172) avec des moyens de joint de compression supérieurs (188) et montés en interaction dans une seule ouverture cylindrique (174), les pistons définissant ensemble avec leur extrémité supérieure fermé et l’ouverture ladite chambre de combustion (174), caractérisé en ce que les extrémités supérieures des pistons ont chacune une partie centrale courbée de forme concave (180, 182) et des parties de bords supérieurs périphériques respectives (186) qui sont espacées radialement de la paroi de l’ouverture pour définir une largeur maximum de passage (192); en ce que ladite chambre d’air (184) se trouve dans la zone périphérique de l’extrémité supérieure de chaque piston entre la partie des bords périphériques et les moyens de joint de compression de chaque piston et en ce que la distance entre les parties des bords périphériques supérieurs des pistons définit un passage de largeur minimum (190) lorsque la chambre de combustion est à son volume minimum.

**Patentansprüche**

Oeffnung des Durchgangs zwischen Beginn und Beendigung jedes Verbrennungsvorgangs von einer kleineren Abmessung zu einer grosseren zu verändern.


4. Verfahren nach Anspruch 1, gekennzeichnet durch das Einspritzen mindestens eines Teiles des Brennstoffes einer Charge direkt in die Brennkammer während der Kompression der Charge aber nicht später als 30 bis 40° vor Beginn der Verbrennung, und durch Aenderung des Brennstoff- zu Luftverhältnisses durch Verändern der Brennstoffmenge jeder Charge ohne die gesamte Luftmenge zu verändern.

5. Verfahren nach Anspruch 1, gekennzeichnet durch Einspritzen der ganzen Brennstoffmenge jeder Charge unter Druck in die Brennkammer während des Mittelleits des Einlass- und Kompressionsvorgangs; und durch Aenderung des Brennstoff- zu Luftverhältnisses durch Aenderung der Brennstoffmenge jeder Charge ohne die gesamte Luftmenge zu verändern.


8. Maschine nach Anspruch 6 oder 7, dadurch gekennzeichnet, dass das Verhältnis von Luftkammervolumen zu minimalen Brennkammervolumen zwischen 0,2 und 1,8 liegt; und dass die Weite des Durchgangs bei seiner minimalen Fläche nicht unter 0,050 Zoll (1,27 mm) und bei seiner maximalen Fläche nicht über 0,20 Zoll (5,08 mm) liegt.


10. Maschine nach Anspruch 9, dadurch gekennzeichnet, dass das Volumen des Durchgangs (Spaltfläche mal Tiefe) zwischen dem 0,10- und dem 0,35-fachen des gesamten Volumens der Luftkammer liegt, wenn die Fläche des Durchgangs ihre minimale Abmessung aufweist.

11. Maschine nach irgendeinem der Ansprüche 6 bis 10, worin der oder die genannten beweglichen Kolben ein einziger Kolben (26) ist der in- und herbeweglich in einer zylindrischen Bohrung (24) angeordnet ist, wobei die Brennkammer (20) zwischen einem geschlossenen Kopfende (40) der Bohrung und dem geschlossenen oberen Ende (38) des Kolbens gebildet ist, und wobei der Kolben eine obere Kompressionsdichung (36) und ein im Ganzen konvex gewölbtes oberes Ende (38) aufweist, dadurch gekennzeichnet, dass das Kopfende der Bohrung eine zur Mitte konvergierende Seitenwand (22) und eine allgemein konvex gewölbte Einfäche (40) aufweist, dass die Luftkammer (42) im Umfangsbereich des Kolbens unterhalb seines oberen Endes zwischen diesem oberen Ende und der.
erwähnten Kompressionsdichtung angeordnet ist; und dass ein radialer Spalt (46) zwischen dem oberen Kolbende und der Zylinderwand gebildet ist, welcher Spalt den erwähnten Durchgang bildet, und dass der Durchgang seine Abmessung ändert, wenn der Kolben sich der konvergierenden Seitenwand nähert oder sich von derselben entfernt.

12. Maschine nach Anspruch 11, dadurch gekennzeichnet, dass die Randfläche (44, 47) des Kolbendes beim Durchgang so geformt ist, dass mit der anliegenden Seitenwand (24) der Zylinderbohrung der Durchgang in Richtung von der Brennkammer weg konvergiert, und dass die von der Brennkammer zum Durchgang führenden Oberflächen (44) konvex gewölbt sind.

13. Maschine nach Anspruch 6, worin der oder die genannten Kolben ein einziger Kolben (146) ist, der hin- und herbeweglich in einer zylindrischen Bohrung (140) angordnet ist, die Brennkammer (150) zwischen einem geschlossenen Kopfende (144) der Bohrung und einem geschlossenen oberen Ende (149) des Kolbens liegt, das Kopfende der Bohrung eine einwärts konvergierende Seitenwand (144) und eine konvex gewölbte Endfläche aufweist und der Kolben eine obere Kompressionsdichtung und eine konkav gewölbte obere Endfläche (149) aufweist, dadurch gekennzeichnet, dass die Luftkammer (158) im Umfangsbereich des Kopfendes der Bohrung angeordnet ist; dass die Länge des Durchgangsbereichs sich längs des Umfangs des Kopfendes der Bohrung erstreckt; dass eine Fläche (162) am Umfang des oberen Endes des Kolbens an der Durchgangsoffnung liegt und dieselbe teilweise anschliesst wenn die Brennkammer minimales Volumen aufweist und von dem Durchgang entfernt ist, wenn die Brennkammer größer ist als bei minimalem Volumen.

14. Maschine nach Anspruch 6, worin der oder die genannten Kolben ein Paar von Kolben (170, 172) mit geschlossenem Ende sind die obere Kompressionsdichtungsmittel (188) aufweisen und hin- und herbeweglich in einer einzigen zylindrischen Bohrung angeordnet sind, wobei die Kolben zusammen mit ihren geschlossenen oberen Enden und der Bohrung die genannte Brennkammer (174) begrenzen, dadurch gekennzeichnet, dass jedes obere Kolbenende einen konkav gewölbten Mittelteil (180, 182) und je eine obere Kantenpartie (186) am Umfang aufweist die in radialem Abstand von der Bohrungswand liegt und eine maximale Durchgangsweite (192) bestimmt; dass die genannte Luftkammer (184) im Umfangsbereich des oberen Endes jedes Kolbens zwischen der genannten Umfangskantenpartie und den genannten Kompressionsdichtungsmitteln jedes Kolbens liegt, und dass der Abstand zwischen den genannten Oberen Kolbenumfangskantenpartien eine minimale Durchgangsweite (190) bestimmt, wenn die Brennkammer minimales Volumen aufweist.
FIG. 3
FIG. 11 PRIOR ART

FIG. 12

FIG. 13

FIG. 14