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result, little noise will reach the supporting casing. Because of the lever action, the crank (not shown) has half the throw of the piston stroke and can be a stubby, cam-like unit with large, closely spaced pins having substantial overlap for strength.

The compression ratio can be changed by slightly lengthening or shortening the effective length of the beam **39**. This can be done by the lower pivot plate **42** being attached to a block **43**, mounted slidably in a fixed block **44**, in which block **43** can be moved slidably by a servo motor **45**. The gear **45***a* rotated by servo motor **45** is much longer than the gear **44***a* on the screw **43***b*, which is rotatably attached to block **43** and rotates against threads in block **44**, causing gear **44***a* to slide back and forth on gear **45***a*, as block **43** reciprocates in block **44**. Thus, as a diesel, it could be started at 20:1 ratio and then shifted to a 13:1 ratio for higher efficiency and for less friction and stress on parts. This could also be important to allow use of alternate fuels.

Referring now to FIG. **15**. This same advantage holds true for this alternate design in which the pivot **47'** is between the connecting rod **19** and the piston **22"**.

The needed variation of the length of the beam **39** (shown in phantom), connecting the piston **22"** to the connecting rod **19**, can be accomplished by forming a scotch yoke **40** on the beam end fitting over the wrist-pin **41** of the piston **22"**, or by placing a double pivoting link **42'** between the pivot **47'** on the fulcrum of beam **39'**, with the pivot **42"** being attached to a non-movable pan **46** of the engine and the terminal end of beam **39'** being connected to connecting rod **19** by a pin **47**.

Alternatively and preferably, for heavy duty engines (marine propulsion, power production, etc.) the power take off of piston **22"** could be with a conventional piston rod **39'** being arranged between piston **22"** and a crosshead **20'**, with a connecting rod **19'** between the crosshead **20'** and the crankshaft (not shown).

Double-acting power cylinders when used in the engine of this invention, or any other engine, will be especially important where great power is desired and cooling water is readily available, e.g., for marine use or for power generation. These double-ended, double-acting cylinders of the engine of FIG. **14** and FIG. **15** can be used in all of the designs of this invention, or in any Otto or Diesel engine.

Double Burn "Dwell" Time

Referring now to FIG. **16**. There is shown a schematic transverse sectional view of a crankshaft, two connecting rods **19'** and **19"** and a beam **39** showing a means of providing extra loading and burn "Dwell" time at TDC of a conventional 2-stroke or 4-stroke engine and especially for the engine designs of the present invention, namely **4** stroke engines of FIGS. **1** through **7** and FIGS. **18** through **19**-B. The system of Double Burn Time of FIG. **16** is also important in 2-stroke engines of FIGS. **8** through **10**.

This layout for any engine provides for double the piston **22** turnaround time of a normal engine during the critical burn period. This is because piston **22** TDC occurs at BDC of the crank **48**. At this point, crank pin motion around piston **22** is subtracted from the straightening movement of the connecting rod **19'**, instead of being added to it, as in conventional engines. Reversing the usual action slows piston travel around this point, resulting in more complete combustion and further reducing emissions. This system allows more complete fuel burn and heat energy conversion into peak pressure before the power stroke begins. This enhances power per unit of fuel consumption and brake specific fuel consumption (BSFC), and reduces polluting emissions. It also allows the engine to operate under lean burn conditions.

The extra burn time, provided by the design of FIG. **16**, can be important in the engines of this invention and to any Otto or Diesel cycle engine.

Operation of the engine, constructed and arranged with the additional burning time, would be the same as the other engines of this invention, providing high charge density, low compression-ratios in some designs, with a mean effective pressure higher than conventional engines, but with more combustion time than other engines, while producing even less polluting emissions.

Since the crankshaft **48** in FIG. **16** does no more than transmit torque, its main bearings will be very lightly loaded. As a result, little noise will reach the supporting casing. Because of the lever action, the crank can have as little as half the throw of the piston stroke (depending on the point of the fulcrum), and can be a stubby, cam-like unit with large, closely spaced pins having substantial overlap for strength. This layout provides for twice the combustion time of the engine of this invention during the critical loading and burn period. This is because piston TDC occurs at BDC of the crank.

Constant Speed Engines

Whereas the preponderance of the foregoing specification describes embodiments and representative engines of the present invention which are optimized for duty cycles in vehicles such as automobiles, trucks, buses, tanks, trains and airplanes and describes systems and methods for varying power, torque and speed. The present invention finds useful application for obtaining high power and torque, while maintaining optimum fuel economy and low polluting emissions in less complex systems, such as constant speed engines, for example. FIG. **17** depicts alternate embodiments of the present invention which are representative of constant speed engines such as electric power generators, pumps and compressors, outfitted in accordance with the principles of the present invention.

The Engine of **100**[17]

Referring now to FIG. **17**, there is shown is a schematic presentation of an engine which represents any of the 4-stroke or 2-stroke engines of the present invention, outfitted for constant speed operation. The basic components of the engine **100**, such as compressors **1**, **2** and optional intercoolers **10**, **11**, **12** (shown in phantom) and their necessary associated conduits are preferably, designed for optimum operating parameters, having only the basic components. The various controls, shutter valves, air bypass valves and their associated bypass conduits, such as those in previously described embodiments, are preferably eliminated in order to reduce weight, cost and complexity of operation. In FIG. **17**, the engine **100** is shown, as outfitted, with a first ancillary compressor **1** and a second ancillary compressor **2**, optional intercoolers **10**, **11**, **12** (shown in phantom) and interconnecting conduits, all operating as would be understood with reference to the previous detailed descriptions and operating with two stages of precompression of the charge-air, intercooled or adiabatically compressed.

FIG. **17** shows a preferred setup for power generation with any of the engines of this invention. The power output shaft **20** of the engine **100** is coupled schematically by line **140** to power input shaft **20"** of generator **141** which has

electric power output lines **142**. As the shaft **20** of the engine **100** rotates the shaft **20"** of generator **141**, the amount of electric power produced by generator **141** is detected by sensor **143** and relayed to control unit and governor **144**. With various relays and integrated circuits, their overall purpose is to quantify and adjust the fuel input, power output and speed. To do this, they will:

1) Send messages by line **145** to fuel/air control (not shown) on fuel line **148** and speed control **56**.
2) Spark control by line **149** in order to advance or retard the spark in spark-ignited engines
3) Send messages through lines **146** and **146***b* for engines having fuel injection systems, e.g. for natural gas, gasoline or diesel fuel, or to fuel/air controls.

Control unit **144** also sends signals to control the proportioning valve **201**, (not shown) and to proportioning valves **209***a*, **209***b*, **209***c*, **209***d*, **209***e*, **209***f* and **209***g* shown in FIG. **1**-C to control the amount, if any, of exhaust gases re-circulated by these valves for re-burn in any engine of this invention utilizing this feature. Further explanation of the components and operation with the engines of the present invention is deemed unnecessary, as it would be understood by those skilled in the art having reference to the present disclosure.

The optional intercoolers **10**, **11**, **12** (shown in phantom) are preferably reduced in number or cooling capacity in the compression-ignited engine. This is made possible by a cooler working cycle and by low peak pressures and temperatures in the engines of this invention. The constant speed engine would be designed to produce the combustion chamber temperature desired at TDC of piston **22** which may require some of the air bypass systems and controls of the other engine designs.

The Engine of $100^{17\text{-}B}$

Referring now to FIG. **17**-B. There is shown an engine illustrated as a 2-stroke engine, but representing any of the engines of the present invention, 2-stroke or 4-stroke, which is coupled schematically by line **140** with an electric generator **141**. The engine and arrangements are similar in structure and operation as those shown and described for the engine of FIG. **17**, with the exception that engine of FIG. **17**-B, operating as either 2-stroke or 4-stroke cycle engine **100**, has only a single stage of pre-compression, optionally intercooled by intercoolers **11**, **12** (shown in phantom), of the charge air. As with the engine of FIG. **17**, intercoolers **11**, **12** are preferably reduced in cooling capacity in compression-ignited versions of the engine **100** of this invention. Also, as with the engine **100** of FIG. **17**, the governor, and other controls and the operation of the engine and generator would be understood by those skilled in the art having reference to the present disclosure. This engine also may require some of the bypass systems for compression and intercoolers illustrated for other engine designs of the present invention.

Ultra Low Emissions Engine with Extended Expansion Ratio Operating in Operational Design 11

Referring now to FIG. **18**. There is shown a schematic drawing of a six cylinder dual mode engine 100^{18} operating in a 4-stroke cycle which can be spark, compression or homogeneous charge compression ignited (HCCI).

Brief Description of Operational Designs 11 and 12

Operational Design 11, FIG. **18**. The engine is similar in structure to the 4-stroke engine of FIG. **2** and shows an alternative air compression system utilizing air intake **8** (in phantom) or air intake **8'**. FIG. **18** also shows four intercoolers **10**, **10'**, **11**, and **12**, and dual manifolds **13** and **14**. The alternate air induction system **8'** shown in FIG. **18** supplies unpressurized charge-air to low pressure intake valves **16**-B of the engine of FIG. **18** by providing atmospheric pressure air to the intake runners **32'** and **32"** and to intake valves **16**-B of each power cylinder. Optionally, a second air induction system also shown in FIG. **18** where air inlet **8** inlets atmospheric air to a turbo-charger **1** and cooler system **10** (all in phantom) to boost the pressure and cool the initial charge that is then directed optionally to conduit **32** and low pressure intake valves **16**-B where the charge is received compressed while expelling a large portion of the charge into compressing and cooling system, compressor **2** and intercoolers **11** and **12** and is then admitted to the inlet valve **16**-A of power cylinders **22** after piston **22** has passed point "E" in the last part of the power stroke.

Operation

In this concept, air at atmospheric pressure (from inlet **8'**), or pressure boosted by compressor **1** and cooled by intercooler **10** and through conduit **32**. FIG. **18**, is received by the intake (1^{st}) stroke of piston **22** through intake valve **16**-B and then piston **22** turns around and the compression (2^{nd}) stroke begins. Then, during the compression (2^{nd}) stroke, much of the total charge, now being compressed, is expelled (this is part A of charge) through outlet valve **16**-C (indicated, but not shown) which quickly opens and closes at piston position E into a compression-cooling circuit system consisting of conduits **201**, **202**, intercooler **10'**, compressor **2**, bypass system **3**, **4**, intercoolers **11**, **12**, to manifolds **13**–**14**, conduits **15**-A, later to be injected by high-pressure intake valves **16**-A. That portion of the air charge (part A) is now further compressed and cooled with pressure buildup in manifolds **13**–**14**. (The compression stroke of piston **22** continues toward TDC, now compressing part B of the charge, that remaining in cylinder **7** after piston **22** passes point E and valve **16**-C has opened and closed). The cool dense A charge, now accumulated in the compression-cooling circuit, is then re-injected through high-pressure intake valve **16**-A above piston **22** crown after piston **22** has reached or passed the point (E) during the compression stroke and can be injected as late as during combustion. The charge A has been injected, cylinder compression of the now total air charge (B that remaining in cylinder after passing point E plus A, the injected portion.) continues, which is Operational Design 11. Near TDC of piston **22**, fuel is added, if not present, and the charge is ignited for the power (3^{rd}) stroke followed by the scavenging (4^{th}) stroke, as exhaust valve **17** opens, and another power cycle begins. This produces a denser extended expansion ratio charge with great power, low fuel consumption and low emissions.

The compression ratio of the cylinder is alternatively much greater (with, of course, much smaller combustion chamber compared to the volume of cylinder **7**) due to the necessity of retaining the improved density of the charge. The latter is allowed by the lower heat-of-compression allowed by the low "effective" compression ratio of both A and B portions of charge air or fuel-air. The charge portion A, which is highly compressed and cooled before re-injecting and igniting, has an "effective" compression ratio which can be as low as 2:1 to 5:1 for very low heat-of-compression and producing ultra low polluting emissions, especially NO_x. This is also true for portion B of charge.

In this Operational Design (11), although the "effective" compression is extremely low, the actual compression ratio, which is very high, equals the expansion ratio, with much improved power and efficiency per unit of fuel-air charge.

The system optionally operates in this fashion: The cylinder **7** has, say, an expansion ratio of 20:1 and the "effective" compression ratio is much less. Consider that the cylinder has a compression ratio of 20:1, at two-thirds of piston **22** stroke distance (point E), outlet valve **16**-C opens to expel air charge A at a low compression state since there is a pressure blow-down due to suction of compressor **2** which receives the expelled portion (A) of the charge. Piston **22** continues the compression stroke on B, the remainder of the charge, producing about a 5:1 "effective" compression ratio on B, the portion of charge remaining in the cylinder. The charge A received by the compressor **2**, which is pre-cooled by intercooler **10'**, is highly compressed now and again cooled by intercoolers **11** and **12**. This cool, dense charge is, after pressure build-up, re-injected into cylinder **7** above piston **22** after the piston has passed point E in which outlet valve **16**-C has closed and can be as late as TDC. Therefore, piston **22** is compressing the B portion of charge, that remaining in cylinder after expulsion of a portion A of charge through outlet valve **16**-C. The "effective" compression ratio of that portion is as stated, perhaps about 5:1. The highly compressed cool charge A re-injected could have an "effective" compression ratio of less than 4:1 since the distance from injection (after point E) to TDC is perhaps only 30% of piston travel distance. The total charge (A+B) then ignited near TDC to produce the power stroke, is cool, very dense and has an "effective" compression ratio of about 5:1 or less. Yet, piston **22** is depressed in the power stroke with the density of, and the expansion ratio of a 20:1 to 40:1 combustion ratio engine, depending on the volume of the combustion chamber. This system produces a more dense charge than does a 10:1 compression ratio engine, but with much superior power and efficiency with very low polluting emissions including particulates. This system also eliminates problems with spark-knock and pre-ignition. Fuel-air mixes can be used with diesel oils with spark ignition or HCCI operations.

Operational Design 11-B, FIG. **18**: As in Operational. Design 11, atmospheric or pressure boosted air (the latter cooled by intercooler **10**) is received by cylinder **7**, by intake valve **16**-B during intake (1^{st}) stroke of piston **22**. At BDC, valve **16**-B closes. Then, compression (2^{nd}) stroke begins. Ancillary valve **16**-C, which is alternatively automatic, opens quickly during the compression (2^{nd}) stroke near piston TDC with nearly all of the compressed charge being expelled into a cooling circuit (like that described for Design 11.) Outlet valve **16**-C is then closed. At the same time, high pressure intake valve **16**-A opens (after valve **16**-C closes) and temperature adjusted cooled air, which has been building in pressure by rotating the crank shaft or is stored and is cool and stored in the compression cooling circuit, is now injected into combustion chamber **7'** near end of compression stroke TDC. Intake valve **16**-A now closes. Fuel is added, if not present, and the charge is ignited at or near TDC and the power (3^{rd}) stroke occurs, followed by the scavenging (4^{th}) stroke, as exhaust valve **17** opens and another power cycle is complete. The effective compression ratio in this Design is near zero, perhaps 2:1 or less for extremely low heat-of-compression, thus providing ultra low polluting emissions, but with power and low fuel consumption.

Shown in FIG. **18** is the compression and cooling system and a suggested engine control system, all consisting of an engine control module (ECM) **27**, two shutter valves **3** and **5**, two air bypass valves **4** and **6**, four intercoolers **10**, **10'**, **11** and **12**, two compressors **1** and **2** and a scheme of controlling the pressure, temperature and density by controlling air bypass valves **4** and **6** and shutter valves **3** and **5** and intake valves **16**-B and **16**-A and exhaust valve **17** (not shown.) Also indicated, but not shown, is an ancillary outlet valve **16**-C, positively or automatically activated. As described, the air pressure and temperature can be managed as desired. Air bypass valve **4** is alternatively closed to allow compressor **2** to fully compress the charge and shutter valve **3** is shown slightly open to demonstrate allowing part of the air to flow uncooled (hollow arrows) and some of the air cooled (solid arrows) to the manifolds **13** and **14**, all of which could be controlled by the ECM-**27** in order to provide an air charge at optimum density, temperature, and pressure. The hollow arrow in conduit **121** and **122** show how air bypass valve **4** can be partially opened to allow some of the air to bypass and return to the compressor **2** in order to finely adjust the pressure of the doubly compressed and cooled air charge that is injected in order to adjust the charge density and temperature to that desired. Alternatively, all of the air charge can be directed through the intercoolers **10'**, **11**, and **12** (and **10** for pre-compression), or partly through bypass conduits **121** and **122**, to the manifolds **13** and **14** for fine control of temperature. These controls could provide any air-fuel mixtures desired.

Results:

1) Adjusts the temperature of the entire charge.
2) Thoroughly mixes the charge (fuel, if not present is injected).
3) Increases the density of the charge.
4) Produces an expansion ratio greater than the "effective" compression ratio or with equal compression and expansion ratios.
5) Produces great turbulence.

Advantages:

1) Ultra low emissions.
2) Greater power.
3) Increased fuel economy.
4) Increased engine durability.
5) Multi-fuel capability.

Both designs have expansion ratios greater than the effective compression ratio. In either of the Operational Designs 11 or Design 11-B, blocking valve **16**-E (shown in phantom on conduits **15***a*-A and **15***f*-A in FIG. **18**) could be closed when starting the engine in order to more quickly build-up design pressure in manifolds **13** and **14**. Also, external or stored compressed air could be used to build up operational design pressure for quicker power build-up, The engines of this design may be ignited, even with diesel fuels, by spark, modified HCCI or HCCI as described herein.

Ultra Low Emissions Engine Operating in Operational Design 12

Referring now to FIG. **18**-B, there is shown a schematic drawing of a six-cylinder engine **100**$^{18\text{-}B}$ operating in a 4-stroke cycle engine. The engine, which is depicted in FIG. **18**-B, is similar in structure to the 4-stroke engine of FIG. **18**. It can be operated in system of Design 11 or Design 11-B, as desired. FIG. **18**-B shows that it has a variant alternative, enhanced air cooling system useful for providing compression-expansion chilled engine charge air or air-fuel mix. It further shows two air compressors **1** and **2**, four intercoolers **10**, **10'**, **11** and **12**, with controls to alter charge density and temperature, each separate from the others, controlled by ECM-**27**. Conduits **121**, **122**, **113** and **114** lead

from the compressor-cooling system to expansion valves **10'** and **10"** of FIG. **18**-B, pressure-drop-distributors **11'** and **11"** and distributor tubes **17***a* and **17'***a*, which tubes distribute expanding air to conduit **27'***a*–**27***f* and to intake valves **16***a*-A and **16***f*-A of engine power cylinders **7***a* and **7***f* of the engine of FIG. **18**-B. The expansion valve **10'**–**10"** passages are optionally variable, and along with bypass value R to conduit X, are controlled by ECM-**27** as indicated to provide management of pressure and temperature levels.

The operation of engine of FIG. **18**-B is the same as the operation of the engine of FIG. **18**, utilizing Operational Designs 11 or 12, with the primary air charge being optionally pre-compressed and cooled and when operating in Design 11 or Design 11-B with a large portion of the charge, always being highly compressed and cooled. Additional benefits are having broader control of pressure and temperature of both portions of the air charges. By utilizing variable expansion valves **10'**, **10"** and bypass system R and X, chilled air temperature, pressure and density output can be variably controlled by ECM-**27** to provide a dense, cooler charge for more power and lower polluting emissions.

The operating parameters and the advantages of this engine are the same as those described for the engine of FIG. **18** and FIG. **18**-B, using either Operational Design 11 or Design 12, with the added benefits provided by compression-expansion cooling of the charge air as described for engine of FIG. **2**-B.

The expansion of the chilled charge to high-pressure intake valve **16**-A can be varied from less than 1 atmospheric pressure to 5–6 atmospheric pressures to no expansion at all, as expansion valve **10'** can be opened to varying sizes or can be bypassed, partly or wholly, for a very dense, cool secondary air or air-fuel charge, always with great turbulence. (The latter may require pressure control valves **14"***a*–**14"***b* on conduits **27'***a* or **27'***f*).

Ultra Low Emissions Engine with In-Cylinder Compression Operating in Operational Design 13

Referring now to FIG. **18**-C, there is shown a schematic drawing of a six cylinder engine **100**$^{18\text{-}C}$ operating in a 4-stroke cycle which can be spark, compression or homogeneous charge compression ignited (HCCI). This new cycle engine can produce and utilize an air or air-fuel charge, having the density (compactness) of a 40:1 compression ratio and greater, with an "effective" compression ratio as low as 2:1 to 5:1 with very low fuel consumption and peak temperatures with ultra low polluting emissions. In this engine, except for initial pre-compression, all compression occurs in-cylinder.

Brief Description

The engine is similar in structure to the 4-stroke engine of FIG. **18** which shows and describes a charge air compression system with three stages of compression. With the exception of optional pre-compression of the primary air charge by compressor **1**, all compression occurs within the power cylinder, with charge cooling of the first two stages for a denser than normal charge and with an extended expansion process. The system utilizes air intake port **8** (in phantom) or air intake port **8'**. FIG. **18**-C engine also shows four intercoolers **10**, **10'**, **11** and **12** and dual manifolds **13** and **14**. One air induction system of using inlet port **8'** shown in FIG. **18**-C supplies unpressurized charge air to intake valves **16**-B of the engine of FIG. **18**-C by providing atmospheric pressure air to the intake conduits **32'** and **32"**, which then distributes the low pressure air to intake valves **16**-B of each power cylinder. An alternate air induction system also shown in FIG. **18**-C is where air inlet **8** inlets atmospheric air to a turbo-charger **1** and an intercooler **10** system (all in phantom) to pre-compress and cool the entire air charge that is admitted to cylinder **7** by the intake valves **16***a*-B–**16***f*-B of power cylinders **7***a*–**7***f*.

Referring now to FIG. **18**-D: In this concept (Engine of FIG. **18**-C), air, atmospheric or pressure boosted by turbo-charger **1**, and which has been cooled by intercooler **10**, optionally with temperature and pressure adjusted by control valves **5** and **6**, is received through intake valve **16**-B by cylinder **7** during the intake (1st) stroke of piston **22** while piston **22** travels from G to A in cylinder **7** of FIG. **18**-D. During the compression (2nd) stroke which now occurs, the major portion of the charge, portion A, being compressed, is expelled through outlet valve **16**-C of FIG. **18**-D. Valve **16**-C opens and quickly closes at the appropriate piston position points EO_1 and EC in FIG. **18**-D. Now part of the charge has been compressed through valve **16**-C, FIG. **18**-C into a cooling circuit system, composed of conduits **200**, **201**, **202**, intercoolers **10'**, **11** and **12**, conduits **109**, **113** and **114**, manifolds **13**, **14** and conduits **15***a*-A–**15***f*-A and, after pressure build-up is returned to intake valves **16***a*-A–**16***f*-A of cylinders **7***a*–**7***f* (FIGS. **18**-C and **18**-D).

During the compression (2nd) stroke, at the point EC that outlet valve **16**-C closed, FIG. **18**-D, outlet valve **16**-D opens for pressure blow-down as piston **22** travels from position EC to position F in FIG. **18**-D. At point F of piston travel, the in-cylinder pressure being significantly reduced by Blow-Down and optionally vacuum of suction line **16**-DC_2-A or B, valve **16**-D closes and intake valve **16***a*-A–**16***f*-A opens and closes quickly, re-injecting the A portion of the air or air-fuel charge into cylinder **7** after piston point F. With valve **16**-A now closed, compression continues and near TDC, fuel is added, if not present, and the charge is ignited for the power (3rd) stroke. At near BDC of piston **22**, exhaust valve **17** opens to expel exhausted gases into exhaust conduit **18** of FIG. **18**-C, which optionally conveys the exhaust gases to inlet of turbo charger **1** which optionally pre-compresses the incoming charge air.

An alternate means of operating outlet valve **16**-C is as follows: Intake valve **16**-B opens during the intake (1st) stroke of piston **22** while piston **22** travels from G to A in cylinder **7** of FIG. **18**-D. During the compression (2nd) stroke which now occurs, the major portion of the charge, portion A, being compressed, is expelled through automatic outlet valve **16**-C of FIG. **18**-D. Valve **16**-C opens at EO_2 near piston BDC and closes at the appropriate piston position point EC in FIG. **18**-D. Now, the major part of the charge has been compressed through valve **16**-C, FIG. **18**-C into a cooling circuit system, composed of conduits **200**, **201**, **202**, intercoolers **10'**, **11** and **12**, conduits **109**, **113** and **114**, manifolds **13**, **14** and conduits **15***a*-A–**15***f*-A and, after pressure build-up is returned to intake valves **16***a*-A–**16***f*-A of cylinders **7***a*–**7***f* (FIGS. **18**-C and **18**-D). During compression stroke of piston **22** at piston position EC, automatic valve **16**-C closes and is held closed through the remainder of the compression stroke and through the scavenging and air intake strokes. As piston **22** turns around to begin the compression stroke at point EO_2, valve **16**-C is again opened by pressure differential. As the compression stroke takes place, valve **16**-C again closes at position EC and is again held closed until the next compression begins. At point EO_2, valve **16**-C becomes automatic.

During the compression (2nd) stroke, at the point EC that outlet valve **16**-C closed, FIG. **18**-D, outlet valve **16**-D opens for pressure blow-down as piston **22** travels from

position EC to position F in FIG. **18**-D. At point F of piston travel, the in-cylinder pressure being significantly reduced by Blow-Down and optionally vacuum of suction line **16**-DC**2**-A or B, valve **16**-D closes and intake valve **16***a*-A–**16***f*-A opens and closes quickly, re-injecting the A portion of the air or air-fuel charge into cylinder **7** at piston point F. With valve **16**-A now closed, compression continues and near TDC, fuel is added, if not present, and the charge is ignited for the power (3^{rd}) stroke. At near BDC of piston **22**, exhaust valve **17** opens to expel exhausted gases into exhaust conduit **18** of FIG. **18**-C, which optionally conveys the exhaust gases to inlet of turbo charger **1** which optionally pre-compresses the incoming charge air.

Operation of Engine FIG. **18**-C Shown Schematically in FIG. **18**-D

1) Intake valve **16**-B opens. The piston performs the intake (1^{st}) stroke going from G to A which is BDC.
2) Piston **22** reverses and begins compression (2^{nd}) stroke.
3) At point EO_2 outlet valve **16**-C opens and closes at point EC during the compression stroke.
4) Outlet valve **16**-D opens at point EC as soon as outlet valve **16**-C is closed.
5) Piston **22** continues for pressure blow-down between point EC and point F optionally with suction on conduit **16**-DC_2-A or B.
6) At point F, outlet valve **16**-D closes and intake valve **16**-A opens, the latter injecting the now cooled, compressed charge above piston **22** crown and quickly closes as piston **22** continues the compression stroke.
7) At near TDC, fuel is added, if not present, and the charge is ignited.
8) The expansion (3^{rd}) stroke occurs.
9) Piston **22** reverses and scavenges the cylinder through valve **17** and conduit **18** by the (4^{th}) stroke to complete one cycle.
10) Piston **22** draws in fresh air and at BDC, outlet valve **16**-C begins to function by pressure differential.

Operating Principles

When outlet valve **16**-C opens at point EO_2, a large portion of the air charge is compressed into the cooling storage circuit while piston **22** travels between points EO_2 and EC (where valve **16**-C closes) to cool and build proper pressure charge for re-injecting into cylinder **7** above piston **22** at point F. Between points EC and F, the pressure in cylinder blows down optionally out through valve **16**-D, aided by vacuum line **16**-DC**2**-A. (At this point [EC], the cylinder pressure must be reduced significantly in order to allow introduction of the compressed air or fuel-air [cooled or chilled by valve **16**-A, being optionally an expansion valve], at point F). If the air charge contains fuel, the discharged air-fuel charge through outlet valve **16**-D is directed by conduit 16-DC to conduit **16**-DC**2** which goes alternatively to A) suction side of turbo-charger, to B) suction conduit **105** or to C) intake port **8'** of conduit **32"**–**32'**. In any case, the fuel-air charge cannot be expelled into the atmosphere. Some of the work of compressing the part of the charge expelled could be restored with little back-pressure in cylinder **7**. Some of the alternative systems are: (See FIG. **18**-C).

1) Directing conduit **16**-DC to drive a section of a dual turbine on the turbo charger (conduit **16**-DC**3**.)
2) Directing conduit 16-DC to empty into exhaust conduit **18** by way of conduit **16**-DC-**4** to help drive the single turbine-turbo charger.
3) Directing conduit **16**-DC to a turbine (T) geared to the engine crank shaft or drive shaft by way of conduit **16**-DC-**5**.
4) Directing conduit **16**-DC to the atmosphere by conduit **16**-DC-**6**.

A means of quickly building operating pressure is by closing optional shut-off valves **16***a*-E to **16***f*-E as the rotating crank shaft quickly builds pressure. Alternatively, the engine is quickly started by stored compressed air as is the case in some large engines.

Additional operating principles are:

1) Assume the volume of the cylinder divided by the volume of the combustion chamber is 20:1. (We will use these figures for simple calculations, but actually the combustion chamber is preferably made smaller, perhaps 40:1).
2) Therefore, the expansion ratio in this estimation will always be 20:1.
3) An air charge is received by cylinder **7** through intake valve **16**-B (1^{st} stroke). Piston **22** reverses and begins compression (2^{nd} stroke). Assume the compression ratio is 14:1 at point EO and the charge is pressed into valve **16**-C cooling system., FIG. **18**-D. There, valve **16**-C closes at point EC and valve **16**-D opens.
4) A pressure blow-down occurs between points EC and F, optionally aided by vacuum to make room for re-injection of charge after pressure builds.
5) Assume the pressure now in-cylinder at point F is negligible.
6) When intake valve **16**-A opens at F, as valve **16**-D closes, the cylinder above piston **22** at point F is quickly filled with air or air-fuel charge which has the density of a charge which is close to 14:1 compression ratio, but has little or no heat-of-compression.
7) This cooled, dense charge now in combustion chamber of cylinder **7** is compressed further with a low "effective" compression ratio (perhaps 2:1 to 5:1) in the reduced volume—"diminutive" combustion chamber where it can be compacted to a much denser state than that of conventional engines with, again, small amount of heat-of-compression.
8) At or near piston TDC, the air-fuel charge is ignited for the full expansion which for this basic assumption is a 20:1 expansion ratio.
9) The scavenging (4^{th}) piston stroke completes one working cycle.
10) If the injected charge has a density of 12:1 to 14:1 compression ratio is cool and is further compressed by 6:1 compression ratio, the density of the charge is 12×6 for a density of 72:1 or 14×6 for a density equal to an 84:1 compression ratio with very small heat-of-compression.
11) Therefore, the denser charge is inherently more powerful and efficient than a normal engine's charge and having great turbulence, produces much lower amounts of polluting emissions.
12) Although the charge weight has been reduced, the greater density coupled with the extended expansion ratio provides improved efficiency that will offset the waste of energy required to compress that portion of the charge lost in the "blow-down" of pressure at point EC. In this new working cycle, there is a net gain in both power and efficiency.
13) Some of the energy lost in the cylinder blow-down process can be recovered by directing the expelled portion of charge to the turbo-charger **1** drive inlet, or to drive turbine T, as shown in FIG. **18**-C. If the fuel is pre-mixed with air, the charge expelled to point F_0 must be returned to the intake system as described herein.

Shown in FIG. **18**, FIG. **18**-B and FIG. **18**-C is a suggested engine control system, which also pertains to FIG. **18**-D. This control system consists of an engine control

module (ECM) **27**, two shutter valves **3** and **5**, two air bypass valves **4** and **6** and a scheme of controlling the pressure, temperature and density by controlling air bypass valves **4** and **6** and shutter valves **3** and **5**. A control example illustrates that air bypass valve **4** is partly closed to optionally re-circulate part of the compressed charge. Shutter valve **3** is slightly open, allowing part of the air to flow uncooled (hollow arrows) and some of the air cooled (solid arrows) to the manifolds **13** and **14**, all of which is preferably controlled by the ECM-**27** in order to provide an air charge at optimum density, temperature and pressure. The hollow arrow in conduit **120** shows how air bypass valve **4** can be partially opened to allow some of the air to bypass and return in order to finely adjust the pressure of the super-compressed air charge that is injected to adjust the charge density, temperature and perhaps a stoicheometric air fuel mixture. Alternatively, all of the super-compressed portion of the air charge can be directed through the intercoolers **10'**, **11** and **12** to the manifolds **13** and **14**.

Outlet valve **16**-C should be large to reduce fluid flow friction while open and intake valve **16**-A is optionally doubled in size or number. In one optional design, the cylinder **7** diameter may be smaller in respect to the piston **22** stroke length. This provides farther piston travel to allow a longer travel time between points A and E and to give more time for valve **16**-C to open and close. In addition, the smaller diameter cylinder allows more charging and burning time as piston **22** travels from F to G and returns from G to A, the latter producing an extended expansion ratio.

Alternatively, double burn (dwell) time is allowed by utilizing the cylinder-piston drive arrangement shown in FIG. **16** as described herein.

Results and advantages of the engine **100**$^{18\text{-}C}$ and FIG. **18**-D with the primary air charge being boosted in pressure.

Results

1) Low peak temperatures and pressures
2) Homogeneous combustion charge
3) A denser, cooler charge
4) Greater efficiency
5) Greater engine durability

Advantages

1) Lower polluting emissions
2) Greater power per unit of charge
3) Greater torque
4) Increased vehicle range
5) Multi-fuel capabilities
6) Longer engine life

Homogeneous Charge Compression Ignition (HCCI) Basic Operating Principals

The basic HCCI operation, that of controlling charge temperatures and timely ignition, is similar for both 4-stroke engines (FIG. **1** to **7** and FIG. **18**, **18**-B and FIG. **18**-C) and for 2-stroke engines (FIG. **8**, **9** and **10**). In the 2-stroke engine, the primary air charge used generally comes through ports **11** in the wall of cylinder **12**. The valving is by ports **11** being closed and opened by piston **22**. In 4-stroke engines, the primary air charge comes from intake valve **16**-B. In both 2-stroke and 4-stroke engines, the supplementary air charge generally comes through intake valve **16**-A.

For engines designed for HCCI operations using Methods A through G and engine Operational Designs 1 through 13, as depicted and described herein, for the present invention, are recommended.

Operational Designs 1 through 6 pertain particularly to 4-stroke engines of FIG. **1**, FIG. **2**-B, FIG. **2**-C, FIG. **3** to FIG. **5** and FIG. 7. Engine of FIG. **2**-D utilizes Operational Design 7 in either 4-stroke or 2-stroke engines. Operational Designs 8 through 10 pertain particularly to 2-stroke engines of FIG. **8** through FIG. **10**. Operational Designs 11, 12 and 13 pertain particularly to 4-stroke engines of FIG. **18**, FIG. **18**-B and **18**-C.

Description of subsequent Methods, A through G of HCCI operation will be limited almost entirely to the methods of control of charge density, temperatures and the means and timing of charge ignition, with reliance on the instructions for operating the engine as being explicit in the instructions and suggestions for the HCCI methods and also with reliance on references to the various engine designs and their corresponding Engine Operational Designs.

The spark ignition operation of any of the Operational Designs requires similar control of charge density, temperature and ignition timing with much less stringent control of temperatures, the most important being that the temperature of the fuel-air mix must be lower than auto-ignition until the point of ignition, with ignition generally occurring a few degrees before piston TDC. The chief advantage of this system of spark ignition being that low auto-ignition fuels like diesel oils can selectively be compressed to a much denser and more powerful charge, although already mixed with air, than that of the Diesel cycle engine, while also eliminating pre-mixed burning. The latter is the cause of diesel smoke and and smaller particulates. In addition, this system of keeping the combustion and peak temperatures low greatly reduces or eliminates NO_X and some other polluting emissions.

HCCI and Spark Ignition Methods A–G

Referring now to FIG. **19**, FIG. **19**-B and FIG. **19**-C, there is shown schematic transverse sectional views of preparation or pre-combustion chambers **38'** and combustion chamber **38** and **38**-B with some variance of shapes in the different figures with associated inlet conduits, injector pump, ducts, jackets and valving for fluids (gaseous or liquid) and electrical elements, all pertaining to producing and maintaining temperature desired for gaseous or liquid fuel operation of the engines of this invention, 2-stroke, 4-stroke, or rotary, or for any other internal combustion engine for providing spark or compression ignition, modified HCCI or true HCCI in a timely fashion, and for which methods are described herein. All components named will be identified and described in regard to the method of ignition to which they pertain.

The cool, dense air-fuel charge, either premixed or instantaneously homogenized, of the engine of this invention lends itself readily to conventional spark or compression ignition. The charge can also be ignited by modified HCCI or true HCCI systems as described herein. Since ignition by spark or compression is well known in the art, the following describes only the latter two systems. HCCI is accomplished in this manner: FIG. **19** illustrates primary air at atmospheric pressure, or pressure boosted and cooled, entering cylinder **7** through channel **18'** on the intake (1^{st}) stroke through intake valve **16**-B with a second cool or optionally chilled high pressure air charge entering later during the compression stroke through inlet **8'** and intake valve **16**-A where the two charges mix above piston **22**. Temperature and pressure sensors **21'** and **21"** and optional pressures sensors therein signal ECM-**27** to adjust flows to the proper order to produce an air-fuel mixture which has proper density and is at a temperature at near piston TDC, preferably at or just below that which would cause the charge to auto-ignite at piston **22** TDC, depending upon the particular method of charge ignition selected.

Engine Control Module ECM-27 of FIG. 2 or FIG. 2-C, along with compressor(s) 2 and optionally 1 and coolers 10, 10' and/or 11 and 12 with variable valves 3, 4, 5 and 6 on conduits as shown, along with pressure, temperature, oxygen, crank angle, and other sensors, provide a system, in conjunction with heat-of-compression, for controlling the charge pressure, density and temperatures, each separately, within the cylinder at the point and time desired for ignition. Intake valves 16-B and 16-A which can be hydraulically or electrically operated, as shown in FIG. 19, FIG. 19-B, FIG. 19-C and FIG. 19-D, managed by ECM-27 with appropriate activators, can also control the time and quantity of high and low pressure charge admitted to cylinders in order to aid in turbulence, density, pressure and temperature management. Alternatively, control of temperatures in the combustion chamber may be enhanced by the use of compression-expansion chilling of air and air-fuel charges, as depicted and described in and for FIG. 2-B, FIG. 2-C, FIG. 2D, FIG. 11-B, FIG. 18-B and FIG. 18-C. This system directs compressed air to expansion valve 10' pressure-drop-distributor 11, conduit or evaporator tube 27'*a*–27*f*, leading to inlet ports 18' and 8' and valves 16-B or 16-A of engine of FIG. 19, FIG. 19-B, FIG. 19-C and FIG. 19-D, as seen in FIG. 2-C and FIG. 2-B, respectively. Alternatively, the cool or chilled air may be directed to a chamber cooler system, consisting of either conduit 36-B, valve 40-B or conduit 42-B, both of FIG. 19-B, FIG. 19-C and FIG. 19-D, which conduits form cooling and heating jackets for combustion chamber 38', 38 and 38-B.

The pressure and density of the contents of cylinder 7 is adjusted to provide the torque and power required of the engine for the present duty cycle. A computer in the ECM-27 calculates the values received preferably from at least two sensors for each combustion chamber for monitoring temperature (21') and pressure (21"). It then computes the mean values in a single cylinder or for all cylinders of an engine and signals controls in the ECM to arrange the opening and closing of shutter and by-pass valves to adjust the density and temperature desired in each. In either engine of FIGS. 2-B, 2-C, 11-B, 18, 18-B, 18-C, 18-D, 19, 19-B, 19-C and 19-D, the optionally variable expansion valve 10' (see description of FIGS. 2-B, 2-C, FIGS. 11-B, 18-B, 18-C, 19-B and 19-C) and with control valves 3, 4, 5 and 6 and compressor speed all controlled by ECM-27, produces and maintains, in conjunction with heat-of-compression, the desired charge density and temperature at piston near or at TDC. Other sensors known and used in the art can monitor pressure oxygen, crank angle, etc. values and signal ECM-27 to adjust controls for best engine performance, with the ignition point preferably being 10–15 degrees before or at piston TDC. Closer temperature control may be achieved by utilizing compressed heated gases from conduit 36, FIG. 1-C or from conduit 118' of FIG. 2, with temperatures adjusted as described, being ducted through conduits 40, 40-B or 36 or 36-B in chambers 38' 38 or 38-B of FIGS. 19, 19-B, 19-C and 19-D. Also utilizing the cooling or expansion chilling or even heating of cylinder walls as described and depicted for FIG. 6 and FIG. 5 by use of expansion valves 4', FIG. 5 as well as expansion valve 10' and associated apparatus in FIGS. 2-B, 2-C, 11-B, 18-B, 18-C, 19-B, 19-C and 19-D. all controlled by ECM-27 and activators, all in conjunction with heat-of-compression, can assist in fine control of temperature at piston TDC. Heating elements 37, 39' in FIGS. 19, 19-B, 19-C and 19-D, also controlled by ECM-27, are used to regulate charge temperature at piston TDC.

In FIGS. 19-B, 19-C and 19-D, conduits 36-B and 41-B with their corresponding optional valves, 40-B and 42-B are alternatively, both or singularly, for injecting ignition enhancing pilot fluids (in which case conduits 36-B and/or 41-B open and end in combustion chamber 38-B or in preparation chamber 38) or one or both are conduits and jackets to receive and utilize fluids to assist in adjusting charge temperatures to facilitate spark, HCCI or modified HCCI engine operation.

Preparation

(1) The fuel is preferably mixed with high air content by carburation of one or both incoming air streams (supercharging secondary air stream for 2-stroke designs), as described herein, for the engines of the present invention (Operational Designs 1 through 13) and for Otto and Diesel Cycle engines.

(2) It can also have a well mixed air-fuel charge by central-body injection (similar to throttle-body injection) of fuel.

(3) The fuel may be injected into the air stream(s) before entering the cylinder 7.

(4) The fuel may be injected through the valve ports 18' and/or 8' with or without accompanying air.

(5) The fuel may be injected directly into the cylinder, where the air is already present with the tremendous turbulence created by the injection of the high pressure secondary air charge into the cylinder, causing instantaneous homogenization of the charge. The supplemental, secondary air charge injection systems of FIGS. 3, 4, 7, 9, 10, 18, 18-B and 18-C, which can be supplied alternatively through an expansion chiller valve 10' and accompanying chilling apparatus, can be utilized to cool and sweep out the space between piston and cylinder wall, especially above the top piston ring at piston TDC or perhaps all of piston ring crevices to reduce unburned hydrocarbons (HC) and prevent formation of formaldehyde emissions. A gentle blast of compressed air, alternatively through one or multiple ports 17-B of the Figures having such ports, could be used to prevent removing the oil-film from cylinder walls.

HCCI—Method A, Modified HCCI Operational Design 14, FIG. 19, FIG. 19-B Heated Pre-Combustion Chamber

Intake valve 16-B takes in a primary charge, piston 22 reverses direction and begins the compression stroke. Sometime during the compression (2^{nd}) stroke a supercharging fuel-air charge is injected by valve 16-A and compression continues. Now, with the air-fuel charge homogenized at the proper density and the temperature being near, but below auto ignition near piston TDC, the ignition system works in this manner: At near piston TDC, the charge is ignited by spark or for HCCI. Pre-combustion or preparation chamber 38' has an optional blocking valve 21 (FIG. 19 and FIG. 19-B) for HCCI, which separates chamber 38' from combustion chamber 38 and with piston 22 at or near TDC, blocking valve quickly opens. The air-fuel mix in combustion chamber 38 is instantly compressed into empty pre-combustion chamber 38', heating elements 37 and/or 39 immediately raise the temperature of the propulsion-fuel mixture above the auto-ignition temperature of the fuel-air mixture. It almost instantly combusts, expanding into combustion chamber 38, further mixing and catalyzing combustion of the entire fuel-air charge. This causes piston 22 (or rotor FIGS. 24–34) to be pressed into the power (3^{rd}) stroke which is followed by the scavenging (4^{th}) piston stroke.

For spark ignition, item 37 is a sparking plug and preparation chamber 38' and valve 21 are eliminated in all FIGS.

19–19-C and Methods 1–7. In spark ignition, the fuel-air charge is simply kept somewhere below auto-ignition temperature, making the system also feasible for diesel fuel since the charge can be kept below that which would cause auto-ignition (below 8:1 compression ratio temperature).

In Operational Design 1 to Design 6, as described for 4-stroke engines of FIGS. **1** through **7** engines, and for 2-stroke engines of FIG. **8** through FIG. **10**, the four principal Operational Designs are here reiterated briefly:

Design 1: Intake valving and combustion chamber are as normal for a normal compression ratio with a normal expansion ratio.

Design 2: Intake valving is as normal, but the combustion chamber is enlarged for a subnormal compression ratio, but still with an expansion ratio equal to the compression ratio.

Design 3: The intake or exhaust valve closes early or late to capture a small air-fuel charge for a low "effective" compression ratio and an extended expansion ratio.

Design 4: The exhaust valve stays open during the compression stroke to near piston TDC, expelling most of the fresh primary charge from cylinder **7** or **22** (FIGS. **2–10**), which is to cool both chambers **38'** and/or **38** and to produce an "effective" compression ratio of perhaps 2:1 or less with an extended expansion ratio. At near piston TDC, the entire working charge is injected. This produces an "effective" ratio of 2:1 to zero.

In all cases, a supplementary cool, dense air charge is injected, preferably late, during the compression (2^{nd}) stroke. At piston TDC, the charge is ignited for the power (3^{rd}) stroke, followed by the scavenging (4^{th}) stroke to complete a power cycle.

Method A, Detailed Operation

Piston **22** has just completed the scavenging (4^{th}) stroke and is at TDC. Intake valve **16**-B opens as piston **22** begins the intake (1^{st}) stroke, taking in our atmospheric or pressure boosted and cooled air or air-fuel charge. Piston **22** reverses and with insulated valve **21** closed to temporarily prevent flammable air-fuel propulsion charge entering pre-combustion chamber **38'** prematurely, piston **22** begins and continues the compression (2^{nd}) stroke toward TDC. (The primary air-fuel charge would optionally produce a sub-standard or sub-normal compression ratio charge for low heat-of-compression for that portion of the charge).

During the compression (2^{nd}) stroke a supplemental, supercharging, highly compressed, dense, temperature-adjusted air charge is injected into cylinder **7** above piston **22**. The supplemental, secondary air charge can be added at any point after the port intake valve **16**-B closes on the intake (1^{st}) stroke and can be as late as during charge combustion. The supplemental, secondary charge can be injected at such point in the compression stroke that it will also have a very low compression ratio in all Operational Designs, again for very low heat-of-compression. In addition, the supplemental, secondary air charge can be either a major or minor portion of the entire air-fuel charge versus the primary charge which has the same capabilities. (Sensors in chamber **38'** have signaled ECM-**27** information which it uses to use control valves **3**, **4**, **5** and **6** and optionally valve **10'**, including the heat-of-compression to adjust charge temperatures near piston TDC to be just below auto-ignition temperature. Compression continues until piston is very near TDC. At that point, blocking valve **21** opens quickly and receives the compressed fuel-air propulsion charge being compressed into chamber **38'**. Heating elements **37** and/or **39** plus the heat-of-compression raise the temperature of the fuel-air mix above the auto-ignition of the fuel, causing the charge to begin oxidation. (If the temperature in pre-combustion can be controlled sufficiently, blocking valve **21** may be eliminated.) The hot swirling charge expands past valve **21** into combustion chamber **38**, catalyzing the rapid oxidation of the entire fuel-air charge. The expanding charge depresses piston **22**, or rotor **2** or **2**-B, into the power (3^{rd}) stroke. At BDC, piston **22** reverses and scavenges the cylinder with (4^{th}) stroke in preparation for another working cycle. This completes one power cycle of the engine.

The preferred method of operation is for the primary air charge to be boosted in pressure. All Operational Designs, except Design 1 and Design 2, have selectively extended expansion ratios, with low effective compression ratios. When operating in HCCI methods, valve **21**, opening to preparation chamber **38'**, may stay open through the piston power (3^{rd}) stroke, the exhaust (4^{th}) stroke and the piston (1^{st}) stroke, closing at the start of the start of the compression (2^{nd}) stroke. Should proper HCCI temperatures be controllable or spark ignition be employed, valve **21** may be eliminated.

Note that in Operational Design 4, the cool primary charge is expelled through valve **16**-C (not shown) which can open into chamber **38** or **38'** or through exhaust valve **17** and is used only to cool chambers **38** and **38'** after which the expelled charge is returned, if fuel is present, to suction conduit **110** of compressor **2** or to suction port **8'** of FIG. **2**. If Operational Design 4, as described for engine of FIG. **2**, FIG. **2**-B, FIG. **2**-C, FIG. **8**-FIG. **10** or FIGS. **24–32**, is utilized, the primary air charge may be free of fuel since that air is used only for cooling cylinder **7** or **12**, preparation chamber **38'** and/or combustion chamber **38** or **38**-B, the latter of FIG. **19**-C or FIG. **19**-D. As the cooling charge without fuel is pressed through outlet valve **16**-C (not shown), or through valve **40** or **42** of FIG. **19**-C, it is expelled into the engine exhaust or into the atmosphere.

HCCI—Method B, Operational Design 15 Hot, Non-Flamable Gas Injection

An initial air-fuel propulsion charge is prepared and drawn into cylinder **7** on the intake (1^{st}) stroke and is compressed by piston **22** in the second stroke after piston turnaround. The temperature of a supplemental supercharging air charge plus heat-of-compression has been adjusted to produce at piston TDC somewhat less than auto-ignition temperature for (a) spark ignition and only slightly less than auto-ignition temperature at near TDC of piston **22** for (b) modified HCCI operation, as in Method A. At any point found to be most appropriate during the compression (2^{nd}) stroke, intake valve **16**-A opens and closes quickly, injecting the supplemental temperature-adjusted air-fuel charge. Inlet valve **40** restrains hot compressed gas (from conduit **36** of FIG. **1**-C and/or from conduit **118'** of FIG. **2** through FIG. **2**-D, FIGS. **11**, **11**-B, FIGS. **17**, **18**, **18**-B and **18**-C) to insulated conduit **36**, of FIGS. **19**-B, **19**-C or **19**-D, which leads to preparation chamber **38'** or combustion chamber **38** or **38**-B. Heating elements **37** and **39** within preparation chamber **38'** or **38**-B can optionally be present and heated in order to help manage charge temperature with heat and other sensors within chamber **38'** or **38**-B and controlled by ECM-**27**. The compression (2^{nd}) stoke of piston **22** nears completion and the propulsion air-fuel charge is now compressed above piston **22** and fills combustion chamber **38** and/or preparation chamber **38'**.

Near TDC of piston **22**, optional valve **40** or **40**-B, if present, opens and closes quickly. The charge of hot, highly

compressed gas from conduit **36** or **36**-B, which can be part of (a) the engines secondary highly compressed air charge from conduit **109**, FIG. **2**, with part or all of the heat of compression retained, or the charge may be further heated by heater **116**' in FIGS. **2**, **2**-B, **2**-C, **11**, **11**-B, **18**, **18**-B and **18**-C or (b) recycled hot exhaust gas, or a mixture thereof from conduit **203***i* of FIG. **1**-C or other nonflammable gases. This hot, compressed gas, which has been optionally super-heated by heating elements **37** and/or **39**, perhaps assisted by compressor **2**' and by heater **108** in FIG. **1**-C, blasts into the preparation chamber **38**', mixing and instantly catalyzing the ignition by raising the temperature of the swirling charge above the auto-ignition temperature of the propulsion fuel. The ignited charge now expands into combustion chamber **38** where the homogenized air-fuel charge is still mixing from the injection by valve **16**-A. With the ignited charge being expanded from chamber **38**' and mixed with the rest of the charge in combustion chamber **38**, combustion will be broadly spontaneous and more complete with very low unburned hydrocarbons (HC) and low CO, CO_2, NO_X and particulates. As mentioned earlier, part of the high pressure supplemental, secondary air charge, sans fuel, may be directed to conduit **36** FIG. **19**, FIG. **19**-B, FIG. **19**-C and FIG. **19**-D from conduit **109** FIG. **2** to FIG. **2**-D by way of conduit **111**' to compressor **114**' and heater/cooler **116**' (both optional), and/or, wholly or partly, through bypass systems **112**' and **113**', **115**' and **117**'. It is then directed to conduit **118**' which leads to charge distributor (similar to **11**' of FIG. **11**-B) where it divides and conveys part or all of the hot air charge, or combined with hot effluent from conduit **203***i*, FIG. **1**-C, to conduit **36**, FIG. **19** or **36**-B of FIG. **19**-B or FIG. **19**-C of each of the engines cylinders, in order to catalyze ignition of the propulsion charge near TDC.

Alternatively, for non-flammable gas, valve **40** on conduit **36** of FIG. **13**, FIG. **19**, FIG. **19**-B and FIG. **19**-C, which can be close to pre-combustion chamber **38**', can be kept closed to restrain the hot gas until piston **22** nears TDC in the compression stroke, preferably at 10–15 degrees before piston TDC. At that point, valve **40** is opened to inject a small amount of the hot gas, which originates from conduit **203***i*, conduit **109**, FIG. **2**, FIG. **2**-B or FIG. **2**-C, through conduits **111**' and optional compressor **114**' and heater **116**' and conduit **118**' or from any other source, to enter and mix with the small amount of air-fuel within the pre-combustion chamber **38**'. The hot gas instantly raises the temperature of the air-fuel mix in the pre-combustion chamber **38**' catalyzing the ignition whereby the burning mixture expands into the combustion chamber **38**, catalyzing and further oxidizing the entire charge.

Alternatively, the gases can originate from conduit **203***i*, as in FIG. **1**-C from exhaust, from a mixture of hot exhaust and air or any other appropriate gas. The gas is compressed by compressor **2**', if necessary, or any other type compressor. If the gas is not hot enough, it is heated by passing the gas through heater **108** (**108** can optionally be a cooler if charge is too hot) or bypassed, wholly or in part, through bypass system **203***h*' and **203***h*, FIG. **1**-C. The hot gases can be mixed with engine fuel within pre-combustion chamber **38**', ignited and expanded into combustion chamber **38** as described in number **3** above.

Direct Combustion Chamber Ignition

Alternatively, pre-combustion chamber **38**' is eliminated in this system and for HCCI. Hot gases are now directed through insulated conduit **203***i*, FIG. **1**-C (or through conduit **118**', FIGS. **2**-**2**-D) which is fitted with an effluent distributor (not shown), of which a distributor tube goes directly to, and can intrude into combustion chamber **38** or **38**-B, FIGS. **19**, **19**-B, **19**-C and **19**-D of each power cylinder of the engine. A valve **40** on conduit **36** opens and closes quickly at perhaps 10–15 degrees, before piston TDC, to inject the compressed hot gases directly into the turbulent air-fuel charge in combustion chamber **38**. As stated, these hot gases can be a portion of the engines high-pressure supplemental, secondary air charge (without fuel) with the heat-of-compression retained and heated or cooled, if needed, or they can be re-circulated hot or heated exhaust gases, heated air or a mixture thereof from conduit **36**, FIG. **19**, FIG. **19**-B and FIG. **19**-C. When the hot gases enter combustion chamber **38**, they disperse throughout the charge, raising the temperature of the fuel-air charge above the auto-ignition temperature of the charge, catalyzing the ignition of the charge which then expands driving the piston. Any compressed hot, flammable or non-flammable gas would be a good catalyst for low auto-ignition fuels, such as gasoline or diesel fuel. If diesel propulsion fuel were used, the New Cycle Engines of this invention would compact the air-fuel charge to a far greater density than would a current diesel engine with the air-fuel charge temperature being lower than the auto-ignition temperature of diesel oils-air mixture. (Diesel oil-air mix would auto-ignite at a compression ratio of about 8:1 in any normal diesel or current technology HCCI engine, reducing power and efficiency. As already stated, compressed natural gas, with or without air, should be a good pilot fuel for diesel oil fuels, but preferably without being mixed with air.

Optionally, the hot gases may be injected into either preparation chamber **38**' or combustion chamber **38** from two or more injection lines. For example, conduit **41**-B of FIG. **19**-B and valve **42**-B and other entrance systems may be utilized to surround the propulsion fuel to initiate oxidation, especially useful in FIG. **19**-B, FIG. **19**-C and FIG. **19**-D arrangement.

(a) In all options, ignition occurs and the power (3^{rd}) stroke now takes place.

(b) At the end of the power stroke, piston **22** rises in the scavenging (4^{th}) stroke to scavenge the cylinder **7**.

(c) Another cycle begins.

(d) In all options, the engine's fuel-air charge is near auto-ignition at near piston TDC before ignition is catalyzed. For optional spark ignition, low-auto-ignition fuels need only to be below auto-ignition temperatures before spark ignition.

The described method for producing extremely dense charge while suppressing temperature to below auto-ignition temperatures allows spark ignition for diesel oils without smoke and other pollutants. This is because smoke in Diesel engines is caused by pre-mixed burning and in this system the air and fuel are a homogeneous mixture. The fuel-air mixture is ignited by (a) spark plug **37** in pre-combustion chamber **38**' as in FIG. **19**, FIG. **19**-B and FIG. **19**-C or (b) the plugs can be used with the electrodes exposed directly to the combustion chamber **38** or **38**-B as in conventional, spark ignited engines and is shown if FIG. **19**, FIG. **19**-B, FIG. **19**-C and FIG. **19**-D. In each instance, spark plugs are optional and for an emergency backup system except where spark ignition is more desirable.

A good system of supplying hot gases for catalyzing ignition in any of the engines of this invention is depicted in FIG. **1**-C. Various proportioning valves operated and controlled by ECM-**27** remain closed or open, fully or partly, according to desired results, to take either fresh air from air intake **9** or re-circulated exhaust gas from exhaust conduit **18** or any percentage (0 to 100 percent) ad infinitum of each

for conveying to conduit **36** or **36**-B of pre-combustion chamber **38**' or combustion chamber **38** respectively, for catalyzing charge ignition. The ensuing mix or pure gas is taken through conduits **203***d* and **203***e* and passed through proportioning valve **203***f* to pass, wholly or partly, through compressor **2**' or, wholly or partly, through bypass conduit **203***h*' to proportioning valve **203***g*' and caused to pass wholly or partly through heater/cooler **108** or pass, wholly or partly, through bypass conduit **203***h*, both routes leading to conduit **203***i* which in turn leads to effluent distributor and distributor tubes (not shown) leading to conduit **36** or **36**-B of each of the cylinders of the engine design being selected. Other suitable gases, such as natural gas (perhaps CNG) can be directed into conduit **111** of FIG. **2** or conduits **203***d* and **203***e*, FIG. **1**-C either optionally or directing to conduit **36** or **36**-B of FIG. **19** and FIG. **19**-B respectively. The various valves are controlled by ECM-**27** and are adjusted to provide the gas with the concentration and temperature desired for proper ignition catalyzation, the various conditions of which are related to ECM by proper sensors throughout the system.

Proper sensors in each engine cylinder inform ECM-**27** of the conditions in individual cylinders and ECM-**27** makes adjustments, such as temperatures and crank angles, for timely ignition in each.

Further instruction for engine operation for HCCI Method B is found under HCCI Method A with references to the advantages of the various engine designs. The HCCI ignition system only has been described above. The engine working cycle is described in HCCI Method A, utilizing any of Operational Designs 1 through 13.

HCCI operation of the engines of this invention optionally produces an extended expansion ratio in Operational Design 3 through Operational Design 13 for improved fuel economy and low polluting emissions with great power. Because of the low temperature working cycle and homogenized air-fuel charge, all thirteen Operational Designs produce extremely low polluting emissions.

The advantages of this new engine cycle are:

Higher efficiency.

Greater power and torque.

Longer engine life.

Multi-fuel capabilities.

Greater reliability.

Method C—Modified HCCI, Operational Design 16 Flammable Fluid (Gas or Liquid) Pilot Fuel

The air intake (1^{st}) stroke, the compression (2^{nd}) stroke has occurred and the supplementary air-fuel charge has been injected. With the air-fuel propulsion charge homogenized and the temperature of the mix when compressed at near piston TDC being adjusted to near auto-ignition temperature, with combustion chamber **38**, **38**-B and preparation chamber **38**' or **38**'B filled with propulsion fuel above piston **22**, the system works in this manner: The fuel air charge is ignited by (a) spark as in conventional engines and (b) for HCCI operation. Conduit **41** with fuel injector **42** (both in phantom), receives and contains pilot fuel or air-fuel mix, perhaps from a common fuel-rail, which can be any liquid or flammable gaseous product, such as CNG-air mix, dimethyl ether-air mix or preferably CNG or pilot fuel or dimethyl ether alone. All would be appropriate as pilot fuel for diesel-air propulsion fuel, as all have a higher auto-ignition temperature than does diesel oil-air mix. Both natural gas and dimethyl ether can be heated to an even higher temperature, if injected devoid of air.

At near completion of compression (2^{nd}) stroke and near TDC of piston (preferably 10–15 degrees before), a small amount of heated pilot fuel in line **41**, with temperature adjusted to above auto-ignition temperature of the propulsion fuel-air mix, but below auto-ignition of the pilot fuel, is injected by fuel injection pump **42** into preparation chamber **38**', the fuel injected having been further heated, if needed, by heater **37** and/or **39**' or by a heater of any type. (such as **108** in FIG. **1**-C and **116**' in FIGS. **2** through **2**-D and FIGS. **17**, **18**, **18**-B and **18**-C). This higher temperature pilot fuel then ignites the small quantity of propulsion fuel by raising its temperature in chamber **38**' which then, in turn, ignites the pilot fuel and together expands into combustion chamber **38**, raising the temperature of the air-fuel propulsion mix, instantly catalyzing the reaction by quickly raising the temperature to above auto-ignition level. In the compact combustion chamber, with the fuel-air premixed (homogenized) and with the turbulence of the new engine system, the combustion will be far more complete than in a compression ignited (CI) engine where much of the fuel burns prior to mixing. The expanding charge drives piston **22** in the power (3^{rd}) stroke which is followed by the scavenging (4^{th}) stroke of piston **22**.

An ideal hot flammable gas for igniting low ignition fuels would appear to be natural gas, sans air, which can be heated to temperatures higher than the auto-ignition temperature of pure gasoline or diesel oils or other low auto-ignition fuels. When injected into gasoline or diesel oil and air mixtures, the higher temperature natural gas will ignite the lower auto-igniting gasoline or diesel oil and air mixture, which in turn will ignite the natural gas pilot fuel, supplying any oxygen needed for burning the pilot fuel. The pilot fuel can always be heated to a higher temperature if it is not mixed with air before injecting into propulsion fuel. (In this working cycle, the temperature of the propulsion fuel-air mixture in the combustion chamber at TDC will be lower than its auto-ignition temperature at the time of the hot pilot fuel injection.)

If the pilot fuel is a flammable hot gas, it originates from a storage tank or supply pipe line, item **1** and **1***a*, FIGS. **20** or from conduit **43** of FIG. **21**. From there, it proceeds as suggested in method B for non-flammable gas. The flammable gas, which is hotter than the propulsion mix, catalyzes the ignition of the propulsion gas which, in turn, catalyzes ignition of the pilot gas which has a higher auto-ignition temperature.

Direct Combustion Chamber Ignition

Alternatively, with piston **22** near TDC, with preparation chamber eliminated, the charge is ignited by spark or alternatively, conduit **41** is extended directly into and can intrude fully and directly into combustion chamber **38**. Then at the proper moment, a charge of the pure pilot fuel or air-fuel mix in line **41** is injected optionally by injector **42** into the combustion chamber **38** and mixed with the air-fuel propulsion mix within the chamber **38**. Ignition is catalyzed by temperature increase by the higher temperature of the higher auto-ignition temperature pilot fuel, catalyzing ignition of the lower auto-ignition temperature propulsion fuel, (pilot fuel could be dimethyl ether or natural gas (CNG) for diesel or gasoline fueled engines), whereupon the propulsion fuel ignites the pilot fuel and this combusting charge expands to drive piston **22** in the power stroke. The scavenging stroke then occurs, completing a power cycle.

Alternatively, or in addition, for either pre-combustion chamber **38**' or for combustion chamber **38**, **38**-B injection systems other than conduit **41** and injector pump **42** are employed. For example, conduit **36**, **36**-B and injector **40** or