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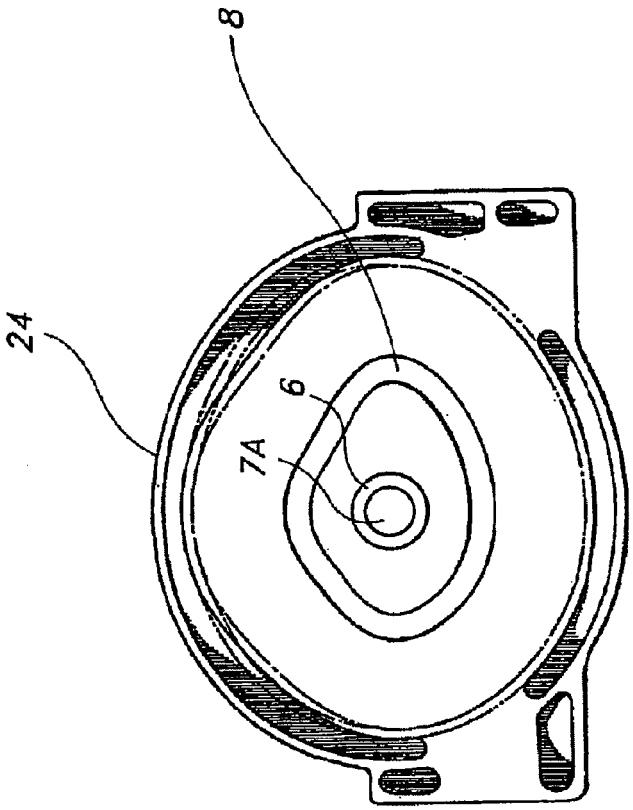


FIG 32

COLD AIR SUPER-CHARGED INTERNAL COMBUSTION ENGINE, WORKING CYCLE AND METHOD

CROSS-REFERENCE TO RELATED APPLICATIONS

This application is a continuation-in-part of application Ser. No. 10/206,847, filed Jul. 26, 2002; which is a continuation-in-part of application Ser. No. 09/632,739, filed on Aug. 4, 2000; which is a continuation of application Ser. No. 08/863,103, filed May 23, 1997, now U.S. Pat. No. 6,279,550; which is a continuation-in-part of application Ser. No. 08/841,488, filed on Apr. 23, 1997 (now abandoned); which application claims the benefit of provisional application Nos. 60/022,102, filed on Jul., 17, 1996; 60/023,460, filed on Aug. 6, 1996; 60/029,260, filed on Oct. 25, 1996; and 60/040,630, filed on Mar. 7, 1997; which is a continuation-in-part of Ser. No. 09/757,772, filed on Jan. 9, 2001 (now abandoned); which application claims the benefit of provisional application No. 60/175,597, filed on Jan. 11, 2000; which is a continuation-in-part of application Ser. No. 10/029,224, filed on Dec. 20,2001 (now abandoned); which application claims the benefit of provisional application Nos. 60/259,727, filed on Jan. 4, 2001; 60/278,030, filed on Mar. 22, 2001; 60/284,711, filed on Apr. 18, 2001; 60/296, 620, filed on Jun. 7, 2001; 60/297,663, filed on Jun. 12, 2001; 60/340,728, filed on Oct. 22, 2001; 60/340,609, filed on Dec. 14, 2001; and 60/374,392, filed Apr. 22, 2002.

The present invention constitutes a new working cycle for internal combustion engines, with methods and apparatuses for managing combustion charge density, temperatures, pressures and turbulence (among other characteristics). At least one embodiment describes a cold air super-charged internal combustion engine in which a super-charging portion of air is compressed, cooled or chilled and injected late in the compression process.

Initial air charge for 4-stroke engines and for epitrochoid (Wankel) type rotary and straight rotary engines: A sub-normal compression ratio or a low "effective" compression ratio initial air charge (A) is received on the intake process, which during the compression process, produces only a fraction of the heat-of-compression as that produced by a conventional engine.

Super-charging air: During the compression process, a dense, cooled or chilled supplementary air charge (B) is injected into the engine, adding density and turbulence above that of conventional engines and producing a low compression ratio or a low "effective" compression ratio for this portion of air charge also. Compression of the combined charge continues and near piston or rotor top dead center (TDC) position, the air charge, being mixed with fuel, is ignited for the power pulse, followed by scavenging, air intake and compression process to complete one power cycle.

For 2-stroke engines: In 2-stroke engines, reciprocating, epitrochoid-action or straight rotary, the cold supplementary air charge (B) is injected during the compression process after a low compression ratio engine or a low "effective" compression ratio process engine has been scavenged and charged with fresh air (A). Compression continues and near piston or rotor top dead center (TDC) position, the fuel-air charge is ignited, producing the power pulse, scavenging, fresh air charging and compression process, thus completing one power cycle.

The charge density is selectively much greater than that of state-of-art engines, providing greater power and torque with increased efficiency, while producing ultra low polluting emissions.

BACKGROUND OF INVENTION

It is well known that as the expansion ratio, as compared to charge density of an internal combustion engine is increased, more energy is extracted from the combustion gases and converted to kinetic energy, increasing the thermodynamic efficiency of the engine. It is further understood that increasing air charge density increases both power and fuel economy due to further thermodynamic improvements. The objectives for an efficient engine are to provide a high-density charge, to begin combustion at maximum density and then to expand the gases as far as possible against a piston, vane or rotor lobe.

It is further known that if the air-fuel charge of an engine is homogeneous, the elimination of cycle-to-cycle variations in components, (pressures, air-fuel ratios, temperatures, etc.), will greatly improve engine performance.

It is also well known that if an air-fuel mixture has a high air content, and because fuel is fully mixed with air before combustion, the fuel bums cooler bringing the bum temperature below the threshold at which it would damage the pistons.

It is also an established principle that the lower the temperature of a standard density charge is at the time of ignition, the lower the comparative peak temperature and pressure produced which further reduces polluting emissions, especially NO_x.

Now, the compression ratio of current engine technology sets the level to which a piston "compacts" a mix of air and fuel before the charge is ignited and hence sets the limits of power and efficiency. If the ratio is too high in gasoline or gas engines, the fuel ignites itself as it does in a Diesel engine. Although efficiency is high with a Diesel engine, polluting emissions, including particulates (smoke and nano-particles, all caused by pre-mixed fuel-bum) make it a health hazard.

The engine of this invention is capable of producing the same "compactness" of charge and an even denser charge than that of the Diesel engine, but without having a high compression ratio or producing excessively high temperature. The charge is introduced and managed in such a manner that regardless of the charge density, the temperature of the charge can be kept below auto-ignition temperature of the fuel until the point for ignition. The homogeneous charge then bums cooler and more completely for truly clean power production.

Described herein is a unique engine, which, within its combustion chambers, produces a lighter than normal or denser than normal, temperature-adjusted homogeneous air-fuel mixture. The temperature of the charge is so managed that at 10–15 degrees before piston top dead center (TDC), the charge can be very low and optionally below auto-ignition temperature, and it is at that point that the charge is ignited by spark, heat infusion or heat profusion or by catalyst for high torque, power and efficiency with ultra low polluting emissions.

Gasoline and Natural Gas Auto-Ignition Engine Concept Under Development by Lawrence Livermore National Laboratories (LLNL), et al.

Discussion of a Novel Engine Concept

HCCI is an abbreviation of "Homogeneous Charge Compression Ignition." Scientists have discovered that at some engine speeds and loads, the combustion in internal combustion engines became more stable and that an engine ran

smoother when the fuel was auto-ignited, instead of being ignited by sparkplug. As the name implies, the homogeneous ("well mixed") charge of air and fuel is ignited by compression heat.

Homogeneous Charge Compression Ignition, or HCCI, is a relatively new combustion technology. It is a hybrid of the traditional spark ignition (SI) and the compression ignition process (such as the Diesel engine). A project at the UC Berkeley Combustion Analysis Laboratory (CAL) is to focus on the experimental side of HCCI. Data collected there is being used to validate the computational models generated by engineers at Lawrence Livermore National Lab (LLNL). Entec Engine Corporation, Atlanta, Ga. is an integral part of the continuing effort to make practical an HCCI engine to provide low emission, high efficiency power generation as well as for transportation.

Unlike a traditional SI or Diesel engine, true HCCI combustion takes place spontaneously and homogeneously without flame propagation. This eliminates heterogeneous air/fuel mixture regions. In addition, HCCI is a lean combustion process. These conditions translate to a lower local flame temperature which lowers the amount of NO_X produced in the process. NO_X is a gas that is believed to be responsible for the creation of ozone (O3).

The Principal of the HCCI Engine Concept

As mentioned above, the HCCI-engine can be seen as a hybrid of the SI-engine and the Diesel engine. First, we will describe the SI and Diesel engines.

In the SI-engine, a homogeneous mixture of fuel and air is ignited at the end of the compression stroke by a spark. The spark causes a flame kernel that grows and propagates throughout the combustion chamber. By controlling the mixture flow to the engine with a throttle plate, the engine load (torque) is changed. The mixture ratio between air and fuel is kept almost constant at all loads.

In the Diesel engine, pure air is compressed. The fuel is injected under high pressure at the end of the compression stroke into the hot compressed air. The fuel is vaporized and mixed partially before self-ignition occurs. The load is adjusted by varying the amount of fuel injected.

In the HCCI-engine, homogeneous air-fuel mixture is compressed so that auto-ignition occurs when the piston is near the top dead center position (TDC). A high compression ratio is necessary in order to ensure auto-ignition. Very lean mixtures have to be used in order to get slow chemistry that reduces the combustion rate. Diluted mixtures can be achieved by using a high air-fuel ratio or by Exhaust Gas Recycling (EGR). Varying the amount of fuel controls the load. Like the Diesel engine, there is no throttle plate i.e., the engine will always get maximum amount of air flow as-the engine is un-throttled.

Advantages of the HCCI Engine

The HCCI-engine is always un-throttled. A high compression ratio is used and the combustion is fast. This provides high efficiency at low loads, compared to a SI-engine that has low efficiency at part load. According to computational models generated by engineers at LLNL, an HCCI engine will reduce fuel consumption to one half of that in an ordinary automobile engine. The formation of nitrogen-oxides is strongly dependent on combustion temperature. Higher temperature gives higher amounts of NO_X. Therefore, since the combustion charge is homogeneous and a very lean mixture is used, the combustion temperature becomes very low, which results in very low amounts of NO_X. LLNL predicts that the HCCI engine will also virtually eliminate soot (particulates) and NO_X.

Disadvantages of the Current HCCI Engine Concept as Being Developed by LLNL, et al.

The control of the combustion is more difficult in the HCCI engine than in the SI or Diesel engines. The HCCI engine provides no direct control of the start of combustion. In other words, there is no "triggering" mechanism as a spark or an injection of fuels into a super-hot air-charge, as in conventional engines. The start of combustion depends on several parameters. The strongest ones are the compression ratio and the air charge inlet temperature. Also, there is no means of separately controlling charge density and temperature. By adjusting these parameters in "the right way," it is possible to control the start of combustion to a desired moment. Another current disadvantage in their concept is high levels of hydrocarbons (HC), which is unburned fuel. The low combustion temperature causes the fuel to burn incompletely.

Potential of the Current HCCI Engine Concept

An appropriate field of operation is power plants where the engines operate with constant speed. Present concepts of the HCCI-engine would compete favorably with natural gas driven SI-engines due to the higher efficiency and lower NO_X emissions. One interesting concept would be to use HCCI combustion at part load conditions and SI combustion at high loads in a car engine. In this way, the fuel consumption would be reduced significantly. Researchers say if the emissions standards should rise and the problem with the HC emissions could be solved, the HCCI engine would be able to compete favorably with the Diesel engine, since the Diesel combustion causes high NO_X, soot particulates and other emissions.

Why HCCI?

The modern conventional SI engine, fitted with a three-way catalyst, can be seen as a very clean engine, but it suffers from poor part load efficiency. As mentioned earlier, this is mainly due to the throttling. Engines in passenger cars operate most of the time at light and part load conditions. For some shorter periods of time, at overtaking and acceleration, they run at high loads, but they seldom run at high loads for any long periods. This means that the overall efficiency at normal driving conditions becomes very low.

The Diesel engine has a much higher part load efficiency than the SI engine. Instead, the Diesel engine fights with great smoke and NO_X problems. Soot (particulates) is mainly formed in the fuel rich regions and NO_X in the hot stoichiometric regions. Due to these mechanisms, it is difficult to reduce both smoke and NO_X simultaneously through combustion improvement. Today, there is no well working exhaust after treatment that takes away both soot and NO_X.

The HCCI engine has much higher part load efficiency than the SI engine and is comparable to the Diesel engine. It has no problem with NO_X and soot (particulates) formation like the Diesel engine. In summary, the HCCI-engine beats the SI engine regarding the efficiency and the Diesel engine regarding the emissions.

The foregoing discussion of the anticipated operating features show that when the HCCI system is successfully developed, according to the present plans of various labs and

engine research groups, it will have a low-density air charge for low efficiency and power for diesel fuel, but will be clean burning and have the efficiency of the Diesel engine if using gasoline or natural gas fuel.

Advantages of the Engine of this Present Invention, the Gasoline, Natural Gas, Hydrogen & New Diesel Fueled, Spark or Auto-Ignition Engine

The present invention described herein promises to make practical the HCCI-engine and provides all of the hoped for advantages stated for the futuristic HCCI-engine concept. In addition, the present invention provides many other engine improvements not anticipated for the HCCI-engine e.g., greater power and efficiency for diesel operation, whether ignited by spark or as an HCCI engine.

The engine of this present invention provides a homogenized air-fuel charge, which can always be much denser than that of the Diesel engine and, in some designs, has an extended expansion ratio, yet with compression temperatures lower than those which would cause auto-ignition prematurely. It can be ignited by spark, compression, modified HCCI or true HCCI. Other advantages are no-knock operation, multi-fuel capabilities, double burning "dwell" time near TDC with improved power, torque, efficiency and durability and greatly reduced NO_x, CO, CO_2, formaldehyde, HC and particulates. Knock-free operation allows reduction or elimination of the gasoline additive methyl tertiary-butyl ether (MTBE) which is currently polluting ground water.

The present invention describes a means of selectively managing the density, pressure and temperature of an air or air-fuel charge with each feature being separately adjustable with regard to the other features. This selective management provides the same reduction in NO_x, particulates and other polluting emissions as projected for HCCI, while providing greater power, efficiency and engine durability.

The density of the final air-fuel charge can be much greater even than that of the Diesel engine and yet has a temperature low enough to prevent knock or pre-auto-ignition. For example, natural gas-air mix auto-ignites at a compression ratio of 19:1 or 20:1, while diesel fuel-air mix auto-ignites at the temperature produced by a compression ratio of about 8:1. Therefore, using diesel fuel in a HCCI engine, under development by LLNL, et al., would greatly reduce its efficiency and power. But, in the present invention, the charge can be "compacted" to an effective compression ratio of as much as 40:1 or greater, with a temperature lower than that normally produced by an 8:1 compression ratio. The engine of this invention can then ignite the charge for greater than "Diesel" power and efficiency.

Currently, according to researchers, the greatest problems in making the HCCI engine feasible are timing of ignition and the problem of temperature control in regard to compression ratio. In their HCCI engine concept, the temperature of the compressed charge can not be kept low, thus severely limiting power and efficiency in HCCI for all fuels. The present invention allows the control of the temperature and density of the charge separately, for much greater steady-state power density and fuel economy, with ultra-low emissions and timely ignition control for all fuels, liquid or gaseous.

Described herein are several alternate means of timely igniting the charge of the present invented engine e.g., a) spark ignited, b) compression ignited, c) by modified HCCI, and d) by true HCCI operation.

SUMMARY OF THE INVENTION

Briefly described, the present invention comprises an internal combustion engine system, (including methods and apparatuses) optionally operating in a first mode and, optionally and selectively, in a second mode and optionally in a preferred third mode for managing combustion charge densities, temperatures, pressures and turbulence in order to produce a true mastery within the power cylinder to increase fuel economy, power, and torque while minimizing polluting emissions. In its preferred embodiments, the method includes the steps of (i) producing a primary air charge and optionally, a secondary air charge (ii) controlling the temperature, density and pressure of the air charges, (iii) transferring an air charge to a power cylinder of the engine in one stage for mode 1 operation and in two stages for mode 2 and mode 3 operation. The primary air charge is supplemented by the secondary air charge in producing mode 2 and mode 3 operation. This system is such that an air charge having a weight and density selected from a range of weight and density levels, ranging from less than atmospheric weight and density to a heavier-than-atmospheric weight and density, is introduced into the power cylinder, and (iv) then compressing the air charge at a normal, or a lower-than-normal compression ratio, (v) optionally introducing a secondary air or air-fuel charge into the same cylinder during the compression stroke, (vi) compressing the total charge, (vii) causing a pre-determined quantity of charge-air and fuel to produce a combustible mixture, (viii) causing the mixture to be ignited within the power cylinder and (ix) allowing the combustion gas to expand against a piston operable in the power cylinder with the expansion ratio of the power cylinder being equal to, or substantially greater than the compression ratio of the power cylinders of the engine. In addition to other advantages, the invented method is capable of producing mean effective [cylinder] pressures (mep) in a range ranging from lower-than-normal to higher-than-normal. In the preferred embodiments, the mean effective cylinder pressure is selectively variable (and selectively varied) throughout the mentioned range during the operation of the engine. In an alternate embodiment related to constant speed operation, the mean effective cylinder pressure is selected from the range and the engine is configured, in accordance with the present invention, so that the mean effective cylinder pressure range is limited, being varied only in the amount required for producing the power, torque and RPM of the duty cycle for which the engine is designed.

In its preferred embodiments, the apparatus of the present invention provides a reciprocating internal combustion engine with at least one atmospheric air intake port for providing charge air to two different cylinder inlet ports at different pressures. The apparatus also provides at least one ancillary compressor for pre-compressing a portion of the air charge, an intercooler, or heat exchanger, through or around which, the compressed air can be directed for temperature adjustment. The apparatus further provides power cylinders in which the combustion gas is ignited and expanded, a piston operable in each power cylinder, connected to a crankshaft by a connecting link for rotating the crankshaft in response to reciprocation of each piston and a transfer conduit communicating the compressor outlet to a control valve and to the intercooler or heat exchanger. Also provided is a transfer manifold communicating the intercooler, or bypass system, with the power cylinders, through which manifold the compressed charge is transferred to enter the power cylinders. The apparatus further provides one intake port or valve controlling admission of the low pressure (primary) air charge to the power cylinder, an intake valve

controlling admission of the compressed (secondary) charge from the transfer manifold to said power cylinders, and an exhaust valve controlling discharge of the exhaust gases from said power cylinders.

For the 4-stroke engine of this invention, one of two intake valves of the power cylinders is timed to operate so that a charge of air, which is lighter, equal to, or heavier than normal, can be received and maintained within a transfer manifold, when required. This charge of air can then be introduced into the power cylinder during the intake (primary) piston stroke with the intake valve closing at piston BDC, or closing substantially before piston BDC position or with the intake valve closing at some point after BDC, or near the end of the compression stroke to provide, in a first mode of operation, either a normal, a low, or a zero compression ratio. This establishes that the expansion ratio can be equal to or greater than the compression ratio of the power cylinders. In this design, in a second mode or third mode of operation, a second intake valve can open during the intake stroke or preferably during the compression stroke, or as late as near TDC, at the point or after the piston has reached the point where the first intake valve closes on either the intake or the compression stroke and closes quickly in order to inject a temperature adjusted high-pressure secondary air charge.

Mode 1 Operation: The 2-stroke engine of this invention, also operating in optional triple modes, introduces lightly compressed scavenging and charging air into primary ports at piston BDC and is the sole air charge in mode 1 operation.

For mode 2 or mode 3 operation, the intake valves of the power cylinders are timed to operate so that a high pressure supercharging air charge is cool and maintained within the transfer manifold, alternatively expansion chilled, and introduced into the power cylinder, during or near the end of the compression stroke, at such a time that the power cylinder has been scavenged and charged by air received from ports which are now closed and the exhaust valve has closed normally or late. This establishes that the expansion ratio of the engine can be equal to or greater than the compression ratio of the power cylinders. Means are provided in both 4-stroke and 2-stroke for causing fuel to be mixed with the air charge to produce a combustible mixture. The combustion chambers of the power cylinders are sized with respect to the displaced volume of the power cylinder in some designs to produce an initial low compression ratio so that the exploded combustion gas can be expanded to a volume equal to the compression ratio of the power cylinder of the engine.

In other 2-stroke designs, the exhaust valve is closed late in order to produce an expansion ratio greater than the "effective" compression ratio.

For mode 3 operation, modes 1 and 2 are combined.

For both 4-stroke and 2-stroke engines, fuel being present, ignition of the charge occurs at near piston TDC and the power pulse occurs as fully described hereafter.

Some of the advantages of the present invention over existing internal combustion engines are, as stated earlier, that it can provide a normal compression ratio or an "effective" compression ratio lower than the expansion ratio of the engine and can selectively provide a mean effective cylinder pressure higher than the conventional engine arrangement with the same or lower maximum cylinder pressure and temperature than that of prior art engines. In addition, the invention, in some designs, alternatively provides for double the burn time over existing engines by providing "Dwell"-burn time at piston TDC.

Because charge density, temperature and pressure are managed separately, light-load operation is practical, even for extended periods, with no sacrifice of fuel economy. The new working cycle is applicable to 2-stroke or 4-stroke engines, spark-ignited, compression-ignited and modified or true HCCI. For all engines, the weight of the charge can be greatly increased without the usual problems of high peak temperatures and pressures with the usual attendant problem of combustion detonation and pre-ignition. (Even in compression-ignited Diesel engines, a heavier, cooler, more turbulent air charge provides low peak cylinder pressure for a given expansion ratio and allows richer, smoke-limited air-fuel ratio giving increased power with lower particulate and NO_X emissions.) Modified HCCI or true HCCI operation will provide greater than "diesel efficiency", using gasoline, natural gas, hydrogen or diesel fuels, while limiting NO_X and particulate emissions (both smoke and nano-particle) but with diesel and hydrogen fuel performance being most benefited. Compression work is reduced due to reduced heat transfer during the compression process. Engine durability is improved because of an overall cooler working cycle and a cooler than normal exhaust and, in some duty cycles, by lower RPM, allowed by greater torque. Due to the ability to greatly increase air-fuel charge density and turbulence, even above that of diesel engines, while controlling the temperature and keeping it below auto-ignition temperature using diesel oils, allows spark ignition for diesel type oils, thus eliminating smoke particles and NO_X.

All of the objects, features and advantages of the present invention cannot be briefly stated in this summary, but will be understood by reference to the following specifications and the accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

Embodiments of the cold air supercharged internal combustion engines, according to the invention, will now be described, by way of example, with reference to the accompanying drawings.

FIG. 1 is a perspective view (with portions in cross-section) of the cylinder block and head of a six cylinder internal combustion engine. It operates in a 4-stroke cycle, and represents a first embodiment of apparatus of the present invention from which a method of operation can be performed and will be described. Among its other components, this embodiment is seen as having an ancillary high performance compressor, with one atmospheric charge-air intake duct and dual intake air routes, one of which is low pressure and one which is high pressure. Both lead to the same power cylinder, a temperature adjusting system with bypass systems and valves for controlling charge-air pressures, density and temperature.

FIG. 1C is a schematic view of an exhaust and an air intake system of an engine, showing a means of re-burning exhaust gases or showing a means of directing exhaust gases to an ancillary compressor and temperature adjusting system for use as a charge diluent or as an ignition catalyst.

FIG. 2 is a schematic drawing of a six cylinder, 4-stroke engine representing yet another embodiment of the apparatus of the present invention, from which yet another method of operation can be performed and will be described. FIG. 2 depicts three alternative systems (two in phantom lines) of inducting a low pressure primary air charge, one of which is by dividing the flow of the primary compressor to supply low-pressure air to the engine and to supply a first stage of compression for the high-pressure air supply. Among its

other components, this embodiment is seen as having three air coolers and dual manifolds and the means of controlling the temperature, density and pressure of the charge by an engine control module and by valving variations.

FIG. **2**B is a schematic drawing of a six cylinder, 4-stroke engine, representing yet another embodiment of the apparatus of the present invention, from which yet another method of operation can be performed and will be described. It depicts three alternative systems (two in phantom lines) of inducting a low pressure primary air charge, one of which is by dividing the flow of the primary compressor to supply both low-pressure air to the engine and to supply a first stage of compression for the high-pressure air supply. Among its other components, this embodiment is seen as having three air coolers and dual manifolds and the means of controlling the temperature, density and pressure of the charge by an engine control module and by valving variations. In addition, FIG. **2**B shows schematically another embodiment in which the engines secondary-charge air or air-fuel charge is optionally wholly or partially compression chilled before being introduced to power cylinders *7a* to *7f* to provide a cooler charge and lower polluting emissions.

FIG. **2**C is a schematic drawing of a six cylinder, 4-stroke engine, representing yet another embodiment of the apparatus of the present invention, from which yet another method of operation can be performed and will be described. Among its other components, this embodiment is seen as having three air coolers and dual manifolds and the means of controlling the temperature, density and pressure of the charge by an engine control module and by valving variations. In addition, FIG. **2**C shows schematically another embodiment in which the engine primary-charge air or air-fuel charge, or air charge only, is, wholly or partially. thermodynamically (compression) chilled before being introduced to power cylinders *7a* to *7f* to provide a cooler charge, greater fuel economy, engine durability and lower polluting emissions. In this design, a cool very dense secondary air charge is introduced during mode 2 operation without any expansion by expansion valves.

FIG. **2**D is a schematic drawing of a six cylinder, 4-stroke engine, representing yet another embodiment of the apparatus of the present invention, also adaptable to 2-stroke engines. It represents another method of operation which can be performed, using alternative systems of inducting a high pressure chilled supercharging air charge, which will be described. Among its other components, this embodiment is seen as having three air coolers and dual manifolds and the means of controlling the temperature, density and pressure of the charge by an engine control module and by valving variations. In addition, FIG. **2**D shows schematically an embodiment in which the engine's sole charge air or air-fuel charge is alternatively, wholly or partially, compression chilled before being introduced to power cylinders *7a* to *7f* to provide greater charge density, fuel economy, durability and lower polluting emissions.

FIG. **3** is a part sectional view through one power cylinder of a 4 stroke engine at the two intake valves, showing an alternative method (adaptable to other embodiments of the present invention) of high pressure air charging. Further, it shows a means of better mixing of fuel and air and of passing air between piston and cylinder wall, in order to prevent formation of formaldehyde and CO, of sweeping unburned fuel out of piston ring crevices for reducing HC, and of closing bottom inlet ports by piston movement.

FIG. **4** is a part sectional view through one power cylinder of a 4 stroke engine at the two intake valves, showing an alternative method (adaptable to other embodiments of the present invention) of high pressure air charging and showing another means of mixing fuel and air and of passing air between cylinder wall and piston, in order to prevent formation of formaldehyde and CO. It further shows another method for sweeping unburned fuel out of piston ring crevices and for closing bottom inlet ports while popette valve is closing.

FIG. **5** is a perspective view (with portions in cross-section) of the cylinder block and head of an internal combustion engine, operating in a 4-stroke cycle, and representing a preferred embodiment of the apparatus of the present invention from which a first method of operation can be performed and will be described. Among its other components, this embodiment is seen as having an auxiliary cooling system to alternatively provide engine air cooling or compression-expansion cooling during such time the liquid cooling system, if used, should be damaged.

FIG. **6** is a cross-section of a partial engine block and a cylinder liner. The liner is fitted with air-passageways containing splines or metal screens arranged for air cooling or compression-expansion cooling of the cylinder, also useful for controlling the temperature of charge-air in HCCI system of FIGS. **19**-B, **19**-C and **19**-D.

FIG. **7** is a perspective view (with portions in cross section) of the cylinder block and head, similar to the engine of FIG. **3**, operating in a 4-stroke cycle and representing a second embodiment of the apparatus of the present invention, from which a second method of operation can be performed and will be described. Among its other components, this embodiment is seen as showing a means of controlling engine temperatures by pumping air or compression-chilled air throughout passages in the engine, at such time liquid cooling system, if used, has failed. A system, similar to FIG. **3**, is shown for mixing air charge and for preventing formaldehyde, CO and unburned fuel (HC).

FIG. **8** is a perspective view (with portions in cross-section) of the cylinder block and head of an internal combustion engine, operating in a 2-stroke cycle, and representing a first 2-stroke embodiment of the apparatus of the present invention, from which still another method of operation can be performed and will be described. Among its other components, this embodiment is seen as having high pressure inlet valve to cylinder and also inlet scavenging and charging cylinder ports leading from a low-pressure air-compressor and having an air or a compression-expansion cooling system and conduits and valves to adjust engine temperature according to the invention.

FIG. **9** is a perspective view (with portions in cross-section) of the cylinder block and head of an internal combustion engine operating in a 2-stroke cycle, and representing a second 2-stroke embodiment of the apparatus of the present invention, from which still another method of operation of can be performed and will be described. Among its other components, this embodiment is seen as having an automatic valve and means for closing bottom scavenging and charging inlet ports and upper inlet port, while the popette intake valve is closing. In addition, it is seen as having one high pressure charge-air route to a power cylinder to supply supplementary charge air after compression has begun. Also shown is a common air-rail and cylinder port leading from a compressor (not shown) with low pressure air intake ports and Roots type compressor (not shown) to a power cylinder for scavenging and receiving primary charging air to the cylinder at piston turn around. It is also seen as having an exhaust valve and conduits to the

atmosphere from the power cylinder and having control valving means and compressed air expansion cooler system for auxiliary cooling the engine at such time that any liquid cooling system might be compromised.

FIG. **10** is a part sectional view through one power cylinder of a 2-stroke engine at the intake and exhaust valves, showing an alternative method (adaptable to other embodiments of the present invention) of injecting a secondary air charge during or near the end of the compression stroke, for increasing charge density and for sweeping out unburned HC and formaldehydes. It also shows an alternative method for preventing charge-air back flow, for automatically adjusting the charge pressure-ratio of the cylinder during the air charging process. There is also shown means for providing auxiliary engine cooling during any loss of engine liquid coolant. There is also shown inlet ports from a Roots type compressor (the compressor is not shown) for scavenging and supplying primary charging of the cylinder at the piston turn-a-round.

FIG. **11** is a schematic view of a six cylinder engine having both low and high pressure charge air supplied by one or two stages of compression and the pressure being amplified by a pressure amplifier. It illustrates how one compressor and a pressure amplifier can supply both low, or low boosted, and high pressure air, for operation of a constant speed engine of this invention, such as for power generation, pumping, etc.

FIG. **11**-B is a schematic view of a six cylinder engine having both low and high pressure charge air, supplied by one or two stages of compression by ordinary compressors, and the pressure then being amplified by a pressure amplifier. This illustrates how one compressor and a pressure amplifier can supply low or low boosted, and high pressure air for operation of a constant speed engine of this invention, such as for power generation, pumping, etc. FIG. **11**-B further illustrates partial or whole compression-expansion chilled cooling of supplementary charge air for the engines of this invention or charge air for any Otto or Diesel Cycle or turbine engine.

FIG. **12** is a pressure—volume diagram (a compilation of four diagrams) of a high speed diesel engine compared to the extended expansion process engines of this invention. These diagrams show three stages of intercooled compression and a fourth stage of uncooled compression.

FIG. **13** is a schematic transverse sectional view of a pre-combustion chamber, a combustion chamber, associated fuel and air inlet ducts, and inlet and exhaust valving, suggested for gaseous or liquid fuel operation for the engine of this invention or for any other internal combustion engine.

FIG. **14** is a part sectional view through one cylinder of an engine, showing an alternate construction, whereby there is supplied two firing strokes each revolution of the shaft for a 2-stroke engine and one firing stroke each revolution of the shaft for a 4-stroke engine. This engine has a beam which pivots on its lower extremity, a connecting rod which is joined mid-point of the beam and is fitted to the crankshaft of the engine, whereby a means is provided for varying the compression ratio of the engine at will.

FIG. **15** is a part sectional view through one cylinder of an engine, showing an alternate construction, whereby there is supplied two firing strokes each crankshaft revolution for a 2-stroke engine and one firing stroke each revolution of the shaft for a 4-stroke engine. A beam connecting the connecting rod and the piston pivots at a point between the piston and the piston connecting rod, which is attached to the crankshaft of the engine. Also shown is an alternate preferred means of power take-off from the piston by a conventional piston rod, cross-head and connecting rod arrangement.

FIG. **16** is a part sectional view through one cylinder of an engine, showing a means of providing extra burn-time each firing stroke in a 2-stroke or 4-stroke engine.

FIG. **17** is a schematic drawing, representing any of the engines of the present invention depicted in an alternate embodiment, which is configured to operate as a constant speed engine. This constant speed engine embodiment of the present invention is shown as including both a primary and an ancillary compressor with optional intercoolers for providing two stages of pre-compressed charge air, either optionally intercooled or adiabatically compressed.

FIG. **17**-B is a schematic drawing representing any of the engines of the present invention, depicting a constant-speed engine in accordance with an alternate embodiment of the present invention, in which there is provided a single compressor with optional intercoolers, providing a single stage of pre-compressed charge-air, either optionally intercooled or adiabatically compressed

FIG. **18** illustrates a variant design of the split-cycle engine, in which a primary charge at boosted or atmospheric pressure air can supply two more stages of compression, one in the firing cylinder, another in an ancillary compressor. The charge in the compressor is temperature adjusted and then injected into the power cylinder during the last part or the end of the compression stroke near TDC or even during combustion.

FIG. **18**-B illustrates a variant design of the split-cycle engine, in which a primary charge at boosted or atmospheric pressure air can supply two more stages of compression, one in the firing cylinder, another in an ancillary compressor, whose charge is temperature adjusted and then injected into the power cylinder during the last part or near the end of the compression stroke or even during combustion. In addition, this figure illustrates a compression-expansion chill cooling system for alternative partially or wholly chilling the charge air or air-fuel for greater density, lower polluting emissions and improved engine durability.

FIG. **18**-C illustrates a variant design of the split-cycle engine that provides a primary air charge at boosted or atmospheric pressure. The engine supplies two more stages of compression, with the two final compression stages being accomplished within the cylinder whose charge is intermittently cooled, then re-injected into the power cylinder, during the last part of the compression stroke, providing a low effective compression ratio with very dense air charge and an extended expansion ratio.

FIG. **18**-D illustrates schematically the operation of the engine of FIG. **18**-C.

FIG. **18**-E illustrates the valve layout of the power cylinder of the engine of FIG. **18**-C.

FIG. **19** is a schematic transverse sectional view of a preparation or pre-combustion chamber and a combustion chamber with associated heating elements, inlet conduits, injector pump, ducts, and valving for air and/or fuel suggested for gaseous or liquid fuel operation for the engines of this invention or for any other internal combustion engine for providing spark or compression ignition, modified HCCI, or true HCCI operation.

FIG. **19**-B is a schematic transverse sectional view of a pre-combustion chamber and a combustion chamber with associated inlet conduits, ducts and valving for optional compression-expansion chill air cooling and/or water cool-

ing as well as optional electric heating elements and fluid conduits, having the purpose of producing and controlling charge temperatures to facilitate homogeneous charge compression ignition (HCCI) for the engines of this invention or for any other internal combustion engine and for facilitating spark, or compression ignition or modified HCCI in any engine.

FIG. **19**-C illustrates a hemispheric-plus (perhaps 300 degrees, more or less), shaped combustion chamber **38**-B, containing all of the conduits, ducts, valves and controls pictured and described for FIG. **19**-B which are operated and controlled in the manner suggested for FIG. **19**-B system. The shape of the combustion chamber allows control of charge temperatures by the charge being surrounded by combustion chamber **38** and piston crown.

FIG. **19**-D illustrates a hemispheric-plus (perhaps 300 degrees, more or less) shaped combustion chamber **38**-B containing all of the conduits, ducts, valves and controls pictured and described for FIG. **19**-C which are operated and controlled in the manner suggested for FIG. **19**-C system. The shape of the combustion chamber allow control of charge temperatures by the charge being surrounded by combustion chamber **38**-B and piston crown. Piston projection performs the last stage of compression for auto-ignition of fuel-air charge in HCCI system.

FIG. **20** is a schematic drawing, depicting a new co-generation system, providing means of extracting work from compressed natural gas (CNG), or hydrogen (CH_2), or any other compressed gas, when contained at high pressure in a storage tank or pipeline. Illustrated is a means of utilizing gas (in this case CNG or CH_2), pressure-drop-distributor and evaporator tube(s) to expand the gas, to provide (compression) chilling for a refrigerator or air conditioner and to chill charge air for internal combustion engines or for a rotary turbine. When the gas is flammable, the expanding gas is then collectively directed, by separate distribution tubes, to where the gas is combusted to power an internal combustion engine, rotary turbine, or gas fired absorption chiller, or to heat a furnace for a steam turbine, or for a space heater or any other purpose needed.

FIG. **21** is a schematic drawing, depicting a means of preparation and presentation of liquefied gases for the extraction of work from either liquefied natural gas (LNG), hydrogen (LH_2) or any liquefied gas. The boil-off gas from the top of the tank or boil-off gas from a container of pumped LNG or LH_2 is directed to send the LNG or LH_2 through outlet conduit **43** to inlet **4** of flexible duct **4'**—FIG. **23** to first cool superconductor electric cable **3** or directly to conduit **2** of FIG. **20**, then to a metering or expansion valve and to pressure drop distributor and distributor tube(s). This will cool a chiller-refrigerator, air-conditioner or cool charge air for an internal combustion engine or rotary turbine. The expanded gas is then channeled collectively, or in separate tubes, to fuel an internal combustion engine, a gas turbine combustion system, a gas fired absorption chiller, or a furnace for a steam turbine, or a space heater burner thereby ultimately consuming the gas to provide additional work.

FIG. **22** is a list of the numbered components in FIG. **21**.

FIG. **23** is a two-dimensional drawing depicting a means of extracting work from liquefied natural gas (LNG), or liquefied hydrogen (LH_2) after first using it for cryo-cooling electric superconductor cables. This constitutes a new co-generation system. At the end of its use in cooling, the pumped LNG or LH_2 is directed to send the LNG or LH_2 through an optional pressurizer unit and then to conduit **2** of FIG. **20** to a metering or expansion valve, then to pressure drop distributor and distributor tube(s). This will cool a chiller-refrigerator, an air-conditioner or a cooling charge air for internal combustion engines or rotary turbine(s). The same expanded gas is then channeled collectively, or in separate tubes, to fuel an internal combustion engine system, a gas turbine combustion system, a gas fired absorption chiller, or a furnace for a steam turbine or a space heater burner, thus, ultimately burning the gas to provide further work.

FIG. **24** is a perspective view (with portions in cross section or shown schematically) of a cold air supercharged rotary internal combustion engine in which the rotor rotates in a epitrochoid (Wankel) type motion, operating in a novel working cycle characterized by having a cool or chilled and more dense fuel-air charge for greater power and torque and efficiency and describes methods of supercharging the engine during the compression process and obtaining the advantages described.

FIG. **24**-B is a perspective view of the engine of FIG. **24**, showing the combustion chamber in firing position with supercharge port closed by rotor.

FIG. **25** is a perspective view (with portions in cross section or are shown schematically) of a rotary internal combustion engine in which the rotor rotates in a epitrochoid (Wankel) type motion and describes operating in a novel working cycle, characterized by double compressing and cooling of the fuel-air charge and describes methods of increasing charge fuel-air density and reducing its overall temperatures.

FIG. **25**-B is a perspective view of the engine of FIG. **25**, showing alternate valving systems with the rotor shown in firing position.

FIG. **26** is a perspective view (with portions shown cut away) of a cold air supercharged rotary internal combustion in which the rotor(s) rotates on a central axis, operating in a unique working cycle, characterized by reducing compression temperatures, increasing power, torque and efficiency with ultra low polluting emissions.

FIG. **27** is a cross sectional view of a compression section of the engine of FIG. **26**, showing means and methods of reducing and varying "effective" compression ratios and of selectively extending the expansion ratio and of cold air supercharging the engine during the compression process and relieving valve tips from rubbing friction.

FIG. **28** is a cross sectional view of the expansion rotor and combustion chambers of the engine of FIG. **26**, showing means and methods of sealing the combustion chambers and obtaining three power pulses for each rotor rotation

FIG. **29** is a perspective view of one of two compressor section end-plates, showing track that valve guide rollers follow to prevent valve tips from rubbing the inside of the stator of the engine of FIG. **26**.

FIG. **30** is a perspective view (with portions in cross section and schematically) of a cold air supercharged rotary internal combustion engine in which rotors rotate on a central axis , the vane tips traveling, guided in an elliptical path and showing means and methods of supercharging the engine during the compression process after the engine has received an initial low "effective" compression ratio fuel-air charge.

FIG. **31** is a perspective view (with portions in cross section and schematically), showing means and methods by which an air charge is received by an engine and cooled during the compression process is cooled and compressed and cooled a second time and then introduced into the combustion chamber and fired with an extended expansion ratio.

FIG. **32** is a perspective view of one of two end-plates of the stator of engines of both FIG. **30** and FIG. **31**, showing the "raceway" or guide groove for the vane guide rollers.

DETAILED DESCRIPTIONS OF THE DRAWINGS

With reference now in greater detail to the drawings, a plurality of alternate preferred embodiments of the apparatus of the cold air supercharged internal combustion engine **100** of the present invention are depicted. Like components will be represented by like numerals throughout the several views and, in some but, not all circumstances, as is deemed necessary (due to the large number of embodiments), similar but, alternate components will be represented by super-scripted numerals (e.g. 100^1). When there are a plurality of similar components, the plurality is often times referenced herein (e.g., six cylinders *7a–7f*), even though fewer than all components are visible in the drawing. Also, components which are common among multiple cylinders are sometimes written with reference solely to the common numeral, for ease of drafting—e.g. piston *22a–22f*=>piston **22**. In an effort to facilitate the understanding of the plurality of embodiments, (but not to limit the disclosure) some, but not all, sections of this Detailed Description are subtitled to reference the system or sub-system detailed in the subject section.

The invented system of the present invention is, perhaps, best presented by reference to the method(s) of managing combustion charge densities, temperatures, pressures and turbulence; and the following description attempts to describe the preferred methods of the present invention by association with and in conjunction with apparatuses configured for and operated in accordance with the alternate, preferred methods.

Some, but not necessarily all, of the system components that are common to two or more of the herein depicted embodiments include a crankshaft **20**, to which are mounted connecting rods *19a–19f*, to each of which is mounted a piston *22a–22f*; each piston traveling within a power cylinder *7a–7f*; air being introduced into cylinder **7** or **12** through inlet ports controlled by intake valves **16**-A and **16**-B, or inlet ports **11**, and air being exhausted from the cylinders through exhaust ports controlled by exhaust valves **17** or **8**-B and conduit **18**. The interaction, modification and operation of these and such other components as are deemed necessary to an understanding of the various embodiments of the present invention are expressed below.

The Engine 100^1 of FIG. **1** Operating in Operational Designs 1–6

Referring now to FIG. **1**, there is shown a six cylinder reciprocating internal combustion engine 100^1 optionally operating in modes 1, 2 or a steady state method 3 and having one atmospheric air intake **9**, FIG. **1**. It is so constructed and arranged that a compressor **2** receives charge air from manifold **14**-B through openings **8**-B (shown in FIG. **2**) and conduit **8'** which air enters through common air intake duct **9** of FIG. **3**. Intake conduits *15a*-C to *15f*-C distributes the low pressure air to the intake valves **16**-B of each power cylinder. This arrangement allows the provision of air to intake valves **16**-A and **16**-B at different pressure levels since the charge air from conduit **15**-A is selectively pressurized by compressor **2**. Intake conduits **15**-A, **15**-C, in which all of the cylinders (only one (**7**) of which is shown in a sectional view) *7a–7f* and associated pistons *22a–22f* operate in a 4-stroke cycle and all power cylinders are used for producing power to a common crankshaft **20** via connecting rods *19a–19f*, respectively. A compressor **2**, in FIG. **1** and FIG. **2**, shows a Lysholm type rotary compressor which, with air conduits, as shown selectively, supplies pressurized air to one or more cylinder intake valves **16**A. An air inlet conduit **8'** receives atmospheric air from a low-pressure manifold **14**-B which receives atmospheric air through inlet **9**, selectively compresses and cools the charge which then goes to intake valve **16**-B by way of conduit **32**, FIG. **2**. Inlet conduits **15**-A, **15**-C and **32**, FIG. **2** separately supply air charge at atmospheric pressure and air which has been compressed to a higher pressure to separate intake valves **16**-B and **16**-A, opening to the same cylinder *7a–7f* (for example, shown here opening to cylinder *7f*). Intercoolers **10**, **11** and **12** and control valves **3**, **4**, **5** and **6** are used in the preferred embodiments to control the air charge density, weight, temperature and pressure of the air charge going to intake valve **16**-A, while compressor **1**, intercooler **10** and conduit **32** optionally and selectively convey charge air to intake valve **16**-B. The intake valves *16a*-B-*16f*-B, which receive air through manifold **14**-B and intake conduits *15a*-C to *15f*-C or through conduit **8'** and conduit **32**, are timed to control the compression ratio of the engine 100^1 in Operational Designs 3–6. The combustion chambers are sized to establish the compression ratio of the engine in Designs 1 and 2. Because of noticeable similarities between the engine 100^1 of FIG. **1** and that of FIG. **2** (where the atmospheric air inlet **9** system has been shown in phantom, for informational value), reference will be made as deemed helpful to FIG. **2** for certain common components.

The engine 100^1 shown in FIG. **1** is characterized by the ability to provide a normal expansion ratio in Operational Designs 1 and 2 and an extended expansion ratio and a low effective compression ratio in Operational Designs 3 through 6, with Design 4 having an effective compression ratio of near zero, perhaps 2:1 or less, and all capable of producing a combustion charge varying in weight from substandard to heavier-than-normal and capable of selectively providing a mean effective cylinder pressure higher than can the conventional arrangement in normal engines with lower maximum cylinder temperature in comparison to conventional engines. Engine Control Module (ECM) (refer to FIG. **2**) and variable valves **3**, **4**, **5**, and **6** on conduits, as shown, provide a system for controlling the charge pressure, density, temperature and mean and peak pressure within the cylinder, which allows greater fuel economy, production of greater power and torque, with low polluting emissions for spark ignited, compression ignited, modified HCCI, or true HCCI engines. In alternate embodiments, a variable valve timing system with the ECM-**27** can also control the time of opening and closing of the intake valves **16**-A and/or **16**-B to further provide an improved management of conditions in the combustion chambers to allow for a flatter torque curve and higher power, with low levels of both fuel consumption and polluting emissions. In another alternate embodiment, as shown in FIG. **2**-B, FIG. **2**-C, FIG. **2**-D, FIG. **11**-B, FIG. **18**-B, FIG. **20**, FIG. **21** and FIG. **23**, the engines of this invention and any current technology Otto or Diesel Cycle engine, or any other internal combustion engine, can utilize compression-expansion chilled charge air for greater power and efficiency and to lessen polluting emissions. (A design for Otto and Diesel Cycle engines, using refrigerated charge air, is depicted in and described for FIG. **2**-D).

Brief Description of Triple Mode Operation

The new cycle engine 100^1 of FIG. **1** is a high efficiency engine that attains both high power and torque, with low fuel

consumption and low polluting emissions. The new cycle is an external compression type combustion cycle. In this cycle, part of the intake air (all of which is compressed in the power cylinders in conventional engines) is selectively compressed by an ancillary compressor **2**. The temperature rise at the end of compression can be adjusted by use of air coolers **10**, **11** and **12**, which cool the intake air and by the late cylinder injection of temperature-adjusted air, and alternatively, by using chilled charge air and always with a low compression ratio in Operational Designs 2–6. While operating optionally in mode 1, the valve **16**-A, which injects a supplementary secondary air-fuel charge in mode 2 and mode 3 can be deactivated and/or compressor **2** is "waste gated" by bypass valves **3**, **4**, **5** and **6** in FIG. **2**, being opened. (Optionally, high-pressure valve **16**-A is a simple air or gas injection valve similar to a natural gas fuel injector which is closed during mode 1 operation.)

Optional operational modes for the engines of this invention assures continued reliable engine operation in mode 1 at such a time that ancillary compressor(s) may fail when operating in mode 2 or mode 3. By cutting out the supercharging air supply section, the engine continues to operate in mode 1 with power equal to conventional engines and with improved fuel economy. (This pertains to Operational Designs 1, 2, 3 and 6 with Design 4 being excepted.) Thus, mode 1 is especially essential for military vehicle or power generation duty.

Mode 1

During operation, a primary air charge is supplied by piston **22** intake (1^{st}) stroke to the cylinder **7** through intake valve **16**-B at atmospheric pressure or air which has been increased in pressure by perhaps one-half to one atmosphere through an atmospheric air inlet **9** or through inlet **8** of FIG. **2**, which is preferably carbureted. For mode 1 operation, the charge is then compressed (2^{nd} stroke) in the cylinder **7**, fuel added if not present, and ignited at the appropriate point near TDC for the power (3^{rd}) stroke, followed by the scavenging (4^{th}) stroke, providing high fuel economy and low polluting emissions. This completes one power cycle.

Mode 2

When more power is desired for mode 2 or mode 3 operation, a secondary air charge (mandatory at all times for Design 4), originating from air conduit **8'** and high pressure manifold (**13** and **14** in FIG. **2**), is introduced into the power cylinder **7** by a second intake valve **16**-A, which introduces a higher pressure temperature adjusted air or air-fuel charge after the first intake valve, **16**-B has closed. It is then injected either during the intake stroke, or preferably after compression has begun, or at any time during the compression stroke, or near TDC. This is in order to adjust the temperature and increase charge density and turbulence when needed. After the secondary air charge has been injected, intake valve **16**-A, which can be operated hydraulically or electrically or in some designs by cam, quickly closes. Compression (2^{nd}) stroke continues and the charge is ignited near TDC. Piston **22** expands in the power (3^{rd}) stroke followed by the scavenging 4^{th} stroke. This completes one power cycle.

Mode 3 (Steady-State)

For mode 3 operation, modes 1 and 2 are combined, simplifying operation with the high-pressure secondary charge being injected without interruption into cylinder **7** any time after low-pressure intake valve **16**-B closes as in mode 1, and preferably injected during the compression stroke. The compression stroke continues to near TDC. The charge is ignited for the power (3^{rd}) stroke followed the scavenging (4^{th}) stroke. This completes one power cycle in mode 3 operation.

Alternatively, a one-way valve (one type of which is shown as **2'** in FIGS. **7**, **9** and **10**) can be utilized to provide a constant or a variable initial "pressure ratio" in the cylinder **12**, while improving turbulence. In this alternate method of operation, valve **2'** would close when the pressure in the cylinder **12** nearly equates or exceeds the pressure in conduit B, FIGS. **7**, **9** and **10** or conduit **15**-A in FIGS. **1** and **2**, isolating valve **16**-A from cylinders **7** or **12** and giving valve **16**-A adequate time to close. Thus, the pressure in conduit B or **15**-A, FIG. **1**, controlled by compressor speed, along with valves **3**, **4**, **5** and **6** and the ultimate compression ratio would regulate the pressure, density, temperature and turbulence of the combustion charge.

The air pressure, supplied to intake runner-conduit **15**-A (FIG. **1**), is produced at a high level, and intake valve **16**-A is, in alternate embodiments, replaced by a fast-acting, more controllable valve, such as, but not limited to a high speed valve (**21'**, FIGS. **3**, **4**, **5**, **7** and **10**), which valve is, preferably, either mechanically, hydraulically, electrically or vacuum operated alternatively under the control of an engine control module (ECM)-**27**. In such an embodiment, a small or large, dense, temperature-adjusted, high-pressure charge, with or without accompanying fuel, can, selectively, be injected (perhaps by a gas-fuel type injector) at any time during the compression stroke, or even during the combustion process (always after intake valve **16**-B has closed) in order to adjust the charge temperature and density, to reduce peak and overall combustion temperatures and to complete the desired charge mixing before combustion.

Alternatively, conduit **15**-A, FIG. **1** and FIG. **2** may be fitted with compression-expansion valves **10'** as illustrated in FIG. **2**-B, FIG. **2**-D, FIG. **11**-B and FIG. **18**-B to maximize the cooling abilities of the high pressure charge going to intake valve **16**-A, or in one design, FIG. **2**-C, the chilled charge is primary and is injected by intake valve **16**-B. (Preferably when the air charge is injected by expansion valve **10'**, the injection takes place during the intake stroke as in FIG. **2**-C). Also, the expansion valve **10'** can be made variable in size, and/or bypass system R and X (adjustable by ECM-**27**, FIG. **2**-B, FIG. **2**-C, and FIG. **2**-D) in order to control the cooling and power characteristics of the injected charge, as needed.

In order to maximize the chill cooling effect, the differential pressure of the cylinder and the incoming charge must be significantly in favor of the incoming charge with the cylinder pressure being the lower.

Alternatively, inlet ducts **17**-A or **17**-B of FIGS. **3**, **4**, **7** and **10** are placed in the cylinder wall so that part of the high-pressure charge air is injected between cylinder wall **7** and piston **22**, preferably above the top piston ring at TDC. This will provide cooling for the engine and sweep the area clean of unburned hydrocarbons and also prevent the formation of formaldehyde

High pressure inlet conduits **15**-A in the engines of FIG. **1**, FIG. **2**, FIG. **3** and FIG. **4** and high pressure conduits B in FIG. **5**, FIG. **7**, FIG. **8**, FIG. **9**, FIG. **10**, FIG. **11**, and conduit **9'** in FIG. **11**-B and conduit **8'** in FIG. **13**, conduits **15***a*-A, **15***f*-A to valve **16***a*-E and **16***f*-E of FIG. **18**, and conduit **8'** in FIG. **19** and FIG. **19**-B, in alternative designs are fitted with adjustable expansion valves **10'** as in FIG. **2**-B, FIG. **2**-D, FIG. **11**-B and FIG. **18**-B to refrigerate the