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Wednesday, 21 December, 2005 12:33:12 PM
Clerk, U.S. District Court, ILCD

being present, the charge is ignited by spark or compression and the power stroke takes place.

When more power is desired, a secondary air charge from conduit **15**-A can be introduced into the power cylinder at the time of, or after closure of exhaust valve(s) **17***a* during the compression stroke, by intake valve **16**-A which introduces a higher pressure air charge, and quickly closes, in order to increase the charge density. Alternatively, the primary air charge may be boosted to a higher pressure by adjusting air bypass valve **6** to send more air through compressor **2**, by increasing the speed of compressor **2** or by changing the setting on the control valve **25** on the conduit **15**-B which alternatively supplies the low pressure primary air charge to intake valve **16**-B. The temperature, pressure, amount and point of injection of a secondary charge, if added, is adjusted to produce the desired results.

For light-load operation an intake valve disabler **31** (there are several on the market, for example, Eaton Corp. and Cadillac) can disable intake valve **16**-A when light-load operation does not require a high mean effective cylinder pressure. Alternatively, during the time the low pressure air to intake valve **16**-B is supplied by conduit **15**-B the air bypass valve (ABV) **6** can be opened to recirculate some of the charge-air back through the compressor **2** in order to relieve the compressor of compression work during light-load operation. Additionally, and preferably, air bypass valve **4** can re-circulate part or all of the air pumped by compressor **1** back to the inlet of compressor **1** on demand in order to reduce pressure and density of the secondary charge going through intake valve **16**-A.

One suggested, preferred method of operation of the new-cycle engine **100**^{9} is thus:

1. Intake air at greater than atmospheric pressure that has been compressed by at least one compressor **2** and has had its temperature adjusted by bypass systems or charge-air cooler **10**, is introduced into the cylinder **7** through intake valve **16**-B, which is opened by a small lobe on cam **21**-B at near bottom dead center, at the end of the power stroke (perhaps at bottom dead center) after exhaust valve(s) **17**, **17'** have opened earlier say, at 40° before bottom dead center, for exhaust blowdown. The exhaust valves remain open after bottom dead center for further scavenging of the cylinder **7**. The intake valve **16**-B closes at near bottom dead center.
2. After the power stroke is complete and cylinder **7** is filled with fresh charge, the exhaust valve(s) **17** is left open for a period of time after the piston has passed bottom dead center (with intake valve **16**-B now closed) in order to further scavenge the power cylinder with the fresh air charge present and further, in order to establish a low compression ratio of the engine, the compression ratio being established by the displaced cylinder volume remaining at the point of the exhaust valve **17** closure, being divided by the volume of the combustion chamber.
3. With the cylinder **7** now filled with fresh air, the compression (2nd) stroke continues and, at some point the exhaust valve **17** is closed and compression begins for a small compression ratio. This makes it possible to lessen the temperature rise during the compression stroke. Compression continues, fuel is added if not present, and the charge is fired a the appropriate point near top dead center and the power stroke occurs.
4. (a) Alternatively, when greater power is required, a secondary compressed, temperature-adjusted air charge is injected into the cylinder **7** by intake valve **16**-A opening and quickly closing during the compression stroke at the point at which the exhaust valve closes, or later in the stroke, to produce a more dense charge in order to provide the torque and power desired of the engine.

   (b) When even greater power is required, the secondary air charge can be increased in density and weight by being passed through one or more intercoolers **10**, **11** and **12** and by increasing compressor speed or by cutting in another stage of auxiliary compression or by passing more of the charge air through the operational compressors.
5. Near bottom dead center of the piston position, exhaust valves **17**, **17'** open and the cylinder is efficiently scavenged by blowdown and by the air injected by primary intake valve **16**-B.

Detailed Description of the Operation of the Engine **100**° of FIG. **9**

Near the end of the power (1st) stroke of the piston **22**, perhaps at about 40° before bottom dead center position of piston **22**, the exhaust valves **17** open for exhaust blowdown, shortly after low pressure air flows through air conduit **32** from conduit **106** and optional shut-off valve **33** and compressor **2** or alternatively through air conduit **15**-B supplied by a pressure regulator valve **25** from compressed air line **15**-A (as shown in FIG. **9**, and FIG. **10**), through an intake valve **16**-B into the cylinder **7**. Intake valve **16**-B closes shortly after bottom dead center or, perhaps at bottom dead center. Exhaust valves **17** remain open during the first part of the compression (2nd) stroke of piston **22**. The cylinder **7** is now efficiently scavenged by blowdown and by loop scavenging and at any point during the compression stroke, the cylinder **7**, now filled with fresh air, the exhaust valves **17**, **17'** can close. But since a low compression ratio is desired, the exhaust valves **17**, **17'** can be held open until the piston has reached the point that is desired to establish the compression ratio. At, or after the time exhaust valves **17***a* and **17***a'* are closed, a secondary charge of high pressure, temperature adjusted air which has been compressed by a compressor(s) can be injected by intake valve **16**-A into the same cylinder, after which intake valve **16**-A closes. In addition, when very high torque and power is needed, the density of the secondary charge-air can be greatly increased by cutting-in compressor **2** or by increasing the speed of compressor **2**, if already compressing, as in FIG. **9**, directing more air through compressors **1** and/or **2** by valves **4** and/or **6**, and by routing the charge wholly or in part through intercoolers **10**, **11** and **12**.

In this system, regardless of the point the exhaust valve is closed to establish the compression ratio, the primary fresh air charge trapped in the cylinder **7** will be lighter than normal and the compression ratio will be lower than normal, therefore, if needed, a highly compressed, temperature adjusted air charge can be injected at exhaust valve closure or later in the stroke, to provide a heavier than normal charge but with the temperature rise being restrained by the cooled charge and the short compression stroke. This produces a greater than normal mean effective cylinder pressure when combusted for great torque and power but still with an expansion ratio greater than the compression ratio.

For light-load operation a shut-off valve, or a valve disabler **31** (in phantom) on the high pressure intake valve could temporarily restrain the intake air, or hold the valve **16**-A closed. This would add to the fuel economy of the engine. Alternatively, if compressor **2** is not supplying air to conduit **32** and intake valve **16**-B, during light-load operation the shutter valve **5** could be closed and the air bypass valve **6** can be opened so that air pumped by compressor **2**

would be returned in part or wholly to the inlet conduit of the compressor **2** with little or no compression taking place there.

An ancillary automatic intake valve **26**, FIG. **10**, can be arranged, as shown in FIG. **10**, to prevent any back-flow of charge-air into conduit **15**-A if the cylinder **7** pressure should approximate or exceed the pressure in conduit **15**-A during the compression stroke of piston **22** before the closure of intake valve **16**-A.

Alternatively, the ancillary automatic valve **26** of FIG. **10** could be used to provide a constant or a variable pressure ratio in cylinder **7**. In this case valve **16**-A would be kept open to near top dead center and the closure time of valve **26** would be adjusted by the pressure differential in cylinder **7**, controlled by valves **3, 4, 5** and **6** by compressor(s) output and by any throttle valve present. The automatic valve **26** could be of the spring-retracted disc type and could be fabricated of metal or ceramics.

Fuel can be carbureted, injected in a throttle body **56**, shown in FIG. **15** through FIG. **17** and item **56** in FIG. **19** and FIG. **20**, or the fuel can be injected into the inlet stream of air, injected into a pre-combustion chamber (similar to that seen in FIG. **21**) or, injected through intake valves **16**-A, or it may be injected directly into the combustion chamber at point x during the exhaust-compression stroke, at the time or after the piston **22** has passed point x in the compression stroke. The fuel can also be injected later and in the case of diesel operation can be injected at the usual point for diesel oil injection, perhaps into a pre-combustion chamber or directly into the combustion chamber, perhaps as FIG. **21**, or directly onto a glow plug. After the temperature-and-density-adjusting-air charge has been injected, if used, compression of the charge continues and with fuel present, is ignited at the opportune time for the expansion stroke. (The compression ratio is established by the displaced volume of the cylinder remaining after point x has been reached, being divided by the volume of the combustion chamber. The expansion ratio is determined by dividing the cylinders total clearance volume by the volume of the combustion chamber.)

Now the fuel-air charge is ignited and the power (2nd) stroke of piston **22** takes place as the combusted gases expand. Near bottom dead center of the power stroke the exhaust valve(s) **17, 17'** open and the cylinder **7** is efficiently scavenged by blowdown and by loop scavenging at the end of the power stroke and largely during the piston **22** turn-around time.

It can be seen that the later the point in the compression stroke that point x is reached (the later the exhaust valve is closed), the lower is the compression ratio of the engine and the less the charge is heated during compression.

It can also be seen that the later the temperature-density-adjusting charge is introduced, the less work will be required of the engine to compress the charge, the later part of which has received some compression already by compressor **1** and/or by an ancillary compressor **2**. In some cases where the load is light and fuel economy important the ancillary compressor could be bypassed with the secondary air charge perhaps eliminated temporarily and the total charge weight could be less than that of a conventional engine and with the extended expansion ratio produce even better fuel economy.

During light-load operation of this 2-stroke cycle engine (FIG. **9** and FIG. **9**-B) such as vehicle cruising or light-load power generation, the secondary air charge can be eliminated by disabling high pressure intake valve **16**-A temporarily (several valve disabling systems Eton, Cadillac, etc.) or air can be shut off to intake valve **16**-A and the engine still produce greater fuel economy and power with the air charge being supplied by compressor **2** or **1** through conduits **15**-A, **110, 32** and intake valve **16**-B.

The Engine $100^{9\text{-}B}$ of FIG. 9-B

FIG. **9**-B is a schematic representation of a six-cylinder reciprocating internal combustion engine $100^{9\text{-}B}$ which is for the most part identical to the engine 100^{9} of FIG. **9**. The characteristics and operation and structure of the engine $100^{9\text{-}B}$ of FIG. **9**-B are substantially similar to the engine 100^{9} of FIG. **9** and, except as necessary to point out specific points of distinction, such characteristics, operation and structure are not repeated here. Reference should be made to the sections on characteristics, structure and operations (both brief and detailed) previously presented with respect to the engine 100^{9} of FIG. **9**.

The major point of distinction between engine 100^{9} and engine $100^{9\text{-}B}$ is that engine $100^{9\text{-}B}$ represents an embodiment of the engine 100^{9} wherein the compressors **1,2** are of alternate types. That is, in $100^{9\text{-}B}$, the primary compressor **1** is shown as a Lysholm rotary compressor (as opposed to the reciprocating-type compressor of engine 100^{9}) and the secondary compressor **2** is of the turbo-type (as opposed to the Lysholm-type of 100^{9}). Although conduit **32** from conduit **110** (designated as **106** in FIG. **9**) and optional shut-off valve **33** is shown supplying intake valves **16**-B of only two cylinders of the engine, it is understood that other intake runners (not shown) distribute air from conduit **110** to the remainder of the intake valves **16**-B of the engine, or that conduit **32** supplies an "air box" or manifolds which distribute the air to all of the intake valves **16**-B.

Referring now to FIG. **10** there is shown the same engine and the same operating system as described for the engines of FIG. **9** and FIG. **9**-B, but has an optional added feature in that the secondary intake valve **16**-A has an ancillary valve **26** which is automatic to prevent charge-air back-flow from cylinder **7**. This feature will prevent any back-flow from occurring during the compression stroke of the engine of this invention, should the cylinder pressure approximate or exceed the pressure in conduit **15**-A before the intake valve **16**-A was fully closed. (This optional automatic valve **26** could be of the spring-retracted disc type, or could be any type of one-way valve.) An automatic valve at this place could be used to regulate the pressure ratio in cylinder **7** during the compression of the charge. In this case intake valve **16**-A could be kept open to near top dead center, valve **26** automatically closing the intake below valve **16**-A during compression, ignition and power stroke of the charge. Furthermore, the use of automatic valve **26** would allow the pressure ratio of the engine to be adjusted by simply adjusting the pressure in conduit **15**-A, with intake valve **16**-A being kept open to near top dead center of piston **22**. The ancillary valve **26**, if present, would also impart a tangentially oriented swirl turbulence to the combustion charge as would also, shrouding of intake valve **16**-A.

The Engine 100^{11} of FIG. 11

Referring now to FIG. **11**, there is shown a six cylinder reciprocating internal combustion engine 100^{11} with one atmospheric air intake, in which all of the cylinders ***7a–7f*** (only one (***7f***) of which is shown in a sectional view) and associated pistons ***22a–22f*** operate in a 2-stroke cycle and all power cylinders are used for producing power to a common crankshaft **20** via connecting rods ***19a–19f*** respectively. A primary compressor **1**, in this figure a double-acting

reciprocating type, is shown which, with an air conduits, as shown, supplies pressurized air to one or more cylinder intake valves **16***a* and **16***b*. A secondary compressor **2** of the Lysholm type is shown in series with compressor **1**. An air inlet **8** and associated inlet conduit and manifolds **13** and **14** supply air charge which has been compressed to a higher than atmospheric pressure, to a cylinder intake conduit **15** which supplies charge-air to two intake valves, which intake valves **16***a* and **16***b* operate independently of each other but open into the same cylinder. Intercoolers **10**, **11** and **12** and control valves **3**, **4**, **5** and **6** are used to help control the air charge density, weight, temperature and pressure. The intake valves are timed to control the compression ratio of the engine. The combustion chambers are sized to establish the expansion ratio of the engine.

The engine **100** of FIG. **8**, FIG. **9**, FIG. **10** and FIG. **11** have cam shafts **21** fitted with cams and are arranged to rotate at engine crankshaft speed in order to supply one power stroke for each power piston for each crankshaft rotation.

The engine **100**^{11} shown in FIG. 11 is characterized by a more extensive expansion process, a low compression ratio and capable of producing a combustion charge varying in weight from lighter-than-normal to heavier-than-normal and capable of selectively providing a mean effective cylinder pressure higher than can the conventional arrangement in normal engines, but having similar or lower maximum cylinder pressure. Engine control module (ECM) **27** 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 at all RPM, with low polluting emissions for both spark and compression ignited 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* or **16***b* or both, to further provide an improved management of conditions in the combustion chamber to allow for a flatter torque curve, and higher power, with low levels of both fuel consumption and polluting emissions.

Brief Description of Operation of Engine **100**^{11} Shown in FIG. **11**

The new cycle engine **100**^{11} of FIG. **11** 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 compressed by an ancillary compressor. The temperature rise at the end of compression can be suppressed by use of air coolers, which cools the intake air, and by a shorter compression stroke.

During operation air is supplied to the power cylinder **7** at a pressure which has been increased by perhaps from one-third to several atmospheres, or greater through an air intake conduit **15**. Valve **16***b* opens by pressure on the top of the valve stem from a very small lobe on cam **21**-A for a short period of time near bottom dead center position of piston **22** in order to scavenge the cylinder and provide fresh charge-air. Exhaust valves **17**, **17'** open for exhaust blowdown slightly before intake valve **16***b* opens to admit scavenging air. The cylinder **7** is efficiently scavenged mostly during the turnaround time of piston **22**. During the first part of the compression stroke, perhaps as early as 10–20° after bottom dead center of piston **22** position, the first intake valve **16***b* closes, at a later time the exhaust valve **17**, **17'** closes, at which point compression of the fresh air charge starts, which establishes the compression ratio of the engine. At the point the exhaust valves **17**, **17'** closes or any point later, the second intake valve **16***a* and perhaps **16***b*, by a second lobe **21**-C is, preferably, opened to introduce more of the temperature and density adjusted charge, if needed.

An intake valve disabler **31** in FIG. **10** (there are several on the market, for example, Eaton Corp. and Cadillac) can disable intake valve **16***a* when light-load operation does not require a high mean effective cylinder pressure. Alternatively, the air bypass valve (ABV) **6** is opened wholly or partially to re-circulate some or all of the charge-air back through the compressor **2** in order to relieve the compressor of compression work during light-load operation. Additionally, air bypass valve **4** can re-circulate part or all of the air pumped by compressor 1 on demand in order to reduce charge pressure and density.

One suggested, preferred method of operation of the new cycle engine **100**^{11} is thus:

1. Intake air at greater than atmospheric pressure that has been compressed by at least one compressor and has had its temperature adjusted by bypass systems and charge-air coolers are introduced into the cylinder **7** through intake valve **16***b* which is opened by a very small lobe **21**-D on cam **21**-A at or near bottom dead center of piston **22** at the end of the power-stroke, as exhaust valve(s) **17***a*, **17***a'* have opened a little earlier (perhaps 40° before bottom dead center) for exhaust blowdown. The exhaust valve **17** remains open through bottom dead center for efficient scavenging of the cylinder **7** by blowdown and loop scavenging. Intake valve **16***b* closes as the fresh high-pressure charge very quickly scavenges the cylinder **7**.
2. After the power stroke is complete the exhaust valves **17** are left open for a period of time after the piston has passed bottom dead center (with intake valve **16***b* now closed) in order to continue to scavenge the power cylinder with the fresh air charge and father, in order to establish a low compression ratio of the engine, the compression ratio being established by the displaced cylinder volume remaining at the point of the exhaust valve **17** closure being divided by the volume of the combustion chamber.
3. With the cylinder **7** now filled with fresh air which is near atmospheric pressure, the compression (2nd) stroke continues and, at the point the exhaust valve is closed, compression begins for a small compression ratio. This makes it possible to lessen the temperature rise during the compression stroke. Compression continues, fuel is added, if not present, and the charge is fired at the appropriate point near top dead center and the power-stroke occurs.
4. (a) Alternatively, at any point deemed appropriate at the time or after the exhaust valve has closed and compression of the charge has begun, a secondary density and temperature-adjusted air charge can be injected through intake valve **16***a*and perhaps by a second lobe **21**-C on cam **21**-A, through intake valve **16***b*. Compression continues with the secondary air charge injection, fuel is added, if not present, the charge is ignited and combustion produces a large expansion of the combusted gases producing great energy. This energy is turned into high torque and power by the engine.

   (b) When even greater power is required, the air charge can be increased in density and weight by being passed through one or more intercoolers and by increasing

compressor speed or by cutting in a second stage **2** of auxiliary compression, FIG. **11**. Alternatively, the timing of closing exhaust valve **17** and of the opening of intake valve **16***a* could be altered temporarily to close earlier and to open earlier, respectively, for a larger charge.

5. Near bottom dead center of the piston, exhaust valves **17**, **17'** open and the cylinder is scavenged by blowdown and by the air injected by primary intake valve **16***b*.

Detailed Description of the Operation of the Engine **100**[11] of FIG. **11**

Near the end of the power (1st) stroke of the piston **22**, perhaps at about 40° before bottom dead center position of piston **22**, the exhaust valves **17** open for exhaust blowdown, shortly after, high pressure air flows through air conduit **15** from manifold **13** and **14**, as shown in FIG. **11**, through an intake valve **16***b* into the cylinder **7**, the cylinder **7** is scavenged, intake valve **16***b* closes. (Intake valve head **30** can be recessed as shown in FIG. **11** in order to form a pipe-like opening into cylinder **7** so that when the charge-air is highly compressed, and as much as 500–530 psi is feasible, the small lobe **21**-D on cam **21**-A of intake valve **16***b* lets in a small blast of the high pressure air which is directed downward for loop scavenging, during or just after piston **22** turnaround at bottom dead center piston position.) Exhaust valves **17** remain open during the first part of the compression (2nd) stroke of piston **22**. The cylinder **7** is now efficiently scavenged by blowdown and by loop scavenging and at any point during the compression stroke, the cylinder **7**, now being filled with fresh air, the exhaust valves **17**, **17'** can close. But since a low compression ratio is desired, the exhaust valves **17**, **17'** can be held open until the piston has reached the point that is desired to establish the compression ratio. At, or after the time exhaust valve **17** closed, a secondary charge of high pressure temperature adjusted air which has been compressed by compressor **1** and/or **2** can be injected by the second intake valve **16***a* and, if desired, by another lobe **21**-C (in phantom) on the first valve **16***b* into the same cylinder. (When high torque and power is needed, the density of the charge-air can be greatly increased by increasing the speed of the primary compressor **1** or by cutting in another stage of compression, as in compressor **2**, FIG. **11**, and routing the charge through aftercoolers **10**, **11** and **12**. Also the speed of compressor **2** can be increased to shove in more charge on the back end.) Compression would continue, for a small compression ratio, fuel would be added, if not present, the charge would be ignited and the gases expanded against piston **22** for the power stroke.

For light-load operation, a shut-off valve (or a valve disabler **31** shown in FIG. **10** on the intake valve **16**-A) could temporarily restrain the intake air, or hold the intake valve **16***a* closed. This would add to the fuel economy of the engine. Alternatively, during light-load operation the shutter valve **5** could be closed and the air bypass valve **6** opened so that air pumped by compressor **2** would be returned to the inlet conduit of the compressor **2** without any compression taking place. In the same manner valves **3** and **4** could return part of the air being pumped through back to the intake **106** of compressor **1**.

The ancillary automatic intake valve **26**, FIG. **10**, which can be of the spring-returned disc type, can be arranged, as shown in FIG. **10**, to prevent any back-flow of charge-air into conduit **15** if the cylinder pressure should equal or exceed the pressure in conduit **15** during the compression stroke of piston **22** before intake valve **16***a* had closed completely. (As in other engine designs herein presented, the optional automatic valve **26** shown in FIG. **10** can be utilized to control the pressure ratio of this engine. If the intake valve **16***a* is kept open to near top dead center, the closure of valve **26** and the pressure ratio of cylinder **7** would be controlled by control valves **3**, **4**, **5** and **6** and by compressor speed and by any throttle valve present.) Automatic valve **26** would seal the intake from conduit **15** during the last part of the compression stroke, ignition of the charge and during the power stroke.

Fuel can be carbureted, injected in a throttle body **56** in FIG. **15** through FIG. **17**, and **56** in FIG. **19** and **20**, or the fuel can be injected into the inlet stream of air, or injected into a pre-combustion chamber or, injected through intake valves **16***a*, **16***b*, (the latter during its second opening by lobe **21**-C on cam **21**-A), or it may be injected directly into the combustion chamber at or past point x in the exhaust-compression stroke. The fuel can also be injected later and in the case of diesel operation can be injected at the usual point for diesel oil injection, perhaps into a pre-combustion chamber or directly into the combustion chamber or directly onto a glow plug. After the temperature-and-density-adjusting-air charge has been injected, if used, compression of the charge continues and with fuel present, is ignited at the opportune time for the expansion stroke. (The compression ratio is established by the displaced volume of the cylinder remaining after point x (at exhaust valve closure) has been reached, being divided by the volume of the combustion chamber. The expansion ratio is determined by dividing the cylinders total clearance volume by the volume of the combustion chamber.) Now the fuel-air charge has been ignited and the power stroke of piston **22** takes place as the combusted gases expand. Near bottom dead center of the power stroke, the exhaust valve(s) **17** opens and the cylinder **7** is efficiently scavenged, first by blowdown, then by loop scavenging by air from intake valve **16***b* at the end of the power stroke or shortly after.

It can be seen that the later the point in the compression stroke that point x (the later the exhaust valve is closed) is reached, the lower is the compression ratio of the engine and the less the charge is heated during compression.

It can also be seen that the later the temperature-density-adjusting charge is introduced, the less work will be required of the engine to compress the charge, the later part of which has received some compression already by compressor **1** and/or by an ancillary compressor **2**. In some cases where the load is light and fuel economy important the ancillary compressor could be bypassed with the secondary air charge perhaps eliminated temporarily and the total charge weight could be less than that of a conventional engine.

Referring now to FIG. **12** there is shown a pressure-volume diagram for a high-speedDiesel engine compared to the engines of this invention, showing three stages of intercooled compression and a fourth stage of uncooled compression indicating a compression ratio of approximately 2:1, which arrangement is suggested for optimum power, with efficiency for the engine of this invention. (The charts of FIG. **13** and FIG. **14** show some of the improvements of the engine of this invention over current heavy-duty 2-stroke and 4-stroke engines.)

There are several features that improve the thermal efficiency of the engine of this invention. Greater power to weight ratios will provide a smaller engine with less frictional losses. The extended expansion ratio results in higher thermodynamic cycle efficiency, which is shown in theoretical considerations. There are also definite efficiency gains in a "staged" compression process even with external

compressors with associated piping, intercoolers and aftercoolers,etc. There is a very significant energy savings when air is compressed in intercooled stages. Less energy is used in compressing a charge to 500 psi in **2**, **3** or **4** intercooled stages than is used to compress the hot charge to the same 500 psi in a conventional engine. A normal engine uses approximately 20% of its own energy produced to compress its own air charge. Calculations show a significant energy savings in an engine if the air is compressed in aftercooled stages. Compressing a charge in only two stages to 531 psi (a 13:1 compression ratio) reduces the energy used by 15.8% over compressing to the same 531 psi level in a single stage as does the Otto and the Diesel Cycle engines. Three stages of intercooled compression raises the savings to 18%. This is the ideal. Degradation from the ideal should not exceed 25% which leaves a 13.5% energy savings. This 13.5% energy savings times the 20% of a normal engine's power used for compressing its own charge, is a 2.7% efficiency improvement by the compression process alone. This is one of the advantages of this engine which adds to the other thermal efficiency improvements. The low compression ratio, along with the large expansion ratio provides improvements in efficiency, torque, power and durability while lowering polluting emissions.

Note 1—In FIG. **12** the travel distance of the line for engine B on the horizontal coordinate indicates the theoretical volume at the greater density. The density is kept at that level at the actual combustion chamber volume (as shown by dashed line V) regardless of the density, by pumping in more charge at the back-side.

Referring now to FIG. **13**, there is shown a chart which compares various operating parameters of the engine of this invention (B) with the operation parameters of a popular heavy-duty, 2-stroke diesel cycle engine (A).

The parameters shown for engine A are the normal operating parameters for that engine, e.g., compression ratio, combustion temperatures, charge density, etc. The parameters chosen to illustrate for engine (B) are given at two different lower "nominal" compression ratios with corresponding "effective" compression ratios, intercooled and uncooled, for two different levels of power output. The columns showing charge densities and expansion ratios indicate the improvements in steady state power density improvements for engine B even at a substantially lower nominal compression ratio and an effective compression ratio as low as 2:1 as shown in FIG. **10**. The columns showing low temperatures at the end of combustion, and the column showing extended expansion ratios indicate much lower polluting emissions. Indicated power improvements of engine (B) over engine (A) even at the lower nominal compression ratio are no less than 50%.

Referring now to FIG. **14** there is shown a chart which compares the various operating parameters of the engine of this invention (B) with the operating parameters of a popular heavy-duty 4-stroke diesel engine (A).

When comparisons similar to those of FIG. **13** are made, steady state power and density improvements are much higher since engine (B) fires the denser charge twice as often as engine A for an indicated steady-state power density improvement of 180% over engine (A).

Referring now to FIG. **15**, there is shown a schematic drawing of an engine representing the engines of FIGS. **5–7**, and FIGS. **9–10** with a separate air cooler **10** for ancillary compressor **2**, with the primary compressor **1** supplying two manifolds **13** and **14** and having separate air coolers **11** and **12** and charge-air conduits **114** and **115**, and having each manifold having three cylinder air intake runners **15***a*–**15***c*, **15***d*–**15***f*, respectively. The engine of FIG. **15** operates the same as the engines of FIGS. **5–7** and FIGS. **9–10** and here shows suggested valving positions for shutter valve and air bypass valves for supplying the manifolds **13** and **14** with an air charge optimum for light-load operation of the engine of FIGS. **5–7** and FIGS. **9–10**. For light-load operation, the shutter valve **5** can be closed and the air bypass valve **6** of compressor **2** (if compressor **2** is not supplying primary air charge directly to conduit **32** and intake valve **15**-B) can be opened fully or partially so that part or all of the intake air of compressor **2** can be returned to the intake of compressor **2** with little or no compression occurring there. Also, the shutter valve **3** of compressor **1** can be closed, passing the air charge away from the coolers **11** and **12**, the air bypass valve **4** would be closed to prevent re-circulation of the now compressed and heated air back through compressor **1** and whose shutter valve **3** and air bypass valves are both directing the air charge uncooled into manifold **13** and **14** for a low density heated charge for light-load operation. Preferably compressor **2** would be kept operative in order to supply the primary air charge through conduits **110**, **32** and intake valve **16**-B for a more economical scavenging-charging system.

Referring now to FIG. **16**, there is shown suggested valve positions for supplying manifolds **13** and **14** with an air charge optimum for medium-load operation for engines of FIG. **16** or for the engines of FIGS. **5–7** and FIGS. **9–10**. For medium-load operation the shutter valve **5** of compressor **2** is closed and the air bypass valve **6** would be opened to pass the air charge uncooled and without compression to the intake of compressor I where closed shutter valve **3** and closed air bypass valve **4** directs the air charge now compressed by compressor **1** past the intercoolers to manifolds **13** and **14** with the air compressed and heated by compressor **1**, for medium-load operation.

Referring now to FIG. **17**, there is shown a suggested scenario for providing the engine of FIG. **17** or for the engines of FIGS. **5–7** and FIGS. **9–10** with a high density air charge for heavy duty, high power output operation. FIG. **17** shows both shutter valves **3** and **5** open and both air bypass valves **4** and **6** closed completely so that the primary stage of compression is operative and a second stage of compression has been made operative for maximum compression of the charge and the entire air charge is being passed through the intercoolers **10**, **11** and **12** to produce a cooled, very high density air charge to manifolds **13** and **14** and to the engines power cylinders for heavy-load operation. This produces a very high mean effective cylinder pressure for high torque and power with maximum cylinder pressure being the same as, or lower than that of normal engines.

Referring now to FIG. **18**, there is shown a schematic drawing of an alternative type of auxiliary compressor **2'** for the engines of FIGS. **5–7** and FIGS. **9–10** and for any other engine of this invention and a system of providing a system for cutting out the auxiliary compressor when high charge pressure and density is not needed. For relieving compressor **2'** of work, (if the air compressed by compressor **2** does not go directly to conduit **32** and valve **16**-B to supply the primary air charge) shutter valve **5** is closed and air bypass valve **6** is opened so that air pumped through compressor **2'** can re-circulate through compressor **2**, thus relieving the compressor of compression work.

Referring now to FIG. **19**, there is shown a schematic drawing of the engines of FIGS. **5–7** and FIGS. **9–10**, illustrating means of controlling charge-air density, temperature and pressure by varying directions of air flow through various electronic or vacuum operated valves and their conduits.

FIG. **19** also shows the various charge-air paths possible by using hollow arrows to indicate heated air paths and solid arrows to indicate the more dense intercooled air paths thereby indicating how charge-air temperatures can be thermastatically or electronically controlled by dividing the air charge into two different paths. Alternatively, all of the air charge can be directed past the air coolers or all can be directed through the air coolers, as shown in FIG. **19**. Also, FIG. **19** shows how the pressure output of compressor **1** and compressor **2** can be varied by partially or fully opening air bypass valves **4** and **6** or by completely closing one or both of these control valves. An engine control module (ECM) **27** is suggested for controlling the various operating parameters of the engines of this invention.

Referring now to FIG. **20**, there is shown is a schematic drawing depicting an alternate arrangement in which an electric motor **34** preferably drives the compressor(s) of any of the engines of the present invention.

Charge-air Cooler Bypass (ACB) "Shutter Valve" Control

In this section are described aspects of preferred control components which find application in connection with any of the engines (4-stroke and 2-stroke) of the present invention.

Outline: Valves **3** and **5** are charge-air-cooler bypass solenoid (ACB) valves. In charge-air cooler bypass control, the intake air is switched between two routes by valves **3** and **5**, independently of each other: either (a) valve **5** directs the flow from compressor **2** directly to the intake conduit of compressor **1** or (b) through the charge-air cooler **10** before flowing to the intake conduit of compressor **1**. Valve **3** directs the flow from compressor **1** either (a) to the conduit **111**/**121**/**122** leading directly to the intake manifolds **13** and **14** or (b) it passes the air charge through charge-air coolers **11** and **12** before it flows to manifolds **13** and **14**.

An engine control module (ECM) **27** can control the air cooler bypass valves **3** and **5**. The bypass valves may be a shutter type valve to pass all or none of the air charge in either direction or valves **3** and **5** may be of a helical solenoid or other type of valve which can pass part of the air charge through bypass conduits **121** and **122** and part through air coolers **10**, **11** and **12** for fine control of the temperature and density of the air charge. The ECM could receive signals from sensors such as an engine coolant sensor, a crankshaft position sensor, throttle position sensor, camshaft position sensor, a manifold absolute pressure sensor and a heated oxygen sensor.

Air Bypass Valve (ABV) Control

Outline: To provide optimum air charging pressure for differing engine operations conditions, the ECM **27** can send signals to control air bypass valves **4** and **6**. These valves could be on-off solenoid valves, possibly vacuum operated, or they could be helical solenoids or other type of valve which could open part way or all way in order to re-circulate part or all of the air charge back through the inlets **110** and **8** of compressors **1** and **2** in order to reduce or eliminate entirely the pumping pressure of either compressor **1** or compressor **2**, or both. Similar arrangements of air pressure control could be used for additional stages of air compression if additional stages are used.

The operation could be thus: The ABV valves **4** and **6** can be controlled by signals from the ECM **27** to control the opening angle of valves **4** and **6** to provide the optimum air charging pressures for various engine loads and duty cycles. When ABV **6** is opened partially some of the air pumped through compressor **2** is passed back into the intake **8** of compressor **2** to reduce compression pressure. When ABV **6** is opened fully all of the charge of compressor **2** is passed back through compressor **2**, thus compressor **2** only pumps the charge through with no pressure increase. The system can work the same for valve **4** which could bypass some of the air charge pumped by compressor **1** back into the intake conduit **110** of compressor **1** in order to reduce air charge density.

With this arrangement, combined with the arrangement of ECM **27** control of charge-air cooler bypass system for variable valves **3** and **5**, the temperature, density, pressure and turbulence of the charge-air can be managed to produce the desired power and torque levels and emissions characteristics in the power cylinder of the engine.

Engine conditions that could be monitored by ECM **27** in order to effect proper engine conditions in regard to control of ABV valves **4** and **6** could include a throttle position sensor (or fuel injection activity sensor), intake air temperature sensor at various points, manifold absolute pressure sensor, camshaft position sensor, crankshaft position sensor, exhaust temperature sensor, a heated oxygen sensor and/or other sensory inputs known to be used in internal combustion engines.

The ECM **27** can control both the shutter valves **3** and **5** and the air bypass valves **4** and **6** in order to maintain the optimum air charging density pressure and temperature at all engine operating duty cycles.

Alternate Combustion Systems

Referring now to FIG. **21**, there is shown a schematic transverse view of a pre-combustion chamber **38'**, a combustion chamber **38**, a piston crown **22** and associated fuel inlet **36**, a sparking plug **37**, an air or air/fuel mixture inlet **8'** duct, intake valve **16**, an exhaust duct **18'** an exhaust valve **17** suggested for liquid or gaseous fuel operation for the engines of this invention or for any other internal combustion engine.

There are many choices of systems for compression or spark ignition combustion for the engine of this invention, as shown in FIG. **1** through FIG. **33**. Every fuel from avgas to heavy diesel fuels, including the alcohols and gaseous fuels can be spark ignited (SI) in this engine. One good SI system would be similar to the system shown in FIG. **21** for compressed natural gas, propane, hydrogen, gasoline, alcohols or diesel fuel. In this system, an extremely fuel rich mixture constituting the entire fuel charge is, preferably, injected into the pre-combustion chamber **38'**. The fuel could be injected through fuel duct **36** with or without air blast injection, the air charge, some of which can accompany the fuel charge would be compressed into the pre-combustion chamber **38'** by piston **22** during the compression stroke. Additional air with or without additional fuel, could be introduced into the cylinder proper either on the intake stroke or on the compression stroke through intake conduit **8'**. In either case the second combustion stage in the cylinder proper would be with a lean mixture.

The two-stage combustion system shown in FIG. **21** will operate in this manner:

1. Pre-Combustion (first stage)

Pre-combustion occurs in the pre-combustion chamber **38'** when fuel in an amount much in excess of the amount of oxygen present is injected and ignited (injector not shown). This oxygen deficiency along with the cooler, turbulent charge and lower peak temperatures and pressures greatly

reduces the formation of oxides of nitrogen. The combination of the hot pre-combustion chamber wall and intense turbulence promotes more complete combustion.

2. Post-Combustion (second stage) Post-combustion takes place at lower pressure and relatively low temperature conditions in the space above the piston in the cylinder as the gases expand from the first stage pre-combustion chamber into the cylinder proper. If there is additional fuel in the cylinder proper, the leaner mixture is ignited by this plasma-like blast from the pre-combustion chamber. The low temperature and the admixture of burned gases prevent any further formation of oxides of nitrogen. Excess air, a strong swirling action, and the extended expansion process assure more complete combustion of carbon monoxide, hydrocarbons, and carbon.

The results of the engine of this invention using the pre-combustion chamber **38'** of FIG. **21** are: higher thermal efficiencies due to the greater expansion, along with a cooler exhaust and a lower level of polluting emissions including oxides of nitrogen, and in addition for diesel fuels, lower aromatics and particulates.

Referring now to FIG. **22** there is shown a schematic transverse sectional view of an optional cylinder of the engine of this invention which will convert the 2-stroke engine of FIGS. **8** through **33** to a one-stroke cycle engine and will convert the 4-stroke engines of FIG. **1** through FIG. **7** and FIG. **33** to operate in a 2-stroke cycle.

By building any 2-stroke engine with all power cylinders double acting, the power to weight ratio can be doubled over the basic engine. One end of the cylinder fires and the other end is scavenged on each stroke for a nominal one stroke cycle engine in the engines of FIG. **8** through FIG. **33**. Use of double-acting power cylinders in the 4-stroke engine of FIG. **1** through FIG. **7** and FIG. **33** converts the engine to a 2-stroke engine because one end of the cylinder is scavenged and one end is fired during each crankshaft rotation.

In the design of FIG. **22** needed variation of beam **39** length is accomplished by the beam end forming a scotch yoke **40** and fitting over the wrist pin **41** of the piston.

The double ended piston **22"** can be linked to the end of a vertical beam **39** that pivots at the lower end **42**. A connecting rod **19'** is joined between the midpoint of the beam and the crankshaft **20'**.

Since the crankshaft **20'** itself 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 (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** and 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 less friction and stress on parts. This could also be important to allow use of alternate fuels.

Referring now to FIG. **23**: These same advantages hold true for the alternate design (FIG. **23**) 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 part **46** of the engine and the terminal end of beam **39'** being connected to connecting rod **19** by a pin **47**.

Alternately 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 will be especially of importance 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 can be used in all of the designs of this invention.

Referring now to FIG. **24**: 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 burn time of a conventional 2-stroke or 4-stroke engine.

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'** top dead center (TDC) occurs at bottom dead center (BDC) of the crank **48**. At this point, crankpin motion around piston **22'** top dead center 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.

The extra burn time provided by the design of FIG. **24** 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 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. **24** itself 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 also provides for nearly twice the combustion time of a conventional engine during the critical burn period. This is because piston top dead center occurs at bottom dead center (BDC) of the crank.

The Engine 100^{25} of FIG. **25**

Referring now to FIG. **25** of the drawings, there is shown a six cylinder reciprocating internal combustion engine in which all of the cylinders **7***a*–**7***f* (only one (**7***f*) of which is shown in a sectional view) and associated pistons **22***a*–**22***f* are adapted to operate in a 2-stroke cycle and all cylinders are used for producing power to a common crankshaft **20** via connecting rods **19***a*–**19***f*, respectively. A compressor **2** supplies air to scavenging ports **52** by way of optional shut-off valve **33**-M and conduit **32** and to cylinder charge inlet

valve(s) **16** and **16'** by way of conduits **15**. The engine of FIG. **25** is adapted to operate in a 2-stroke cycle so as to produce six power strokes per revolution of the crankshaft **20**. To this end, compressor **1** takes in an air charge which may have been previously subjected to compressing to a higher pressure, via an admission control valves **5** and **6** through an intake conduit **110**, leading from compressor **2** by way of intercooler **10** or bypass conduit **104** and shutter valve **5**. During operation of the engine of FIG. **25**, compressor **2** receives atmospheric air through inlet opening **8**, pre-compresses the air charge into conduit **101** leading to control shutter valve **5** which in response to signals from the ECM **27**, to shutter valve **5** and air bypass valve **6**, will direct the compressed charge through intercooler **10** or through cooler bypass conduits **104** to compressor **1**. The air charge is compressed within compressor **1** by its associated piston **131**, and the compressed air charge is forced through an outlet into a high pressure transfer conduit **109** which leads to control shutter valve **3** which, if open, directs the air through intercooler **11** and **12** to manifolds **13** and **14** or, if closed, through a conduit and air bypass valve **4** which can direct part of the air charge back through inlet conduit **104** of the compressor **1**, or valve **4** if fully closed, directs all of the charge from compressor **1**, in response to signals from the engine control module (ECM) **27**, through the intercoolers **11** and **12** or through the bypass conduit **111/121/122** into manifolds **13** and **14**. Manifolds **13** and **14** are constructed and arranged to distribute the compressed air charge by means of branch conduits **15***a*–**15***f* to inlet valves **16** and **16'** of the cylinder **7***a*, and to the remaining five power cylinders **7***b*–**7***f*. In an alternate embodiment, instead of providing scavenging air through conduit **32'**, scavending air is provided through shut-off valve **49** and conduit **50** and pressure reducing valve **25** to air box **51**, through conduits **125***a*–**125***f* to scavenging ports **52***a*–**52***f*.

The engine **100**[25] shown in FIG. **25** has a camshaft which is arranged to be driven at the same speed as the crankshaft in order to supply one working stroke per revolution for the power pistons. The compressor can be reciprocating, comprised of one or more stages of compression with one or more double-acting cylinders, one is shown, **1** in FIG. **25**. The compressor can be driven by associated connecting rods **19***g* to crankshaft **20** which can have throws of different lengths for different length piston strokes for the air compressor(s) than those of the power pistons. In addition, compressor **1** can be driven by a second crankshaft (not shown) which is driven by a gear meshing with a step-up gear mounted on the common crankshaft. The ancillary rotary compressor, a Lysholm type is shown **2**, can be driven by a V-pulley being rotated by a ribbed V-belt and has a step-up gear arranged between the V-pulley and the compressor drive shaft. The rotary compressor **2** could also have a variable speed, or two speed drive, as in some aircraft engines.

The operation of engine **100**[25] shown in FIG. **25** is thus: Charge-air is inducted into the inlet opening **8** of compressor **2**. From there it is pumped through the compressor **2** where it is directed by shutter valve **5** through the intercooler **10** or through a conduit to air bypass valve **6** where it is directed to the inlet of compressor **1**. The charge is then pumped by compressor **1** through the outlet valve to shutter valve **3** which directs the air charge either through intercoolers **11** and **12**, to manifolds **13** and **14** or into a conduit leading to air bypass valve **4** which can direct a part of the charge back through the inlet of compressor **1** or valve **4** directs the charge wholly or partially to the shutter valve **3** which directs the charge wholly or partially through intercoolers **13** and **14**, or directly to manifolds **13** and **14** which distributes the temperature-adjusted charge-air to cylinder **7** inlet valves **16** and **16'** to each power cylinder of the engine. An off-and-on control valve (not shown) and conduit **32'** directs air to air box **51** and to scavenging ports **52***a*–**52***f* in the bottom of cylinders **7***a*–**7***f*. In the alternates embodiment (shown in phantom in FIG. **25**), the scavenging air is directed through pressure reduction valve **25**, arranged on conduit **50** to provide and adjust scavenging air pressure from compression **1**. Another option to reducing the manifold air pressure for scavenging the cylinders **7***a*–**7***f* is to use the manifold air through conduit **50**, air box **51** and intake ports **52***a*–**52***f* without reducing the pressure from manifolds **13** and **14**. The air would be used at full pressure for scavenging by the scavenging ports **52***a*–**52***f* in FIG. **25** and through inlet port **52"** and exhaust port **52'** in FIG. **30**, which ports **52***a*–**52***f*, **52'** and **52"** would be constructed much smaller than normally done. In this instance, although the scavenging ports were smaller-than-normal, the higher-than-normal pressure scavenging air would be very efficient. Several means of scavenging the cylinders are suggested herein. FIG. **26** illustrates more clearly (although in phantom) the preferred system of supplying low pressure scavenging air. Conduit **32'** and valve **33** (shown in phantom in FIG. **26**) channels air from conduit **110** from compressor **2** to conduit **50** which supplies scavenging air to air box **51**.

The engine control module (ECM) **27** (see, for example, FIG. **26**) controls valves **3**, **4**, **5**, and **6** in order to adjust the pressure, temperature and density of the charge going to the combustion chambers and valve **25**, and can selectively direct a portion, a portion at a reduced pressure of the air charge to scavenging ports **52** and can control valve **53** and valves **49'** to open or close to select the mode of scavenging desired. The ECM can also control a variable-valve-happening control system to adjust the valve opening time and duration of opening time of inlet valves **16** and exhaust valves **17** in relationship to the degree or angle of rotation of crankshaft **20**, in order to adjust the compression ratio of the engine for optimum performance in regard to power, torque, fuel economy, fuel characteristics and to scavenging mode desired.

The preferred operation of the power cylinders shown in FIG. **25** is in this manner:

After blowdown and scavenging of the cylinder **7** has taken place the cylinder is now filled with fresh air, and piston **22** has closed exhaust ports **52** and the piston **22** is in its scavenging-charging stroke and is rising with the exhaust valve **17** still open, at any point, perhaps as early as 120 to 90 degrees before top dead center, the exhaust valve **17** is closed to establish the compression ratio and begin compression, intake valve **16**, **16'** are opened at that time or later in order to produce the desired charge density and weight desired, the compressed air charge or fuel air mixture is injected through intake valve **16**, **16'**, intake valve **16**, **16'** is then closed. Compression of the charge which started at point X, the point where exhaust valve **17** was closed, continues with the compression ratio being established by the cylinder clearance volume remaining at point x, divided by the combustion chamber volume. Fuel can be injected into the secondary compressed air stream being injected into the combustion chamber or injected into a pre-combustion chamber (one is shown in FIG. **21**) or may be injected directly into the combustion chamber. After the closure of intake valve **16**, **16'**, the fuel or more fuel can be injected into the midst of the charge swirl for a stratified charge combustion process, or as in compression ignited engines fuel can be injected directly into the combustion chamber, perhaps

directly onto a glow plug, if suggested pre-combustion chamber is used or not, and can be injected continuously during part of the expansion stroke for a mostly constant pressure combustion process.

The fuel-air mixture is ignited by spark plug, by compression ignition, or by glow plug at the point deemed most efficient, preferably before top dead center of the compression stroke of piston **22**. The expansion stroke of piston **22** takes place as the expanding gases force the piston toward bottom dead center. Near the end of the power stroke, perhaps about 40° before bottom dead center, scavenging ports **52** are uncovered, near the same time exhaust valve(s) **17** in the engine head are opened and a rapid blowdown and scavenging takes place in any of four ways as shown in FIG. **27**, FIG. **28**, FIG. **29** and FIG. **30**. In any case the exhaust valves **17**, **17'** remain open past bottom dead center and for a significant part of the scavenging-charge-adjusting stroke in order to establish the engines compression ratio.

Referring now to FIG. **26**, there is shown a schematic drawing showing an engine similar in structure and operation to the engine **100**^{25} of FIG. **25**, having two compressors, but differing in that compressor **1** is depicted as a Lysholm rotary compressor, and compressor **2** is depicted as a turbo compressor, and having one air cooler for the secondary compressor, two air coolers for the primary compressor, dual manifolds, with shutter controls, air bypass controls and conduits for different air paths. Also shown is an engine control module (ECM) **27** which can control charge and scavenging air pressures, density and temperatures in order to effect the desired output and emissions characteristics of the engine. Alternate sources of scavenging air are shown, the preferred one being from conduit **110** by way of conduit **32'**. Air paths are shown by arrows, hollow arrows for uncooled compressed air and solid arrows for cooled denser air. Also shown are air bypass valves (in this case both closed) which, with the shutter valves (one of which is closed and one of which is partly open, the latter to allow cooling of part of the charge), can control the charge temperature, weight and density as required for best engine performance.

Referring now to FIG. **27**, there is shown one system of efficient scavenging of the exhausted products of the engine of FIG. **25**;

Scavenging System A (FIG. **27**)

Blowdown of exhaust occurs at from about 40° before bottom dead center to perhaps 40–50° after bottom dead center, with exhaust valves **17** opening at approximately the same time the ports **52** are opened and remaining open after bottom ports are closed by piston **22**, and closing later causing a low compression ratio.

Scavenging air can be supplied from a manifold with perhaps a pressure-reducing valve **25** on conduit **50** or, preferably scavenging air can be supplied from conduit **32'** from ancillary compressor **2**, (shown in phantom). In this case, bottom ports **52** open shortly before exhaust valves **17** open. Blowdown occurs through bottom ports **52** out through bottom exhaust conduit and valve **53** to main exhaust pipe **18**, at same time or shortly after, exhaust valves **17** open and blowdown of the exhaust occurs both at the top of the cylinder through exhaust valves **53** and **17**, and through exhaust manifold **18'** and pipe **18** to the atmosphere. The exhaust valve **17** then stays open through a significant part of the 2nd or exhaust-charge stroke for additional scavenging, this part by positive displacement. During this scavenging-charging stroke the exhaust valve **17** may be closed at any point after the first 20 percent of piston **22** travel. Now at any point with cylinder **7** being now filled with fresh air, exhaust valve **17** can close and intake valve **16'** open to admit pressurized air which has its temperature adjusted to what is deemed proper. The later in the exhaust-charging stroke the exhaust valve **17** is closed, the lower is the compression ratio of the engine established. If closed early enough the effective compression ratio can be as much as 13 or 16 to 1, if closed later the effective compression ratio can be as low as 2:1. At any point after exhaust valve **17** has closed, and the compression ratio has been established, and before piston **22** has reached top dead center, the air charge, with temperature density and pressure adjusted may be introduced by opening and then closing intake valve **16**. All of the operating parameters suggested would depend on the duty cycle of the engines, e.g., power requirements, efficiency, emissions considerations and the fuel used.

An engine control module (ECM) **27** is shown with connections to the critical control valves of the engine which can be adjusted according the conditions signaled to the ECM **27** from various sensors in the engine.

Referring now to FIG. **28**, there is shown a second system of efficiently scavenging the engine of FIG. **25**;

Scavenging System B (FIG. **28**)

Exhaust blowdown occurs through exhaust valves **17** only, with scavenging air being supplied by compressor **2** by way of conduit **32'**, or alternatively from manifolds **13** and **14** through conduits **50** past control valve **49** and optional pressure control **25** into air box **51** and through scavenging ports **52** in the bottom of the cylinders **7**, up through the cylinder **7**, out exhaust valves **17** and through exhaust pipe **18**, with valve **53** being closed. In this system as piston **22** approaches bottom dead center in the power expansion stroke, ports **52** would be uncovered by piston **22** and as blowdown occurs pressurized air would be injected through all bottom ports **52** and would sweep combusted products through exhaust valves **17** which open perhaps before ports **52** for the exhaust blowdown. The bottom ports can be constructed to open at perhaps 40° before bottom dead center and could close at the same point after piston begins its second stroke. The exhaust valves **17** could remain open after bottom ports **52** are closed to aid in scavenging by positive displacement by piston **22** and to establish the desired compression ratio which is established by the point at which exhaust valves **17** close.

During this scavenging-charging stroke of piston **22** the cylinder **7** being now filled with fresh air, the exhaust valve **17** may be closed at any point after the first **20** percent or so of piston **22** travel. Now at any point exhaust valve **17** can close and intake valve **16** can open to admit highly pressurized air which has its temperature and density adjusted to what is deemed proper. The later in the exhaust-charging stroke the exhaust valve **17** is closed, the lower is the effective compression ratio of the engine established. If closed early enough the effective compression ratio can be as much as 13 or 19 to 1, if closed later the effective compression ratio can be as low as 2:1. All of the operating parameters suggested would depend on the duty cycle of the engines, e.g., power requirements, efficiency and emissions considerations and the fuel used.

An engine control module **27** is suggested for use as shown for controlling the various operating conditions desired and as signaled from the engine's various sensors.

Referring now to FIG. **29**, there is shown a third efficient system of scavenging the engine of FIG. **25**;

Scavenging System C (FIG. **29**)

This scavenging system would be that shut off valves **49'** would be closed, (or valves **25** and **49** could be eliminated),

with bottom ports opened to the atmosphere by valve **53**, one inlet valve **16** leading from manifolds **13** and **14** to cylinder **7** could be opened for a very short period of time by a cam, perhaps by a small lobe on a cam that has a large lobe to open the same valve (as **21**-C in FIG. 11) at a different crank angle, at the same time ports **52** were uncovered by piston **22** and exhaust valves **17** were opened. The high pressure air would quickly sweep combusted gases through ports **52** and exhaust valves **17**, through their respective exhaust pipes **17** and **17'** to the atmosphere. The intake valve **16** would close quickly, no later than the time exhaust ports **52** closed. The exhaust valve would remain open for further scavenging and for the reduction of the compression ratio of the engine. Alternatively bottom exhaust valves **53** would be closed and as bottom ports **52** were uncovered by piston **22**, exhaust valves **17** would also open earlier for blowdown, air from the airbox **51** supplied by conduit **32** would blow into ports **52** and scavenge the cylinder **7** through exhaust valves **17**.

During this scavenging-charging stroke the exhaust valve **17** is closed at a point after the first 20 percent or so of piston **22** travel. At any point after exhaust valve **17** has closed, the cylinder **7** being now filled with fresh air, and the compression ratio having been established, and before piston **22** has reached top dead center, additional (secondary) air charge, with temperature density and pressure adjusted is introduced when needed by opening a second intake valve **16** and/or by another lobe **21**-C on the same cam (see **21**-C, FIG. **11**) opening the same intake valve again. All of the operating parameters suggested would depend on the duty cycle of the engines, e.g., power requirements, efficiency and emissions considerations and the fuel used. The later in the exhaust-charging stroke the exhaust valve **17** is closed, the lower is the compression ratio of the engine established. If closed early enough the effective compression ratio can be as much as 13:1 or 22:1, if closed later the effective compression ratio can be as low as 2:1.

An engine control module could control all of the conditions required of the engine.

Referring now to FIG. **30**, there is shown a fourth system of efficient scavenging the engine of FIG. **25**.

Scavenging System D (FIG. **30**)

In this system exhaust blowdown occurs through the top exhaust valves **17** and through part of the bottom scavenging ports **52'** which open just before bottom dead center, perhaps 40°, and simultaneously with or just after the top exhaust valves open. At the time bottom ports **52'** are opened, or shortly after, exhaust valves **17** are also opened, or, valve **53** leading to bottom exhaust line **18** is already open, and exhaust blowdown occurs over the next, 40° or so after bottom dead center, with scavenging air being injected through at least one of the bottom ports **52"** which has been constructed to receive pressurized air from air box **55** supplied by conduit **32'** or **50** at such a time the ports **52'** are opened by piston **22** and the pressure in cylinder **7** has dropped below the pressure in air-box **55**. After ports **52'** are closed, exhaust valves remain open through a significant part of the second or exhaust-charge stroke of piston **22** for additional scavenging by positive displacement and in order to establish a low compression ratio.

During this scavenging-charging stroke the cylinder **7** being now filled with fresh air, exhaust valve **17** may be closed at any point after the first 20 percent or so of piston **22** travel. Now at any point exhaust valve **17** can close to establish the compression ratio and inlet valve **16** can open to admit a secondary pressurized air charge which has its temperature and pressure adjusted to what is deemed proper. The later in the exhaust-charging stroke the exhaust valve **17** is closed, the lower is the compression ratio of the engine established. If closed early enough the effective compression ratio can be as much as 13:1 or 22:1, if closed later the effective compression ratio can be as low as 2:1. All of the operating parameters suggested would depend on the duty cycle of the engines, e.g., power requirements, efficiency and emissions considerations and the type of fuel used, and can be controlled by an engine control module which receives signals relating conditions in certain engine areas and which are relayed to the ECM **27**.

Referring to FIG. **31**, there is shown a schematic drawing depicting an alternate arrangement in which an electric motor **34** preferably drives the air compressors of an engine similar to that of FIG. **25**.

Referring now to FIG. **32**, there is shown a schematic drawing showing the 2-stroke engine of FIG. **25** and FIG. **26** and having only one compressor **1** for supplying both scavenging and charge-air. Also shown are a shutter valve **3** and an air bypass valve **4**, valves **16** and **17** controlling charge and scavenging air and valves **53** and **53'** for releasing exhaust blowdown out of the cylinder bottom ports **52** through exhaust conduit **18** to the atmosphere. Thus the engine of FIG. **32** can perform all of the feats described for the engine of FIG. **25** and described for the engine of FIG. **25**, FIG. **26**, FIG. **27**, FIG. **28**, FIG. **29**, FIG. **30** and FIG. **32**. Also shown is an engine control module (ECM) **27**, and connections to various valves in order to manage the charge and scavenging air temperature, density, weight and pressure, and the pressure and path of the scavenging air to achieve the desired results from the engine. Arrows show the paths possible for the heated (hollow arrows) air and the cooled (solid arrows), air, and for the charge-air to pass through the air bypass valve **4**, all in order to adjust air pressure, density, weight and temperature for optimum engine performance.

The Engine **100**^{33} System of FIG. **33**

Referring now to FIG. **33**, there is illustrated a six cylinder internal combustion engine in which part of the cylinders **62** through **65** are used for producing power and two of the cylinders, cylinders **66** and **67**, are used for compressing the air necessary to operate the engine. A supercharger **57**, in this case preferably a Lysholm type, is used to boost the atmospheric pressure air received through air intake **8'**, before the air enters compressor cylinders **66** and **67**. A shutter valve **3'** and air bypass valve **4'** re-circulate the charge-air back through compressor **57** when both are opens to lessen compressor work and reduce charge densities for light-load operation. When air bypass valve **4'** is closed shutter valve **3'** can open or close to send the air charge to the cylinders cooled or uncooled, respectively, in order to manage combustion temperatures and temperatures for optimum performance.

The second stage of compression is transferred from compression cylinders **66** and **67** through conduits **201**, **202** to shutter valve **4"** which, when closed, sends the compressed charge through conduit **204** and intercooler **11** and conduit **205** to the engine manifold **58'** in a cooled condition. If opened, shutter valve **4"** directs the charge away from cooler **11** through conduit **203** and **205** to the power cylinders without cooling.

By having its camshaft arranged to rotate at one-half crankshaft speed, the engine **100**^{33} operates in a 4-stroke cycle, with a low compression ratio, an extended expansion