United States Patent

Ginter

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[45] **Mar. 28, 1972**

[54] ENGINE SYSTEM AND THERMOGENERATOR THEREFOR

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[73] Assignee: **Ginter Corporation,** Portland, Oreg.

[22] Filed: **Mar. 18, 1969**

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[52] **U.S. Cl.** **60/39.26,** 60/39.3, 60/39.6, 60/39.63, 60/39.65

[51] **Int. Cl.** **F02g 1/02,** F02d 1/06, F02d 35/02

[58] **Field of Search** 60/39.6, 39.26, 39.3, 39.65

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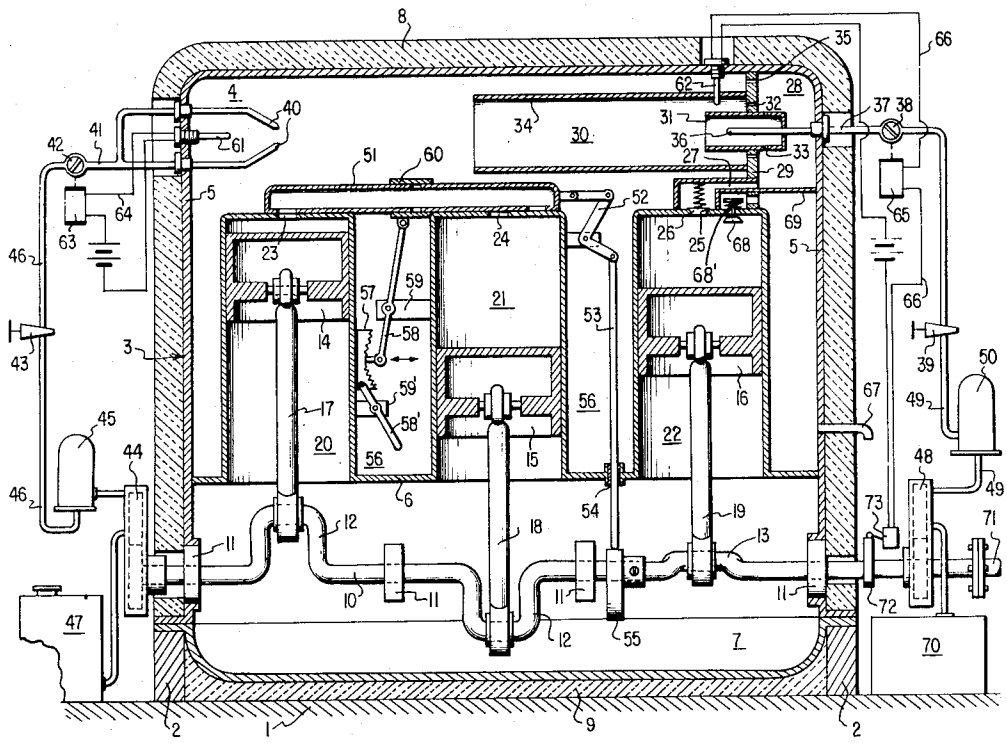
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Primary Examiner—Mark M. Newman
Assistant Examiner—R. B. Cox
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[57] **ABSTRACT**

This invention combines ICE and ECE principles in a single engine which provides working fluid at pressures and temperatures which are nearly constant, after start-up, in which fuel is burned in response to a decrease in fluid pressure as fluid is withdrawn while at the same time resupplying air compressed adiabatically to provide ignition temperature. When products of combustion are above permissible operating temperature water is supplied to absorb the entire heat of combustion becoming superheated steam to double the working fluid, without external cooling.

32 Claims, 13 Drawing Figures



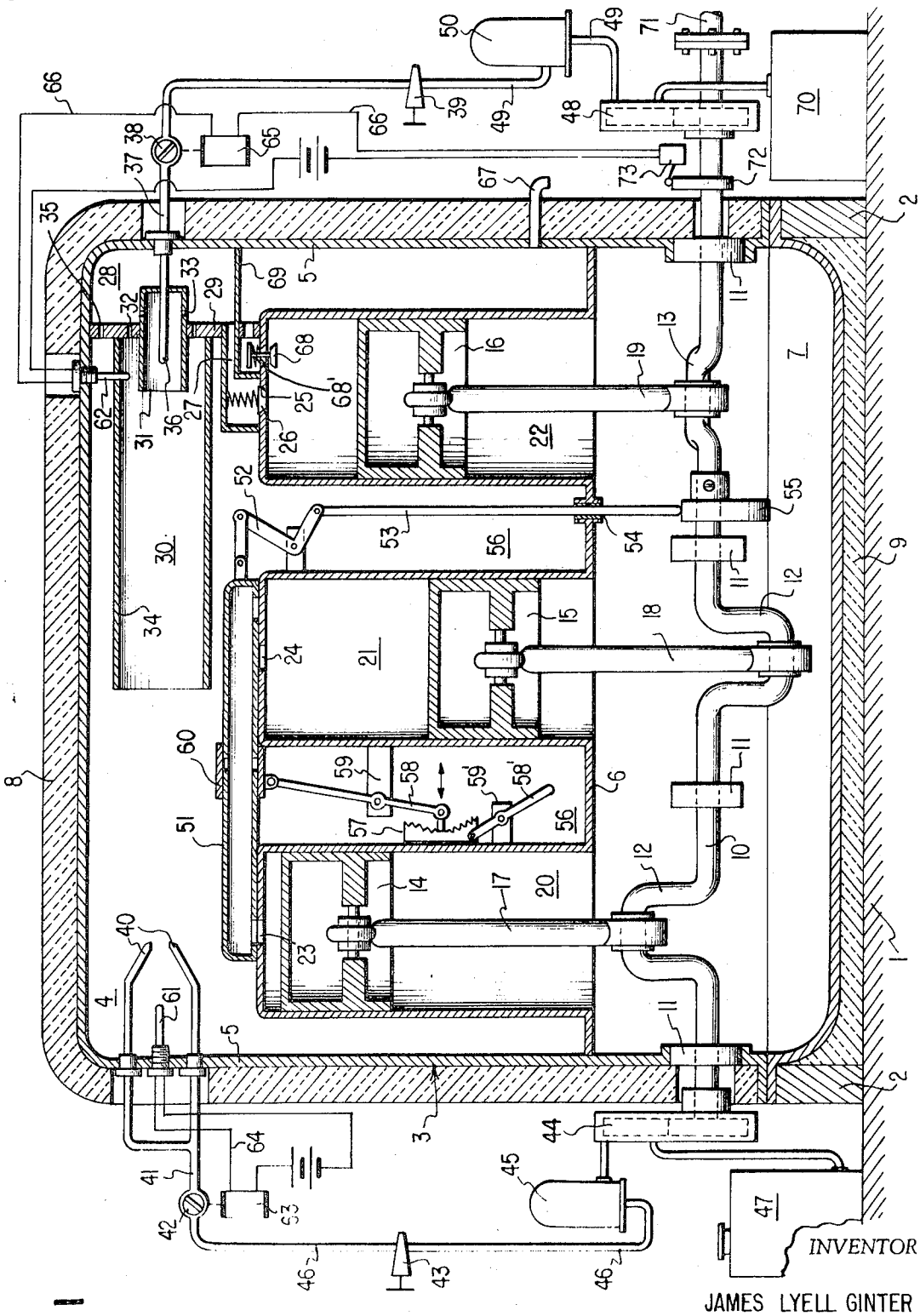


FIG. 1

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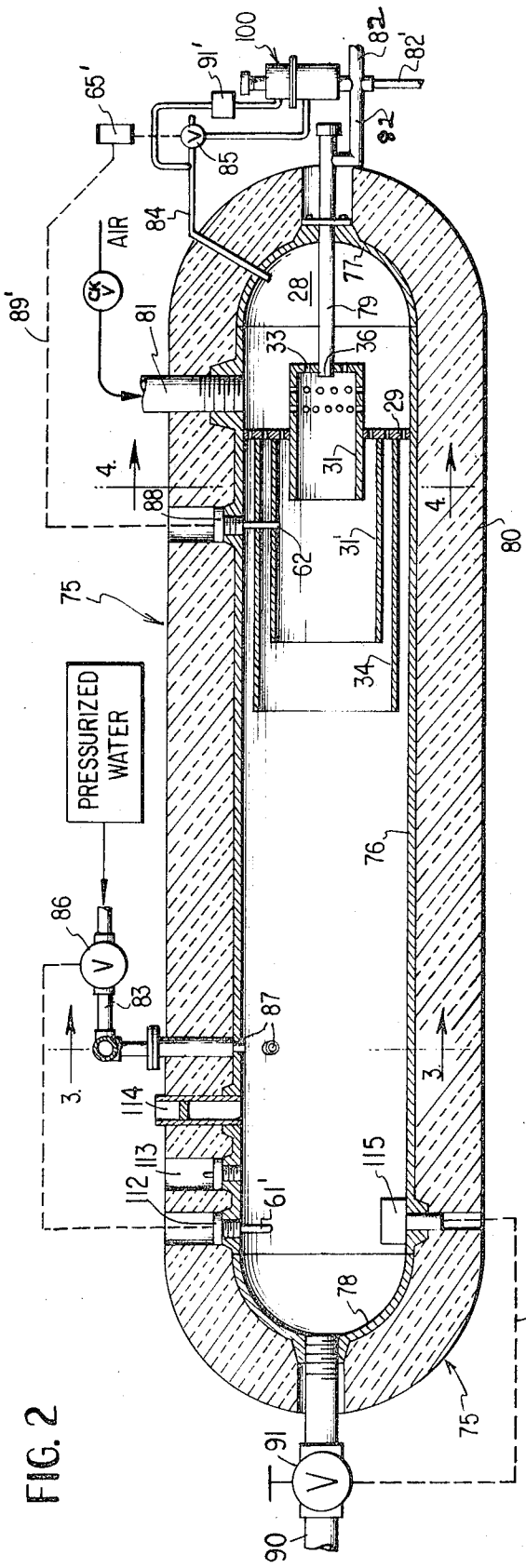


FIG. 2

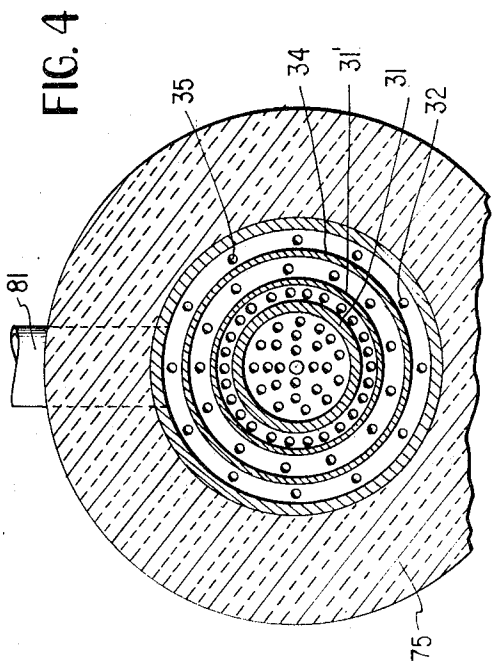


FIG. 4

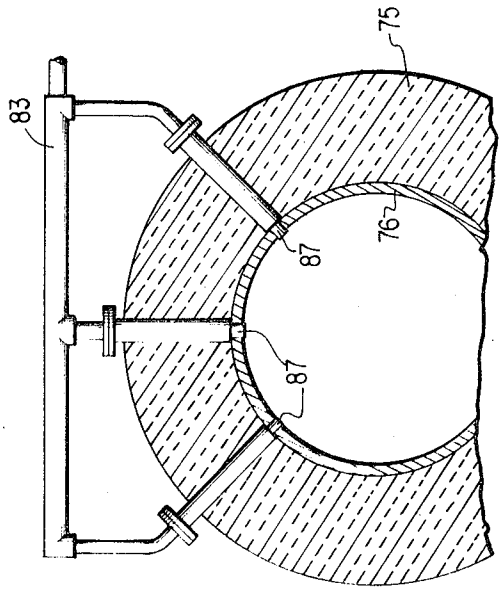


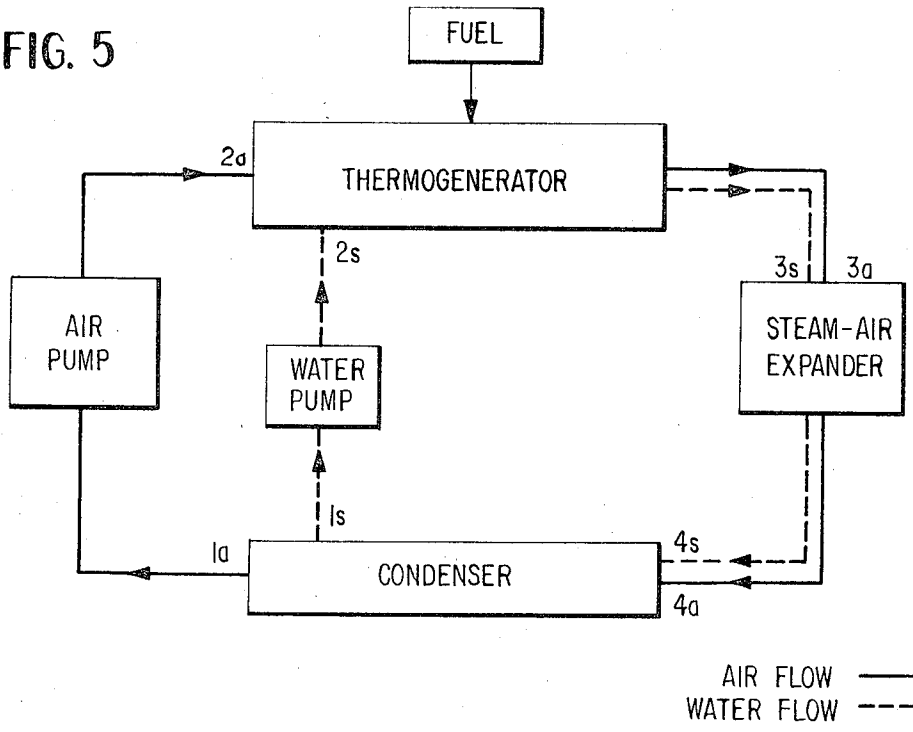
FIG. 3

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FIG. 5



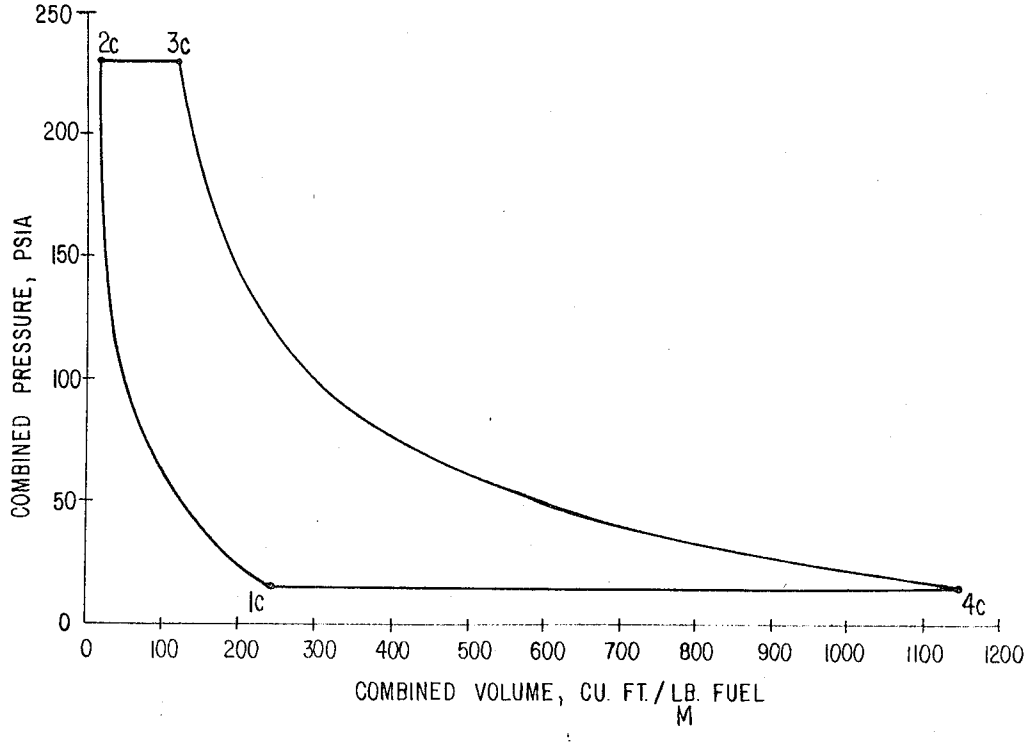
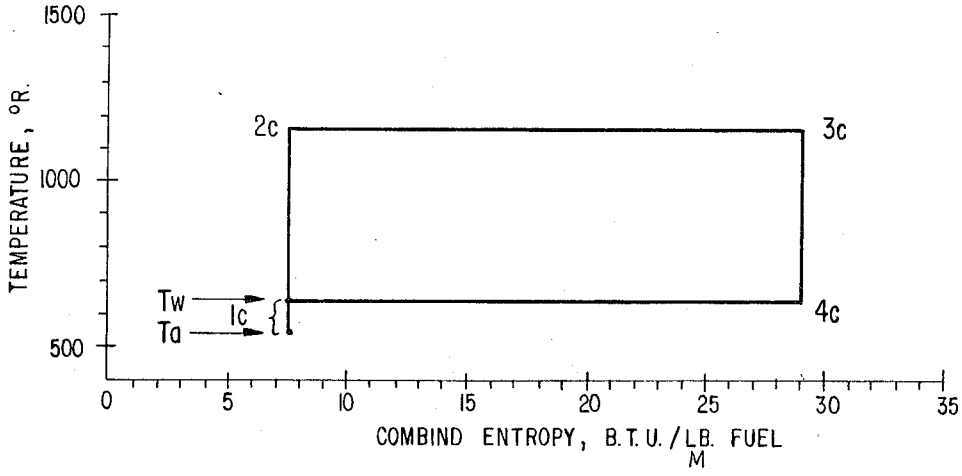


FIG. 6 COMBINED P-V DIAGRAM (AIR-WATER CYCLE)
BASED ON ONE POUND OF FUEL

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FIG. 7 COMBINED T-S DIAGRAM (AIR-WATER CYCLE)
BASED ON ONE POUND OF FUEL



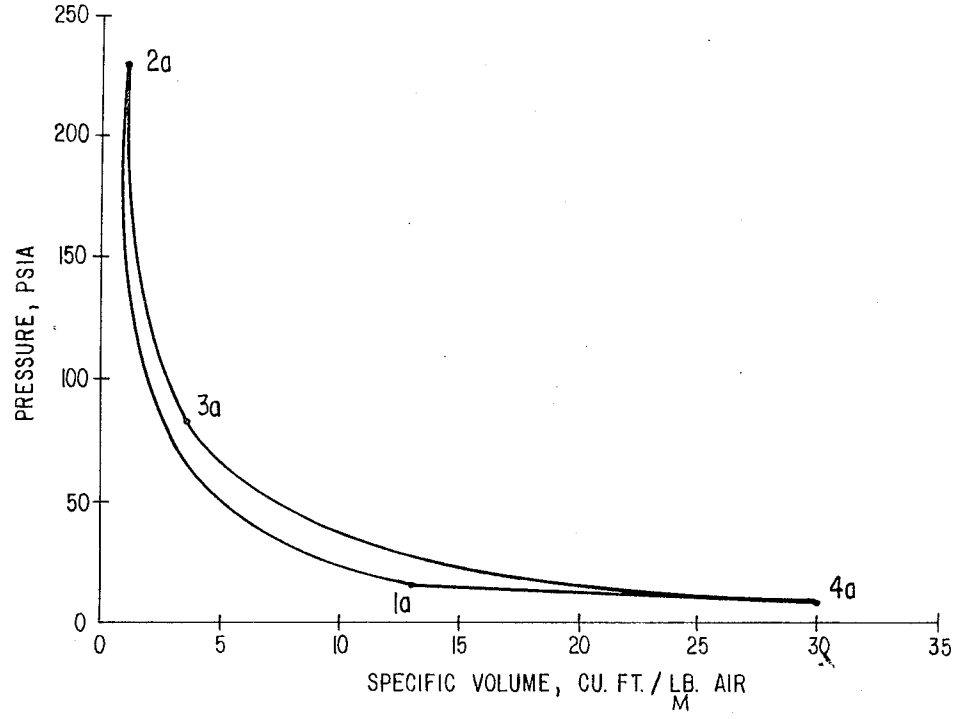
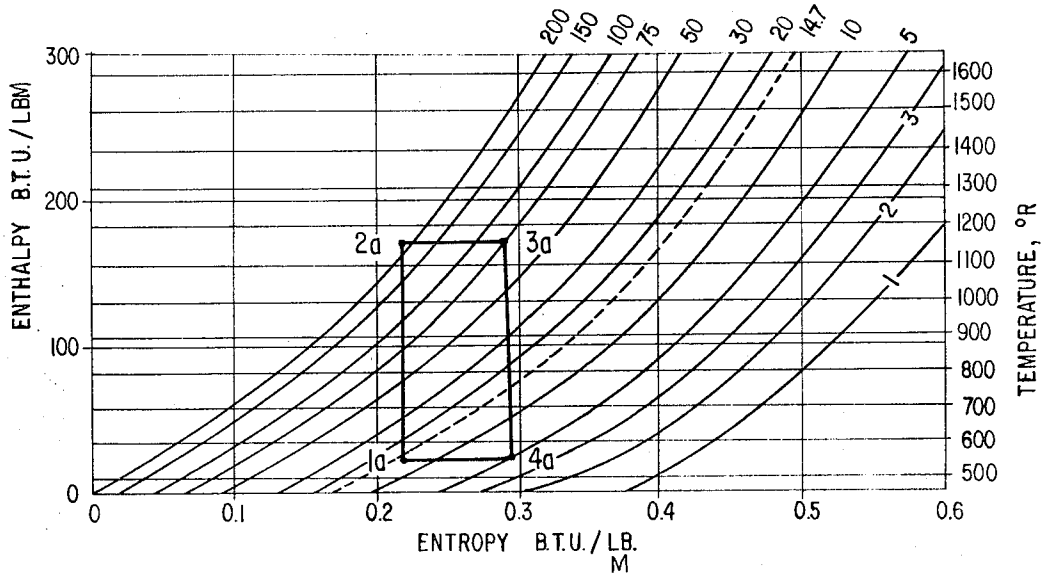


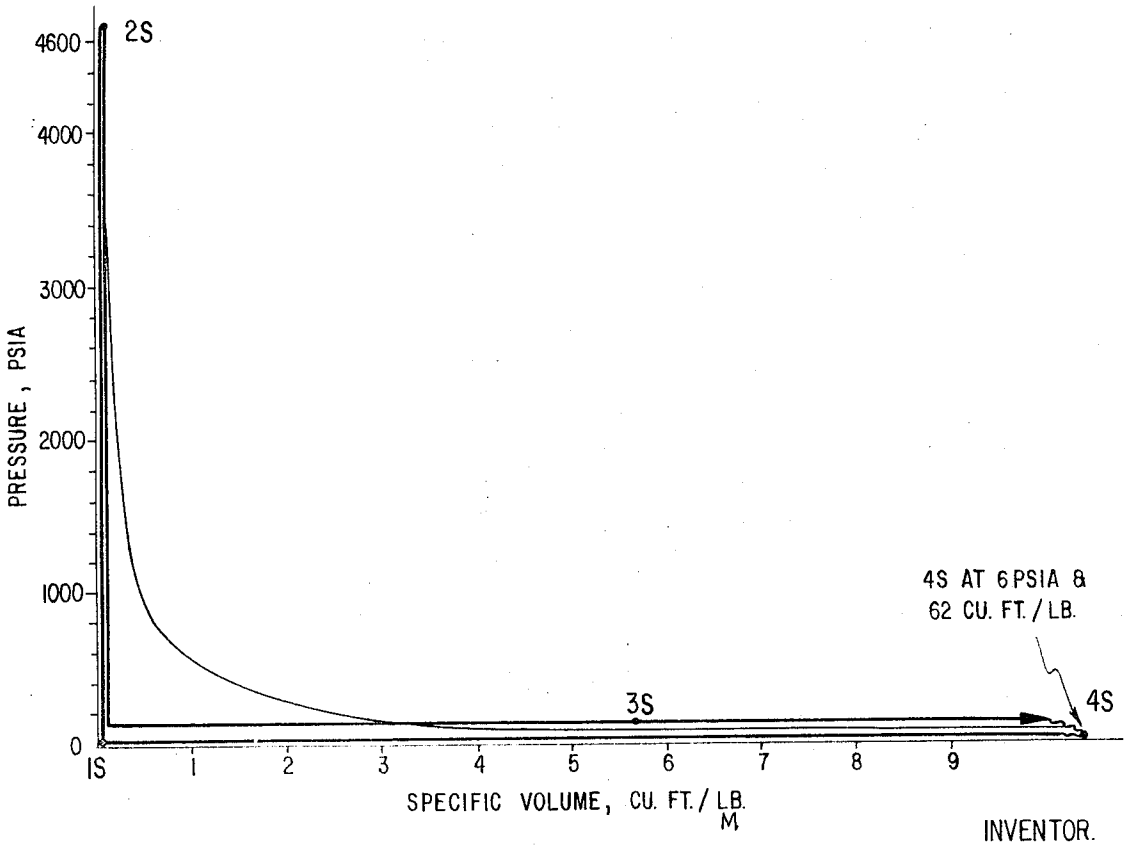
FIG 8 P-V DIAGRAM FOR AIR CYCLE

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FIG. 9 H-S DIAGRAM (T-S) FOR AIR CYCLE.





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FIG. 10 P-V DIAGRAM FOR WATER CYCLE.

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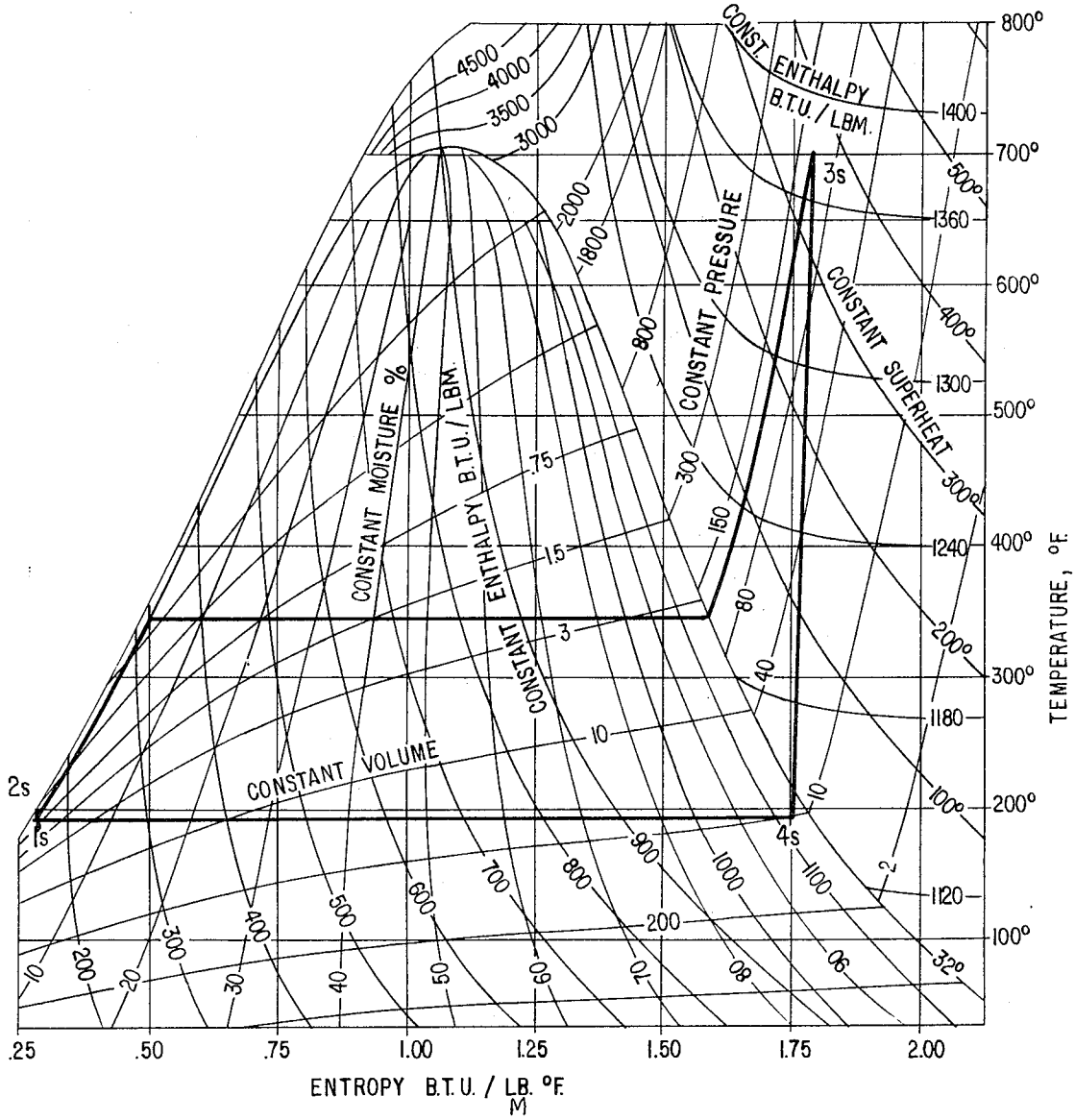


FIG. 11 TEMPERATURE—ENTROPY CHART FOR WATER CYCLE

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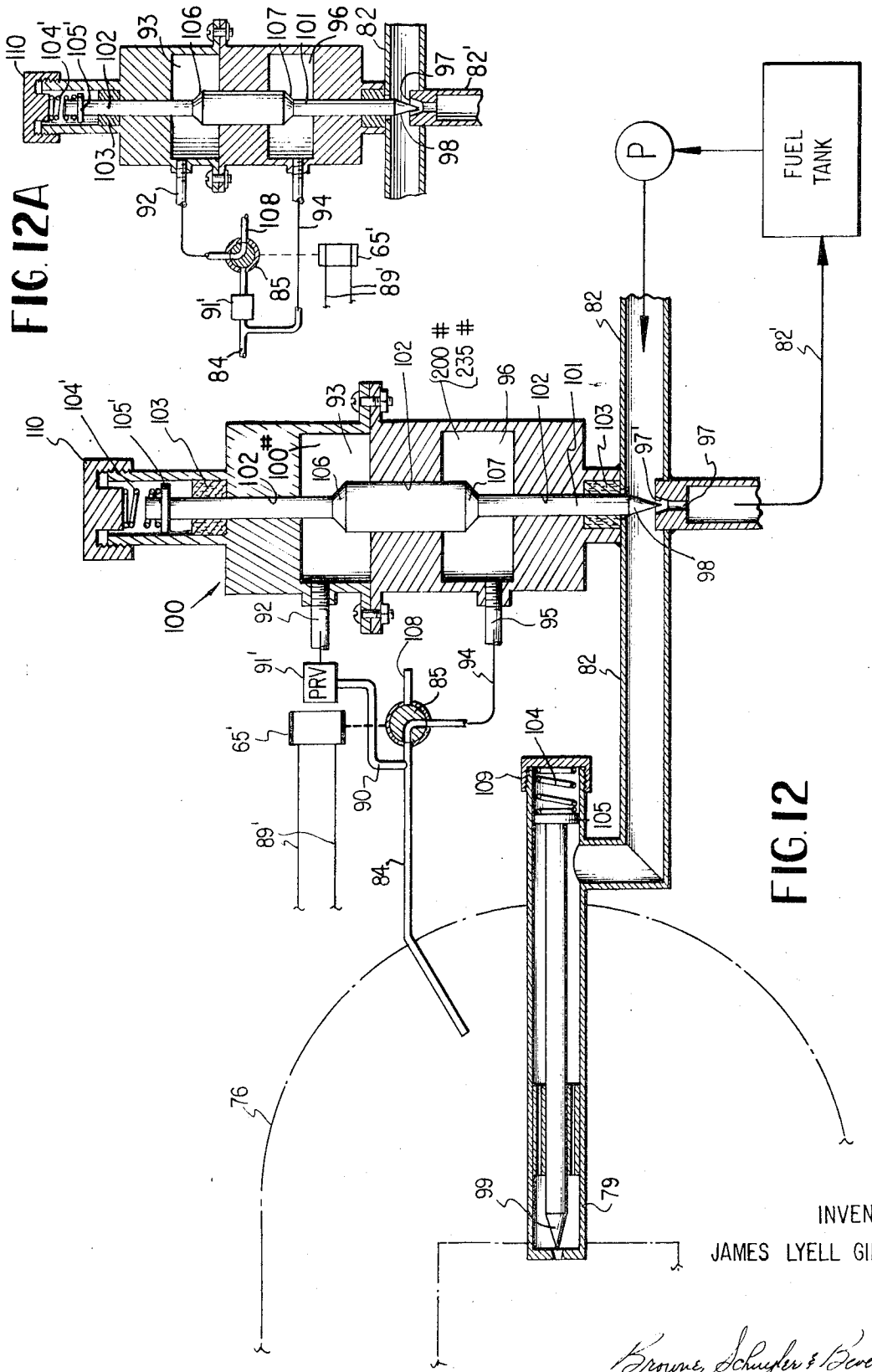


FIG. 12A

FIG. 12

ENGINE SYSTEM AND THERMOGENERATOR THEREFOR

SUMMARY

Briefly, air is supplied by the operation of a positive displacement compressor until a temperature above the ignition temperature for the fuel is reached. This is stored in a chamber. Upon decrease of chamber pressure, as when the engine is operated the compressor is caused to cycle to restore pressure, and fuel is burned in the chamber. The hot charge of gas and steam is held within an insulated chamber against loss of heat while any decrease of pressure as fluid is withdrawn is sensed to proportionally inject fuel, with cooling water during the burning of the fuel, so that fuel and water are both cutoff when the temperature and pressure are again restored. The combustion chamber is designed to provide a conical burning zone surrounded by excess air for complete combustion. The engine operates on a new cycle, in part as an external combustion engine supplied with superheated steam and in part as an internal combustion engine, compression and expansion strokes being in separate cylinders with the expansible fluid about double the fluid compressed. Engine cooling is not desired and the working fluid portions are actually insulated to prevent loss of thermal energy. Working controls are variable but automatic when set.

PRIOR DEVICES

Two general types of internal combustion engines are referred to as constant volume and constant pressure. The Otto cycle engines operate by explosion of a volatile fuel in a volume of compressed air near top dead center while diesel cycle engines burn the fuel in a modified cycle, burning being approximately described as at constant pressure. External combustion engines are exemplified by steam engines and turbines and some forms of gas turbines.

It has been known to supply a gas engine with a fluid heated and compressed from an external source and to operate various motor devices from energy stored in this compressed gas. These are classified as external combustion engines. It is also known to burn fuel in a chamber and exhaust the combustion products into a working cylinder, sometimes with the injection of water in accordance with the rising temperature. These may also be classified as external combustion engines. Some other devices have been proposed in which combustion chambers are cooled by addition of water internally rather than employing external cooling. Still another form of apparatus has been proposed for operation on fuel injected into a combustion cylinder as the temperature falls, having means to terminate fuel injection when the pressure reaches a desired value.

Each of these prior systems has encountered difficulties which have prevented their general adoption as a power source for the operation of prime movers. Among these difficulties have been the inability of such an engine to meet sudden demand and/or to maintain a constant working temperature or pressure as may be required for efficient operation of an engine. Furthermore, control of such engines has been inefficient, and the ability of the gas generator to maintain itself in standby condition has been wholly inadequate. In all practically applied engine configurations the requirement for cooling the confining walls of the work cylinders has resulted in loss of efficiency and a number of other disadvantages previously inherent in ICE engines. These are avoided or mitigated by this invention which produces working fluid without air or liquid external cooling, using a portion of the working fluid itself to reduce combustion temperatures to containable limits, but using the coolant to double the volume of working fluid without mechanical compression, by converting heated excess gas temperature to steam pressure.

OBJECTIVES

It is one object of the invention to provide a new compression, burning and power cycle constituting a new thermodynamic process and an engine to utilize the process.

It is also an object of this invention to provide a thermogenerator in which the fuel-air ratio is appropriately selected and in which the products of combustion have a volume approximating the volume of steam generated by the injection of water to cool the resulting mixture to a containable working temperature in which the weight of water employed approximates the weight of the combustion products, including the excess of air.

It is also an object to provide improved controls for combustion, cooling, and storage of an engine working fluid, under constant time, temperature and pressure of burning controlled to prevent wall-quenching of process and with increased reaction time to complete combustion and avoid formation of smog-producing products.

Objects and features as summarized above are achieved in methods and apparatus illustrated in the accompanying drawings in which:

FIG. 1 is a diagrammatic longitudinal section of one form of engine according to this invention;

FIG. 2 is a longitudinal section of one form of combustion and storage chamber useful in an engine according to FIG. 1;

FIG. 3 is a section taken along line 3—3 of FIG. 2;

FIG. 4 is a section taken along line 4—4 of FIG. 2;

FIG. 5 is a block diagram illustrating the thermodynamic process;

FIG. 6 is a P-V diagram for a binary fluid comprising air and water in two phases;

FIG. 7 is a T–S diagram for the fluid as in FIG. 6;

FIG. 8 is a P–V diagram for air alone entering into the diagram of FIG. 6;

FIG. 9 is a T–S diagram for the air alone entering into the diagram of FIG. 7;

FIG. 10 is a P–V diagram for water alone;

FIG. 11 is a T–S diagram for water alone;

FIG. 12 is a sectional elevation of a valve for fuel injection proportional to pressure drop and engine r.p.m., for alternative use in the system of FIG. 1 or the generator of FIG. 2; and

FIG. 12A illustrates a preferred form of control for fuel injection.

GENERAL DISCUSSION

Internal combustion Otto engines have wide acceptability because of their relative versatility in converting fuel to variable power with a minimum of equipment and moderate efficiency. Constant pressure type engines such as those classified as having a diesel cycle are less effective in ratio of power to weight and in availability for sudden spurts of output power. All such engines, of either constant volume or constant pressure type, have the disadvantage that the combustion gases are cooled by chamber wall cooling by heat-absorbing means which exhausts this heat to the surroundings without doing useful work. In the gasoline engine, these cooled walls and cylinder heads generally utilize a water jacket.

The relatively cold walls quench combustion in immediately adjacent layers and produce smog-forming products.

According to accepted thermodynamic principles a considerable degree of waste heat will be a necessary consequence of exhausting the used gases to the atmosphere at a considerable remaining pressure, thereby containing a quantity of heat according to that temperature in a relationship to absolute zero which limits effective utilization of fuel to the range of 10 to 34 percent efficiency. Certain specialized engines can achieve somewhat higher efficiency under favorable conditions. In stationary engines a degree of improvement is possible, particularly using the diesel cycle, inasmuch as a heavier machine can be used and a longer work stroke in relation to the residual volume in the cylinder at top dead center, and for other reasons. Under the best prior practice, there remains this cold-wall source of contamination as well as a wasteful cooling effect at confining walls which must be sufficiently cooled to keep them within temperature limits for adequate strength. While use of exotic materials in recent

years suggests the possibility of confining wall temperatures something like half the temperature of the burning gases, this is not yet a practical expedient for either mobile or stationary engines. The problem of wall cooling is inherent in gas turbine practice and is generally solved by the use of a large excess of input air so as to dilute combustion products sufficiently to reduce the working fluid temperature to a value which can be withstood by the turbine blades, etc. The working fluid is thus increased in volume and exhausted still retaining added compressional energy without useful result in respect to the air for added dilution purposes. Accordingly, no present engines are effective to convert combustion product excess temperature to wholly useful working fluid.

Steam engine practice recognizes the disadvantages of employment of a vapor such as steam which has a high condensation temperature both at the beginning and the end of a work stroke. This generally requires the exhaust of steam at a temperature sufficiently high to prevent water accumulation in the cylinder at the end of a work stroke with a loss of efficiency. When steam is admitted to a cylinder during the first portion of a work stroke and exhausted after expansion, a large cylinder must be employed to obtain substantial work output during the latter portions of the stroke, as by the use of double or triple expansion cylinder arrangements. One means of avoiding condensation of the steam on the walls of the cylinder at the latter portions of each stroke followed by re-evaporation of the water is by the use of a configuration now generally referred to as "uniflow." These condensation problems make steam a less ideal working fluid than combustion gases associated with the burning of a volatile fuel in air. At the same time, it is now recognized that containment of combustion gases within metallic enclosures during the work process requires the walls to withstand high temperatures caused by heat transfer, avoided only by external cooling with its consequent loss to the ambient air of a considerable portion of the energy of fuel combustion in addition to the Carnot cycle losses.

This invention provides a means for utilizing the advantages of internal combustion of fuel in a compressed gas while at the same time avoiding previously required cooling of the containing walls in order to bring the combustion products into the range of temperatures which they can safely withstand under pressure. This is accomplished by the injection of water into a special combustion chamber proportioned and arranged such that the combustion products do not touch an exterior wall until the temperature has been reduced to a safe value. Thus no wall needs to be cooled and all walls can be insulated. Principles of gas flow in jet engines are incorporated into the design of the combustion chamber by bringing the compressed air in at one end through a combustion head in a simple fire tube which is supplied internally with about half of the input compressed air. At least one additional tube surrounds the fire tube, referred to as the heat shield since this tube protects the outer walls of the combustion chamber against accidental irregularities in the flow of the combustion products prior to their cooling. Ordinarily a second combustion cone tube surrounds the first fire tube within the heat shield and extends from the combustion head longitudinally of the chamber so that the terminations of the three tubes confine the flow of gases to a diverging cone longitudinally concentric with the combustion chamber. Before the combustion gases can reach the heat shield they are cooled by water injected in an atomized spray from nozzles near the exit end of the combustion chamber. The amount of water employed is exactly sufficient to reduce the combustion product temperature to a design limit. This may either be the working temperature of a steadily maintained pressure and temperature of compression of the input air, or may be at a higher value in order to supply a reserve of energy, or may be at a still higher temperature such that a higher degree of superheat is retained in the steam evolved. It will be appreciated that the higher this temperature during conditions of its use the higher is the theoretical thermodynamic efficiency of an engine which draws its working fluid from this chamber.

While some previous schemes have been devised for cooling combustion products and generating therefrom steam these have not generally been adapted to the efficient operation of a piston engine or to the use in an engine operating on essentially a diesel cycle or a self-ignition ratio of compressed air to intake air. In the present invention this is achieved first through the use of a common crankshaft to which connecting rods are attached for operation of, or operation by, pistons of compressor cylinders and work cylinders which are discrete from each other and separated from the combustion chamber itself. A crankshaft for the work engine comprising usually two, four, or six cylinders, may be continuous with the crankshaft which drives the compressor pistons, generally equal in number to half the number of work cylinders, or a separate crankshaft may drive the compressor coupled through suitable gear means with the crankshaft from which output work is derived as the work pistons operate. For the purpose of illustration, it will be seen that the principle is exemplified in FIG. 1 utilizing one compression cylinder and two work cylinders of like size, which arrangement is approximately ideal when the weight of water injected to cool the combustion gases is equal to the combined weight of the fuel and compressed air input. This provides a doubled volume or a PV increased by a factor or approximately two to thus double the working fluid available for operating the work pistons over that which would be available from the combustion products. This doubling of fluid at usable temperatures is made possible in large measure by the elimination of the wall cooling since this cooling is now used for the manufacture of the doubled volume of working fluid at some maximum temperature which can be readily tolerated by the walls of the working chambers.

Ideally, this engine has an insulation preventing any loss of heat by radiation or convection. Practically, a loss must be tolerated, although it is reduced as much as possible by selection of a lightweight, high efficiency cellular material of several inches thickness or an equivalent insulation of other type. The engine configuration may be arranged for minimum surface exposure to aid in retaining adiabatic temperature conditions resulting from compression of 12:1 or more, generally in the order of 16:1 so as to permit a considerable temperature loss in the storage chamber before self-ignition is lost. At start-up, or after long standing, temperature is restorable by further compression of input air until a set value is reached permitting fuel injection to cause combustion. The fuel employed may be any hydrocarbon, preferably in the liquid state for easy injection, and a minimum temperature setting somewhat above the continuous burning point is used. Injected fuel is instantly vaporized at the chamber temperature, or within the nozzle which quickly becomes heated to the chamber temperature. Airflow from the compressor is unidirectional for storage of successive charges by use of check valving, and compressed air is held in the chamber until exhausted (with or without combustion) to drive work pistons, which then drive the compressor to resupply the required air and temperature. A large percentage expansion occurs on burning the fuel to generate a working fluid of high specific energy, but produces a combustion temperature too high for storage or use without cooling. Water converted to steam at the compressed air temperature can exactly absorb the heat of combustion while a little more than doubling the PV of the fluid in the chamber.

A thermostatic control of water injection can be variously adjusted to leave a fluid temperature substantially the same as that developed by adiabatic compression of the input air. If the input air is at 60° F. and a 16:1 or higher ratio is selected a pressure develops in a fully insulated chamber approximating 235 p.s.i.a. During start-up the compression will be less than adiabatic since the walls of the compressor and storage chamber will be below the value ultimately reached on complete charging to the 235 p.s.i.a. value, and because the air forced into the chamber mixes with air already there at a lower temperature. Air will at first be delivered over a considerable percentage of each compression stroke against the small back pressure in the chamber as air accumulates until

the fuel firing limit (set by a first thermostat) is reached, at which time fuel can be injected. Initial pump-up of ignition temperature may be accomplished by opening the working fluid valve to the work cylinders, manually or otherwise, as in a normal opening when power is desired. Compressed air then drives the work engine to compress further air which is then accumulated at progressively higher temperature until the ignition temperature is reached. This firing limit will normally be set well below the temperature obtainable by compression to assure ignition under normal working conditions down to some limiting low temperature of input air, such as 0° F. or −40° F., according to climate.

The starter motor is energized from a source adequate to pump up to 235 p.s.i.a. or more which will provide input air at least 650° F. from the compressor to the combustion chamber around the fuel nozzle. This pump-up time may be shortened or lengthened depending upon the relative magnitudes of the steady and sudden power demands as the work engine is loaded in normal service, the higher ratios of sudden-to-steady power requiring larger storage capacity.

When the combustion chamber reaches a temperature for fuel ignition, fuel is injected from a high pressure source coupled, for example, to the crankshaft, either to inject a fixed slug of fuel intermittently according to crank rotation or in proportion to crank speed. Therefore, the air supplied is in a fixed ratio to the fuel supplied and may be in the order of 5 to 25 percent above that required for combustion of the fuel. Below ignition temperature no fuel is supplied, so no accumulation of unburned fuel can occur in the chamber, and the fuel-air ratio becomes constant very quickly after burning commences.

At a compression ratio of 12:1, for example, the pressure in the chamber reaches about 176 p.s.i.a. and the temperature for adiabatic compression from 60° air would be about 600° F. At a more generally used ratio of 16:1 the chamber reaches about 235 p.s.i.a. and 700° F. and at 20:1 the chamber reaches about 295 p.s.i.a. at a temperature of about 800° F. If a higher temperature is set for opening the injected fuel value a higher stored compression ratio will be reached before ignition.

Upon igniting fuel the temperature rises to about 3,000° to 3,800° F. in the flame area, and this temperature spreads down the fire chamber to the second thermostat which controls water injection in proportion to temperature in excess of the selected setting. If the first thermostat permits fuel injection at about 600° F., the second thermostat controls water injection to bring combustion products at 3,000°–3,800° F. down to a usable temperature such as 1,000° F. while at the same time doubling the volume of working fluid.

A higher temperature of superheated steam and combustion products can be utilized during continuous operation of the machine without endangering the walls due to the dynamic flow conditions of the combustion chamber, later described. This permits a higher thermal efficiency for the work engine merely by setting the second thermostat to delay water injection to a temperature higher than 1,000° F. A theoretical efficiency of 46 percent in output work relative to input energy as stored in the chamber corresponding to a chamber temperature of 1,600° F. at a pressure in the range of 300 to 400 p.s.i.a. may be achieved in a practical engine design.

An engine as thus constructed performs in succession the functions of intake and and compression, enthalpy or PV increase in a combustion chamber, a work-performing expansion, and exhaust in four steps generally corresponding to the steps of a diesel engine except that intake-compression, energization (combustion and steaming) and expansion-exhaust each occurs in a separate chamber under separate control as to pressure and temperature of operation, wide limits being permissible for each variable, which can be altered during a particular power run by changing a thermostat setting or a pressure-responsive control.

A manual power throttle may be coupled with the fuel injection control. It is preferably applied to control the rate or the duration of fluid supply to the work cylinders. A rate control may govern the speed at a constant torque or a combination of torque and speed. If the duration of fluid flow to each cylinder is controlled the resulting fluid delivery to the cylinders is similarly controlled within the limits of resupply of compressed air. During variation of the engine r.p.m. the resupply of air is of course varied proportionally. Fuel rate may also be varied within the aforesaid range of fuel cutoff or admission necessary to assure full combustion. Variation of water injection to control the working temperature is an alternative governing factor on the output power, it being of interest to note that increased power output controlled in that manner is accompanied by increased thermal efficiency as the working temperature rises within limits of the wall temperature.

A main control for the fuel injection is by linkage of the fuel control valve to a proportional pressure sensor in the after portion of the combustion chamber, preferably near the entrance to the distributor valve admitting fluid to the respective work cylinders according to crank rotation. A two cylinder work engine, or any multiple thereof, may have a common sliding valve open to either of two cylinders oppositely disposed about the shaft. A valve governs admission of fluid to a sliding valve chamber, being preferably coupled to the manual control or to a speed control mechanism arranged to decrease either the flow rate into the slide valve or the duration of the flow, generally being limited to withdraw no more consumed input air than is supplied by the compressor, regardless of its temperature or pressure, except for short periods or power spurts where stored air is utilized in excess of that currently pumped.

As a simple control the fluid throttle opens whenever there is a need for power, here illustratively by a manual control lever and/or by a pressure decrease below a limit, furnishing fluid to the work cylinders and driving the compression piston to exactly make up the air exhausted. When on standby throttle setting, pressure in the chamber may decrease to a set limit such as 220 p.s.i.a., (if 220–235 p.s.i.a. is selected as the normal working pressure range in the combustion chamber) the fuel throttle commences to open. During start-up fuel flow is inhibited by a sensed temperature below the first thermostat setting. When temperature is above this setting and pressure falls below 220 the fuel throttle opens progressively to full fuel flow at 200 p.s.i.a.

Upon fuel injection the temperature then rises until water is injected to restore the temperature to a value below the upper limit set on the second thermostat. Fuel burned raises the pressure to progressively cut off the fuel injection between 220 and 235 p.s.i.a. This pressure rise continues as the water is injected to restore temperature to a safe operating temperature as set in the second thermostat, leaving a pressure considerably in excess of that which would have been developed by the combustion products, mainly CO_2 and H_2O generated by burning the fuel and the excess air used. This additional pressure will occur simultaneously with fuel burning under steady operating conditions, thus to shorten the duration of the fuel injection or to reduce its rate. In standby operation, when the manual throttle override is not set up, the pressure sensor is operative to cause recycling to restore pressure.

Each control may be made rapidly intermittent, or slow and variable to establish a smooth and steady flow both of fuel and water for steady engine operation. In either case fuel injection is proportioned to the amount of pressure drop or the frequency of detected pressure drops if fixed fuel slugs are repeatedly injected into the chamber to restore the temperature and pressure of the operating fluid, all needed intake air being supplied in fixed amounts according to engine rotation. Lesser fuel injection rates result in lesser heating of the combustion products and lesser injection of water. Water injected is thus in nearly exact relationship to fuel injected, although the ratio varies according to the upper temperature limit selected, in predictable relation to the input temperature of the compressed air.

By way of example, it requires about 14 pounds of air to fully burn 1 pound of diesel fuel of 18,000 B.t.u. pounds and a

temperature rise results which requires about 13 pounds of water to restore to the value before fuel injection. These Figures provide a little more than a doubling of the PV, or the P if a full restoration to the compression temperature is achieved. By injecting water sufficient to reduce the temperature to about 1,000° F., rather than the assumed 700° F. from the compressor, a lesser weight of water will be required than the combined weight of air and combustion products.

The control of fuel injected according to decrease in a stored gas pressure and the control of water injected according to the rise in temperature of the chamber provide the only essential variants in this engine cycle, and with these controls a new work cycle is achieved, conditioned upon the compression of input air proportioned to engine rotation. Output torque, however, is variable, as is storage temperature and pressure by changing proportioning linkages.

A number of significant features are to be noted from the above general description. An engine operating as described operates on a stored working fluid after the starter is disconnected. The engine is instantly restartable from the working fluid stored. This fluid is stored at a temperature well above the working temperature required for self-ignition of the fuel but will then slowly cool down. Storage is at a pressure here assumed to be 235 p.s.i.a., but could be adjusted to store at a value such as 500 p.s.i.a. after start-up.

If temperature of the stored compressed air (with or without combustion products) should coast down by loss of heat, a pressure decrease will also occur and the engine (if in standby setting, since the manual fluid valve is not operated when power is not needed) will recycle beginning at a pressure decrease to about 220 p.s.i.a., or as otherwise set for a higher storage pressure, and the fluid valve will open to admit working fluid to the work cylinders to recycle the compressor.

When fluid at 200–220 p.s.i.a. is withdrawn to compress further air a rise in specific enthalpy, that is, a higher pressure-temperature product occurs. Since a lower specific energy source produces a higher specific energy result it is necessary that a larger volume of withdrawn gas be utilized than is supplied as compressed input air of higher pressure and temperature. This is effected by use of two work cylinders for each compression cylinder, or of otherwise increased displacement relative to compressor displacement, such as by cylinders or crank throws of different magnitude. When a high superheat is used the ratio may be less than 2:1, for example, 5:3.

Recycling to increase temperature can occur with a decrease of overall pressure when only the temperature in the vicinity of the fuel injector is considered rather than the overall temperature-pressure of the chamber, because more gas is removed than is restored, until fuel is burned.

If a locally adequate ignition temperature prevails, even momentarily upon air injection, and the fluid throttle is open, fuel is injected and the temperature is immediately restored, with considerable additional pressure, so as to cut off the throttle (if in standby setting). When the amount of fuel burned is sufficient to raise the temperature to a setting such as 1,000° F. water is injected to restore the temperature to its assigned limit. Obviously, this will not occur on a rise in pressure merely to accompany fuel burning needed to turn off the fuel, and water injection will be confined to runs in which power is withdrawn via the shaft.

DETAILED DESCRIPTION

Referring now to the drawings and FIG. 1 thereof, there is shown schematically an engine according to the present invention in simplified form, exhaust valving, for example, being omitted for clarity of showing, wherein base 1 is any support on which the engine is mounted. Supports 2 extend from base 1 to the engine housing shown at 3, which comprises a closure for the working portions of the engine adapted to retain therein pressure and temperature according to the operating conditions for the engine. The casing 3 may be domed at the top and preferably is made without sharp corners which would provide stress points for excessive strain tending to damage the casing under conditions of use. In some configurations casing 3 may be in the form of an ellipsoid possibly modified for economy of space. Any number of cylinders may be enclosed, generally two work cylinders for one compressor cylinder, or cylinders of differing size may provide 60 to 100 percent more work than compression displacement.

There is provided a pressure dome 4 which is made sufficiently large to accommodate those elements comprising a combustion chamber. Alternatively, this chamber may be as in FIG. 2. Dome 4 has first and second walls 5 and a bottom closure 6 which may be either at the lower ends of the pistons, as later described, at the cylinder head level, or at the bottom of the engine.

The lower side of casing 3 is closed by a suitable member 7' comprising pan space 7 provided with either negative or positive pressure with respect to the atmosphere, or may be held substantially at atmospheric pressure if the dome 4 is closed at the bottom by members such as 6. In either case dome 4 comprises a storage chamber of volume several times greater than the displacement of the work cylinders in one revolution of the crankshaft. It is insulated for a temperature which is elevated some 1,000° F. or more with respect to atmospheric temperature and at a pressure likewise elevated to 235 p.s.i.a., for example.

Since casing 3 is held at an elevated pressure and temperature, preferably including the cylinder heads, cylinders, and crankcase, insulation shown at 8 surrounds the entire casing 3 and pan 7'. This insulation is preferably of high quality and low density suitable for use at temperatures up to 1,800° F. and is made sufficiently thick to maintain heat heat storage in dome 4 above 600° F. when the engine is not resupplying compressed air or heated by combustion.

End walls of casing 3 are pierced to provide access for crankshaft 10 in bearings 11 in a configuration adapted for dynamic balance of the work pistons and the compression pistons. If only three cylinders are used, as diagrammatically shown, power cranks 12 are illustratively at 180° separation on crankshaft 10 while compressor crank 13 is at an intermediate angle, although it could be at any desired fixed orientation relative to power cranks 12. Whenever the crankshaft 10 is continuously powered, as where three or four power pistons are operating, different crank orientations about the shaft 10 may be at 120° or 90° separation. Pistons 14 and 15 are power pistons while piston 16 is a compressor piston, preferably of similar size and configuration, although either the work pistons 14 and 15 or the compressor piston 16 may be made double-acting by suitable porting and end closure of the cylinders. Connecting rods 17, 18 and 19 connect pistons 14, 15 and 16, respectively, to cranks 12 and 13 for reciprocation in cylinders 20, 21 and 22.

At the top of cylinder 20 there is a port 23 corresponding to port 24 at the top of cylinder 21, these ports being covered or uncovered according to the operation of a suitable valve arrangement, illustratively a slide valve. Port 25 at the top of the compressor cylinder 22 has a check valve 26 suitably spring urged into engagement so that when the piston 16 rises to complete the compression stroke check valve 26 opens to pass compressed air from cylinder 22 by way of air passage 27 to an air chamber header 28 in the upper portion of the casing forming one end of dome 4. A similar check valve oppositely operable admits intake air to cylinder 22.

According to FIG. 1 illustration of the invention a combustion chamber is disposed within dome 4 across the top of the engine comprising cylinders 20–22. According to other embodiments of the invention this combustion chamber may be a cylinder as in FIGS. 2 and 3 or may be formed as one of the four cylinders of a conventional four, six or eight cylinder engine block, with changes in closures and fittings, by eliminating one piston therefrom and providing air and fuel inlet at one end and water injection at the other end, in which case it will usually be desirable to supply an auxiliary air pressure chamber for storage of operating pressure during protracted periods of non-use of the engine.

A combustion chamber **30** is provided with a perforated head separating the body of the chamber **30** from the air header **28**, with perforations therein proportioned to provide a distributed flow of compressed air into and along the axial dimension of the combustion chamber. A more detailed description of a suitable combustion chamber will be provided in connection with FIG. **2**. A relatively long chamber **30** terminates at header **29** for the admission of compressed air from cylinder **22** and header space **28**. Header **29** preferably has a fire tube **31** extending therethrough from space **28**, tube **31** being a relatively small tube in both diameter and length in relation to chamber **30** and is preferably supplied with parallel-flowing air from perforations **32** through header **29** and with internal longitudinally flowing air from perforations **33** into fire tube **31** which communicates directly with header space **28**. A further tube **34** extends from header **29** longitudinally of chamber **30** and has perforations **35** therearound for receiving air from space **28**.

One illustration found to provide the appropriate distribution of air, fuel, and combustion products about half of the input air to the combustion chamber enters by way of tube **31** partly radially and partly axially to provide mixing, and a smaller portion from areas around the fire tube by way of perforations **32** and a still smaller portion around tube **34** from perforations **35**, In addition, a further tube shield **31'** may surround tube **31** within tube **34**, extending a distance down the length of the chamber **30** beyond tube **31** less than the length of tube **34**. In this case air will be distributed from perforations **32** both inside and outside of this additional tube. This air distribution is appropriately in the proportions one-half, one-quarter, one-eighth and one-eighth.

Apparatus comprising pistoned cylinder **22**, check valve **26**, air space **28** and fire chamber **30** within the pressure dome **4** comprises means for supplying compressed air at a pressure taken normally at 235 p.s.i.a. and 700° F. This temperature is a result of compression, heat losses and absorption being neglected, and burning of fuel not considered. Fuel burned of course further elevates both pressure and temperature. Clearance above piston **16** may be made substantially zero by appropriate shaping of the cylinder head to permit escape of compressed air through port **25**. If it is assumed that the piston has a 4-inch stroke it will be seen that piston **16** provides a compression ratio of **16:1** when it reaches a position one-quarter inch from the cylinder head, and would accordingly provide a 16:1 compression of intake air for about one-quarter inch of stroke. When the starting pressure in space **28** is less than **16** atmospheres air delivery from cylinder **22** is of course over a proportionately higher fraction of the compression stroke.

One means for operating crank **13** to initially supply about a 16:1 adiabatic compression is by use of a conventional starter arrangement connected to the crankshaft at either end of casing **3**. This starter is not shown in FIG. **1** inasmuch as it is a conventional arrangement operated, for example, by an electric starter motor having sufficient capacity to provide of the order of 100 revolutions of crankshaft **10**. It will be appreciated that the space in pressure dome **4** will be filled with compressed air and that this air will be heated nearly adiabatically except for initial losses of heat due to the warming up of the metallic portions comprising confining walls of the dome. Suitable connection between the starting motor and a temperature limit control for feeding fuel for self-ignition may include a conventional holding circuit keeping the starter motor in operation until a temperature is detected in the vicinity of the fire tube **31** adequate for commencing combustion of the selected fuel. When initial losses are substantial and a considerable departure from adiabatic compression occurs a pressure of 235 p.s.i.a. may be built up without ignition temperature being reached. In this case manual opening of the fluid throttle permits exhaust of portions of pumped air.

Fuel is supplied for this engine by way of an injector **36** connected with a line **37** through wall **5** to a suitable throttle **38** which is mechanically linked or electrically connected for response to pressure and temperature conditions within dome **4** such that operation of crankshaft **10** will cause fuel injection proportioned to air input whenever the temperature in the fire chamber is adequate for combustion. The maximum rate of fuel injection may be controlled by way of a needle valve **39** in series with the throttle **38**, or may otherwise be controlled proportionally to a power demand signal or as in a diesel engine in which crankshaft operation injects a small fixed quantity of fuel through nozzle **36** in response to rotation except when inhibited herein by an overriding thermostatic control to prevent the injection of fuel when the temperature is insufficient for combustion.

Conditions for either steady or intermittent burning of fuel in the fire tube **31** are established, by providing means for establishing air pressure at a self-ignition temperature for a particular fuel selected and means for supplying fuel at a higher pressure through nozzle **36**. A very much higher pressure is used to thoroughly atomize the fuel spray. Typical combustion temperatures for liquid hydrocarbon fuels are about 3,000° to 3,800° F. where normally a small excess of air is supplied. Larger quantities of excess air would of course reduce the resulting temperature but would not greatly affect the actual temperature of burning or the ignition temperature. A main feature of this invention is avoidance of chamber-wall-cooling of the products of combustion, it being the objective to convert as large as possible a percentage of this hot combustion product into a useful work fluid. Water proportioned precisely to the quantity of fuel burned is injected by way of nozzles **40** into combustion chamber **30** preferably near the outlet end thereof within dome **4**. A water line **41** supplies nozzles **40** by way of a water throttle **42** and a rate control needle valve **43**, preferably supplied by a pump **44** attached to crankshaft **10**. A pressure equalizing dome **45** receives water from pump **44** and water line sections **46** convey this water to throttle **42** which then controls the timing and duration of water injection. This injection is made just sufficient to neutralize any degree of excess temperature in the chamber **30** which is selected as appropriate to the structural materials employed in tube **34**, pistons **14**, **15**, cylinders **20**, **21** and casing **3**. Upon entry into chamber **30** water is instantly vaporized and converted to superheated steam.

A positive displacement pump such as the gear pump shown at **44** permits development of a sufficient pressure in water lines **46** to overcome the pressure in dome **4** during start-up by the starter motor and to atomize water ejected from nozzles **40**. Similarly, a fuel pump **48** may operate from crankshaft **10** to receive fuel by lines **49** and pressure equalizing chamber **50** to needle valve **39**, which controls the rate of injection of fuel according to this illustrated means for injecting fuel.

Work pistons **14** and **15** are powered whenever steam and the products of combustion mingled in dome **4** are admitted to cylinders **20** and **21** by way of ports **23** and **24**. Various valve arrangements can be employed for reciprocating pistons **14** and **15**, as by a slide valve illustrated at **51** as a moving ported tube dimensioned to cover one or the other of ports **23** or **24** in accordance with the position of crankshaft **10**, linked by way of bell crank **52**, push rod **53** and cam **55** through stuffing box **54** in order to control the position of valve **51** with respect to the dead center positions of the work pistons. According to the illustration of FIG. **1** dome **4** communicates with the region surrounding cylinders **20–22** shown at **56** to provide a compact dome of considerable volume with respect to the displacement of each piston. A suitable ratio is in range of 30:1 or 100:1, the higher ratio providing a larger reserve of standby power for getting the engine into operation from standby, but requiring a longer operation of the starter for initial pumping up of air pressure to a self-igniting temperature.

For convenience of illustration, space **56** lying between cylinders **20** and **21** connects to and is supplied with the same pressure as dome **4**, and is accordingly a suitable location for a pressure sensor or control device. Sensor **57** is mounted to respond to a decreased static pressure, being for example an air filled bellows linked to arm **58** on support **59** for moving slide valve **60** attached to slide valve **51** to admit working fluid

to the distributing slide valve **51**, being illustratively a tube in which the slide **60** is a fitting surrounding a ported tube portion of **51**, being sufficiently movable to provide access to the interior of slide valve **51** whenever the pressure in space **56** decreases below a specified minimum. This minimum is adjustably set at the pressure sensing device **57**.

Device **57** may be of conventional type and may be mounted in any convenient location so as to control entry of high pressure working fluid into cylinders **20** and **21** by way of distributing slide valve **51**. Furthermore, valve **51** is manually settable in a manner to override the automatic pressure control by use of a lever **58'** on pivot **59'** which moves the base of sensor **57** so as to wholly or variably open valve **60**. This mechanism is a schematic main control for the engine and establishes operation for power production as well as a recycling standby operation of the crankshaft whenever the pressure stored in dome **4** falls below a suitably adjusted lower limit. It is adjustable to provide intermittent operation of the engine to resupply compressed air during standby, or continuous operation when output work is desired. Numerous pressure sensitive devices operable at typical engine working temperatures are available, for mounting as illustrated, or otherwise responsive to pressure decreases in dome **4**. Separate manual throttle opening means may be utilized.

Whenever crankshaft **10** is operated by admission of working fluid to valve **51** and port **23** or **24** compressor piston **16** operates in an exactly proportional manner. This may be by direct connection to the same crankshaft or by a geared connection, not shown. An essential feature is that any operation of pistons **14** and **15** causes a similar operation of piston **16**, proportioned to supply additional compressed air equal in volume to the volume withdrawn to the work cylinders during a power run, but of increased relative volume in a recycling operation when steam is not generated. It may be observed that under the assumed conditions of 700° F. and 235 p.s.i.a. in dome **4** an injection of fuel will elevate this temperature as well as the pressure over the period of duration of fuel injection as controlled by valve **38**. Since temperature would then build up to destructive values a runaway control is required for continued operation.

Thermal runaway is avoided by a proportional injection of water through nozzles **40**, arranged for spraying a fine mist of water transversely of the fire chamber **30** forward of the exit end of the chamber so as to cool the combustion products prior to their reaching thermostat **61**, which then controls the duration and rate of water injection by proportionally adjusting valve **42** within the limit established by needle valve **43**.

Thermostat **62** is preferably located near the air header **29**, within tube **34** so as to respond to the input air temperature at the inlet end of fire chamber **30** and is connected in a manner to prevent fuel injection when this temperature is too low. The importance of thermostat **62** will be appreciated in that a sudden firing with a large excess of fuel in tube **34**, in the event fuel is permitted to be injected without burning, would result in an excessive and rapid rise of temperature counteracted only by the injection of water. Water injection of course greatly increases the volume of working fluid, being about 2 to 1 when the operating temperature in dome **4** is about 700° to 1,000° F. Thermostat **62** thus provides a safety feature preventing overpressures due to accumulated fuel or fuel flooding and may be set, for example, at 600° F. or other suitable temperature such that the injected fuel will surely burn spontaneously and without any electrical igniting device. As an additional safety factor the valve **42** controls the flow of water and can be adjusted to continue the injection of water until the temperature is brought down to some value, such as 700° to 1,000° F., at which time water injection terminates and the pressure in dome **4** remains static until a demand signal is applied to open valve **60** and admit working fluid to slide valve **51** for entry into work cylinders by way of ports **23** or **24**.

Valve **42** may conveniently be operated by a solenoid, a differentially controlled common rail injection system or other electrical motor device in proportion to need for a duration which depends upon the duration of a closed circuit of thermostat **61**. This might be provided in a bimetallic contact element, to energize coil **63** by leads **64** through a suitable power supply illustratively shown as a battery. Similarly, throttle **38** may control fuel injection from a pressurized source at **50** under suitable electrical control similar to **63** being connected to normally open thermostatic contacts **62** in series with a coil **65** having leads **66** to a power supply and a mechanical linkage responsive to rotation of crankshaft **10**. This linkage illustratively includes cam **72**, which may be of centrifugal or electronic type to promptly close a slow-opening switch **73**, which is also in series with a power supply, coil **65** and thermostat **62**, for operation of valve **38**.

Air input to cylinder **22** above piston **16** may be accomplished in a conventional manner as by inlet **67** through casing **3** and insulator **8** to a space separated from dome **4**. Air may enter **22** by biased valve **68** during a downstroke of piston **16**. Inlet air may be confined to a region about port **68'** by a barrier **69** across the upper portion of the inlet space. Equivalently, a tube might connect inlet **67** and valve **68**, suitably insulated at **69** from the high temperature area **28** so that the input air to the compressor will not be externally heated to cause an undue expenditure of work in compressing a fixed volume of input air or a corresponding decrease in air mass input. The fuel tank **70** is illustrated as feeding fuel line **49**. Output shaft **71** is shown suitably connected to crankshaft **10** for providing output power.

Referring now to FIGS. **2**, **3**, and **4**, an alternative form of the combustion chamber or thermogenerator is shown generally at **75**, having an elongated metallic body **76** with domed inlet end **77** and domed outlet end **78**. A fuel burning arrangement of jet style is shown generally at **79** corresponding to the burner of FIG. **1**. Generator **75** is surrounded by insulation **80E** of suitable quality and thickness to retain the heat of compression over a considerable standby period wherein air is admitted by air inlet **81** from a suitable compressor delivering pressures of **12** to **30** atmospheres. A fuel inlet **82** and a water inlet **83** are shown generally as in FIG. **1**, wherein a proportioning valve **91'** governs the rate of fuel delivery and a cutoff valve **85** prevents inflow of fuel when thermostat **62** indicates insufficient temperature. Water inlet nozzles **87** are preferably arranged near the aft end of the generator directed inwardly toward the center or axis thereof at suitable rotations such as 45° with respect to each other. Thermostat **62** is mounted preferably by a replaceable screw-in fitting **88** and may have a mechanical linkage to valve **85**, although this linkage is illustratively by leads **89'** to an electrical operating device **65'**, such as the solenoid **65** of FIG. **1**.

The generator **75** has an output for generated working fluid at **90** controlled as by manual valve **91** leading to work cylinders by way of a suitable distributing valve such as **51** of FIG. **1**. Thermostat **61'** may be mounted by a screw-in connection **112** and has electrical or mechanical linkage to valve **86**. As a conventional safety feature, a pop-off valve **113** and a rupture disc **114** are provided for the thermogenerator main chamber.

Indicated generally at **115** is a pressure sensor or pressurestat with response proportionally to decrease in pressure in the chamber and having an output to valve **97** for control of the opening of the valve progressively as pressure falls, e.g., below 235 p.s.i.a., until full valve opening is attained at a working pressure for the thermogenerator when a full load is applied. Proportional valve **100** may be hydraulic, mechanical, or electrical and operates to provide fuel inlet according to pressure decrease below a specified limit which may be anywhere in the range of useful compressed fluid supplies for an engine, such as 100 p.s.i.a. to 500 p.s.i.a. A sensor and proportional controller can similarly operate on temperature, rather than pressure, inasmuch as a decreasing temperature can be sensed and this decreasing temperature utilized to turn on fuel whenever it is desired to reheat chamber **75** as when new air is brought in by way of inlet **81**. A proportioning gas valve at **91** may be manually controlled by a settable fitting having an overriding control so linked at **111** to a pressurestat **115** that a drop in

pressure in generator **75** causes opening of the gas valve sufficiently to add additional compressed air by way of inlet **81**. One form of control may be a linkage to the sensor which controls the valve from a fixed base point established by a hand control, which, in one position, governs the recycling of the compressor when in standby condition to resupply compressed air as required, and in the opposite setting, provides an output power opening for the gas valve, intermediate settings for the control being adjustable according to power requirements.

A thermogenerator governed as illustrated in FIG. **2** provides means for burning fuel at constant pressure under conditions also approximating constant temperature burning. Both the temperature and the pressure are controllable at will, and are accordingly made suitable to a particular engine design with which the generator is associated. Compressed air input to the thermogenerator, after start-up, is of a constant pressure, determined by controls associated with the thermogenerator. Burning occurs in the chamber immediately following injection of fuel under high pressure and provides idealized burning conditions for efficiency and avoidance of air contaminants in which the fuel mixture may at first be richer than the mixture for complete combustion, additional air being added as burning continues, this air being added circumferentially around the burning fuel and in an amount which ultimately exceeds that necessary for complete combustion of the fuel components. In achieving this more ideal burning than is possible in the usual engine, there is employed, preferably a cylindrical structure having a fire tube, an air tube, and a heat shield, all within the chamber and coaxially arranged with respect to the fuel injection nozzle. A satisfactory burning condition, approaching the ideal, is achieved by the use of a fire tube having both diameter and length related to the diameter and length of the air tube and the heat shield tube, approximately according to the ratio 1:1–1/2:2. Thus, if the fire tube has a diameter of 2 inches, the air tube has a diameter of 3 inches, and the heat shield a diameter of 4 inches, each having similar length proportions, although the length may be increased. As pointed out in FIG. **1**, the air inlet is controlled by perforations in the air header so that about 50 percent of the air enters the chamber through the fire tube, being mixed and pushed along axially thereof. The air tube has an additional 25 percent air entering from the header and the heat shield approximately 12 percent or 13 percent entering through the header outside of the air tube, the remainder entering through the header exterior to the heat shield, entrance thereinto being by way of perforations along the length of the heat shield. Water is supplied according to temperature rise during combustion by way of nozzles **87**.

When a thermogenerator as shown in FIG. **2** is employed in a practical engine, a number of thermodynamic advantages are obtained. These will best be understood by reference to the relationship between an engine cycle, according to this invention, and prior cycles, as graphically illustrated in FIGS. **5–11**. This cycle is a combination of a compressed air work cycle and a steam cycle since both air and steam are present as a working fluid wherein each makes up a portion of the total pressure developed in the thermogenerator. During this discussion, it will be understood that the term "air" is intended to include fuel as combusted by the inlet compressed air together with any excess of air which may be present, and thus includes all of the products of combustion, while the term "steam" refers to water which is injected in the liquid state to become superheated steam, but is used in a work cycle step with also a change of state in which a part of the steam becomes liquid water. The new cycle or process of burning fuel to do work makes use of the combined steam and air, with the exception of the compression process in which air only is involved.

Two heat exchange processes and two work processes are involved in a complete engine diagrammatically illustrated in FIG. **5**. The pressure-volume, or P–V, and entropy relationships are illustrated in the FIGS. **6** through **11**. It may be noted that these diagrams differ from the conventional diagrams for either a steam cycle or a hot gas (air) cycle. Separate diagrams for the several constituents of the combustion products are not shown since these are all closely related to the diagrams for air as illustrated.

In FIG. **5**, it will be noted that two fluids are separately shown as entering into a work operation between points **1**, **2**, **3**, and **4**. Point **1a** designates the inlet to the air compressor while **1s** represents the inlet to a water pump; **2***a* indicates inlet to the thermogenerator for the air and **2***s* indicates inlet to the thermogenerator for the water; **3** indicates combined inlet for the expansion portion of the process, i.e., to the expander, **3***a* representing the inlet for air and **3***s*, the inlet for steam derived from the heat absorption occurring in the thermogenerator as the heat of combustion is removed down to a point of steady containment; and **4** represents combined inlet to the condenser, **4***a* being the air portion and **4s** being the steam portion, assuming that a condenser is employed for purposes of illustration of the complete cycle. In the following discussion, temperature is represented as T, pressure as P, volume as V, enthalpy as H, while entropy is represented as S. It will be understood that the diagrams are given for a particular addition step of the variable conditions as imposed by operator, load, engine, or other factors involved which may vary over any range of conditions practically applicable.

FIG. **6** shows a P–V diagram for the combined air and water components, assuming a volume corresponding to the burning of 1 pound of a particular fuel. On this diagram, progressing from **1***c* to **2***c* represents an equal entropy change referred to as isentropic compression during which the working substance or fluid is compressed from the condition at intake to the condition within the thermogenerator as it operates. This diagram carries the pressure rise only to 230 p.s.i.a. although a range up to 4,000 p.s.i.a. may actually be employed. From **2***c* to **3***c* there is shown a constant pressure process which, in the alternative, can be made a constant temperature process. In this respect, the thermogenerator differs from prior devices in a fundamental aspect since the thermogenerator provides a means for increasing a working fluid either at constant pressure or at constant temperature, the latter being effected by a control of water injection in accordance with a rise in temperature above that limit determined upon as the control limit. It is thus possible for the temperature to be controlled from a level as low as the saturation temperature of water at the thermogenerator pressure to as high as the adiabatic flame temperature of the particular fuel being burned. The practical limit of the discharge temperature from the generator is in turn governed by the material strength of the containing walls at the discharge temperature. This discharge temperature is controlled between suitable limits by variation in the injection of high pressure water which then flashes to steam, the heat of the vaporization and superheat being equated to the heat of combustion of the fuel being burned. The quantity of injected water is thus determined by the desired operating temperature, being less for high superheats, but actually maintaining a fixed operating temperature. The third process in FIG. **6** is from **3***c* to **4***c* and relates to the combined substance, being an isentropic expansion to the exhaust pressure determined by the exhaust pressure wanted for the expansion engine, and determined by a pressure sensor and fuel injection actuator.

FIG. **7** is a temperature-entropy (T–S) diagram for the combined working substance. The state points on this diagram correspond to the state points on the P–V diagram of FIG. **6**. The cycle shown in the diagram of FIG. **7** shows the combined working substance following a Carnot cycle, made possible if a constant temperature is maintained in the thermogenerator, a new feature provided by apparatus according to this invention. The discharge temperature from the thermogenerator can be not only held constant, but can be adjustably varied over wide limits by the selection of a water injection temperature which establishes the output temperature limit.

FIG. **8** is a P–V diagram for the air portion of the working fluid in FIG. **7** and has a process between **1***a* and **2***a* which is

isentropic compression. Between 2*a* and 3*a* is an isothermal heat addition with, in this particular case, a corresponding pressure reduction and volume expansion. The process 3*a* to 4*a* is affected by the presence of steam as in other processes in which there are two differing fluids present. In this case, there is an entropy increase in the air equal to the energy given up by the steam during the steam expansion process. There is, as a result of the expansion and interchange, a corresponding decrease in entropy of the steam.

FIG. 9 shows the T–S diagram for the air cycle component of the process included in diagram of FIG. 7. Between 1*a* and 2*a*, there is shown an isentropic compression which is adiabatic and reversible between the ambient conditions and the operating pressure of the thermogenerator wherein the rise in T is sufficient to cause detonation when fuel is injected. Between 2*a* and 3*a* is a heat addition process which may result in either: (a) rise in T with a decrease in P while increasing V; (b) a constant T with a decrease in P and an increase in V (as in the Carnot cycle); (c) a decrease in T with an increase in P and an increase in V. Between 3*a* and 4*a* is an expansion to approximately atmospheric pressure in which the entropy is approximately constant except for a small entropy increase due to heat exchange between steam and the air, which have differing coefficients of expansion. This diagram assumes there is no heat loss to the boundaries of the chamber in which the process occurs, and this is made possible for this engine operating as described because walls are insulated in sharp contrast to the process occurring in conventional internal combustion engines wherein the walls must be cooled to protect them from the temperature of burning fuel therein. Between 4*a* and 1*a* a heat rejection process is shown in which T drops and P rises to the condition of start in the diagram.

FIGS. 10 and 11 are the corresponding P–V and T–S diagrams for the water cycle. They illustrate water in the liquid, liquid-vapor, and vapor regions since a phase change occurs in the thermogenerator and a second phase change occurs on expansion toward zero pressure in the work engine. FIG. 11 also shows that the water portion of the composite cycle follows the Rankine cycle, except that a small deviation from the Rankine cycle occurs in the expansion process in that there is a small decrease in entropy associated with the expansion 3*s* to 4*s*. Energy removal associated with this decrease is equal to the energy added to the air during process 3*a* to 4*a*, resulting in the above-mentioned small increase in entropy of the air.

VARIABLE AIR-WATER CYCLE

It may be seen that the thermogenerator as shown in FIG. 2, when supplied with compressed air at an ignition temperature for the fuel, therein burned upon injection, being further supplied with a temperature-proportioned water injection settable at a desired limit, and with pressure sensing means operating a fuel valve proportionally responsive to temperature and/or pressure deviations, on the downward side of a preset limit, there results a fuel burning process supplying a working fluid which may be withdrawn at constant temperature and pressure but in which this temperature and this pressure may be varied at the control of an operator over wide limits. This cycle may be designated a "Variable Air-water Cycle." In essence, it is a two-fluid process in which at least one fluid changes between two phases. It is a process in which the two fluids may be characterized as a separated fluid or substance, namely, the products of combustion plus excess air, and water, each having its own partial pressure in the total pressure detected or measured for proportional control of thermogenerator fuel input. This separated substance is isentropically compressed in its two parts and combined, to which heat is then added to the total fluid in a constant pressure of total fluid process, followed by adiabatic expansion of the composite fluid to produce useful work, and a final heat rejection process for the combined fluid occurring at constant total pressure. By reference to FIG. 1, it will be seen that the first fluid portion which is compressed air is introduced at a pressure regulated according to a pressurestat or other regulating device for controlling resupply of working fluid according to that decrease in pressure which may occur either in standby coast-down or upon use of the engine. The second portion of the working fluid is the water compressed and added to the after end of the chamber during or after the burning, but which may be considered as an independent addition of a fluid. The combined air and water has heat added according to fuel burned, the addition being proportioned to input of the air portion and the water being added in proportion to the addition of heat such that the result of the heat addition is the heating of the combined fluid. When considered as an operating device, it can be seen that the fluids are compressed and injected isentropically and the heat is added under conditions to keep the total pressure constant, even though the water and air proportions may vary as may be necessary to keep temperatures constant for differing rates of fluid withdrawn, or standby heat loss. It can also be seen that the work engine withdraws this fluid for adiabatic expansion to perform the desired work without substantial confining wall cooling inasmuch as the walls may be at the mean temperature between work cylinder inlet fluid and outlet fluid. As in other heat engines, the composite fluid is discarded at constant pressure, which may be either atmospheric pressure or a reduced pressure if condensing means are applied to reduce that pressure on the pistons as by condensing the steam.

Such an engine has a cycle with the following advantages in contrast to prior power generator cycles:

a. complete control over wide limits of output T and P at the exhaust of the engine;

b. wide range of variability of T and P at the control of an operator during a run;

c. settable operating conditions without change of engine design;

d. combustion energy converted to a working fluid at substantially 100 percent thermal efficiency;

e. stoichiometric fuel-air mixtures, or richer at the start of combustion, followed by excess peripheral air mixing to assure complete combustion within a closed chamber;

f. absence of moving parts in the vicinity of fuel burning; and

g. absence of cooling walls or variation in the temperature, pressure and surrounding gas temperature during the burning process. By combining the two-phase water system operating upon the Rankine cycle with the air and combustion products operating on the Carnot cycle, advantages are achieved which are not obtainable from the two-cycles separately. While some of these advantages are obtained in the Brayton cycle, that cycle is not adaptable to reciprocating engines whereas the present cycle may be used either in turbines or reciprocating engines.

PREVENTION AND REDUCTION OF SMOG-FORMING ELEMENTS

All types of combustion tend to produce products which react in air to form smog, whether in engines or industrial furnaces, although of different kinds. Internal combustion engines operated with cooled cylinder walls and heads have boundary layer cooling of fuel-air mixtures sufficient to result in small percentages of unburned hydrocarbons emitted during the exhaust stroke. This invention provides means avoiding combustion chamber wall cooling in two distinct ways to keep the burning temperature for the fuel high: first, by flowing hot air around the combustion space inside the exterior wall such that the combustion occurs only within a space heated above ignition temperatures; and second, by shielding the flame with air unmixed with fuel. A heat shield surrounds the burning space throughout the burning area, which shield is insulated and heated to the ignition temperature prior to injection of any fuel since it is surrounded with hot input air. Thus, a hot wall combustion, preferably above 2,000° F., is possible in an engine in which the burning is carried on outside the ac-

tual work cylinders, this combustion chamber being held at a steady high temperature by the fuel burning within an air shield therein.

Recent investigations indicated that CO and other products of partial combustion are inhibited by high temperature burning, preferably well above 3,000° F., and retained for a considerable time after start of burning. At the higher temperatures, more nitrous and nitric oxides are formed, so that neither higher nor lower temperatures are a cure. However, this invention provides a more complete cure by commencing the burning at high temperature, then reducing that temperature for a considerable dwell time and then cooling (after completion of the burning) to a containable temperature by water injection at the permissible temperature limit to thereby assure complete burning of all the hydrocarbons, first in a rich mixture, and then in an excess of air, both to cool the gases below about 3,000° F. for about half of the dwell time in the chamber, and then quenched by water to an operating temperature.

For purposes of efficiency, a hydrocarbon fuel should be burned at a mixture with air a little richer than that required to supply oxygen enough to burn the fuel, i.e., the stoichiometric proportions. This would result in excess CO and more complex products of incomplete combustion. Dilution with further air is necessary, but should be done as a secondary process after some time lag. This invention provides for this progressive supply of air, first in the fire tube, and then in the air tube and heat shield, each separately supplied with unmixed air, which add the air progressively down the chamber length to provide dwell time in a rich mixture and then in a leaner and cooler mixture.

Oxides of nitrogen form more rapidly at higher temperatures, but appear to form more slowly than the combustion occurs so that a control of the rate of dilution by air helps to control the formation of nitrogen oxides. This procedure appears to be compatible with complete and efficient fuel burning to eliminate incomplete combustion products and reduce other products such as nitrogen oxides. After a considerable dwell time at the reduced but effective burning temperature, the products of combustion and excess air are then cooled to the engine working temperature, which may be in the range of 1,000° to 1,800° F., or may be as low as 700° to 800° F. as established by the water injection process limits set by thermostat. Selection of a temperature in a range about 750° F. minimizes both CO and the nitrogen oxides, if time is provided for reaching an equilibrium condition.

This is effected by making the combustion chamber from two to four times the length of the burning zone, the after end being quenched by water to that selected temperature, which further permits a distance separation between the zones of higher temperature and lowered containable temperature thus to minimize interaction. Air flow is progressive down the chamber and large temperature differentials are found to exist continuously for continuous firing, the resultant pressure being derived by phase changes at each end of the chamber, flow being progressive and relatively slow as indicated by a transit time between input of air and outflow which may be several seconds.

A burning as described provides a method of reducing some smog-forming elements while eliminating others, at the same time, providing a complete conversion of fuel energy to fluid energy.

A thermogenerator as in FIGS. **1** and **2** requires a proportional control. In FIG. **12** is illustrated a suitable fuel injector control which is proportional to engine revolutions whenever the compressor is delivering air proportional to engine revolutions and also proportional to work output when the machine is in normal operation. Thermogenerator **75**, generally as in FIG. **2**, has an elongated body **76** supplied with fuel via line **82** and an injector fitment through domed end **77** from a high pressure fuel pump in which pump pressure above a desired minimum relates to engine r.p.m. Fuel is led into the generator via a fuel jet housing **79** having therein a nozzle as in FIG. **1**. The thermogenerator has air inlet tube **81**, fuel and pressurized water lines **82** and **83** and a pressurized gas line **84** to valve **85**, which controls operation of a positive fuel shut-off. A water shut-off valve **86** is in line **83** to nozzles **87**. A fuel positive shut-off valve **85** is provided being under control of a first thermostat mounted at **88** having leads **89'** to an actuator such as **65**. Working fluid generated for operating an engine fills chamber **28** and this is connected to gas pressure line **84** extending by way of a conventional gas pressure reducing valve **91** to inlet **92** into a chamber **93** of the fuel injector according to pressure reduction and positive cutoff by a device shown generally at **100**, including a proportioning valve.

Valve **85** is a conventional three-way valve having a rotor element connecting an inlet and an outlet alternatively to a pressure line **94** extending to inlet **95** for a second chamber **96** of the proportioning valve. This valve accomplishes the function of response to reducing pressure in the chamber **28** whereby the fuel valve is opened if the crankshaft is in operation to operate the pump. A needle valve **97** has a seat **97'** within fuel line **82** and connecting to fuel line **882'** which extends back to the fuel tank. Seat **97'** cooperates with a tapered valve stem portion of the needle valve type shown at **98** which is effective to progressively close the valve, shutting off bypass line **82'** back to the fuel tank. This arrangement permits the fuel pump to operate according to crankshaft rotation and permits bypassing of fuel through the bypass line **882'** whenever it is desired to cut off fuel to the nozzle. This cutoff is progressive and gradual according to positioning of the tapered portion **98** of the valve stem **101**.

A similar needle valve arrangement can also close the fuel nozzle within housing **79** by the provision of a cooperating valve seat and plunger **99** in housing **79**. When valve **97** is closed the full pressure of the fuel pump is supplied via line **82** to the housing **79** and operates against a spring **104** urged against a shoulder **105** secured on the valve stem to operate the closure at point **99**, the spring **104** being adjusted to hold valve **99** closed whenever the pressure within the line **82** is less than a desired minimum, above which valve **99** opens.

Fuel injector **100** has slip joints **102** for a stem **101**. Packings **103** at either end of the injector housing prevent loss of gas pressure from the chambers **93** and **96** through the end portions and prevent fuel from being forced up through the lower end of the valve housing to chamber **96**. A spring **104'** and a shoulder **105'** similar to spring and shoulder **104** and **105** of the injector nozzle housing **79** permit a spring loading of valve stem **101**. Chambers **93** and **96** have therein shoulders **106** and **107**, respectively, which may be oppositely acted upon by the pressures in these closed chambers. Pressure in **96** is governed by the pressure in the chamber **28** conducted by way of line **84**, valve **85** and inlet **95** to chamber **96**. Pressure in chamber **93** is governed by the pressure reducing valve **91** feeding into inlet **92** to chamber **93** and is preferably adjusted at about 100 lb., gauge, for normal operation of the thermogenerator. However, the pressure in chamber **96** varies directly in proportion to the pressure in chamber **28** except when valve **85** is operated to vent gas in chamber **96** by way of **95, 94, 85**, and vent tube **108**. It is therefore possible, when the thermostat operating actuator **65** is indicating too low a temperature for fuel injection, to cause the valve **97,98** to open whereby the fuel is bypassed from the pump back to the supply tank. When the thermostat is actuated to hold line **94** open to line **84** and line **92** to line **84** by way of reducing valve **91** it will be seen that the pressure in chamber **93** aids spring **104'** in forcing the needle valve **97** to be closed. However, as pressure rises in chamber **28**, and is communicated to chamber **96**, valve **97** opens progressively according to the increasing pressure so that pressure in line **82** is varied. Valve **99** then serves as a fuel injector preventor so that fuel will not accidentally drip into the combustion chamber, while at the same time permitting a rate of injection which varies according to the pressure in line **82** when the fuel line pressure is adequate to open valve **99**. Control of the operation of valves **97** and **99** is effected by adjustment of springs **104** and **104'** by

adjustment of spring caps **109** and **110** to effectively vary the spring tension. Alternatively, valve **85** may be placed in line **92** so that the thermostatically controlled actuator **65** operates to aid in closing valve **97** by applying pressure in chamber **93** or assure the opening of valve **97** by venting inlet **92** to the atmosphere.

In FIG. **12A** valve **85** connects line **92** to line **84** by way of reducing valve **91'** when actuator **61'** operates, being shown in the fuel bypass position as when chamber **76** is below ignition temperature. Chamber **93** is vented at **108** and pressure in chamber **96** is sufficient to open valve **97**. As ignition temperature is reached **61'** operates to connect **84** to **93** for controlling valve **97**.

Various controls have been illustrated in diagrammatic form and may be varied in construction and in linkage to the temperature and pressure sensors without departing from the control concepts as here described, the illustrations being not intended as limitations, and the scope of protection being limited only in accordance with the following claims.

I claim:

1. An internal combustion engine comprising

a compressor for elevating the pressure of ambient air received at an intake therefor to a value in the range of 12 atmospheres and higher, the temperature being thereby elevated to the ignition temperature for a selected fuel,

a closed combustion chamber discrete from said compressor and having a connection to said compressor by way of a check valve for preventing backflow,

means insulating said compressor and chamber to prevent loss of heat of compression during periods of storage of compressed air in said chamber,

means for initial driving of said compressor during accumulation in said chamber of air pressure above a predetermined lower limit having a heat of compression sufficient to raise the air to said ignition temperature,

exhaust valve means for said chamber operative according to power demand,

a work engine supplied through said valve means according to said demand,

means coupling said engine to said compressor in driving relation therefor to resupply compressed air in proportion to air exhausted through said valve,

pressure detector means in said chamber adjusted to generate a fuel control signal independently of air supply in response to pressure decrease below a minimum value,

fuel injection means including

a fuel source under pressure,

a spray nozzle directed into said chamber, and

metering means connected for response to said control signal,

air flow control means for causing compressed air from the compressor to mix with said fuel progressively along one dimension of said chamber toward said exhaust valve means,

thermostat means in said chamber responsive to temperature therein above a set temperature maximum for generating a second control signal, and

water injection means including

a water source under pressure,

at least one nozzle directed into said chamber, and

water metering means connected for response to said second signal for injecting water at a rate sufficient to decrease said temperature detected by the thermostat means,

said work engine being supplied with a mixture comprising said compressed air and fuel combustion products and said water converted to steam at pressure above said minimum value and at temperature above said ignition temperature and below said maximum.

2. An engine according to claim **1** said compressor being a displacement device and said work engine comprising a plurality of cylinders having combined displacement exceeding the total displacement of the compressor, said work engine driving a crankshaft connected to drive said compressor.

3. An engine according to claim **1** said compressor and said work engine being positive displacement devices, said means for generating a second control signal being a thermostat adjustably settable in the range of 725° to 1,800° F.

4. An engine according to claim **1** said means for initial driving of said compressor including means preventing operation of said fuel injection means before reaching spontaneous ignition temperature.

5. An engine according to claim **1** including pressure signaling means responsive to pressure less than a second minimum value and means responsive thereto for preventing operation of said fuel injection means before reaching spontaneous ignition temperature.

6. An engine according to claim **1** said compressor and work engine being coupled by common or interconnected crankshafts coupled to pistons in compressor and work engine cylinders.

7. An engine according to claim **1** said metering devices being adjusted to deliver water not substantially less than a weight ratio of seven times that of fuel delivered in response to said fuel metering means.

8. An engine according to claim **1** said metering devices being adjusted to deliver substantially 3 to 8 percent of fuel by weight relative to the air delivered to said chamber and water substantially at least seven times the weight of fuel to prevent average temperature rise in the chamber about 1,800° F.

9. In a reciprocating-piston engine having at least compression and expansion strokes,

a first cylinder having piston means for compressing air, sufficiently for autoignition,

a combustion chamber connected through check valve means for accumulating successive charges of said air when compressed, said chamber means being insulated thermally to retain the heat of compression,

at least a second cylinder having second piston means for receiving gas from said chamber in driving relation to said second piston means,

crank means coupling said first and second piston means for causing first piston operation to charge said chamber whenever said second piston means is operated independently of the pressure in said chamber,

means controlling withdrawal of successive portions of accumulated gas from said chamber to continuously drive said second piston means,

means detecting a decrease in pressure in said chamber below a set value as gas is withdrawn,

fuel injection means exiting into said chamber for continuously inserting fuel in response to said detected decrease in pressure,

means detecting a rise in temperature in said chamber above a predetermined operating temperature range, and

means responsive to detected temperature rise for inserting a quantity of atomized cooling liquid into the chamber proportioned to substantially absorb the heat of combustion of said quantity of fuel.

10. In an engine according to claim **9** said first cylinder and piston means being gas intake and compression means, said second cylinder and piston means being work stroke and exhaust means, said combustion chamber comprising means for separating the combustion engine cycle functions into separate means for effecting intake-compression, combustion-cooling, and work-exhaust, respectively.

11. A compression-ignition fluid fuel engine, comprising

a plurality of variable fluid chambers each enclosing a compression-expansion member connected to a rotative power shaft and in control of gas volume therein according to shaft rotation, said member of a first chamber being driven as a compressor and having a compression factor not less than sufficient to provide autoignition temperature and at least one second chamber member connected in driving relation to said shaft,

a combustion chamber of fixed volume connected between said variable chambers including means detecting a decrease in pressure therein below a setting,

means admitting an oxidizing gas to said first chamber for compression therein in response to shaft rotation,

means passing from said first chamber to said combustion chamber gas compressed therein independently of pressure in said combustion chamber,

outlet means connecting said combustion chamber to one or more other said chambers including variable valve means for controlling outlet of working fluid.

a liquid fuel supply at a pressure exceeding the pressure in the combustion chamber.

fuel injection means connected to said supply being regulated to supply fuel to the combustion chamber in proportion to a said decrease in pressure, independently of temperature therein,

temperature sensing means in said combustion chamber for detecting temperature increases above a set value, and

injection means for injecting a liquid coolant into said combustion chamber in proportion to temperature increase above said set value, thereby to control temperature therein independently of fuel and air supplied thereto.

12. An engine according to claim **11** said temperature being controlled in the range of 500° to 1,800° F.

13. An engine according to claim **11** said instant pressure being in the range of 200 to 800 pounds gauge.

14. An engine according to claim **11** said liquid being water.

15. An engine according to claim **11** including thermal insulation means surrounding said combustion chamber for maintaining stored heat during periods between engine use.

16. An engine according to claim **11** said chambers being thermally insulated from each other to prevent heat flow from said combustion chamber to said first chamber and said second chambers.

17. An engine according to claim **11** said chambers being pistoned and cranked for displacement substantially double the volume change of said first chamber.

18. A compression-ignition engine, comprising

a plurality of pistoned combustion chambers associated together in a block,

a crankcase attached to said block,

a crank within said case having crank arms operatively connected to pistons in said cylinders,

intake valve means for a first cylinder for admitting an air charge upon outstroke of the piston therein as said crankshaft rotates, said first cylinder being proportioned relative to the length of stroke of a piston therein for compression adiabatically to above a self-ignition temperature in said charge,

air outlet means for passing said charge to said combustion chamber, including a unidirectional valve whereby repeated charges are accumulated in said chamber,

pressure sensing means in said chamber,

a fuel supply including injection means for continuously supplying fuel to said charge,

means for operating said injection means in response to a sensed decrease of pressure below a predetermined pressure,

valved conduit means connecting said chamber to another said cylinder for controlling flow of working fluid thereto in response to a work requirement,

temperature sensing means in said chamber,

a vaporizable liquid source under pressure exceeding the pressure in said chamber, and

means connected thereto for controlling injection in proportion to a sensed temperature rise above a predetermined point independently of chamber pressure.

19. A combustion engine, comprising one or more positive displacement air compressor pistons and a number of positive displacement work pistons double the number of said compression pistons, each mounted for reciprocation in a confining cylinder means, crank shaft means connecting said pistons for simultaneous operation according to selected crank orientations,

combustion chamber means exterior to said cylinder means and connected to receive unidirectional air flow from the compressor cylinder means independently of the pressure in said combustion chamber and to supply working fluid to the cylinder means for said work pistons,

means injecting fuel into said chamber means proportioned to said air flow,

means detecting a temperature rise in said chamber means relative a design value,

means injecting water independently of chamber pressure into said chamber proportional to said temperature rise for substantially doubling, upon volatilization, the quantity of working fluid comprising said air and the products of fuel combustion.

20. An engine comprising an external elongated combustion chamber for generating a working fluid for said engine

means supplying compressed air at a fuel ignition temperature in one end of the chamber proportioned to fluid withdrawn,

means injecting fuel axially of said chamber for ignition at said one end,

first means confining flame to a first central zone of the chamber for a first portion of the length of the chamber,

first means admitting a first portion of said compressed air insufficient for complete combustion into said first means for confining the flame,

second means confining flame to a second zone surrounding said first portion and extending therebeyond from said one end,

second means admitting a second portion of said air to substantially complete fuel combustion in said second zone,

third means confining flame to a third zone surrounding said second zone and extending therebeyond along the chamber,

third means admitting a third portion of said air to said third zone sufficient to at least complete combustion in the third zone,

means supplying excess air in said chamber around said third zone for forming a mixture with combustion products beyond said third zone,

means injecting cooling water into said mixture beyond said third zone toward a second end of the chamber, and

means withdrawing portions of the resulting binary working fluid for operating said engine coupled to drive the air compressing means.

21. In a chamber according to claim **20** said second and third flame confining means being generally cylindrical and concentric with said first confining means and each extending beyond the flame confining means therein so as to mix successively added portions of admitted air peripherally around the previously formed zones of combustion.

22. In a chamber according to claim **20** said means injecting water being spaced remotely from said one end to prevent cooling of said mixture until combustion is completed as the mixture passes through the chamber.

23. In a chamber according to claim **22** said spacing being at least substantially equal to the diameter of said third flame confining means, whereby time is provided for establishing equilibrium conditions between excess air and the undesired products of combustion at temperatures according to excess air supplied prior to water cooling.

24. In a chamber according to claim **20** said means injecting water being located proximate to a second end of said chamber and remotely from said third zone such that the time after burning and before water cooling exceeds the time for burning in said three zones.

25. In a chamber according to claim **20** said compressed air being supplied through additional means for converting pulsating air flow to substantially uniform air flow at a velocity small with respect to the speed of flame propagation wherein the mixture of combustion products and excess air is retained at the working fluid pressure and a temperature in excess of chamber peripheral temperatures being surrounded by said excess air.

26. A thermogenerator for supplying a heat engine, comprising

an elongated pressure chamber having air inlet means at one end and combustion product outlet means at the other end thereof,

check valve means in said inlet means operative to admit air when the pressure in said chamber is less than the pressure in said inlet means,

power valve means comprising said outlet means to withdraw working fluid from said chamber in response to valve operation,

air pump means connecting to said inlet means for supplying air compressed to a temperature sufficient for self-ignition of a fuel,

air manifold header means extending substantially across said chamber proximate to the inlet end thereof,

a tube extending from said air manifold means axially of said chamber toward the outlet forming a central fire tube,

injection means for injecting fuel axially along said tube in response to outlet valve operation,

a second tube supported concentrically about said fire tube and extending from said manifold means along said chamber beyond the termination of said fire tube,

a third tube surrounding said second tube and concentric therewith extending from said manifold means at least substantially halfway to the outlet end of said chamber,

perforations through said manifold header means admitting air in adjusted relative proportions respectively to said fire tube, between said fire tube and said second tube, between said second and said third tubes, and exteriorly of said third tube such that air flow into the chamber is predominantly about the injected fuel and distributed axially along the chamber, and

means responsive to operation of said power valve means for causing pressure and temperature to be maintained in the chamber, said temperature being above the ignition temperature for a fuel injected.

27. A thermogenerator according to claim 26 including temperature sensor means within said chamber for providing an indication of a rise in temperature above a predetermined limit, and

liquid injection means coupled to said sensor means having at least one outlet within said chamber for introducing coolant liquid in response to said indication of rise in temperature.

28. A thermogenerator according to claim 27, and means operating the pump means during opening of the power valve means to cause continual air flow in said tubes.

29. A thermogenerator according to claim 27 said liquid injection means being actuated in proportion to a sensed rise in temperature in said chamber.

30. A reciprocating engine of the character disclosed comprising in combination at least one compressor chamber having an outlet to a combustion and working fluid storage chamber, a plurality of expander chambers having controllable valved inlets from a working fluid storage chamber, and a combustion and fluid storage chamber connected to receive output from said compressor chamber outlet and an outlet connecting to said valved inlets, said storage chamber having volume at least exceeding the displacement volume of said expander chambers, said combustion chamber including connections for fuel and water supplied under pressure for injection thereinto at mutually remote locations, said expander chambers being operative to drive an output mechanism operatively coupled for supplying pressurized fuel and water, and said storage chamber including upper limiting temperature sensing means in control of water injected independently of chamber pressure and lower limiting pressure sensing means in control of fuel injected to separately maintain substantially constant temperature and pressure values of said stored working fluid.

31. In an engine including a plurality of cylinders fitted with work pistons for reciprocation therein and at least one compressor cylinder fitted with a piston for reciprocation therein, being connected for operation simultaneously with said work pistons and including ambient air inlet control means for supplying air at a pressure not substantially less than 12 atmospheres, the combination of

an elongated combustion and gas storage chamber of volume exceeding the volume of said cylinders having a first end fitted for receiving said air and a second end fitted for delivering a mixed working fluid comprising combustion products and steam,

means in said first end for injecting axially into said chamber a fuel charge,

shielding barrier means surrounding said fuel injecting means for separating burning injected fuel from outer walls of said chamber,

means providing a portion of said air into said barrier means insufficient for complete burning of said charge and supplying the remainder thereof peripherally therearound in sufficient quantity to complete combustion at a position in the chamber intermediate said ends,

means detecting a rise in temperature in the chamber above a design working value at a point remote from said first end,

means responsive to said detected rise in temperature for injecting into said chamber near said second end an amount of water sufficient to reduce the temperature below said limit,

means effective, after said engine has been placed in operation so as to develop a predetermined pressure value, for preventing further fuel injection in response to a further rise in pressure, including means for continuing fuel injection in response to a decrease in pressure below said value so as to maintain said value of pressure substantially constant,

said means for injecting water according to temperature and said means for injecting fuel according to pressure comprising controls for providing a volume of mixed working fluids at independently controllable constant temperature and pressure values.

32. In an engine comprising separate pistoned air compressor and work fluid expander units interconnected for simultaneous operation and an external combustion chamber of volume to store sufficient energy to operate said expanders and compressor units for at least a plurality of cycles in the absence of further supplied energy,

means in said combustion chamber for detecting a decrease in pressure below a value corresponding to a desired reserve of energy stored,

means causing recycling of said expander and compressor units in accordance with a said detected decrease in pressure,

means responsive to operation of said units to cause injection of fuel thereby to augment said stored energy upon burning of the fuel, and

means responsive to a rise in temperature in said chamber above a desired temperature value for injecting sufficient water therein to restore chamber temperature substantially to said value,

said compressor and expander units being connected to supply combustion air to, and withdraw working fluid from, said chamber, respectively, and being proportioned to provide an adiabatic temperature rise sufficient for self-ignition of said fuel.

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