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Bush et al.

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[54] CIRCUITAL FLOW HOT GAS ENGINES

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[52] **U.S. Cl.** ..... **60/59 R**, 60/36

[51] **Int. Cl.** **F01b 1/00**

[58] **Field of Search** ..... 60/59 R, 59 T, 57 R, 39.6, 60/36

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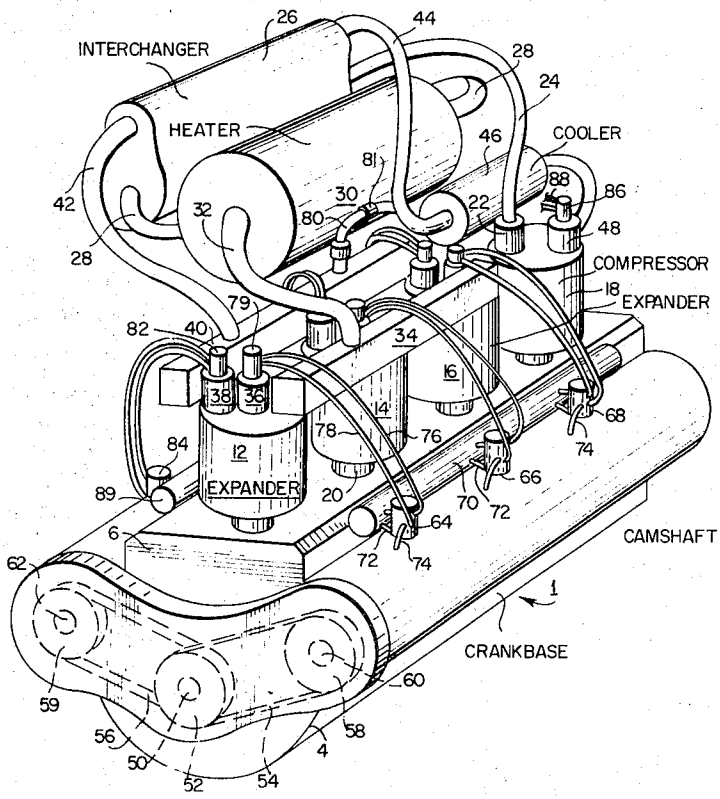
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[57] **ABSTRACT**

A compressor, heater, expander and cooler are connected to form a closed circuit. Valves control the pulsating flow of a working gas about this circuit. An interchanger may be included in the circuit to allow use of a low expansion ratio with a high ratio of extreme temperatures. Relatively large volumes of heater and cooler may be used. Power output is controlled by altering valve timing. Alternately, power output may be controlled by the use of auxiliary chambers and altering the mass of working gas in the circuit.

34 Claims, 6 Drawing Figures



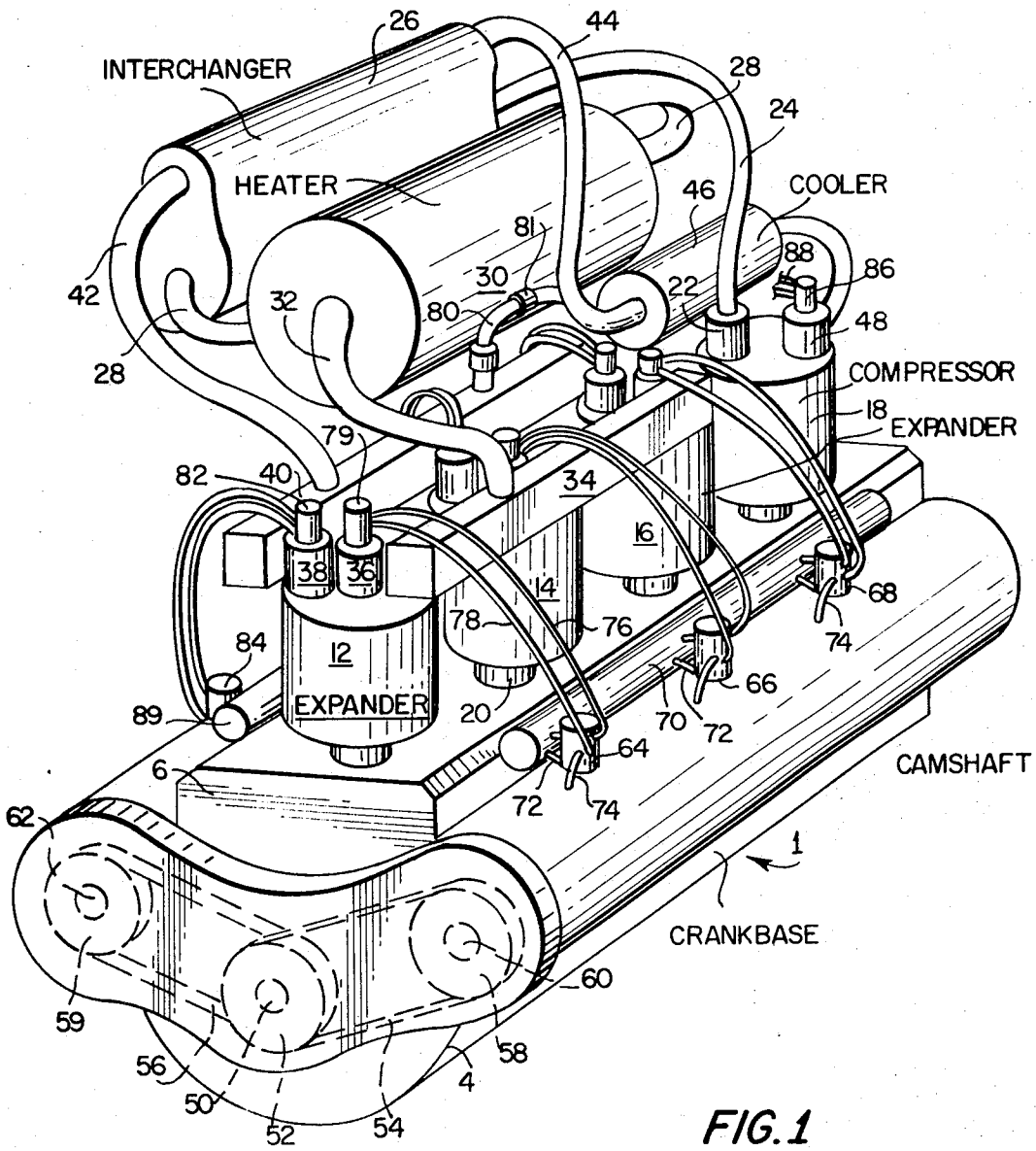


FIG. 1

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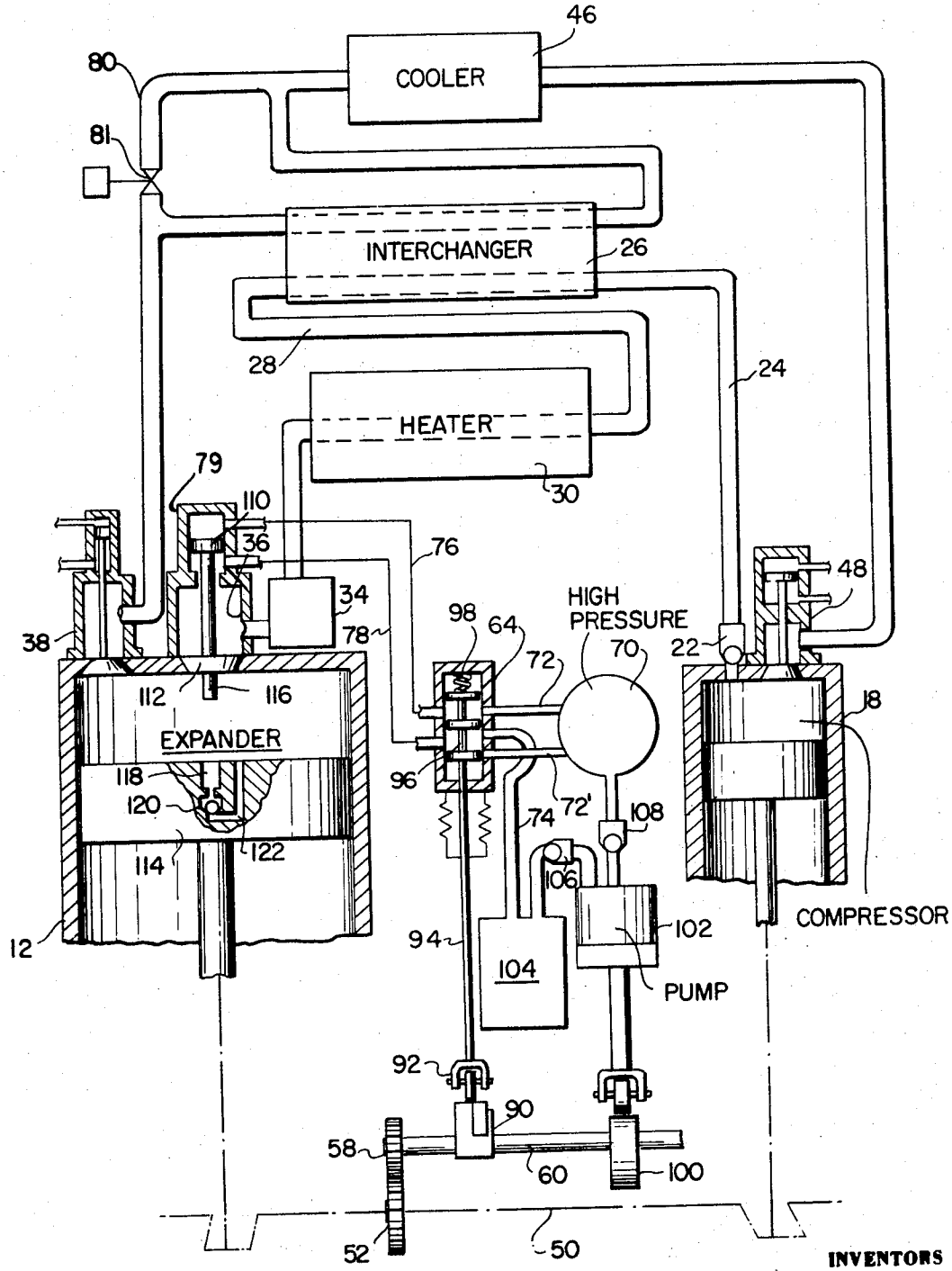


FIG.2

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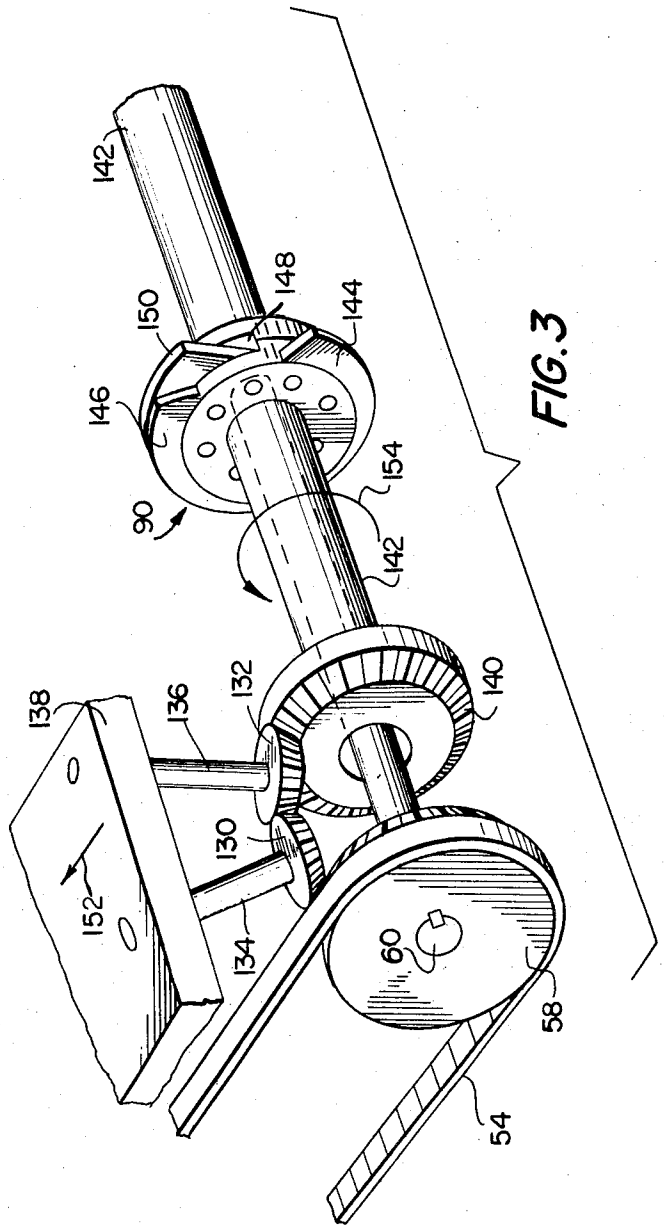


FIG. 3

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FIG. 4

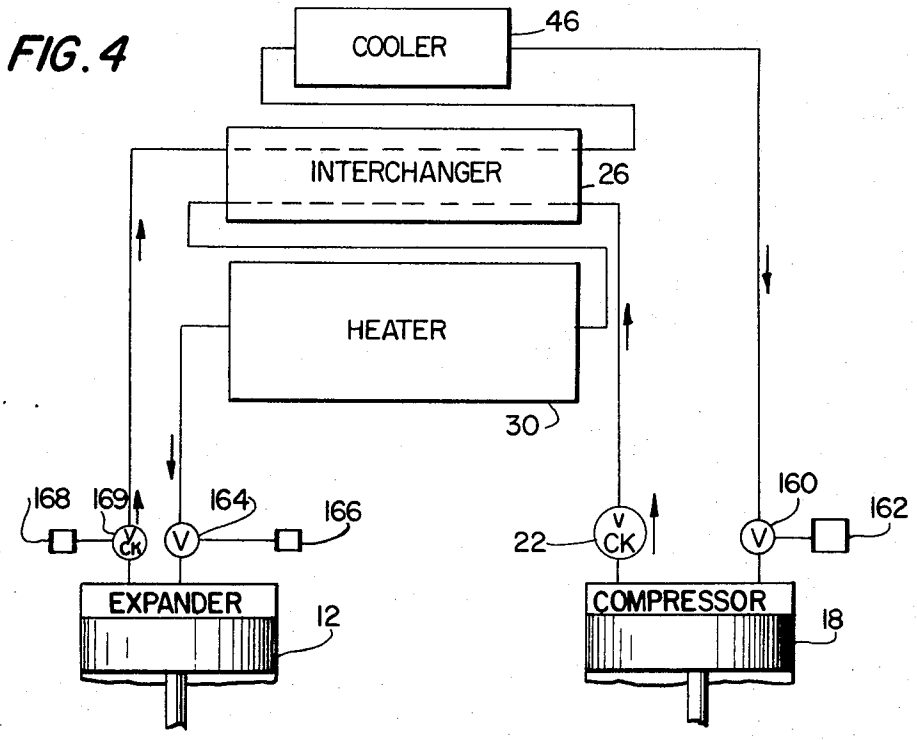


FIG. 5

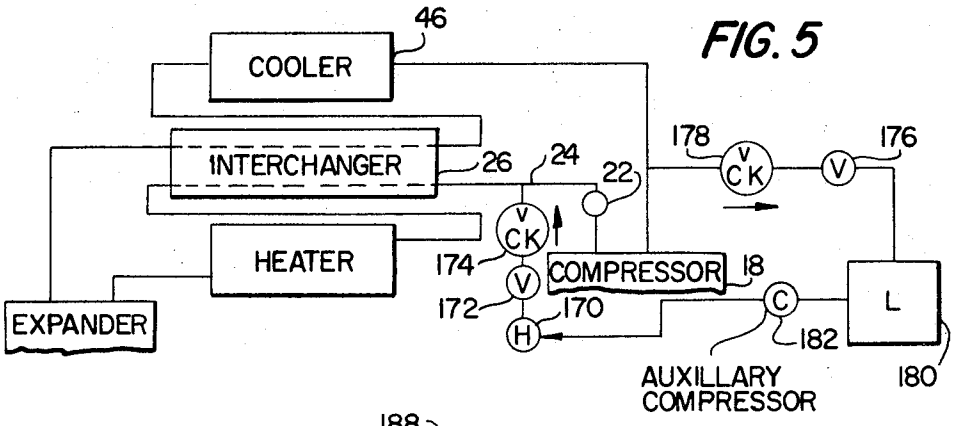
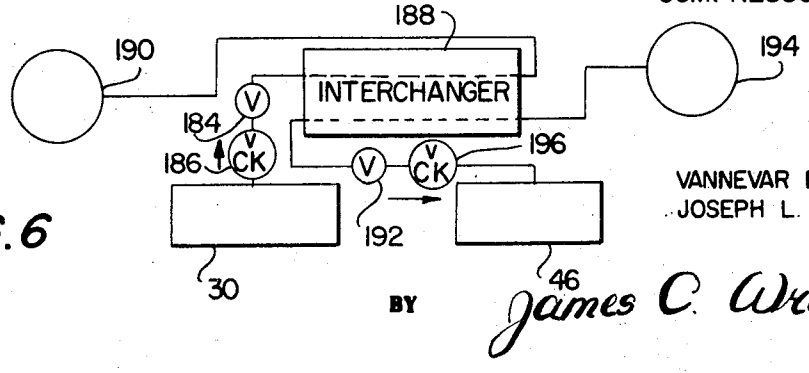


FIG. 6



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CIRCUITAL FLOW HOT GAS ENGINES

BACKGROUND OF THE INVENTION

This invention presents a new form of hot gas engine. Hot gas engines are a form of external combustion engines, that is engines in which heat, to be partially converted into work, is applied externally, not as in engines in which combustion occurs inside of a cylinder.

The usual hot gas engine comprises a gas filled volume and a means by which the bulk of the gas is shifted cyclically, alternately in one direction and then the other direction from one region to another, and an arrangement by which the consequent change of temperature and pressure is caused to perform work, usually by means of a piston operating a crankshaft. There is also a regenerator to cause the changes of gas temperature to occur gradually rather than abruptly.

In another known form of hot gas engine, gas is caused to move in the enclosed volume about a circuit rather than to move cyclically back and forth through the same channel. Compressed gas flows from a compressor through a heater to an expander and then through a cooler to the compressor. The regenerator commonly found on cyclic flow hot gas engines is omitted as no longer necessary. For certain modes of operation, the regenerator is replaced by a heat interchanger.

Conventional hot gas engines may be employed in constant output applications in which power requirements remain uniform. Constant load pumps and generators are some examples of satisfactory applications of constant output engines. However, where power requirements rapidly vary, such as in automotive engines, conventional hot gas engines have been unsatisfactory.

Methods of adjusting power in hot gas engines have concentrated on changing heater temperature or changing the gas charge and hence changing system pressure. Known power control methods have several drawbacks, including inherent slowness of response.

SUMMARY OF THE INVENTION

The broad objectives of the invention are realized in hot gas energy conversion apparatus which heats, expands and compresses gas repetitively in a closed circuit and which controls the energy output by variably controlling flow of working gas around the circuit.

The primary elements in the preferred forms of the new engine are compressors, heaters, expanders and coolers connected in closed circuits, and variable valves between at least two adjacent elements. In one embodiment, an interchanger is added, and the interchanger may be used or bypassed as conditions require.

A gas compressor may be of any of the usual forms of such a device, for example single or multiple reciprocating or rotating devices of variable or uniform volumes. For convenience, a preferred form of the invention is described as a compressor consisting of piston and cylinder, with inlet and exhaust valves.

A heater is preferably in the form of a bundle of tubes in a furnace, so arranged that the working gas enters at one end, is gradually heated to a high temperature as it flows through the heater tubes, and emerges at the other end. The heater internal volume, containing the working gas, is preferably large compared to the active volume, presently to be described. The furnace is designed so that good furnace efficiency may be attained. There are no strict limits on the area available for heat transfer between furnace gases and working gas. That is in contrast with the conventional hot gas engine in which all non-active volumes, such as heater volume, must be held to a minimum in order to secure improved output, thus placing severe restrictions on design.

A cooler is preferably of a type similar to the heater, and also preferably is of relatively large volume. Rejected or unused heat is transferred from the working gas to a cooling medium, usually air or water. As in the heater there is a temperature gradient along the path of working gas flow in the cooler.

One or more gas expanders may be of any form of variable internal volume, but will be described as piston and cylinder units adapted to operate with a crankshaft in the usual manner to produce mechanical power.

A set of valves admits high pressure gas from the heater to an expander for a chosen interval, and exhausts the gas to the cooler after expansion. These valves may be operated by mechanical, electrical or pneumatic methods, and may be controlled by any convenient device, preferably by cams rotating at the speed of the drive shaft. Provisions for altering the timing of the valves are described herein.

The valves may be mechanically operated, as in a conventional internal combustion engine. In this discussion however, it will be assumed that they are pilot operated, that is that the power shaft operates a set of small pilot valves which in turn operate the main valves, or that the power shaft operates a set of contacts, and corresponding solenoids operate the main valves.

A heat interchanger, during certain aspects of engine operation, cools gas flowing from the expander to the cooler and correspondingly heats gas flowing from the compressor to the heater. The two gas flows in the interchanger are separated by relatively thin walls so that the gas entering from the expander can readily give up heat to the gas passing from the compressor to the heater. The use of an interchanger is an important feature since is allows the effective use of a relatively low pressure ratio with a high temperature ratio between heater and cooler. When a low pressure ratio is used, an interchanger is needed; it allows good output and efficiency under those conditions. But if a high pressure ratio is produced, as when the cutoff point is made earlier, and compressor valves are altered in timing, the interchanger is no longer needed, in fact becomes a distinct disadvantage. Hence, it is well to bypass the interchanger when valve control produces low output.

The flow of working gas about a circuit is thus as follows: The compressor draws gas from the cold end of the cooler, compresses it nearly adiabatically, with consequent increase of temperature, and delivers it to the low temperature end of the heater, with the modification that, during certain phases of engine operation, it is further heated on its way to the heater by heat transfer in the interchanger. The gas then flows through the heater, and its temperature rises to a maximum. Working gas flows into an expander during a controlled interval through which the expander volume has increased from the point of minimum volume, near zero,

to cut off volume. During the balance of the increase of volume of the expander, the gas is expanded nearly adiabatically and is correspondingly cooled to a lower temperature and pressure. On the return stroke during the decrease of volume of the expander, the gas is forced into the warm end of the cooler, or, under certain conditions, the gas loses heat in the interchanger before reaching the cooler. The gas then flows through the cooler, and reaches its minimum temperature, to complete the circuit.

The expander and compressor may operate on the same shaft; there may be a number of expanders and compressors operating with a common heater and cooler. The total active volume of compressors will be less than that active volume of expanders. Both expander and compressor cylinders are preferably double acting.

In a specific embodiment of a four cylinder engine, one cylinder is used as a compressor, and the other three are used as expanders. The diameter or stroke, or both, of a compressor may be different from those of an expander. Pistons in the cylinders are connected to a single crankshaft at convenient phase intervals. Each expander has individual valves connected to intake and exhaust headers, which are respectively connected to the heater and to the interchanger.

Inlet valves of the expanders open at the dead center of the pistons when cylinder volume is a minimum, and in the present embodiment remain open for an adjustable fraction of the stroke. A cam arrangement for this purpose is on a camshaft which is directly driven by the power shaft to turn at the same speed, and in the same or opposite direction, by gearing from the power shaft. In this train of gears there is included a differential. When the third shaft of this differential is held stationary, the drive occurs as described. By adjusting the angle of the third shaft, however, the angular relation between the two part cam and dwell may be adjusted. When the elevated cam portion is adjusted, the valve can be made to close at any desired angle of the power shaft. In preferred embodiments, expander intake and exhaust and compressor intake valve timing is varied. Adjusting cam dwell is one example of a convenient method for changing valve timing.

For certain embodiments, both valves of the compressor may be simple check valves. For the present embodiment, it is desirable that the delivery of the compressor be adjustable. For this purpose, the compressor outlet may be a check valve, and the inlet valve may be operated by a cam mechanism such as is described for expander valves. For full delivery, the inlet valve will be open during the entire interval when cylinder volume is increasing, and the check valve will be open during an interval when it is decreasing. For partial delivery, however, the inlet valve will be caused to close before full cylinder volume has been reached. In this condition the gas in the cylinder will be expanded before it is compressed to heater pressure to open the check valve. In an alternate arrangement, the compressor inlet valve may be held open after the cylinder has reached full volume. In this condition, a part of the gas charge flows back to the compressor before compression begins. The compressor cylinder may be of the same diameter and stroke as the expander cylinders; they may be of different diameters.

Adjustment of the expander valves alters the point of cutoff, and the mass circulated per stroke, and hence alters the expansion ratio and the power output. Adjustment of the compressor valves serves to maintain mass balance, that is to ensure that the same mass of gas per cycle is delivered to the heater as is extracted therefrom, thus preserving pressure relations.

It is desirable that maximum temperature of the heater tubes be maintained constant, independent of load, for this improves efficiency. This is provided for by thermostatic control of the furnace.

Preferably, the cylinders of the engine are double acting, that is with operating gas in both ends of the cylinder, as in reciprocating steam engines, rather than single acting as is conventional in internal combustion engines. The maximum power output of an engine, for a given weight and size, depends upon speed, allowable pressures, and allowable forces on bearings. The use of double acting cylinders not only introduces directly a factor of two, but also greatly decreases the load on bearings, for pressures in the two ends of the cylinder tend to partially offset one another. In this connection it should be noted that, while in a nonsupercharged internal combustion engine the maximum cylinder pressure is determined largely by atmospheric pressure and compression ratio, there is no such limit with the present engine. When double acting cylinders are used there is employed a crosshead and crankshaft such as has long been used with reciprocating steam engines. The piston rod emerges from the cylinder through a sleeve or packing. Since there is bound to be some leakage past this sleeve, it is preferable to use a sealed crankbase full of the working gas. A small engine driven pump may be used to transfer working gas from the crankbase to the operating volume or to a reservoir. The crankbase, with its bearings, is lubricated in the usual manner, but there is preferably no oil in the working volume. When helium is used as a working gas, and the crankbase is sealed, there is no tendency for lubricating oil to deteriorate.

The engine herein described differs markedly from the internal combustion engine. That engine is restricted by maximum allowable compression ratios. The maximum pressures used in the latter engines depend on mechanical strength of cylinders and allowable compression ratios, unless they are supercharged. In the present engine, maximum cylinder pressure can be chosen merely from strength considerations. Thus, one can halve cylinder cross section and double pressure to obtain the same output, without increasing load on the bearings. In engine design, it is the maximum load on bearings which is usually determining.

This leads to a second advantage of double acting cylinders. The maximum pressure may be increased by the value equal to the minimum pressure without increasing the maximum load on bearings. This renders it favorable to operate with large temperature ratio and with late cutoff.

In the embodiment discussed above, if it is made double acting, each cylinder is concerned only with either compressor or expander action. In an alternate embodiment a single cylinder or multiple cylinder engine may be used. One end of each cylinder acts as an expander; the other end acts as a compressor. The end containing the piston rod is preferably the compressor.

Even so, the compressor volume, if fully used, is too large for appropriate mass balance. Hence, the compressor side has idle volume. That is, during piston movement the minimum volume is larger on the compressor side than on the expander side. Preferably, a single piston seal is placed near the compressor end of the piston, where the temperature is low. With this configuration, the engine may operate without a high temperature seal.

VALVE OPERATION

A preferred way of altering output is by altering valve timing.

When full output is to be maintained at all times, as might be the case of an engine driving a water pump operating with a fixed head, the valves on the compressor may be simple check valves, such as are commonly used in compressors. When the ratio of compressor to expander volume is correctly chosen, in the light of cutoff point and the temperatures of operation, the gas will be expanded in the expander to approximately the pressure in the cooler at the end of the expander stroke. When the expander exhaust valve is opened at the dead center of the expander cylinder, there is very little pressure across it. It can therefore readily be opened by a small pneumatic actuator.

The engine expander and compressor valves are operated by use of small cylinders and pistons, using the working gas which is controlled by pilot valves. That has some advantages, for example in some types of application the pressures developed in the engine may be utilized to provide pressures in reservoirs, from which the engine valves may be operated. Also, the use of helium in valves, which may be at very high temperatures, avoids problems of oxidation. Helium provides rapid engine valve operation in response to pilot valve changes.

Preferably, there are two small auxiliary reservoirs, in which pressure of working gas is maintained by a small pump. One reservoir is at a pressure close to atmospheric, and the other is at a high pressure. The auxiliary pump for this purpose may be operated by a battery when the engine is idle, to have proper conditions for starting the engine.

Small pilot valves are driven synchronously with the expanders and compressor. The pilot valves selectively connect actuators for expander and compressor valves with the high and low pressure reservoirs to open and close the engine valves. Preferably gas driven actuators operate the compressor intake and expander intake and exhaust valves, for example:

The exhaust valve is constructed with a piston and cylinder with connections to the pilot valve. A chamber below the actuator cylinder has a passage connected to the interchanger. Any leak along the actuator rod merely allows gas to flow to the chamber and to the low pressure side of the system. The actuator cylinder may be of very small diameter if a high pressure is maintained in the small reservoir noted above. By placing the ports at proper points in the actuator cylinder, there is a dashpot effect at the end of the stroke in either direction. Thus, although the engine valve operates very rapidly, undesirable impact is avoided at the end of the stroke.

In a preferred form, the actuator cylinder is operated by a spool type pilot valve which selectively connects ends of the actuator cylinder to the high pressure reservoir and to the low pressure reservoir. The spool is moved by a cam on a shaft which turns at the same rate as the engine. This cam causes the pilot operated expander exhaust valve to move up at one dead center and down at the other. The stem of the spool valve is sealed by a metal bellows, the interior of which is connected to the low pressure reservoir to avoid loss of working gas. A spring causes the rod to follow the cam. The use of a spring at the expander valve actuator is avoided, as valves in the head of the expander are at high temperature. Only a small amount of gas, per revolution flows through the pilot valve; the dimensions hence can be very small. Since the actuator cylinder is very hot, it is desirable to line it with a ceramic coating, finished to a true cylindrical form at operating temperature, as described in copending application Ser. No. 15,462 titled "Piston Sealing", filed Oct. 7, 1970, by Vannevar Bush.

In an alternative procedure and apparatus, two small reservoirs are kept filled with air at appropriate high and low pressures by an auxiliary pump, and the actuator cylinders are operated by air pressure. The actuator cylinder is separated mechanically from the expander valve, except for the valve rod. This allows the actuator cylinder to be kept relatively cool. It can even be lubricated if this is desirable. Bellows can be omitted from the spool type valve.

While the expander exhaust valve is oriented in one direction, namely so that expander pressure tends to hold it closed, the intake valve is oriented in the other direction, so that heater pressure tends to hold it closed.

The inlet valve to the expander opens at dead center and closes at the cutoff point. The construction may be similar to that already described for the exhaust valve. One further point is considered. When the inlet valve opens at dead center, if there is a large difference of pressure differential across it, the actuator cylinder needs to be large to ensure the outward opening of the inlet valve.

This difficulty is avoided in one of several ways. One is as follows. When the closure of the expander exhaust valve is timed to occur just before dead center, the gas remaining in the cylinder will be compressed during the remainder of the stroke, thus reducing the pressure difference tending to hold the inlet valve closed. It may be arranged so that the cylinder pressure will exceed that in the heater, and this excess pressure will itself open the valve.

A second method of ensuring positive action of the intake valve is as follows. A projection of the valve is actually hit by the expander piston as it nears the end of its stroke, thus ensuring very positive opening. The speed of the piston at that time, near dead center, is relatively low, and the impact thus is moderate.

In another valve opening apparatus, a cylindrical projection on the valve enters a closely fitting recess in the piston of the expander as this piston approaches the end of its stroke. The confined, and hence compressed, gas provides a force to open the valve. To avoid a force in the other direction as the piston recedes, a small check valve is added in a passage connecting with the piston recess.

When the engine is started, in cold condition after a period of inactivity, the pressures in heater and cooler are nearly equal, but if the auxiliary small reservoir has been maintained at proper pressures, there is no problem of proper valve action on starting. As the engine is turned over, on starting, by a starting motor, operating pressures are gradually established. The furnace is simultaneously brought up to operating temperature.

When an engine runs constantly at full power, such as might be the case in operating a pump or a generator having a constant load, valve timing may remain uniform throughout the engine operation. Power output of the engine is preferably varied by changing timing of one or more valves. The coordinating of timing of a compressor valve and all expander valves is preferred so that the active volumes of the compressor and the expanders remain in constant ratio. For example, for half load operation the compressor exhaust valve remains a check valve, permitting flow toward the heater when compressor pressure exceeds heater pressure, but the compressor intake valve is held open for only one half of the filling stroke of the compressor or is held open for the first half of the compression stroke. Thus, only half a compressor volume is compressed. At the same time, the expander intake valve is cut off at one half of the full load cut off point so that only one half as much gas is expanded, and the expander exhaust valve is open for only the last half of the exhaust stroke. Thus, the gas is expanded to a low cooler pressure and then compressed to cooler pressure before the expander exhaust valve opens to release the gas to the cooler or heat exchanger.

A second method of operation of the expander exhaust valve at part load is to open the exhaust valve at the point in the expansion where the pressure is equal to the cooler pressure. The valve remains open for the remainder of the expansion stroke and for all of the exhaust stroke. With this method of operation, the pressure difference across the valve is always in the direction to hold the valve closed. For this reason, holding the valve open for the additional time is preferred in both the expander and the compressor.

There are three principal ways in which power output may be varied by changing timing of valves between elements. The effective expander volume may be reduced, the effective compressor volume may be reduced, or both effective volumes may be reduced.

Assume that, at full load, the temperature at the outlet of the heater is T_1 and the pressure P_1. Similarly, the cooler has temperature T_2 and pressure P_2. The volume of the expander up to the point of cut off is V_1, and the active volume of the compressor is V_2. In the steady state, the mass of gas moved from the heater, per cycle, must be equal to the mass of gas moved from the cooler to the heater.

This requires that

$$P_1 V_1/T_1 = P_2 V_2/I_2$$

Pressures alter until this condition is met.

For example, if a temperature ratio $T_1/T_2 = 3$, in terms of absolute temperature, is to be used, and if it is wished to employ a pressure ratio $P_1 / P_2 = 2$ at full load, $V_2 / V_1 = 2/3$; that is the active volume of the compressor should be two-thirds the volume of the expander up to the point of cut off at full load. The actual volume ratio is somewhat larger because of flow losses in the valves, heat transfer to the cylinder walls, clearance volume in the compressor and recompression in the expander clearance volume.

At full load, the expander pressure should equal the cooler pressure at the end of the power stroke. After cut off, the gas in the expander expands by a volume ratio V_1 / V, where V is total expander cylinder volume. Since the expansion is nearly adiabatic, this involves a pressure ratio $(V_1/V)^k$ where k, for helium, is 5/3. Since pressure decreases to one-half of maximum pressure, this requires that $V_1/V = 0.648$.

To decrease power output, the point of cutoff is advanced, the expander inlet valve is caused to close at an earlier point in the cycle. The new value of expander volume at cutoff is designated V_1^1. Pressures alter until

$$P_1^1/P_2^1 = \tfrac{2}{3} (T_1/T_2) (V_1/V_1^1)$$

or, on the assumed temperature ratio,

$$P_1^1/P_2^1 = 2\, V_1/V_1^1$$

Thus, as cutoff is advanced, the pressure ratio rises. When volume at cut off is halved, the pressure ratio is doubled.

The mass of gas in the system is assumed constant. When the volumes of heater and cooler are assumed equal, and each is assumed large compared to cylinder volume,

$$(P_1^1/T_1)+(P_2^1/T_2) = \text{constant}$$

$$(P_1^1/T_1)+(P_2^1/T_2) = (P_1/T_1)+(P_2/T_2)$$

These expressions can be combined to show that, when cut off occurs earlier, the heater pressure rises. In fact, in our example, if expander volume at cut off is halved, the heater pressure will rise to 10/7 of its full load value. This is undesirable, for one wishes to use the maximum allowable heater pressure under conditions of full load.

If the heater volume is made larger than the cooler volume, the situation is somewhat improved, but the pressure will still rise as cutoff is advanced.

As cut off is advanced, pressure ratio increases, so that

$$P_2^1 = (V_1^1/2V_1)\, P_1^1$$

but the expansion is then in the volumetric ratio V_1^1/V, and the pressure in the cylinder at the end of the power stroke is $(V_1^1)^k P_1$.

To ensure against back flow from the cooler to the expander, the exhaust valve of the expander may be a check valve, which is held closed for a proper interval by pressure in an actuator cylinder. Pressure should be applied to hold the exhaust valve closed during an interval from just before the inlet valve opens until the end of the power stroke. A fixed cam operating a pilot valve may be used for this purpose. The exhaust valve will then open only when pressures become equalized.

Altering the operation of the valves of the compressor, the inlet check valves of the compressor are replaced by pilot controlled valves. The active volume of the compressor is reduced to decrease output.

The active volume may be reduced in one of two ways. The inlet valve may close late, that is, after dead center, in which case part of the gas drawn from the

cooler will be forced back into the cooler before the remainder is compressed and is forced into the interchanger. The inlet valve may close early, in which case, the gas in the compressor is first expanded and then compressed before being transferred to the interchanger.

From the expression above

$$(P_1/P_2) = (T_1/T_2)(V_2/V_1)$$

it is evident that the pressure ratio will be decreased when the effective volume of the compressor V_2 is decreased. With fixed expander cutoff, the pressure ratio in the expander will remain constant. Hence, at the end of the expander stroke, the pressure in the expander will be below that in the cooler. To avoid a rush of gas when the expander exhaust valve is opened, it is necessary to delay that opening beyond dead center until the gas in the expander has been compressed to the pressure existing in the cooler. Thus, the timing of two valves needs to be altered to use efficiently the changes of the effective compressor volume for control.

Both of these methods (change of only V_1 or change of only V_2) require changes in pressure ratio for changes in power. Since a significant amount of gas must be compressed or expanded to change the pressure ratio, there is a delay before the power adjusts to a signal to change power. In the preferred method of control, the timing of the expander inlet valve controls V_1 (cut off volume), the timing of the expander exhaust valve controls V (the maximum effective volume of the expander), and the timing of the compressor inlet valve timing controls V_2 (the effective compressor volume).

When the timing of these three valves is adjusted so that V_1, V and V_2 always remain in the same ratios, the power of the engine is changed without any change in P_1 or P_2. This eliminates all delays in changing the power of the engine. With this method of control, the thermodynamic cycle of the gas remains fixed, and power level is set by only changing the mass of gas circulated per stroke. Thus, the thermodynamic efficiency of the engine can be maintained over the entire power range.

Every unit mass of gas flowing about the circuit extracts from the heater an amount of heat proportional to $(T_1 - T_a)$, and deposits in the cooler an amount proportional to $(T_b - T_2)$. With no interchanger, the temperature after compression is T_a, the temperature into the heater, and the temperature after expansion is T_b, the temperature into the cooler. With an interchanger, the temperature after expansion is $T_a + \Delta T$ and the temperature after compression is $T_b - \Delta T$, where ΔT is the heat exchange temperature difference in the interchanger.

The efficiency is thus:

$$\epsilon = 1 - [(T_b - T_2)/(T_1 - T_a)]$$

This may be transformed to

$$\epsilon = 1 - (T_b/T_1) = 1 - (T_2/T_a)$$

Since the pressure ratio remains constant, the temperatures of gas flowing into the cooler and into the heater, and without interchanger, after expansion and compression, T_b and T_a, also remain constant. The efficiency is hence constant during change of load.

The power output, at a given speed, is proportional to

$$V_1(T_1 - T_a)[1 - (T_2/T_a)]$$

and hence is directly proportional to V_1.

All of the above applies when T_1 and T_2 are at full operating values, and when the engine has operated long enough to produce full operating pressures. There is then no appreciable delay when V_1 is altered to change output, for the mass of gas in heater and cooler does not alter. We assume in above that heater and cooler volumes are large compared to cylinder volumes.

The expressions above for output and efficiency hold whenever the pressures remain constant. They hold for large heater and cooler volumes, provided the ratios T_1/T_2 and P_1/P_2, and V_1/V_2 are appropriately chosen. There is an approximation here, for actually the heater pressure changes during expander intake, and the cooler pressure changes during compressor intake. This effect apparently results merely in small change of P_1/P_2 under operating conditions.

But, when the above relations are widely departed from, under starting conditions there is a net transfer of gas between heater and cooler, each cycle, and P_1 and P_2 gradually alter. If pressures at starting are equal, the heater is pumped up, and the work needed for this needs to be considered in examining starting torque.

Under starting conditions, three cases are considered.

The first case involves starting when both temperatures and pressures are at nearly full value. The engine is now, of course, ready to develop full power at once. For a traffic stop, one takes his foot off the accelerator and the engine idles with early cutoff. But one wishes to stop the engine, briefly or for a long period. For a short stop, one would leave the furnace operating under its thermostatic control. For a brief stop, one could stop the engine by leaving the car in gear and applying the brake. This would not be desirable, for the engine might start again on brake release, if valves happen to be in correct position. It is better to have a positive stop controlled by the car key. This would operate a shutoff valve in the line from the heater, or it would move cutoff back to nearly zero position. In either case, on turning the key, the engine would automatically start and idle, or a small action by the starting motor would cause it to do so.

In the second case, the heater temperature is fully established, but cooler and heater pressures are equal. This condition arises when the furnace is left ignited and the engine is stopped for a considerable period, while gas leaks from heater to the cooler past valves and pistons. The effect is reduced if a shut off valve above is used, but valves can be expected to leak somewhat. Automobile operators probably will not wish to shut off their furnace when they make a relatively short stop.

In one example, the ratio of heater volume to total effective expander volume is about 40 in a multicylinder engine. If the accelerator is fully depressed and the cutoff hence is at full load position, the initial negative starting torque is about half of full load torque. That calls for a powerful starting motor, or one well geared down. The pressures change rapidly to develop positive power, and the net torque arrives at zero after only about 10 cycles.

In another example, the accelerator is at idling position, and cutoff is at one fifth of its full load value. The initial negative torque is one fifth of the previous value, but the engine has to turn over for **50** cycles before positive net torque appears. That is not a serious matter. The conditions of this case will not occur often. And if the starting motor turns the engine over at 500 rpm, one would need to wait only a fraction of a minute before the engine would idle when the accelerator was depressed.

The third case is the usual one, starting with a cold furnace and with pressures uniform throughout. The furnace and starting motor are turned on at the same time. The time constant for the furnace heating is about 30 seconds. The engine is turned over at 60 rpm; there is hence one cycle per second.

At first, pumping is in the revere direction, that is to decrease P_1 rather than to increase it. Net torque becomes positive after 27 seconds. The burden on the starting motor is not large. At the end of one minute half of full load torque is available, assuming the accelerator to be fully depressed throughout and assuming the slow starting turnover to continue. Of course, as with present engines, when positive torque appears, the engine speeds up, and the accelerator is released. Thus, nearly full load torque is available after about 40 seconds.

It is of advantage to delay operation of the starting motor until the furnace has heated up a bit. When the motor is started when T_1/T_2 has arrived at 1:5, **1.5**, is very little negative net torque, and positive torque begins after 20 seconds. The sequence readily is made automatic on turning the car key, or a light indicates when the key should be turned from a furnace ignition position to a second motor starting position.

ALTERNATE POWER ADJUSTMENT

There are a number of ways in which the power output of this engine may be controlled with no adjustment of valve timing on either expander or compressor. The engine operates at fixed cutoff, and mass balance is secured by choosing a proper diameter of compressor compared to expander cylinders. Power output is then adjusted by altering the temperatures or the total mass of gas in the working volume. These can be accomplished in several ways.

The fuel supply to the heater may be controlled to vary the maximum temperature in the heater. This is an undesirable method, first because there is a long time lag in response when a change of output is called for, and second because efficiency is reduced when heater temperature is lowered.

Another power-changing method is to alter the total mass of gas in the system, thus altering the pressure during the cycle. This may involve a time lag, but may be employed when such a lag is not of great importance, as in a marine engine.

In one form of mass adjustment as described in U.S. Pat. No. 3,527,049 (FIG. 4), a reservoir is provided filled with working gas. The pressure of this gas is intermediate between maximum and minimum pressures in the working volume. This reservoir is connected to the system, preferably to the cooler, by two pipes. Each pipe includes a check valve and a second valve, preferably, a solenoid operated shutoff. One check valve allows flow only into the reservoir, and the second allows flow only out of the reservoir. When the first solenoid valve is operated, the mass of working gas is decreased, and when the second is operated, it is increased. The power output is thus under control over a wide range. The action of these valves may readily be compounded, by well understood means, so that a given position of a lever or pedal will correspond to a definite working gas mass and hence power output.

In another method for varying system mass, there is supplied a high pressure reservoir, and a low pressure reservoir, containing working gas, which is preferably helium. Pipes from each of the reservoirs, with a valve in each pipe, connect to the system, preferably to the cooler. Opening one valve for an interval increases system mass and output; opening the other decreases system mass and output. An auxiliary pump may be supplied to pump gas from the low to the high pressure reservoir, in order to maintain a desired pressure difference.

Another method, as follows, may be used. A pipe is connected from the heater to a high pressure reservoir, with a check valve in the pipe to allow flow toward the reservoir. A second pipe, with check valve, connects the cooler and a low pressure reservoir, allowing flow from this reservoir to the cooler. These two pipes may be passed through an interchanger to allow heat interchange. Under steady conditions of operation, one reservoir will have a pressure which is approximately equal to maximum system pressure, and the other to minimum system pressure. Change of output can then be obtained as before. There is, however, a limit to the amount of output control which can be obtained in this manner. Consider an example. Let us assume that the system is set to operate with a pressure ratio of 2:1, at full power the HP reservoir will be at maximum pressure, and the LP reservoir at half of that. Assume we now wish to decrease output. Even if the reservoirs are very large, we can decrease maximum operating pressure in the system only to half its previous value. The consequent change in output may not be sufficient, especially if one wishes to go from full power to idling power. Of course, one may combine the two methods of maintaining reservoir pressures.

A preferable method of altering system mass and output is as follows. The control of engine power by changing the mass of gas does not involve a time lag when the new gas is added to the system from a high pressure reservoir into the high pressure part of the system, preferably between the compressor outlet and the interchanger. This method requires a high pressure reservoir which is always at a pressure above the maximum pressure for the engine. An auxiliary compressor with the required high pressure ratio adds to engine weight and consumes power.

OBJECTS OF THE INVENTION

One object of the invention is the provision of hot gas energy conversion methods and apparatus having a heater, expander and compressor connected in a circuit with at least one variable valve controlling flow of working gas around the circuit and thereby varying energy conversion.

Another object of the invention is to provide hot gas engines having gas flowing about a circuit from a compressor to a heater to an expander to a cooler and back

to the compressor, with valves between the elements controlling the flow, and with at least one of the valves being variable.

Another object of the invention is the provision of a hot gas engine having a circuit flow with variable valves controlling the flow of gas around the circuit.

This invention has an another object provision of a hot gas engine having circuitous gas flow with valves operating in adjustable timed relationship with the compressor and with the expander for adjusting power output of the engine.

The invention has as other objects the provision of pilot valves and valve actuators and check valve disabling actuators for controlling hot gas engine valves.

Another object of the invention is the provision of working gas mass varying systems which remove gas from a circuit between a cooler and compressor and which add gas to a circuit between a compressor and heater for reducing or increasing power.

These and other objects of the invention are apparent in the disclosure which includes the foregoing and ongoing specification with the appended claims and the drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a schematic view of a hot gas engine constructed according to the present invention showing a general innerrelationship of the parts.

FIG. 2 is a schematic detail of a hot gas engine showing valve operations.

FIG. 3 is a schematic detail of a cam mechanism for varying valve dwell and timing to control power output of the engine.

FIG. 4 is a schematic detail showing the engine circuit and valve control for an embodiment of the engine of this invention.

FIG. 5 is a schematic detail of a method of controlling power by varying the mass of gas moving through the circuit.

FIG. 6 is a detail of an alternative form of mass-varying power control.

DETAILED DESCRIPTION OF THE DRAWINGS

A hot gas engine is generally referred to by the numeral 1. Four cylinders are mounted on a crankbase 4. Pistons within the cylinders are connected to crossheads within the base 4. Four sets of crosshead guides are mounted internally within crankbase 4 beneath head 6 to constrain crosshead and piston movement to pure reciprocation, eliminating side loading of pistons in the cylinders. Connecting rods connect the crossheads to a crankshaft in the crankbase.

The engine is made up of a compressor, heater, expander, cooler, interchanger which may be removed from the circuit, and valves to control flow around the circuit. Each expander or compressor cylinder has two valves, four valves when double ended. Variable control is provided for the expander and compressor intake valves to operate the engine at a selected power output between full and idling power. The compressor exhaust valves are check valves, and the expander exhaust valves are check valves which are held closed for part of each cycle.

Expander cylinders 12, 14 and 16 and compressor cylinder 18 are thermally insulated from the environment with jackets which conduct heat over external surfaces of the cylinders for providing uniform temperatures throughout. The expander cylinders 12, 14 and 16 and compressor cylinder 18 are spaced from the crankbase head 6 by thermal isolating tubes 20. Preferably, all of the cylinders are double-acting with duplicate sets of valves at opposite ends of the cylinders. For clarity in the present description of the invention, valve systems and working gas supplies are shown only for the connections at upper ends of the cylinders.

Pistons reciprocate and the cylinders operate in a relationship which is selected for kinematic engine balance and design convenience. Timing relationship between the compressor and expanders is not critical. It is not necessary that the compressor be driven at the same speed as the expanders, and it is not necessary that all of the expanders be driven at the same speed. Varying speeds of expanders and compressors may be employed, and the elements may have varied bores and strokes. An important consideration is the volumetric compressor and expander relationship which is described herein.

In a cycle of operation, compressed gas is exhausted from the compressor through valve 22, which is a check valve in a preferred form of the embodiment. Gas passes through conduit 24 into a thin walled channel within heat interchanger 26, and the gas continues through conduit 28 into heater 30.

Heated gas under the high pressure supplied by the compressor leaves the heater through conduit 32 and flows into header 34, from which the gas is valved into the expanders.

When the piston within expander cylinder 12 approaches top dead center, valve 36 is opened, and gas passes into the cylinder 12 as the piston begins its descent. At a selected cut-off point, valve 36 is closed, and the gas entrapped in the upper end of cylinder 12 is expanded, as the piston continues its descent.

Near the maximum volume of the upper end of cylinder 12, valve 38 is opened. As the piston within cylinder 1 ascends, the expanded gas within the upper end of the cylinder flows through valve 38 into exhaust header 40 and exhaust conduit 42, into a second thin walled channel within heat interchanger 26. There the expanded gas gives up heat to the incoming compressed gas before the expanded gas continues its flow through conduit 44 into cooler 46.

While the piston in the upper end of compressor 18 is descending, compressor intake valve 48 is opened, filling the compressor and completing the cycle. During the cycle, the expander intake valves for expanders 14 and 16 are opened at appropriate times in relationship to pistons moving within the cylinders, and the expander exhaust valves are opened in similar timed relationship, for filling the expanders with heated high pressure gas from header 34 and for exhausting expanded gas into header 40.

Expander intake and exhaust valves 36 and 38 and corresponding intake and exhaust valves on expanders 14 and 16 and the intake valve 48 for compressor 18 are operated in timed relation to the cyclic volumetric changes of the cylinders. In one embodiment of the invention, the crankshaft 50, which is driven by the expanders and which drives the compressor, is provided with two gear toothed pulleys 52. Timing belts 54 and

56 drive gear toothed pulleys 58 and 59, which are keyed to cam shafts 60 and 62. The cam shafts in turn drive pilot valves of servo valve assemblies which operate the expander and compressor valves.

Pilot valves 64, 66 and 68 are driven by separate cams on cam shaft 60. A high pressure reservoir 70 supplies a valve-operating gas, preferably helium, the same as the engine working gas, through lines 72 to the pilot valves. Lines 74 connect central portions of the pilot valves to a low pressure reservoir which is located, for example within the crankbase. An auxiliary compressor, driven by one of the cam shafts or the crankshaft, pumps gas from the low pressure reservoir to the high pressure reservoir 70. As the spool piston within valve 64 is driven up and down, lines 76 and 78 are alternately pressurized and exhausted by communication with lines 72 and 74. Thus, the alternating high and low pressures in lines 76 and 78 drive the valve operating piston within cylinder 79 up and down to open and close valve 36 in timed relation with cam shaft 60 and crankshaft 50.

Interchanger bypass 80 may be used between the expander and the cooler by opening valves 81.

Expander exhaust valve 38 is operated by a piston within cylinder 82. The valve operating cylinder 82 is supplied with pressure by pilot valve 84, which has a spool piston driven by cam shaft 62.

All expander intake valves are operated by pilot pistons driven from cam shaft 60. All expander exhaust valves and compressor intake valve 48 are driven by cams on cam shaft 62. Two cam shafts are preferred because power reduction requires different adjustments for the compressor intake and expander exhaust than for the expander intakes. For example, half power adjustment requires approximately quarter revolution adjustment for the expander exhaust valves and compressor intake valve and requires less than a quarter revolution adjustment for the expander intake valve. As shown in FIG. 1, the compressor intake valve 48 is operated by a piston within cylinder 86, which is driven by alternations of pressures in lines 88 as controlled by a pilot valve operated by cam shaft 62. In FIG. 1, 89 generally indicates a high pressure reservoir for supplying the pilot valves driven by cam shaft 62.

FIG. 2 is a schematic representation of the elements shown in FIG. 1. Gas flows from compressor 18 through check valve 22 and interchanger 26 to the heater 30 and thence to header 34 which communicates with valve 36. Expanded gas leaving expander 12 exits through exhaust valve 38 and interchanger 26 on its way to cooler 46 and back to compressor 18 through valve 48, completing the circuit. As shown in the drawings, valve 22 is a check valve permitting flow outward from compressor 18 into conduit 24. The remaining valves are piston operated valves. The operation of valve 36 is described in detail. The operation of valves 38 and 48 are similar to that of 36.

Gears 52 and 58 and an intermediate timing belt drive cam shaft 60 at the same speed as crankshaft 50. An adjustable cam 90 on the cam shaft drives a cam follower 92 on a piston rod 94. A spool piston 96 is mounted on the piston rod within valve 64. Spring 98 ensures positive contact of follower 92 with cam 90. A cam 100 on cam shaft 60 drives a piston pump 102 which takes gas from low pressure reservoir 104 and supplies gas to the high pressure reservoir 70 via check valves 106 and 108.

As shown in FIG. 2, cam 90 has raised follower 92 and spool piston 96 to the upper position. Upper high pressure line 72 is communicated through line 76 with the upper end of valve operating cylinder 80. Pressure is thus applied to the upper surface of piston 110, holding poppet valve 112 seated so that valve 36 is closed.

As cam 90 turns and spring 98 forces piston 96 to its lower position, line 76 is communicated with line 74 for returning gas from the upper end of cylinder 80 to the low pressure reservoir. At the same time, the lower high pressure line 72' is communicated with line 78 for lifting piston 110 and opening poppet valve 112.

Poppet valve 112 opens upward against full system pressure of the compressed and heated working gas. At the time that the valve is opened, piston 114 is approaching top dead center. To provide additional force to open valve 112 against the system pressure, a downward projection 116 is formed on the base of the valve. Projection 116 fits with close tolerance in recess 118, building pressure within the recess to force valve 112 upward. Check valve 120 and fine tube 122 prevent recess from drawing projection 116 downward as piston 114 descends with valve 112 open. A similar problem does not exist in the opening of the expander exhaust valve or the compressor intake valve.

The expander intake valve 36 remains open for the initial part of the expander cycle when volume begins to increase. At a cut off point, valve 36 is closed, and the gas within the expander is expanded. As the piston reaches its lowest point, valve 38 is opened and remains open until the piston 114 approaches the top dead center. Compressor intake valve is open for the entire intake stroke at full power. As shown in FIG. 2, the bores of compressors and expanders may differ. Relative piston position in FIG. 2 is of no significance.

Those valve openings apply for full power operation. When the engine is operated at less than full power, active volumes of the expanders and compressors are reduced, preferably maintaining the volumetric ratio.

A means to control the time of the valve opening is shown in FIG. 3. Cam shaft 60 is driven by a belt 54 and gear 58. The gear has a bevel gear arranged on its rear face for driving a differential unit. Bevel gears 130 and 132 turn freely on slanted axles 134 and 136, which are mounted in slide 138. The large bevel gear on the rear of gear 58 turns a small bevel gear 130, which in turn spins small bevel gear 132, thereby turning bevel gear 140 in the same direction as gear 58. Gear 140 drives hollow shaft 142, which has fixed at intervals therealong cams 144. Each cam has a lifting portion 146 and an adjacent arcuate slot 148. Cam 150, which is attached to the cam shaft 60, extends through slot 148. As long as slide 138 is held fixed, shaft 142 turns with the shaft 60, and cam elements 146 and 150 remain relatively fixed. When slide 138 is moved in the direction of arrow 152, gear 140 and shaft 142 and cam 146 are caused to rotate in the direction of arrow 154, increasing the raised circumferential cam surface and reducing the time over which the pilot valve is depressed for opening the expander intake valve 36.

FIG. 4 is a simplified schematic representation of the engine of the present invention in which compressor 18 has an intake valve 160, which is operated by a sole-

noid 162. Gas flows from compressor 18 through check valve 22, interchanger 26 and heater 30, and through intake valve 164 which is opened by solenoid 166. After gas is expanded in expander 12, solenoid 168 opens exhaust valve 169, flowing gas through interchanger 26 and cooler 46, completing the cycle. Solenoids 162, 166 and 168 are controlled electrically by contacts or magnetic switches operated by cam shafts similar to the gas pilot valve operation.

Rather than controlling power by valve timing, or in addition to controlling power by controlling valve timing, power may be controlled by increasing or decreasing the gas supply in the circuit. As shown in FIG. 5, one preferred method of increasing system power is by maintaining gas at a high pressure in reservoir 170. When increased power is desired, valve 172 is opened and gas flows from high pressure reservoir 170 through check valve 174 into conduit 24 between the compressor 18 and the interchanger 26. System pressure may be reduced by opening valve 176, allowing gas to flow from the circuit conduit between cooler 46 and compressor 18. Gas removed from the circuit flows into low pressure reservoir 180, where it may be pumped by auxiliary compressor 182 into the high pressure reservoir 170.

An alternate form of changing circuit pressure and controlling power is shown in FIG. 6. When a power reduction is required, valve 184 is opened, allowing gas to flow from heater 30 through check valve 186 and interchanger 188 to reservoir 190. When increased power is required, valve 192 is opened, allowing gas to flow from reservoir 194 through interchanger 188 and check valve 196 to cooler 46. An auxiliary compressor, not shown, transfers gas from low pressure reservoir 190 to high pressure reservoir 194. Although power output can be controlled by increasing or decreasing mass within the system, the preferred method of power control is by varying valve timing.

While the invention has been particularly shown and described with reference to a preferred embodiment thereof, it will be understood by those skilled in the art that various changes in form and details may be made therein without departing from the spirit and scope of the invention.

What is claimed is:

1. A hot gas engine comprising a compressor, a heater, an expander, and a cooler in closed circuit connection, valves in the circuit to control flow of working gas in the circuit, and variable control means connected to at least one of the valves for varying operation of the valves and thereby varying output of the engine.

2. The hot gas engine of claim 1 wherein the expander is a cyclically variable volume chamber, and wherein the control means is connected to the chamber for selectively controlling timing of the valves in varied relation to cyclical variations of the chamber volume.

3. The hot gas engine of claim 1 wherein the variable control means is connected to an expander exhaust valve between the expander and the cooler.

4. The hot gas engine of claim 3 wherein the variable control means is connected to an expander intake valve between the heater and the expander.

5. The hot gas engine of claim 1 wherein the variable control means is connected to an expander intake valve between the heater and the expander.

6. The hot gas engine of claim 1 wherein the variable control means is connected to a compressor intake valve between the cooler and the compressor.

7. The hot gas engine of claim 6 wherein the variable control means is connected to an expander exhaust valve between the expander and the cooler.

8. The hot gas engine of claim 6 wherein the variable control means is connected to an expander intake valve between the heater and the expander.

9. The hot gas engine of claim 8 wherein the variable control means is connected to expander intake and exhaust valves.

10. The hot gas engine of claim 1 further comprising a shaft connected to the expander and wherein at least one of the valves comprises a pilot valve, and wherein the variable control means comprises a variable pilot valve operator connected to the shaft.

11. The hot gas engine of claim 1 further comprising an interchanger in the circuit between the expander and cooler and the compressor and heater for removing heat from gas flowing between the expander and the cooler and transferring heat to gas flowing between the compressor and the heater.

12. The hot gas engine of claim 11 further comprising a valve controlled interchanger bypass in the circuit between the expander and the cooler.

13. In hot gas energy conversion apparatus having sequentially connected heater means, expander means, and compressor means for flowing gas between the means and repeatedly heating, expanding and compressing gas and thereby converting energy, the improvement comprising variable valve means connected between at least two sequentially connected means for varying gas flow between the means and for varying energy conversion.

14. The apparatus of claim 13 wherein the variable valve means comprises variable expander intake valve means connected between the heater means and the expander means for selectively varying amount of gas which is expanded in the expander means.

15. The apparatus of claim 14 wherein the variable valve means further comprises variable expander exhaust valve means connected between the expander means and the compressor means for varying the period of gas flow to the cooler from the expander means.

16. The apparatus of claim 15 further comprising check valve means connected between the compressor means and the heater means for permitting free flow of gas toward the heater means.

17. The apparatus of claim 15 wherein the expander means comprises cylinder and piston means, and further comprising shaft means connected to the piston means for receiving energy therefrom, and first and second variable valve operating means connected to the shaft means and respectively connected to the expander intake and exhaust valve means for varying timing of the valve means in relation to movement of the piston means.

18. The apparatus of claim 13 further comprising cooler means sequentially connected between the expander means and the compressor means for cooling gas flowing between the expander means and the compressor means.

19. An engine valve operator comprising a piston connected to an engine valve, a cylinder forming a

chamber with the piston, a gas line connected to the chamber, a pilot valve connected to the gas line, an operator connected to the pilot valve, and connected to an engine shaft, a pressure reservoir connected to the pilot valve for supplying pressurized gas through the pilot valve to the chamber for driving the piston and changing state of an engine valve in a timing sequence in relation to the engine shaft as controlled by the operator.

20. The apparatus of claim **19** wherein the operator has a selectively variable size for selectively varying dwell of the engine valve.

21. The method of operating a hot gas engine comprising compressing, heating, expanding and cooling working gas and variably controlling flow of the working gas in a closed circuit.

22. The method of claim **21** wherein the controlling comprises varying flow of working gas to an expander and thereby varying the expanding step.

23. The method of claim **22** wherein the controlling step comprises varying flow of working gas from an expander and thereby varying the expanding step.

24. The method of claim **21** wherein the controlling step comprises varying flow of working gas from an expander and thereby varying the expanding step.

25. The method of claim **21** wherein the controlling step comprises varying inward flow to a compressor, thereby varying the compressing step.

26. The method of claim **25** wherein the controlling step comprises varying flow of working gas to an expander and from an expander, thereby varying the expanding step.

27. The method of claim **21** wherein the controlling step comprises opening an expander intake valve and admitting gas during a portion of expansion of a cyclically variable expansion chamber, closing the intake valve at varied points in expansion of the chamber and expanding the gas during completion of the expansion of the chamber, and operating the expander intake valve in timed relationship with the cyclic variation of the chamber.

28. The method of claim **21** wherein the expanding step comprises cyclically varying an expander volume, and wherein the controlling step comprises opening an expander exhaust valve at a volumetrically cyclic point between before reaching maximum expander volume and before reaching minimum expander volume and holding the expander exhaust valve open until about minimum expander volume.

29. The method of claim **21** wherein the compressing step comprises cyclically varying a compressor volume, and wherein the controlling step comprises opening an intake valve in a compressor at about minimum compressor volume, holding the compressor intake valve open, and selectively closing the compressor intake valve between a cyclical point before reaching maximum compressor volume and a cyclical point after passing maximum compressor volume.

30. The method of claim **29** wherein the expanding step comprises cyclically varying an expander volume, and wherein the controlling step further comprises opening an expander intake valve at about minimum expander volume, holding the expander intake valve open, selectively closing the expander intake valve at a variable cyclic point between minimum and maximum expander volume, selectively opening an expander exhaust valve at a variable cyclic point between the closing of the expander intake valve and before reaching minimum expander volume holding the exhaust valve open and closing the exhaust valve at about minimum expander volume.

31. The method of claim **21** wherein the controlling step comprises changing open periods of the expander and compressor intake valves thereby changing power.

32. The method of claim **31** wherein the controlling step comprises maintaining a constant ratio of expander and compressor active volumes.

33. The method of claim **21** wherein the controlling step comprises changing open periods of the expander and compressor intake valves, permitting free exhaust from the compressor and permitting free exhaust from the expander during limited cyclic periods.

34. The method of hot gas energy conversion comprising flowing working gas through a compressor, heater and expander and cooler, removing working gas between the cooler and compressor to reduce power, storing working gas at high pressure, and adding working gas between the compressor and heater to increase power.

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