United States Patent

Bennethum

[15] **3,699,770**

[45] **Oct. 24, 1972**

[54] **STIRLING ENGINE CONTROL SYSTEM**

[72] Inventor: **James E. Bennethum,** Rochester, Mich.

[73] Assignee: **General Motors Corporation,** Detroit, Mich.

[22] Filed: **May 27, 1971**

[21] Appl. No.: **147,355**

[52] **U.S. Cl.** **60/24**

[51] **Int. Cl.** **F03g 7/06**

[58] **Field of Search** 60/24

[56] **References Cited**

UNITED STATES PATENTS

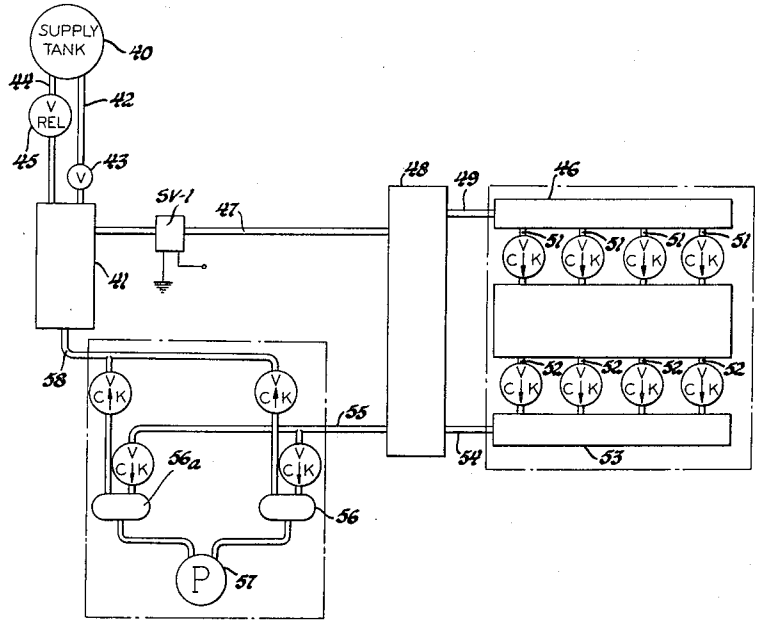
2,616,243	11/1952	Van Weenen	60/24
2,794,315	6/1957	Meijer	60/24
3,372,539	3/1968	Reinhoudt	60/24
3,554,672	1/1971	Brandes	60/24 X

Primary Examiner—Martin P. Schwadron
Assistant Examiner—A. M. Ostrager
Attorney—Jean L. Carpenter and Arthur N. Krein

[57] **ABSTRACT**

A working gas control system for a double acting hot gas engine particularly adapted for vehicle installation and including a manually adjustable system for variably controlling the pressure of the engine working gas to provide for load control of the engine. The system includes a gas reservoir for working gas to supply the engine system with gas at a suitable mean working pressure through an operator actuator control system. A gas supply tank is provided to maintain a predetermined minimum gas pressure in the reservoir.

3 Claims, 5 Drawing Figures



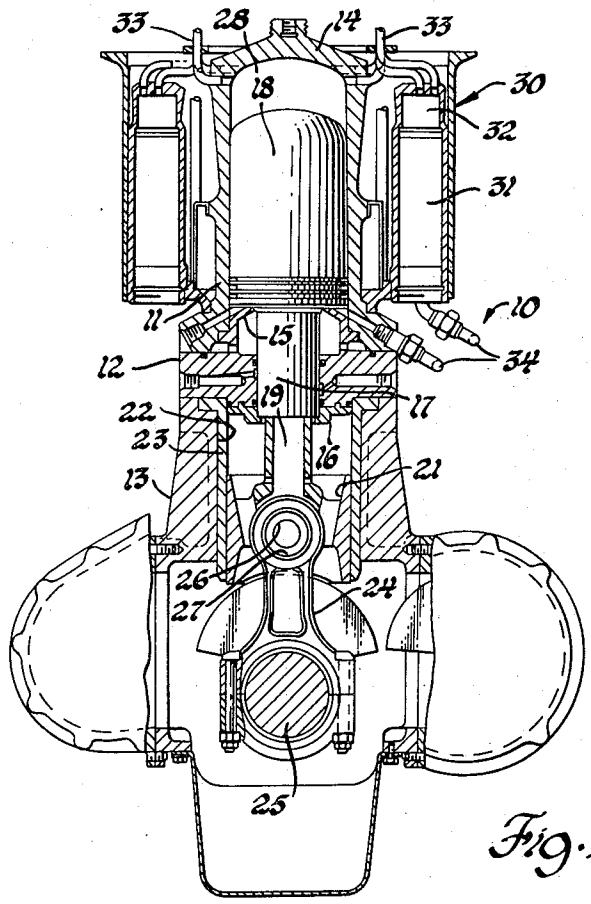


Fig. 1

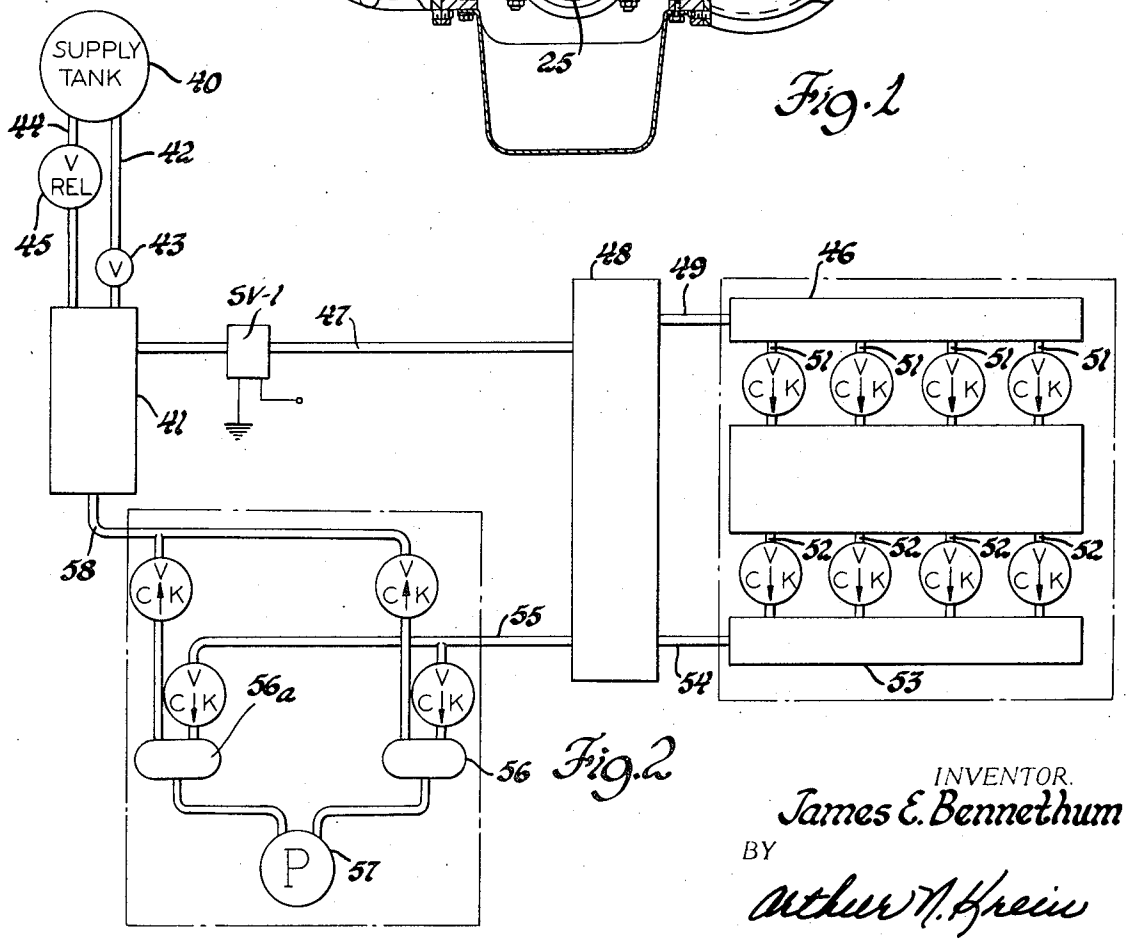


Fig. 2

INVENTOR.
James E. Bennethum
BY
Arthur N. Krein
ATTORNEY

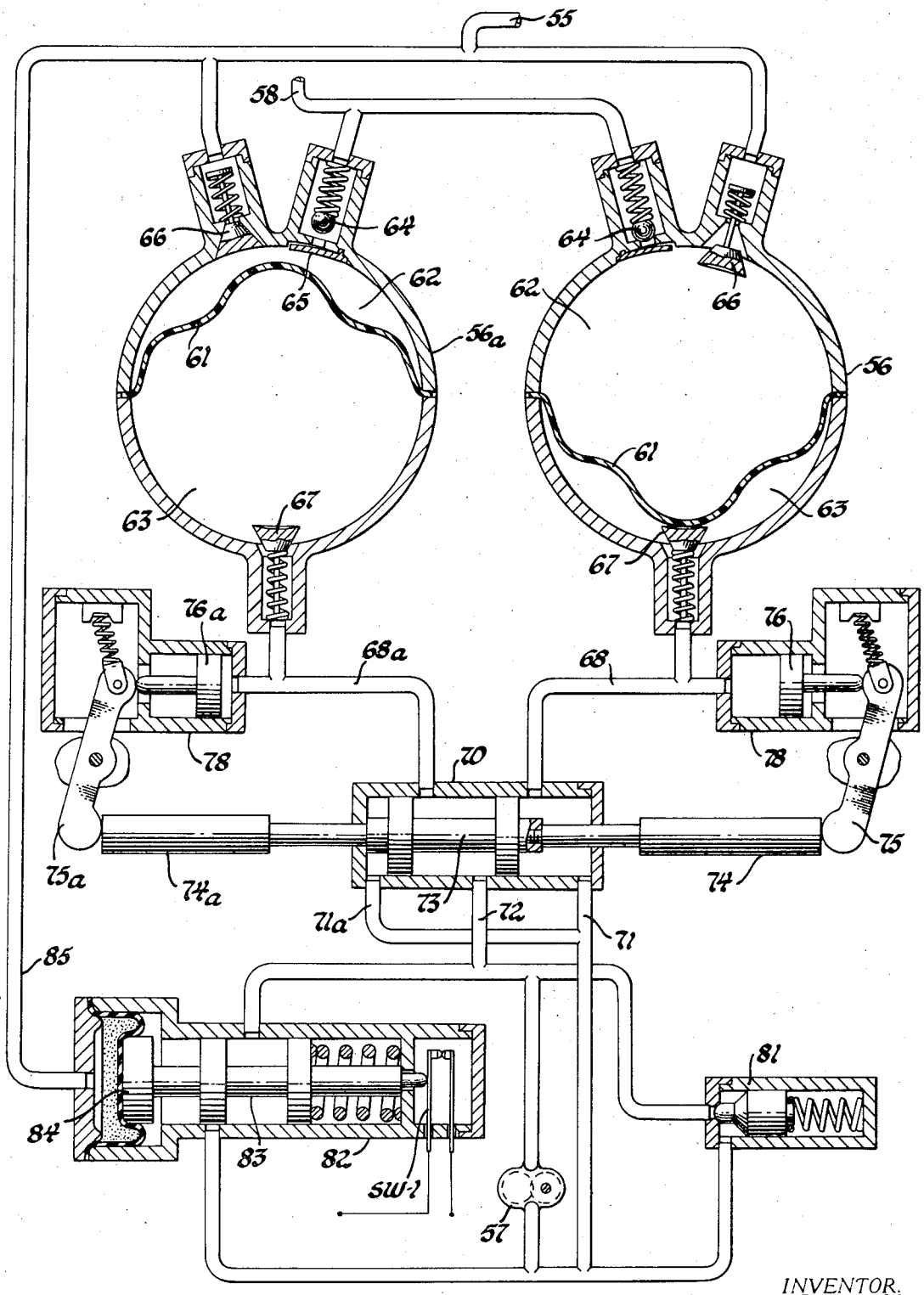


Fig. 3

INVENTOR.
James E. Bennethum
BY
Arthur N. Krein
ATTORNEY

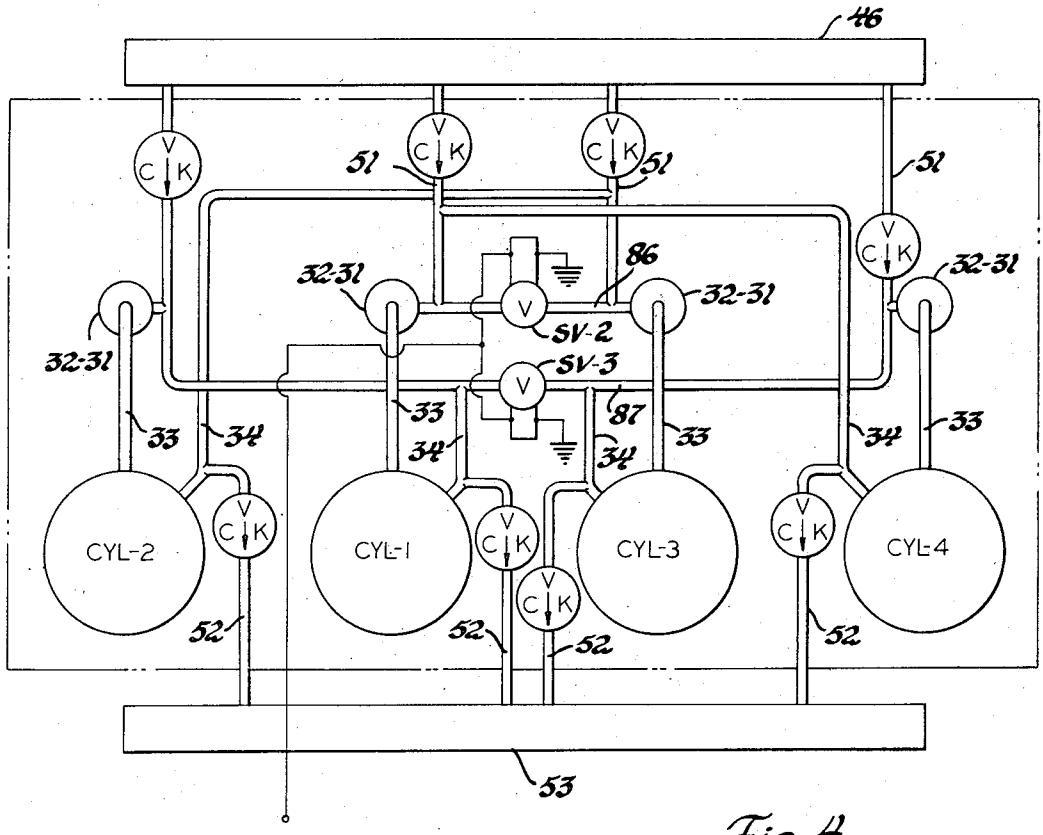


Fig. 4

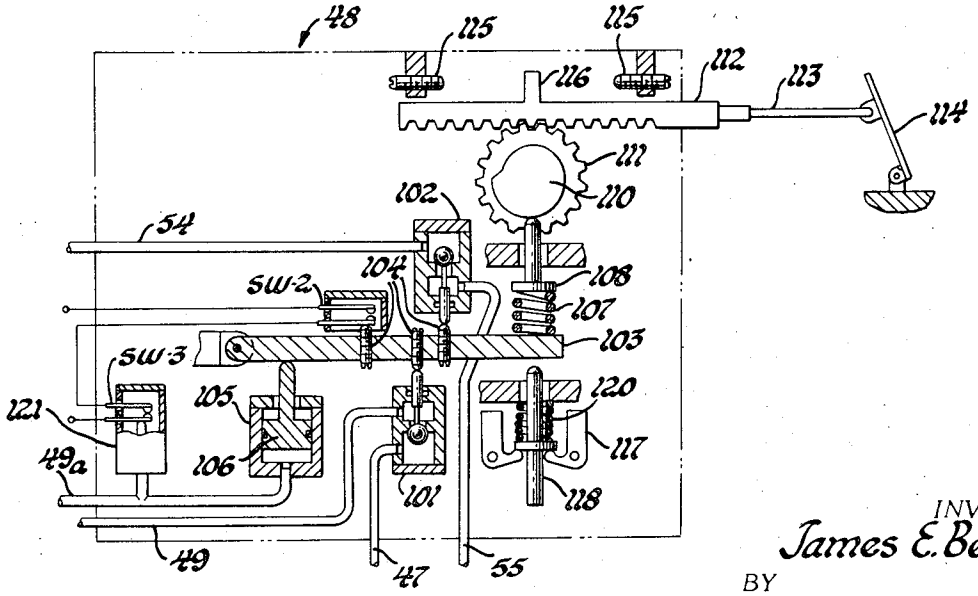


Fig. 5

INVENTOR.
James E. Bennethum
BY
Arthur N. Krein
ATTORNEY

STIRLING ENGINE CONTROL SYSTEM

This invention relates to hot gas engines and more particularly to a gas control system for a closed cycle hot gas engine such as a double acting Stirling cycle engine.

It is known in the art to provide means for controlling hot gas engines of the Stirling cycle type by varying the pressure of the working gas in the engine working spaces. Such pressure variations have been accomplished by supplying gas from external storage means to the engine working space and exhausting gas from the working spaces to the storage means through the operation of suitable transfer and control means including a gas compressor. Normal practice has been to use one small high pressure external gas reservoir to store the working gas and to serve as the source of gas to fill the engine during operation. However, it has been found that the use of such small high pressure reservoirs for the gas is disadvantageous because the high pressure of the gas maintained in the reservoir requires relatively high compressor input power to refill the reservoir with gas discharged from the hot gas engine and the use of a high pressure reservoir did not allow the system to make use of the hot gas engine's own pumping ability to directly refill the gas reservoir from the working spaces of the hot gas engine under certain engine operating conditions.

It is therefore the principal object of this invention to improve hot gas engine control systems whereby the power required for refilling the reservoir with gas exhausted from the engine is reduced.

Another object of this invention is to improve a hot gas engine control system whereby engine response times are small and independent of the engine duty cycle.

These and other objects of the invention are attained by means of an engine control system in which working gas is supplied to the engine by normal gas pressure flow from a supply reservoir through an engine pressure control system; a gas supply tank being provided to supply makeup gas to maintain a minimum gas pressure in the reservoir and to serve as a secondary reservoir to which gas can be bled if the pressure in the main reservoir exceeds a predetermined pressure at engine shutdown. Engine power is varied in a known manner by varying the gas pressure in the engine working space with pressure responsive regulating means being utilized to maintain the desired working space pressure by sensing the mean pressure in the working space and controlling the gas supply and exhaust in accordance with desired pressure requirements. The regulating means includes manually adjustable pressure selection means which are actuated by the vehicle operator such as through a conventional foot pedal to manually select the desired working gas pressure. The variation of this manual selection means thus varies the engine output torque and provides engine operation similar to that of the throttle controlled internal combustion engine.

For a better understanding of the invention, as well as other objects and further features thereof, reference is had to the following detailed description of the invention to be read in connection with the accompanying drawings, wherein:

FIG. 1 is a cross sectional view of a double acting, four-cylinder, in line, hot gas engine of the Stirling cycle type;

FIG. 2 is a schematic illustration of the gas supply and pressure control system for the engine of FIG. 1;

FIG. 3 is a diagrammatic representation of the fluid pumping means in the system of FIG. 2;

FIG. 4 is a schematic illustration of the inlet and exhaust system to the engine of FIG. 1; and,

FIG. 5 is a sectional view showing the pressure control system of the gas supply and pressure control system of FIG. 2.

Referring now to the drawings, a four-cylinder, double acting, in line, hot gas engine, generally designated by reference numeral **10**, includes cylinders **11** mounted on and partially closed at one end by an intermediate block **12** which in turn is supported on an engine crankcase **13**. The cylinders **11** terminate in closed ends **14** at one end thereof and are partially closed at the opposite end by the apertured reduced portion **15** of the intermediate block **12**. Intermediate block **12** with annular apertured end caps **16** secured thereto support part of a rolling seal structure **17** for each piston. Each rolling seal structure may be of the type disclosed in U.S. Pat. No. 3,568,436 issued Mar. 9, 1971 in the names of Francis E. Heffner and Richard R. Toepel.

A non-vented piston **18** is slidably received in each of the cylinders **11** and is suitably connected at one end to the piston rod portion **19** of a crosshead piston **21** slidably received in a crosshead cylinder portion **22** of the crankcase **13**, which may also be provided with crosshead piston liners **23**. Each crosshead piston **21**, in turn, is connected by a connecting rod **24** to a crankshaft **25** journaled in the crankcase **13**, the connecting rods **24** being connected at their upper ends to the crosshead pistons **21** by pins **26** retained in the crosshead pistons by retaining rings **27**.

The interiors of the cylinders **11** are separated by their respective pistons **18** into hot chambers **28** formed at the ends **14** of the cylinders, and cold chambers, not numbered, formed at the opposite ends thereof. Canister-like members **30** encircle each of the cylinders and support therein suitable coolers **31** and regenerators **32**. A heater, not shown, which may be a burner or other suitable heating device, is located above the canisters **30**. A plurality of tubes **33** extend from the ends **14** of the cylinders through the heaters to the regenerators **32** of each cylinder. Connecting tubes **34** connect the coolers **31** of one cylinder with the cold chamber end of the next appropriately phased cylinder.

Thus, four separate gas containing working spaces are formed, each made up of the hot and cold chambers of adjacent cylinders connected through heater tubes **33**, regenerators **32**, coolers **31** and connecting tubes **34** in a known manner as shown in U.S. Pat. No. 3,538,706 issued Nov. 10, 1970 to Richard R. Toepel.

Referring to FIG. 2, there is shown schematically a control system diagram for supplying the working gas to the engine **10**. A supply of working gas is contained in a supply tank **40** and a reservoir **41**, the supply tank **40**, which serves as a secondary reservoir being connected to the reservoir **41**, by a supply conduit **42** with a pressure regulator **43** therein and by a return conduit **44** with a pressure relief valve **45** therein. The reservoir **41** is connected to a manifold type fill header **46** for supplying working gas to the engine by means of a conduit **47** through an engine pressure control regulator,

generally designated **48**, and by conduit **49**. A solenoid shuttoff valve SV–1 is placed in the conduit **47** to shut off the flow of gas from the reservoir to the pressure control regulator **48** when the engine is not in operation and it can be controlled by a conventional switch, not shown.

The fill header **46** is connected by valve controlled conduits **51** to the engine and gases are dumped from the engine through valve controlled conduits **52** to a manifold type dump header **53** which in turn is connected by conduit **54** to the engine pressure control regulator **48** and then by conduit **55** through a plurality of compressor gas chambers or accumulators **56** and **56***a*, and conduit **58** to the reservoir **41**. Compressor gas chambers **56** and **56***a* are actuated by a compressor pump **57**, as described hereinafter.

Referring now to FIG. **3**, there is shown diagrammatically the fluid pumping system for the engine which includes the compressor gas chambers **56** and **56***a* and the compressor pump **57**. As shown, the device includes a pair of compressor gas chambers **56** and **56***a* in the form of accumulators, both of which incorporate movable wall means such as diaphragms **61** which divide these accumulators into first and second chambers **62** and **63**, respectively.

Chambers **62** are connected through spring biased check valves **64** and conduit **58** to the reservoir **41**. The valve openings for the check valves **64** include metal anti-extrusion flaps **65** which prevent damage to the diaphragms while permitting flow to the check valves **64**. The chambers **62** are also connected through spring biased poppet inlet valves **66** and conduit **55** with the engine pressure control regulator **48** and therethrough with the working space of the engine **10**. The valves **64** and **66** are arranged to permit flow from the engine working space to either chamber **62** and from either chamber **62** to the reservoir **41**, but to prevent flow in the opposite direction as well as to prevent flow between the two chambers **62**.

Chambers **63** of the compressor gas chambers **56** and **56***a* are connected through spring biased anti-extrusion devices **67** and conduits **68** and **68***a*, respectively, to spaced portions of the housing **70** of a spool type selector valve. Valve housing **70** also includes drain lines **71** and **71***a* located adjacent its ends and a hydraulic supply line **72** which connects with the output of the compressor pump **57** in the form of an oil pump, the drain lines **71** and **71***a* being connected to the inlet of pump **57**. The compressor pump **57** receives pressurized oil drained from lines **71** and **71***a* and oil bypassed through supply line **72** in a manner described in detail hereinafter.

Reciprocally disposed within valve housing **70** is a valve spool **73** which includes oppositely disposed end portions in the form of actuator rods **74** and **74***a*.

Each of the actuator rods **74** and **74***a* are selectively reciprocated by means of pivotally connected levers **75** and **75***a*, respectively, which in turn are pivoted clockwise and counterclockwise, respectively, by means of hydraulic actuated plungers **76** and **76***a* of pressure actuators **78** hydraulically connected to conduits **68** and **68***a*, respectively.

When the valve spool **73** is in the position shown in FIG. **3**, the valve acts to connect hydraulic supply line **72** with conduit **68***a*, while conduit **68** is connected with drain line **71**. In its second position, in which the spool valve **73** is moved to the right with reference to FIG. **3**, it connects hydraulic supply line **72** with conduit **68** while conduit **68***a* is then connected with the drain line **71***a*.

The hydraulic system also includes a pressure release valve **81** connected with the outlet of the compressor pump **57** and is arranged to bypass fluid therefrom to the compressor pump inlet at a predetermined pressure. A bypass valve **82** is also connected to both the inlet and outlet of the compressor pump **57** and includes a valve spool **83** which is spring biased in an opening direction wherein the pump inlet and outlet are connected. A diaphragm actuator **84**, at one end of spool **83**, is acted on by gas pressure in conduit **55** through connection therewith by a line **85** for a purpose to be subsequently described.

The operation of the above-described compressor system is as follows. Compressor pump **57** in the form of an oil pump is driven intermittently, by an electric drive motor, not shown, the operation of which is controlled by a switch SW–1 actuated by valve spool **83** of bypass valve **82**. When, however, the compressor pump is in operation and the valve spool **73** is in the position shown in FIG. **3**, oil is pumped through line **72** and conduit **68***a* to chamber **63** of the accumulator **56***a*. This forces diaphragm **61** upwardly reducing the size of chamber **62** therein and forcing any gas in this chamber out of accumulator **56***a* and past check valve **64** through conduit **55** to the reservoir **41**.

When the gas in chamber **62** is exhausted and diaphragm **61** bears against the sides of the accumulator **56***a*, the hydraulic pressure in line **68***a* increases to force the plunger **76***a* of pressure actuator **78** associated with this line to the left, as shown in FIG. **3**, to rotate the lever **75***a* counterclockwise to move the spool valve **73** to the right, as shown in this figure, thus connecting line **68***a* with drain line **71***a*. At the same time, line **72** is connected with conduit **68** so that the compressor pump **57** begins to fill chamber **63** of accumulator **56** forcing out of it any gas which may be present in its chamber **62** in the same manner as was done with accumulator **56***a*. When all gas is exhausted from accumulator **56**, the pressure in line **68** will increase, actuating the plunger **76** of pressure actuator **78** connected to this line to rotate the lever **75** associated therewith in a clockwise direction to move the spool valve **73** to the left, to the position as shown in FIG. **3**, after which the above-described cycle is again repeated.

While the above-described pumping action is taking place in one accumulator, the full volume of the remaining accumulator is always available to receive a charge of gas from the engine **10** through the dump header **53**, conduit **54**, engine control system **48** and conduit **55**.

Thus, when accumulator **56***a* is being discharged and the engine is cycled for full load to idle, the complete charge of gas is receivable through conduit **55** and check valve **66** into chamber **62** of accumulator **56**. Since the gas is received under pressure, it easily forces out of chamber **63** the oil contained therein, which passes through conduit **68** and drain line **71** to the inlet of pump **57**.

When the pumping cycle on accumulator **56***a* is then completed, the valve spool **73** is moved as previously described and the discharge of gas from accumulator **56** is begun. At this time, accumulator **56***a* is available to receive an additional charge of gas from the engine working space through conduit **55** and check valve **66** to chamber **62** of the accumulator **56***a*.

Whenever the gas pressure in line **55** drops to a predetermined low pressure, such that further pumping action of the compressor is not required, the reduced pressure on diaphragm actuator **84** permits the valve spool **83** to move to the left, as seen in FIG. **3**, opening the bypass valve **83** and bypassing hydraulic fluid discharged from the pump **57** back to the pump inlet. This stops further gas pumping action until increased inlet pressure in line **55** again closes the bypass valve **82**. In addition, when the spool valve **83** moves to the left, as described above, the contacts of switch SW–1 are opened thus deenergizing the electric motor used to drive the pump **57**.

However, as will be described hereinafter, when dumping of gas from the engine is initiated by an operator to reduce mean pressure in the working spaces of the engine **10**, between, for example, mean pressures of 1,500 to 1,100 pounds per square inch absolute, using a preferred working embodiment of such an engine, the engine **10** dumps through the dump header **53**, regulator **48** and accumulators **56** and **56***a* directly into the reservoir **41**. At this range of gas pressure, depending of course on the actual gas pressure in reservoir **41**, gas entering into chambers **62** of the accumulators **56** and **56***a* would immediately flow back out through the valves **64** to the reservoir **41** and no work output from the compressor pump would be required. Below 1,100 pounds per square inch dump pressure, or otherwise as necessary, the compressor pump **57** is used to pump the gas back into the reservoir **41** in the manner described.

Referring now to FIG. **4**, there is illustrated schematically in greater detail that portion of the gas flow system, outlined in broken lines in FIG. **2**, relating to the interconnection of conduits between the fill header **46**, engine **10** and dump header **53**. As shown, the fill header **46** is connected by a conduit **51** to each of the cylinders **11** (the four cylinders in this figure being further identified as cylinders CYL–1, CYL–2, CYL–3 and CYL–4), of the engine **10** through a cooler **31**, regenerator **32** and heater tubes **33** for each of the cylinders, with each cylinder also being connected with the next cylinder, in the appropriate phasing order, by conduits **34** to provide four working spaces as previously described. Conduits **34** are also in communication with the conduits **52** interconnecting the cylinders to the dump header **53**. Conduits **51** leading to out of phase cylinders CYL–1 and CYL–3 are adapted to be placed in communication with each other by solenoid actuated bypass valve SV–2 in connecting conduit **86**, while out of phase cylinders CYL–2 and CYL–4 are adapted to be placed in communication with each other by solenoid actuated bypass valve SV–3 in connecting conduit **87**.

In order to operate the engine at varying output torques and speeds ranging from idle to full rated load, the pressure control regulator **48**, as actuated by an operator, is used to vary the pressure of the working gas in the working spaces of the engine **10**. The pressure control regulator **48** is connected by the conduit **47** to reservoir **41** in which a supply of working gas, preferably hydrogen, is kept under pressure.

Pressure control regulator **48** controls the supply of gas to the cylinders of the engine **10** through conduit **49** and fill header **46** connected to the engine, as previously described.

Pressure control regulator **48** is also connected to the engine working spaces through conduit **54** and dump header **53** to control the exhaust of gas from the working spaces through these conduits. Exhausted gas is transmitted to gas compressors **56** and **56***a* which increase the pressure of gas if necessary before it is returned to reservoir **41**.

Pressure control regulator **48** may be made in any suitable form; however, a preferred arrangement for such a system is schematically shown in FIG. **5** of the drawings. As shown, the regulator **48** includes a fill valve **101** connecting through line **47** with reservoir **41** and through line **49** to the fill header **46** for the engine **10**. Oppositely disposed from the fill valve is a dump valve **102**. Dump valve **102** connects through conduit **54** with the dump header **53** to the engine working spaces and through line **55** with gas compressors **56** and **56***a*.

A pivotally mounted actuating lever **103** is arranged between these valves **101** and **102** so as to open the fill valve **101** upon a downward movement and open the dump valve **102** upon an upward movement of the lever. Adjusting means such as screws **104** are provided to properly adjust the valve positions. The adjustments are preferably such that all valves are closed with the lever in an intermediate position and the dump valve **102** opens upon a slight upward movement and the fill valve opens upon a slight downward movement of the lever from the position shown.

This system further includes a cylinder **105** carrying a reciprocable piston **106**, one end of which engages the underside of lever **103**. The lower face of piston **106** is acted upon by gas pressure supplied through a capillary line **49***a* connecting the cylinder **105** with the engine working spaces. The capillary line acts to restrict gas flow so that cyclic pressure variations in the engine working spaces are not transmitted to cylinder **105**, but changes in the mean working pressure due to operation of the gas pressure control system are followed by the pressure in the cylinder **105**.

The upper surface of lever **103** is engaged by a spring **107** which is partially compressed by a follower **108** controlled by a rotatably mounted cam **110**. Cam **110** is in turn actuated by a gear **111** engaged by a rack **112** which is mechanically connected through linkage **113** with a manually actuated foot pedal **114**. Adjustable stop means **115** are provided to be engaged by an abutment **116** on the rack to limit travel of the manual actuating means.

Beneath lever **103**, an overspeed governor mechanism is provided comprising rotating flyweights **117** which engage a reciprocable shaft **118** urging the shaft upwardly against a spring **120** to engage the lower surface of lever **103**. Flyweights **117** are rotated at a speed proportional to engine speed through mechanism, not shown.

When the engine is in operation, an increase in engine torque is obtained by moving pedal **114** in a coun-

terclockwise direction. This moves rack **112** leftwardly rotating cam **110** so as to increase the downward force of follower **108** on spring **107**. Assuming the mean working pressure acting on piston **106** is less than that called for, lever **103** is pivoted downward by the bias of the spring, opening valve **101** and permitting pressurized gas to flow from the reservoir **41** to the engine fill header **46** and then to the working spaces of the cylinders **11** of engine **10** to increase the working pressure therein. The increase in mean working pressure is transmitted through line **49***a* to cylinder **105** acting against piston **106** and urging lever **103** upwardly until, when the desired pressure is reached, lever **103** resumes a neutral position closing valve **101** and shutting off the flow of gas.

If a decrease in engine torque is desired, pedal **114** is moved clockwise and through cam **110** reducing the downward bias of spring **107** and permitting piston **106** to move lever **103** upwardly opening dump valve **102**. This permits the exhaust of gas from the working space through dump header **53** and lines **54** and **55** to the gas compressor **56** and **56***a* for return back into reservoir **41**.

In case of an engine overspeed, flyweights **117** urge shaft **118** into engagement with lever **103** moving it upwardly and acting to exhaust working gas from the engine through the opening of dump valve **102**.

The regulator **48** also includes a switch SW–2 actuated by lever **103** and a pressure actuated switch SW–3 in the electrical circuit for solenoid actuated bypass valves SV–2 and SV–3 used to effect rapid reduction of output torque from engine **10**, as shown in FIG. **4**. Switch SW–2 is a normally open switch which is closed, depending upon position of adjusting screw **104**, when lever **103** moves upward for opening dump valve **102**. Switch SW–3, a normally closed switch, is opened by a pressure actuator **121** connected to conduit **49***a* whenever the pressure in this conduit is above a predetermined pressure, for example, above a pressure of 1,100 psia in the embodiment of the system disclosed.

Although the engine control system of the invention can be designed for any size engine or gas pressure conditions, a brief description of the system as applied to the engine disclosed operable under predetermined pressure conditions is as follows:

Fill of gas to the hot gas engine is always from the reservoir **41** through the engine pressure control regulator **48** and aforesaid conduits to the engine **10** by normal pressure flow. When the engine is empty, the reservoir contains a gas, such as hydrogen, at a pressure of 2,000 psia. Sufficient hydrogen is in the reservoir **41** to allow filling the entire engine system with gas at 1,500 psia mean working pressure. If leakage should occur during engine operation, the pressure in the reservoir **41** could drop below 1,200 psia at maximum engine pressure conditions. However, the supply tank **40**, which serves as a secondary reservoir, is provided for makeup and contains sufficient gas therein to maintain a minimum gas pressure in the reservoir of 1,200 psia. The supply tank **40** also acts as a secondary reservoir for gas to bleed from the reservoir **41** if the pressure of the gas therein exceeds 2,000 psia at shutdown of the engine.

During normal engine operation, the operator actuates the foot pedal **114** to control the engine mean gas pressure through the pressure control regulator **48** as previously described. As more power is required, the pedal **114** is depressed further to further open the fill valve **101** resulting in higher mean working pressures in the engine. When the pedal **114** is lifted, a dump is initiated, in the manner previously described, to reduce the mean pressure of the gas in the engine through the discharge of gas therefrom.

As previously described, in the embodiment of the system disclosed, between mean pressures of 1,500 psia and 1,100 psia, the engine dumps gas directly into the reservoir **41**, with the filling or dumping time being limited only by the line sizes of the interconnecting conduits between the various components of the system. Below a pressure of 1,100 psia for the dump gas, the compressor pump **57** is required to pump the gas back into the reservoir, the compressor pump **57** being sized so as to effect rapid pumping of the gas back into the reservoir **41**, when necessary, up to a pressure of 2,000 psia.

While one of the accumulators **56** or **56***a* is connected to the discharge side of the compressor pump to effect compression of the gas contained in the compressor accumulator, the oil chamber of the other accumulator is connected to the inlet of the compressor pump to create a suction within the accumulator chamber to further induce gas flow to the gas chamber of the accumulator from the dump header **53**. The charge of gas thus drawn into this accumulator will then be compressed and forced back into the reservoir **41**. The accumulator chambers are then reversed and the process continues until the operator calls for a fill of gas to the engine or the engine idle conditions have been reached. As previously described, an idle speed governor with flyweights **117** therein is built into the engine pressure regulator to effect dumping at excessive engine speeds.

By making the reservoir **41** about equal in volume to the volume of the working spaces in the engine **10** and its associated piping, optimum operating results can be obtained. With the reservoir thus sized, it has been found that the reservoir need not be too large, as compared to previously known systems, and that the pressure of the gas in the system can be set so that the engine can be filled to the maximum gas pressure condition, so that, under normal conditions, the pressure of the gas remaining in the reservoir is about 200 psi above the minimum cycle gas pressure at full engine load conditions.

However, at full engine load conditions, the maximum working pressure of the gas is considerably higher than the pressure of the gas remaining in the reservoir **41** and allows the gases dumped from the working spaces of the engine to be, in effect, pumped by the engine directly to the reservoir **41** with no extra compression of the gas by work output by the compressor pump **57**. This direct dumping of gas from the engine working spaces can be done until the engine maximum cycle pressure gets to within approximately 200 psi of the pressure of the gas remaining in the reservoir **41**. Below this pressure differential, the gas must be pumped into the reservoir **41** by operation of the compressor pump **57**, but about one-third of the gas in the engine working spaces can be dumped back to the reservoir in this manner.

When compression is required, it is desirable to reduce the power required for operation of the compressor pump **57** to a minimum. This is accomplished by using a small low pressure reservoir **41** and by reducing the pressure differential against which the compressor pump **57** must operate. In the arrangement shown in FIG. **3**, the inlet or suction to the compressor pump **57** is from an accumulator which is being charged with dumped gas from the engine, which gas is already under some pressure. Thus, the oil pump must perform work only to compress the dumped gas to overcome a pressure differential which is that existing between the engine maximum pressure and the pressure in the reservoir **41**.

From the above description of the subject control system, it is apparent that there is provided a control system arrangement requiring the least engine power to operate. This advantage is obtained by means of the following:

The use of a reservoir of about equal volume with that of the engine to take full advantage of the engine's pumping ability.

The use of a separate supply tank which allows the system to be maintained at optimum pressure conditions for extended periods of operation and independent of leakage or system temperature changes.

The use of a pressurized compressor pump inlet which reduces the head requirement to a minimum.

The use of a non-vented piston which reduces the engine gas inventory and therefore the compressor pumping capacity requirement.

The use of the bypass torque control to reduce engine output torque while the compressor system is dumping gas and extend the time available for pumping thus reducing the horsepower requirement of the pump.

What is claimed is:

1. A power control system for a hot gas engine of the type having a plurality of closed variable volume working chambers each having a hot space and a cold space and adapted to contain a working gas, said engine including means for heating and for cooling such working gas and means to transfer such gas between said hot and cold chambers while passing it through a working cycle, said control system comprising: a gas storage reservoir means including a gas reservoir and a supply tank interconnected by a pair of conduits, one of which contains a pressure regulator valve to control the flow of gas from said supply tank to said reservoir and the other containing a one-way check valve connecting said reservoir to said supply tank, first gas transfer means connecting said reservoir with said engine working chambers for unidirectional gas flow from said reservoir to the individual chambers, second gas transfer means connecting said reservoir with said engine working chambers for unidirectional gas flow from the individual chambers to said reservoir, regulating means arranged to control the admission to and exhaust from said working chambers of working gas through said first and second gas transfer means respectively and a gas pump in said second gas transfer means between said regulating means and said reservoir and adapted to aid the return flow of gas to said reservoir when the return flow of gas is below a predetermined pressure relative to the pressure of gas in the reservoir while permitting direct return flow of gas to said reservoir when the return flow of gas is above a predetermined pressure relative to the pressure of gas in the reservoir.

2. A power control system according to claim **1** wherein said gas reservoir is sized in respect to the volumes of said engine working spaces and the components of said first and second gas transfer means such that said volumes can be supplied with gas from said reservoir to the maximum mean working pressure condition from only said reservoir, additional gas from said supply tank flowing to said reservoir at a predetermined pressure below the maximum mean working pressure through said pressure regulator valve.

3. A power control system according to claim **1**, further including bypass valve means interconnecting out of phase pairs of said closed variable volume working chambers to permit the bypassing of gas from said out of phase working chambers to effect rapid reduction of engine output torque thereby and pressure responsive means and manually actuated lever means in said regulating means to effect operation of said bypass valve means.

* * * * *