United States Patent

Knöös

[15] **3,698,182**

[45] **Oct. 17, 1972**

[54] METHOD AND DEVICE FOR HOT GAS ENGINE OR GAS REFRIGERATION MACHINE

[72] Inventor: **Stellan Knöös,** Fregattvagen 2, Taby, Sweden

[22] Filed: **Feb. 16, 1971**

[21] Appl. No.: **115,547**

[30] **Foreign Application Priority Data**

March 2, 1970 Sweden 12640/70

[52] **U.S. Cl.** **60/24,** 62/6

[51] **Int. Cl.** **F01k 7/06**

[58] **Field of Search** 60/24; 62/6

[56] **References Cited**

UNITED STATES PATENTS

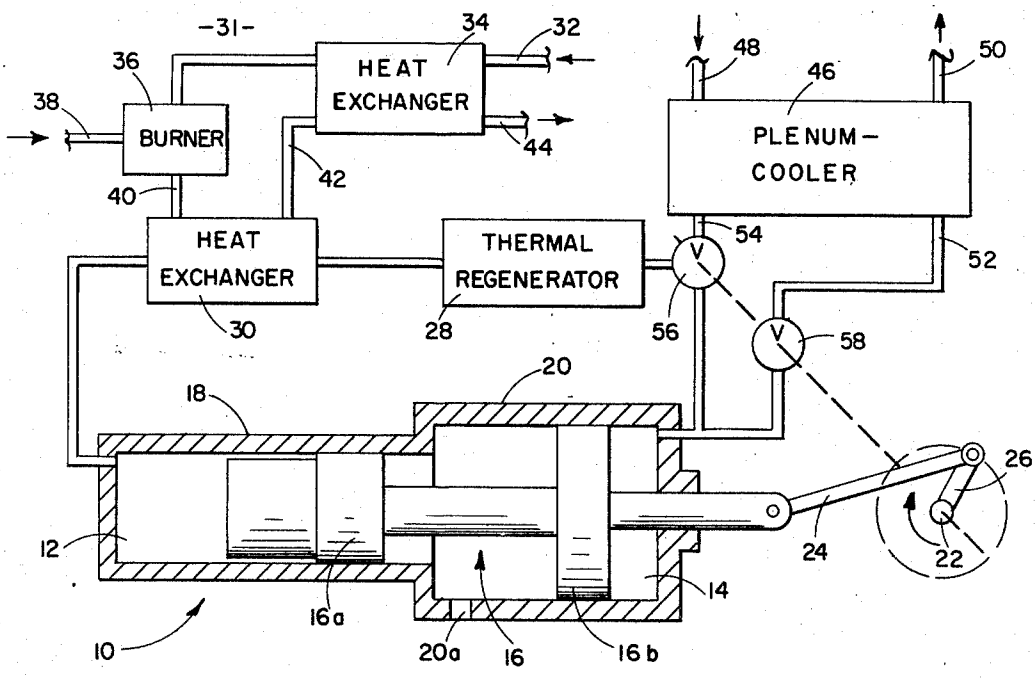
2,067,453	1/1937	Lee	60/24 X
2,657,528	11/1953	Jonkers et al	60/24
2,784,548	3/1957	Fiala	60/24
3,400,281	9/1968	Malik	60/24 X
3,460,344	8/1969	Johnson	60/24
3,552,120	1/1971	Beale	60/24

Primary Examiner—Martin P. Schwadron
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[57] **ABSTRACT**

A closed-cycle gas expansion engine and a closed-cycle or open-cycle gas expansion refrigerator, including a thermal regenerator, devices for transferring heat into the gas or subtracting cold gas, and a piston assembly for developing approximately 180° phase difference in the varying volumes of the warmer and colder gas chambers. A general valve mechanism is employed for properly timed flow of working gas into the primary gas chamber from a plenum chamber containing cooling devices for closed-cycle operation, or for flow of working gas into the primary gas chamber from an ambient atmosphere for open-cycle operation. A valve mechanism, separate or coupled to the first mechanism, is used for proper initiation and termination of gas flow between the primary and secondary chambers in one particular phase of the reciprocating piston motion. Still another valve, which may be integrated with any of the other mentioned valve mechanisms, is used for release of suitably recompressed working gas in the secondary chamber into the plenum chamber with a closed cycle, or for properly timed exhaust of such gas back to the ambient atmosphere with an open cycle.

35 Claims, 23 Drawing Figures



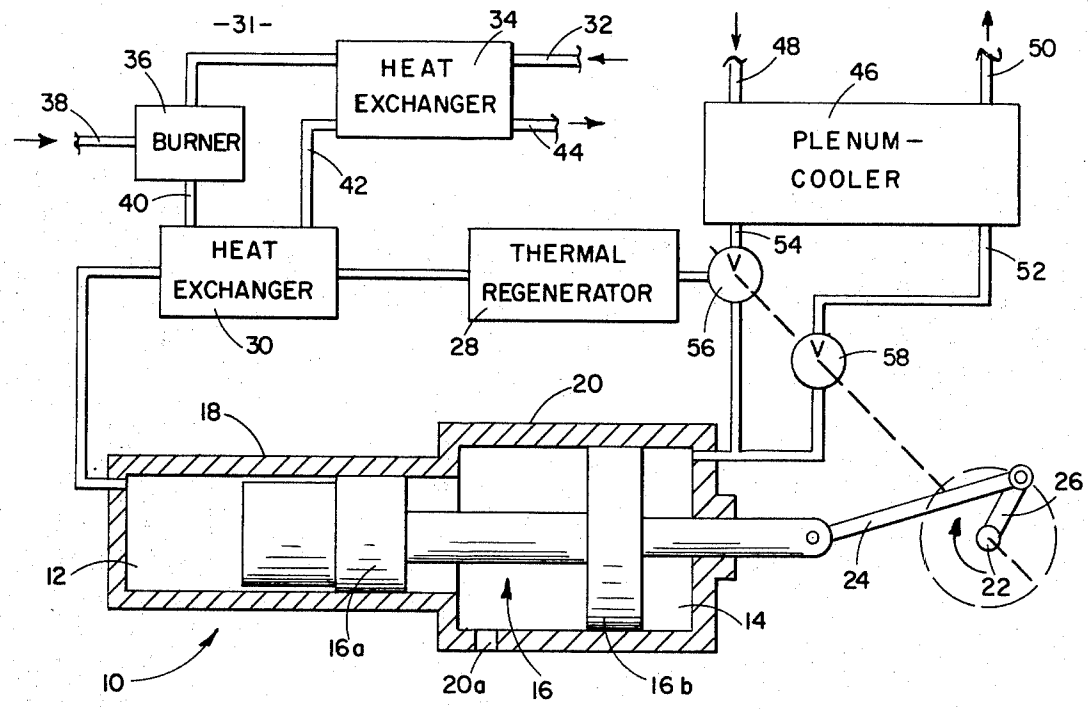


FIG. 1

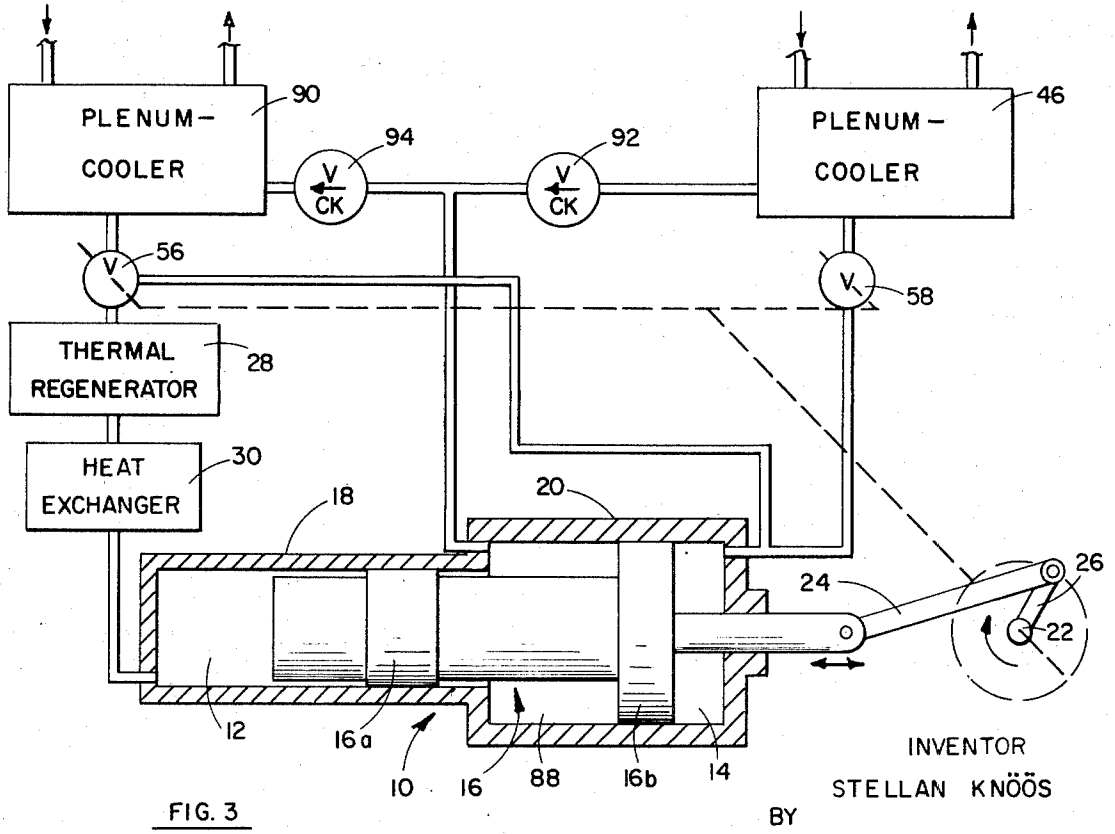


FIG. 3

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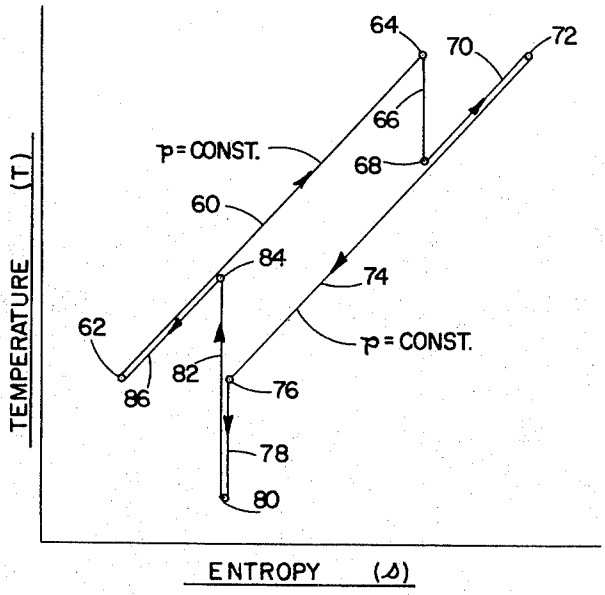


FIG. 2

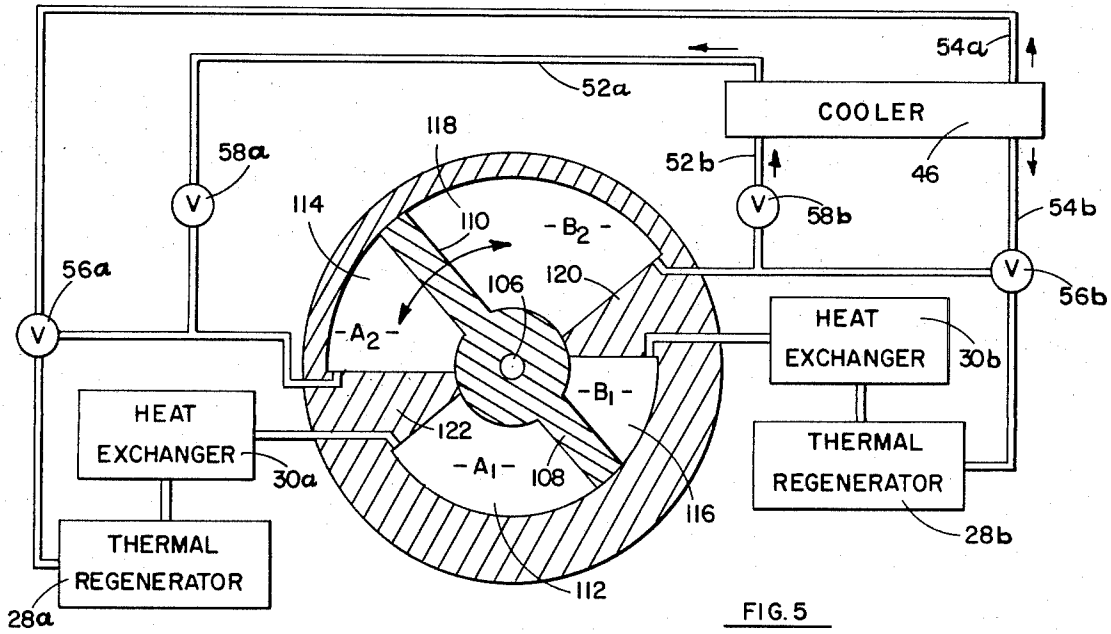


FIG. 5

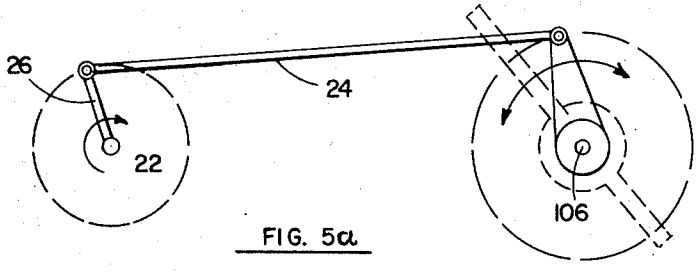


FIG. 5a

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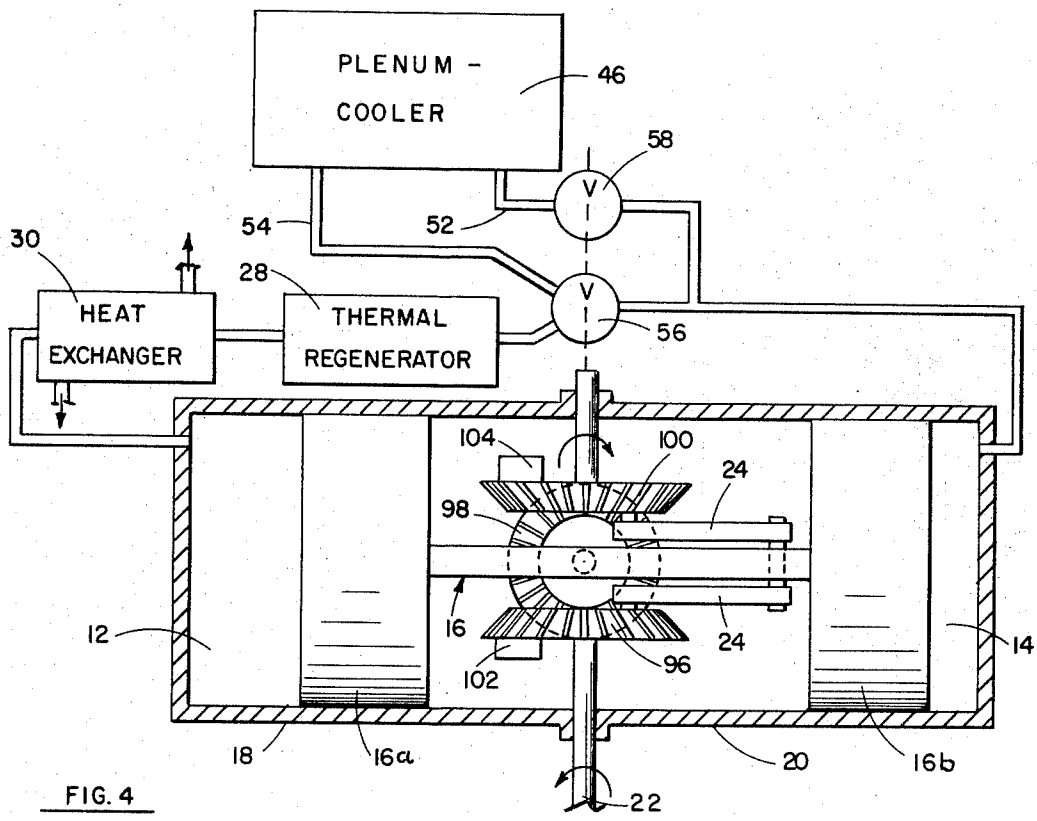


FIG. 4

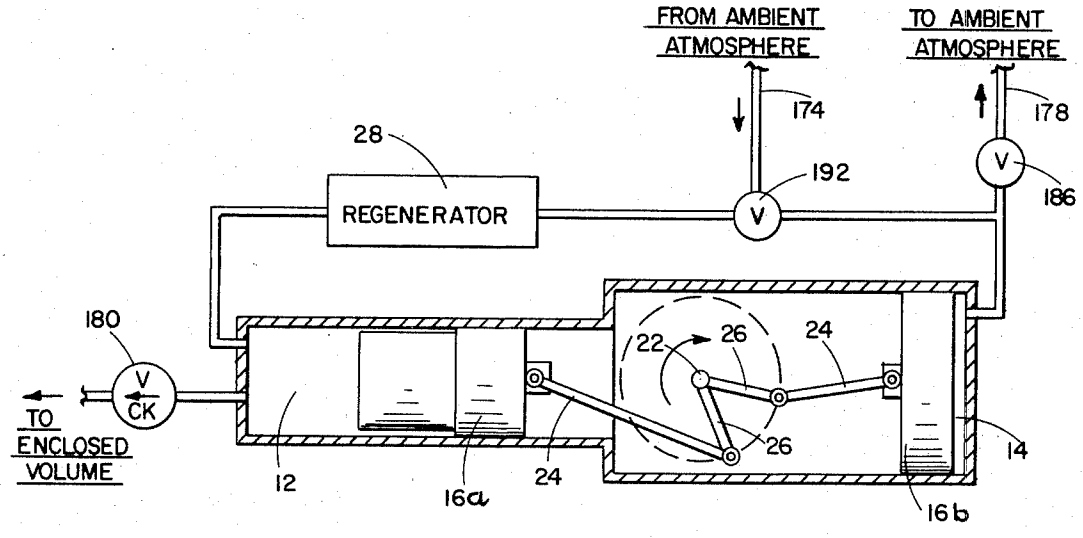


FIG. 11

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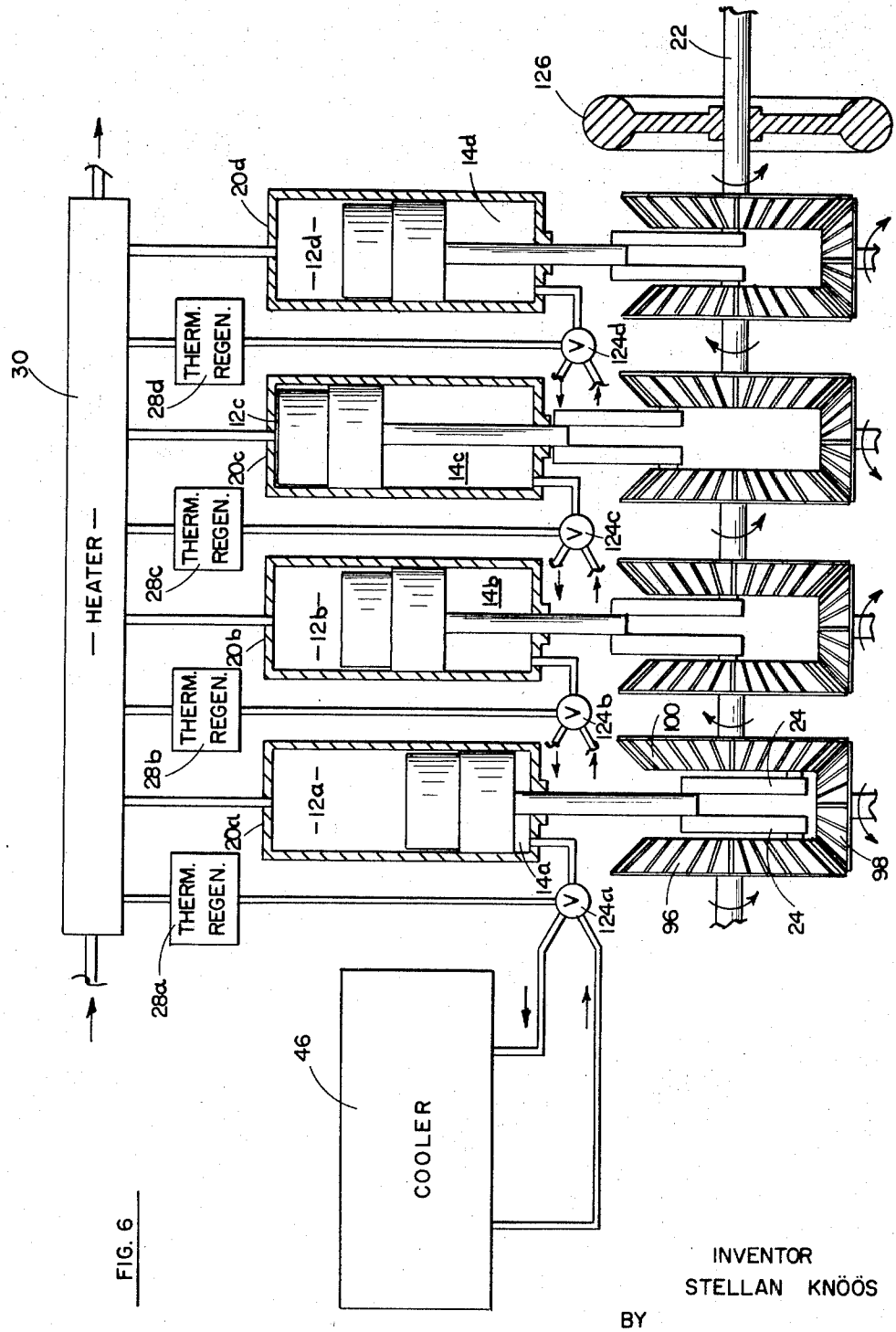


FIG. 6

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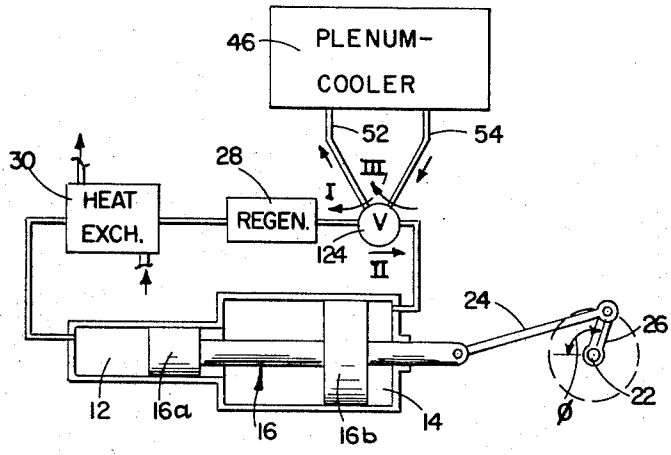


FIG. 7a

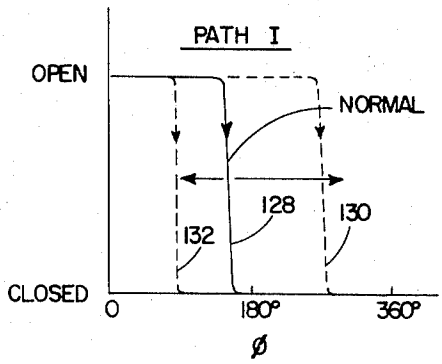


FIG. 7b

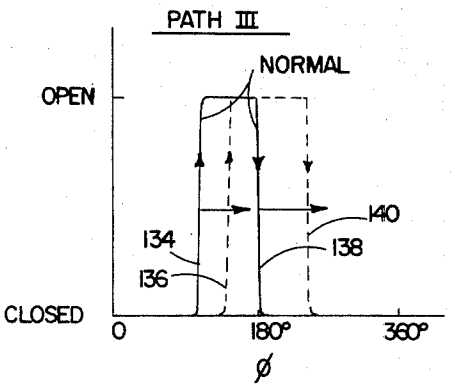


FIG. 7c

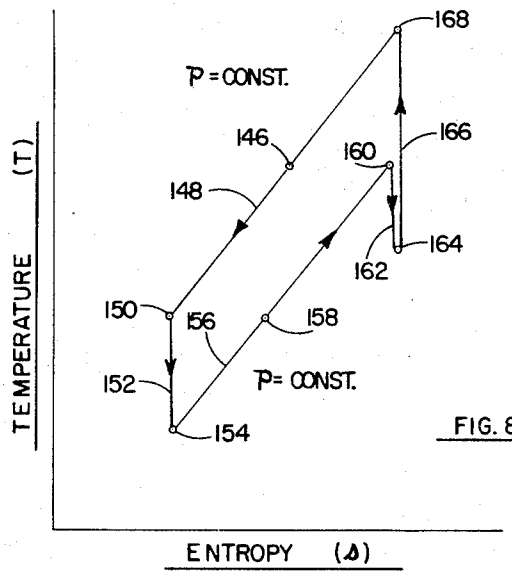


FIG. 8

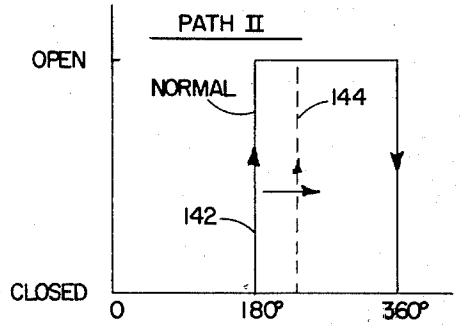


FIG. 7d

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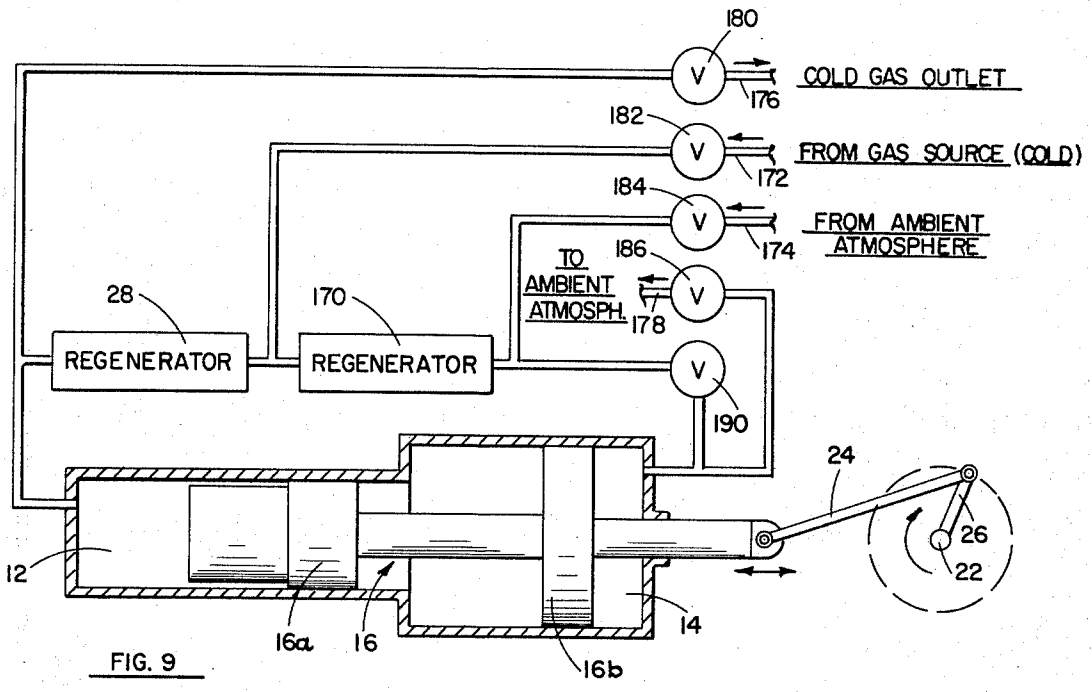


FIG. 9

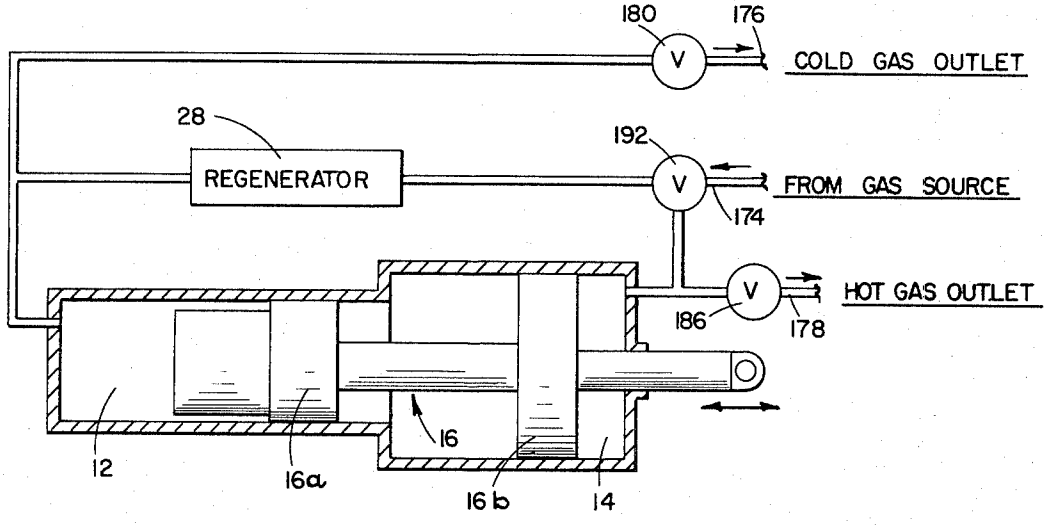


FIG. 10

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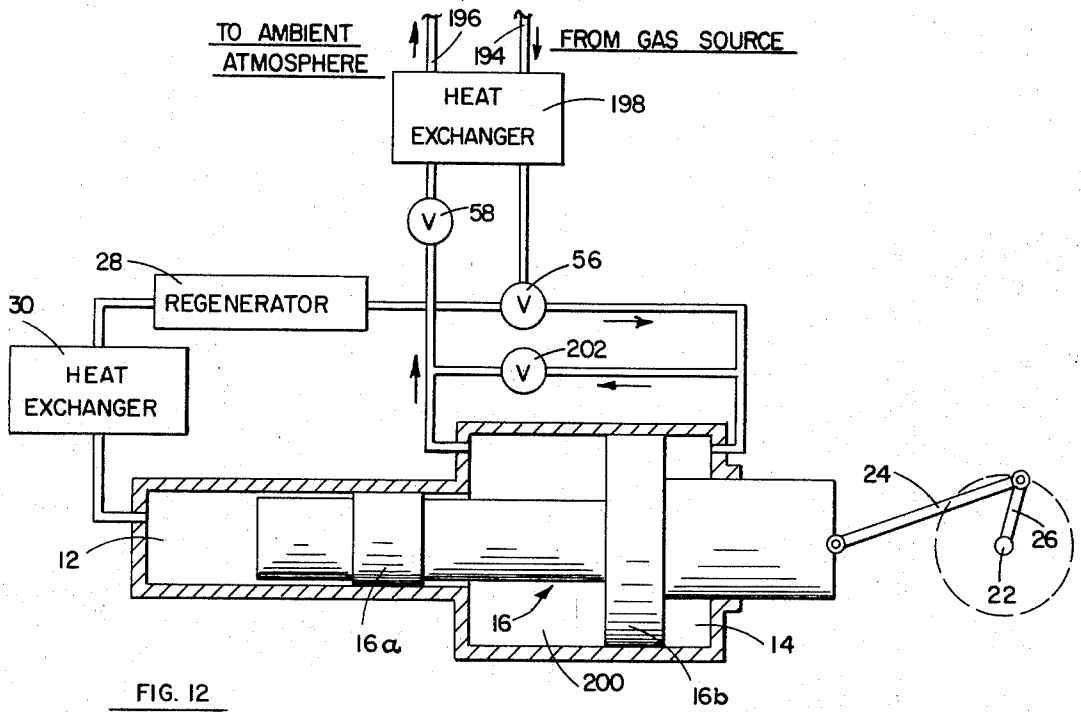


FIG. 12

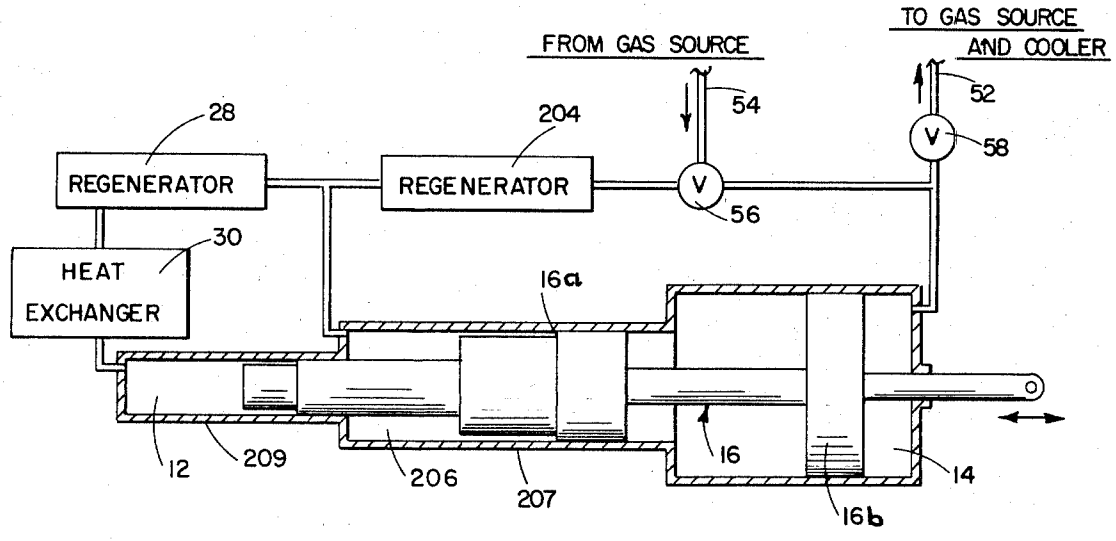


FIG. 13

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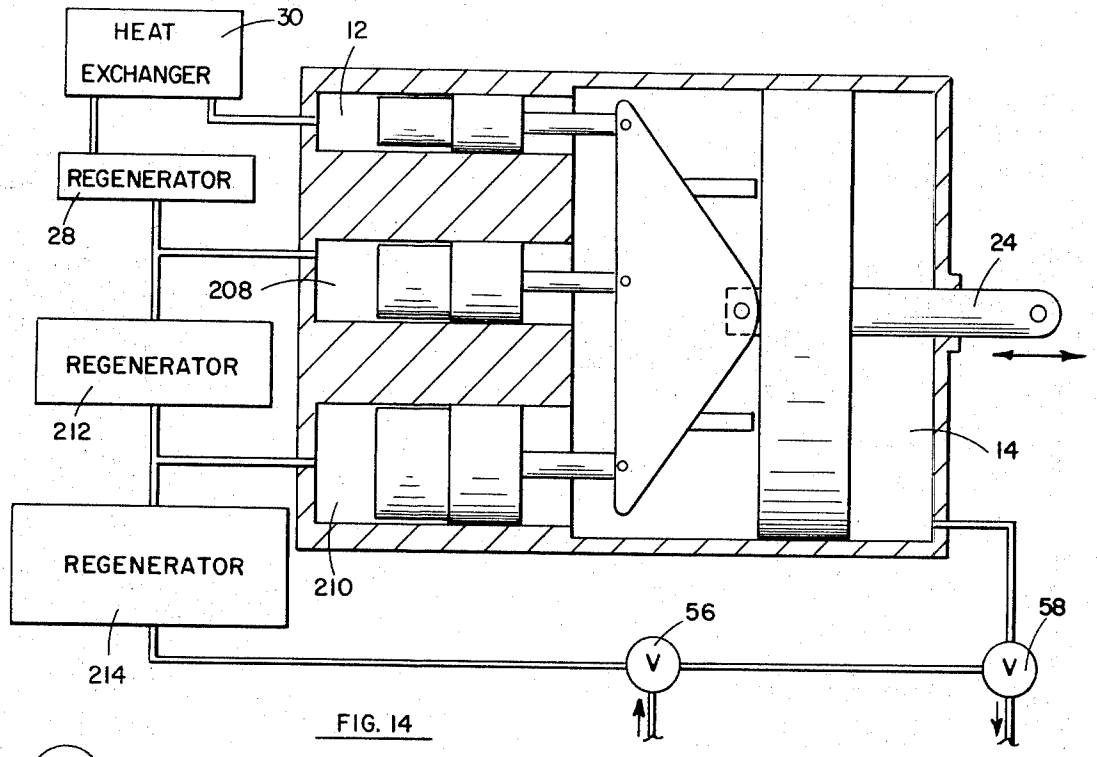


FIG. 14

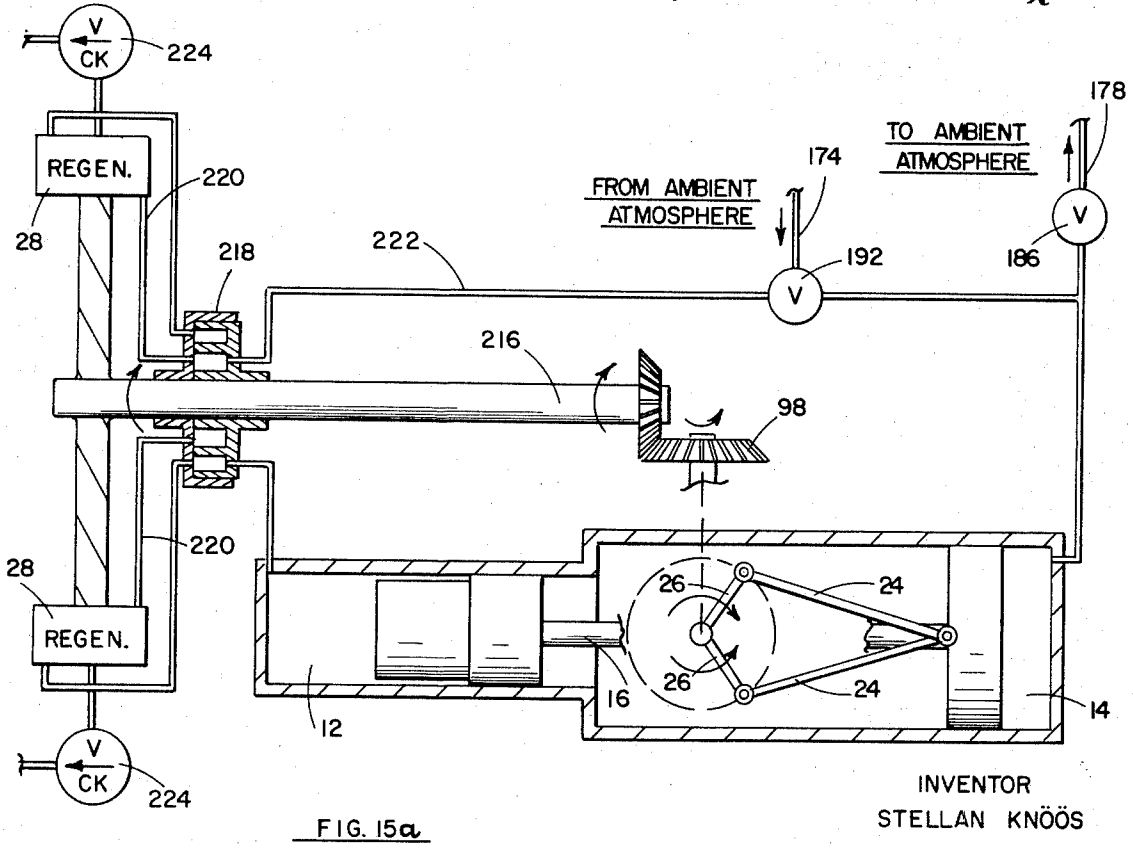


FIG. 15a

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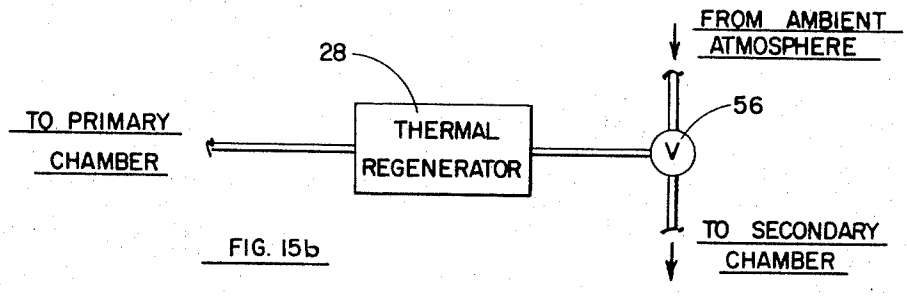


FIG. 15b

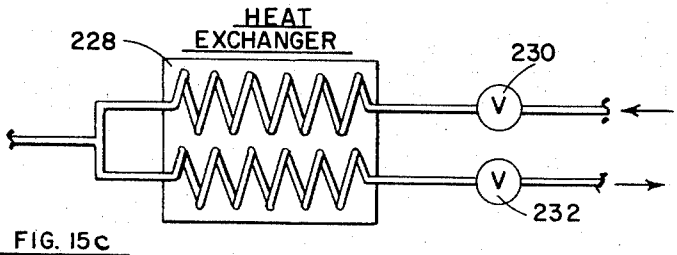


FIG. 15c

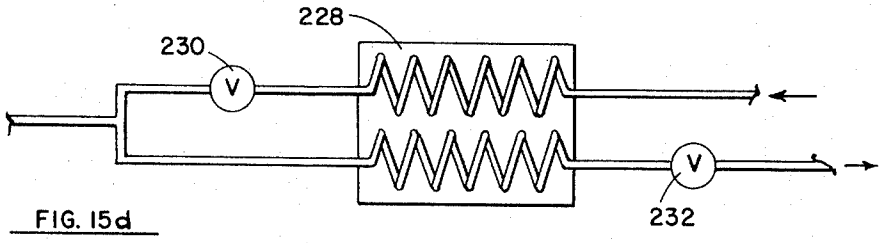


FIG. 15d

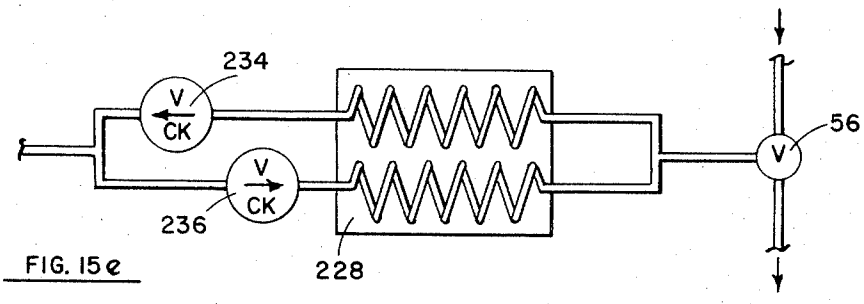


FIG. 15e

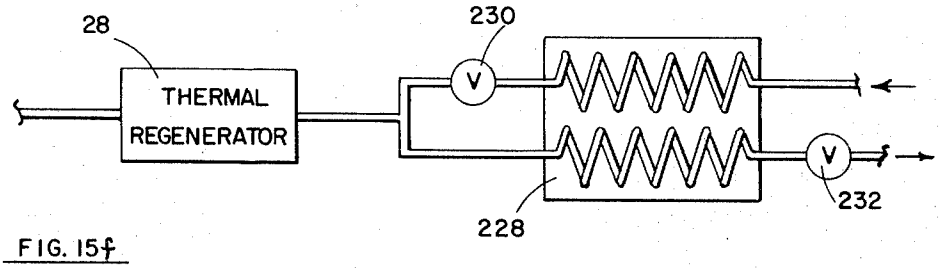


FIG. 15f

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METHOD AND DEVICE FOR HOT GAS ENGINE OR GAS REFRIGERATION MACHINE

BACKGROUND OF THE INVENTION

This invention relates generally to machines for compressing and expanding gas with heat addition and rejection to implement the operation of external combustion gas engines, expansible volume refrigerators or heat pumps, and more particularly to improved means of performing the desired thermodynamic cycles for the working gas in such devices.

Several arrangements are available for performing the Ericsson and the more well-known Stirling cycle for hot gas engines with external-combustion heat addition. The most recent of the Stirling engines employ two pistons reciprocating in a single cylinder approximately 90° out of phase with each other. Stirling engines of this kind are receiving much attention, particularly due to their potentially low-pollution exhaust gas emission, but also due to high thermal efficiency and low noise. They further have the advantage of capability of use with a great number of different liquid and gaseous fuels. The Stirling engines are necessarily equipped with thermal regenerators in order to obtain a high thermodynamic efficiency. Characteristic of present Stirling-cycle engines is a requirement of liquid cooling. The working gas, which could be helium, hydrogen or less preferred air, rejects heat in a cooler placed in immediate connection with the thermal regenerator in the gas line between the hot and colder work chambers. For optimum efficiency, the "dead-space" of the cooler, as well as regenerator and heater "dead-spaces" should be minimum. A compromise generally has to be made between the cooler dead-space and the efficiency of the cooler, by keeping the dead-space moderately small but allowing the temperature difference between the liquid coolant and the working gas to be relatively large, and hence a lower overall temperature ratio and lower thermodynamic efficiency than could otherwise be attained. Contributing reasons why hot gas engines are not in widespread use today are large weight and high mechanical complexity of existing designs, and their estimated high production cost, e.g. as compared to Otto-cycle and Diesel-cycle engines with the same power output. Materials and sealing problems of the past have to a great extent been solved by modern technology, and no large technological barriers exits today to the practical use of hot gas engines, e.g. for automobile propulsion. However, the use of Stirling engines for automobiles has so far not been favored, primarily due to the projected high manufacturing cost of the engine, the relatively poor power/weight ratio, and practical difficulties to be experienced in handling high pressure helium and hydrogen working gases, e.g. in service and overhaul. With the more severe air-pollution problem on hand, the demand of a low-pollution engine is stronger than ever, and the Stirling engine or hot gas engines in general could provide the sought for solution if they could be made practicable.

Modern Stirling engine concepts and associated devices are described, for example, in the following U.S. Pats: No. 3,166,911 to Meijer; No. 3,442,079 to Meijer; No. 3,011,306 to Meijer; No. 3,036,427 to Meijer; No. 3,015,475 to Meijer; No. 3,472,037 to Kohler, and No. 3,458,994 to Heffner. Other hot gas engines are described in U.S. Pats. No. 3,183,662 to Korsgren, No. 3,460,344 to Johnson, No. 3,407,593 to Kelly, No. 3,138,918 to Baker, No. 3,174,276 to Baker, No. 3,385,051 to Kelly, No. 3,080,706 to Flynn and No. 2,564,363 to Horwitz. An interesting invention for a hot gas engine is described in U.S. Pat. No. 3,457,722 to Bush, where the cycle is interrupted during idle periods by the use of valves, and heating and cooling coils are connected into the system. Unfortunately, the device appears to be mechanically complex and uses valves operating in high-temperature environments.

Several gas-expansion refrigerator systems have been devised and built with the Stirling thermodynamic cycle. Such gas-expansion refrigerators operate with a closed thermodynamic cycle resembling the engine cycle, with the difference that mechanical work is added to the gas, heat transferred to the gas at a low temperature (the refrigeration load), and heat rejected at a higher temperature. Stirling-cycle refrigerators have come into widespread practical use, starting a few decades ago, while the corresponding engine still awaits exploitation. Stirling-cycle refrigerators are today capable of providing temperatures considerably below 100°K, and a few systems have demonstrated refrigeration-load capabilities much above 10 kW at these temperatures. Measured overall efficiencies for such systems have been as high as 40 percent of the Carnot efficiency (the theoretical limit). Stirling-cycle refrigerators have been miniaturized and used for cooling, e.g., electro-optical devices. Temperatures below 20°K in multiple-step operation have been demonstrated with helium as the working gas. The refrigeration-load capabilities for these devices have been typically of the order of 1 watt, with the total weight of the refrigerator system of the order of 10 kilograms. As for the Stirling hot-gas engines, the Stirling-cycle refrigerators of today are mechanically relatively complex, primarily due to their use of two pistons with 90° phase difference therebetween. This makes for an expensive design, mitigating against widespread use of this type of refrigerator.

Among other refrigerators using gas-expansion principles with a reciprocating piston can be mentioned the Gifford-McMahon-cycle refrigerators. These devices are related to the Solvay refrigeration cycle, and employ a displacer piston with pneumatically controlled motion, and a separate high-pressure source of working gas, e.g. from a compressor. Irreversible expansion are characteristic for these refrigerators, and the associated entropy production lowers the thermodynamic efficiency to values considerably below the ideal Carnot efficiency. Therefore, the Gifford-McMahon-cycle refrigerators are mostly being used in systems where thermodynamic efficiency and low power consumption are not of primary concern, but rather where reliability and maintenance-free operation are essential, e.g. for airborne and military applications. Refrigerators of this kind are described in the following U.S. Patents: No. 2,966,035 to Gifford; No. 2,906,101 to McMahon; No. 3,119,237 to Gifford; No. 3,188,819 to Hogan.

Other interesting gas-expansion refrigerators are the pulse tube refrigerator by Gifford, No. 3,237,421, and the heat-powered refrigerator of the Vuilleumier type, which has received attention, e.g. for space and air-

borne cryogenic applications for small refrigeration loads. A recent embodiment of the Vuillemier refrigerator is described in U.S. Pat. No. 3,423,948 to Cowan. A great number of gas-expansion refrigerators with counter-current heat exchangers exist as well (reverse Brayton cycle, Claude cycle, etc.). These devices are not of immediate interest with reference to the present invention, since they do not contain a thermal regenerator of the kind normally used in this invention. Only with reference to air dryers and heat pumps would counter-flow heat exchangers be of great interest relative to this invention. Of no immediate interest are Joule-Thomson refrigerators or common two-phase refrigerator systems using freons and other refrigerants, since the present invention is primarily a single-phase (gas) device.

SUMMARY OF THE INVENTION

The present invention provides an improved and simple device for a hot gas engine or a gas-expansion refrigerator, using a new and simple method of performing the expansion step, and if such is desired, recompressing the working gas with a simple reciprocating piston assembly. Basically, simple versions of the invention employ two cylindrical work chambers of volumes which vary harmonically exactly or approximately 180° out of phase by virtue of a single reciprocating piston assembly, or several pistons. Working gas is stored in a plenum chamber for closed-cycle operation. The gas is allowed to flow from the plenum chamber, or from the ambient atmosphere for open-cycle operation, into the so-called primary chamber, first passing through a valve arrangement of suitable kind, though a thermal regenerator, and finally through a heat exchanger (heat for the engine; refrigeration-load heat exchanger for the refrigerator). With cold-gas bleed for the open-cycle refrigerator the heat exchanger may be eliminated, and the heat load presented to the thermal regenerator by the discrepancy in mass of inflow and outflow through the regenerator. The gas is permitted to flow into the primary chamber during a controllable portion of the "forward" piston motion in which the volume of the primary chamber increases. In the "reverse" stroke, gas from the primary chamber is permitted to flow through the mentioned heat exchanger and thermal regenerator, and through the mentioned valve mechanism, or through another separate valve, into the secondary chamber, where the volume is now increasing. This transfer of gas from the primary to secondary chambers is performed in a nearly reversible fashion, in contrast to e.g. Gifford-McMahon devices, by the very important fact that the flow into the secondary chamber is initiated when the volume of this chamber is zero (ideal case) or small, and hence no or little irreversible free expansions will take place. In the subsequent forward stroke the work gas trapped in the secondary chamber is recompressed to the pressure level of gas in the plenum chamber (closed-cycle operation), or to a suitable level (open-cycle operation) for exhaust. The suitably recompressed work gas is permitted to leave the secondary chamber through still another valve mechanism, which may be integrated with one or both of the first mentioned valves, and flows into the plenum chamber, or into the ambient atmosphere for open-cycle operation. For closed-cycle operation, working gas is cooled preferably in the plenum chamber to the original temperature, and can thereafter be used to perform a new work cycle.

The mentioned valve mechanisms could be of various types, e.g., disk valves, sliding linear valves, or rotary valves. Proper synchronization between the valve action and the piston position (or shaft angle) could be achieved in many ways, e.g., with help of cams followers on the rotating shaft, or simply by placing a rotating valve or several rotating valves on the main rotating power shaft, or on a shaft that rotates with the same angular speed as the main shaft for the reciprocating piston motion.

In the transfer flow from the primary to the secondary chambers, the gas pressure should be lower than during injection into the primary chamber in order to obtain the correct thermodynamic overall cycle. This could be accomplished by a proper selection of cross-section area ratios of the primary and secondary chambers, and/or proper termination of the flow into the primary chamber during the forward piston stroke. With injection during the full forward stroke for an engine, as a special case it is possible to employ an effective area ratio 1:1, which case could be of significant practical interest. For the refrigerator, an area ratio of 1:1 must always be accompanied by terminated injection during the forward stroke, at least for cases with no cold-gas bleed from the primary chamber. Secondary chambers with considerably smaller effective cross sectional area than that of the primary chamber can be used in the device of the invention is conjunction with injection only during a small fraction of the forward stroke, but this would be of practical interest primarily only to the engine.

The thermodynamic processes involved with devices of the invention are difficult to describe accurately, in particular since different portions of the work gas undergo different thermodynamic paths, e.g. when described in a temperature-entropy diagram. However, using a simple model for an "average" gas element it is easy to verify that the thermodynamic path is a loop of the desired kind for the closed cycle, with a calculable net amount of mechanical energy that can be extracted for the engine, and a refrigeration-load for the refrigerator. For moderate overall pressure ratios for the work cycle, e.g. ranging from 2 to 5, the computed thermodynamic efficiencies according to ideal models are very high, indicating a performance of devices of the invention that potentially could be brought close to that of an equivalent Carnot cycle. Realistic calculations show efficiencies which in fact are better than for common Stirling-cycle devices, primarily due to superior cooling characteristics and lack of cooler dead-space for devices of this invention.

The mechanical output for the engine of the invention, and the cooling and load characteristics of the refrigerator can be controlled and regulated in a multitude of ways. The invention includes means for providing this control. A simple principle of regulation and control is to connect the plenum chamber (or another special chamber) with any or both of the work chambers during particular phases of the piston motion, and thereby controlling the pressure ratio of the cycle. The plenum chamber is then acting as dead-

space during part of the cycle. The mentioned valve mechanisms can be used for this purpose, e.g. by using three-dimensional cams (with axial) movement) for a disc valve assembly. Alternative schemes could use a separate dead-space chamber or multitude of chambers that fully or stepwise could be connected to the work chambers continuously during the work cycle. As a special example of the first mentioned control technique, we mention termination of the gas flow from the plenum chamber into the primary chamber at an appropriate early position of the piston in its forward stroke. Hereby the amount of work gas entering the device each cycle can be controlled, and as a result thereof also the power output for the engine, or heat-load characteristics of the refrigerator. Maximum power for the engine may be achieved if the injection is terminated, e.g. after the piston has performed three-quarters of the forward stroke. Still another control technique is direct control of the pressure level in the plenum chamber. This could be done by help of hydraulic and pneumatic means, but could be power consuming and therefore is not a favored control technique.

The advantages of the invention will be readily understood from the description of the preferred embodiments thereof. However, we summarize obvious advantages of the closed-cycle engine of the invention in the following list.

1. Possibility of using only one piston or piston assembly, as compared with two pistons in common Stirling-cycle engines. This permits simple mechanical construction, low weight, and low production cost.

2. More efficient cooling system as compared to other hot gas engines. With cooling of the working gas taking place in a separate loop (in the plenum chamber), the residence time for a working gas element in the cooler can be made suitably large as can the heat-transfer area, and the heat-rejection fluxes conveniently low, in certain cases permitting the use of a gaseous coolant (e.g. ambient air). The cooling could therefore be more complete, and hence also the thermodynamic efficiency larger than for conventional Stirling engines.

3. The decoupled cooler does not introduce any dead-space effect, a fact which further increases the efficiency of such an engine.

4. No balancing buffer gas is needed, in contrast to recent Stirling engines employing rhombic-drive mechanisms and high pressures for the working gas. With this invention the power output means could be implemented with a relatively thin piston rod, or simply with the crank mechanism located between the two work chambers.

5. With air as coolant, heated air from the cooler could be used in the combustion process in the burner and heater. Reductions in weight and manufacturing cost can be made with such a scheme, in part from the fact that a single device (e.g. impeller) can be used for propelling this air both through the cooler and through the heater.

6. Possibility of locating the cooler and plenum chamber remote from the work chambers. This could be a most important feature for engines used in automobiles. Possibility of using a filter in the plenum chamber.

7. Simple and efficient control and regulating possibilities.

8. Multiple-cylinder arrangements can use a common cooler and plenum chamber.

9. Multiple-cylinder arrangements can be built mechanically simple with a high power/weight ratio.

10. For certain applications air could be used as working gas. Sealing problems would then be less severe, since gas lost by leakage could simply be replaced from the ambient atmosphere. In common Stirling engines air is a less favored working gas, in part due to difficulties in cooling the gas in the cooler (air has much smaller thermal conductivity than helium or hydrogen).

Advantages of the refrigerators of the invention (closed or open cycles) are summarized as follows

11. Highly efficient cooling system. Superior cooling performance is of primary importance in obtaining low temperatures and refrigeration-load/power ratios.

12. Highly reversible flow. Refrigerators of the invention can be built to give a minimum of irreversible entropy production, resulting in high overall efficiency.

13. Multiple-step staging possible with simple means. Two or more stages can be used for generation of very low temperatures, without addition of valves, and with a simple extension of the piston assembly to include additional work portions.

14. Open-cycle refrigerators can be built with extreme simplicity, without the use of heat exchangers other than the thermal regenerator. Such devices could be used for heat-pump applications and for air-conditioning systems. With modifications they could be used as air dryers, or devices for producing cryogenic liquids. The heat-pump application would be most interesting, since the coefficient of performance and hence the economy would be much better than existing systems using two-phase refrigerator systems with heat exchangers.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a schematic view of a first embodiment of a hot gas engine according to the invention, with a combustion heat source;

FIG. 2 is an idealized temperature-entropy diagram for a closed-cycle hot gas engine as shown in FIG. 1;

FIG. 3 is a diagrammatic view of another embodiment of the invention which is a two-step recompression hot gas engine;

FIG. 4 is a schematic view of a further embodiment of the invention which is a hot gas engine with its crank mechanism nearly perfectly balanced and located between the work chambers;

FIG. 5 is a schematic view of still another embodiment of the invention which is a double-acting hot gas engine with an oscillating shaft and vanes;

FIG. 5*a* is a schematic view of the output drive of the engine of FIG. 5;

FIG. 6 is a schematic view of another embodiment of the invention which is a four-cylinder hot gas engine;

FIGS. 7*a* –7*d* are diagrammatic views illustrating some ways of regulating the power output of the hot gas engine of the invention;

FIG. 8 is an idealized temperature-entropy diagram for a closed-cycle refrigerator of the invention;

FIG. 9 is a schematic view of still another embodiment of the invention which is an open-cycle refrigerator with cold-gas bleedoff;

FIG. 10 is a schematic view of a further embodiment of the invention which is a simple open-cycle refrigerator with cold-gas bleedoff;

FIG. 11 is a schematic view of another embodiment of the invention which is a refrigerator with phase difference between the two pistons operating in the primary and secondary chambers, respectively, and particularly adapted for refrigerators with large cold-gas bleedoff;

FIG. 12 is a schematic view of a further embodiment of the invention which is an open-cycle refrigerator with a primary counter-flow heat exchanger and two-step expansion, without recompression of the working gas;

FIG. 13 is a schematic view of another embodiment of the invention which is a two-step cooling device;

FIG. 14 is a schematic view of still another embodiment of the invention which is a three-step cooling device;

FIG. 15*a* is a schematic view of a further embodiment of the invention which is an air dryer based upon the open-cycle refrigerator principle, and employing centrifugal separation of the condensed liquid in the thermal regenerator; and

FIGS. 15*b*–15*e* are schematic views of alternate (stationary) regenerator embodiments using counter-flow heat-exchangers and valve additions, of particular use for air dryers and heat pumps, but also in particular cases for any of the other closed or open-cycle refrigerators or engines of the invention.

DESCRIPTION OF THE PREFERRED EMBODIMENTS (ENGINE)

Referring now to the drawings, FIG. 1 discloses a hot gas engine generally indicated by numeral **10**. The primary cylinder chamber is denoted **12** and the secondary chamber **14**. The piston **16** has two cylindrical portions of generally different diameters, one cylindrical portion **16***a* moving in the primary chamber **12** formed by cylinder **18**, and the other portion **16**b moving in the secondary chamber **14** formed by cylinder **20**. The piston is connected to rotatable shaft **22** for extraction of mechanical energy by means of connecting rod **24**, and crank **26**. The portion of cylinder **20** between piston portions **16**a and **16**b is maintained at an arbitrary pressure level, e.g., the ambient pressure by means of port **20***a* formed in the cylinder wall. The shaft could be connected to a flywheel arrangement, if desired, that could store mechanical energy and maintain correct reciprocating piston motion even when energy from the gas is not added to the piston assembly.

A thermal regenerator **28** is connected in series with a heat exchanger **30** for addition of heat to the working gas. The regenerator could be a metal matrix, sintered material, a bed of pebbles, packed metal wire-mesh, etc., or a counterflow heat-exchanger with one-way channels (with additional valving to achieve one-way flow in each leg). The regenerator is used for temporary storage of heat of the oscillating working gas. For the engine the regenerator matrix receives thermal energy from the working gas leaving the primary chamber, and adds the same amount (ideally) to gas flowing into the primary chamber; for the refrigerator to be described later, the regenerator receives thermal energy from gas entering the primary chamber, and returns the same amount of energy when the working gas leaves the same chamber. Hence, ideally the regenerator is in no thermal contact with any other material than the working gas.

Ambient air is fed into the heater assembly **31** (which includes burner **36**, e.g. for a liquid petroleum fuel, and heat exchangers **30** and **34**), from line **32**, through heat exchanger **34**, preferably of the counter-flow type, and into the burner **36**. The fuel is fed into the burner through line **38**. The products of combustion in burner **36** are fed through line **40** into the heat exchanger **30**, where heat is transferred to the working gas, which could be helium, hydrogen, or air, etc. FIG. **1** is only schematic, and in practice the burner and heat exchanger **30** could be integrated into a single unit. From the heat exchanger the combustion products are fed through line **42** back into the counter-flow heat exchanger **34**, where further heat is transferred to the incoming air, and finally exhausted through line **44**. The counterflow heat-exchanger **34** could be replaced by a suitable thermal regenerator, e.g. by a rotating regenerator of metal or ceramic material.

The hot gas engine shown in FIG. **1** has a combined cooler and plenum chamber **46**, in which working gas is cooled with help of a coolant circuit, and working gas can be stored at a convenient working pressure for the engine. The coolant (liquid or gaseous) flows into the cooler through line **48** and out of the cooler through line **50**. The working gas enters the cooler **46** through line **52**, and leaves the cooler through line **54**. The volume of the Plenum-cooler **46** is preferably larger in comparison to the swept volume of the piston in the primary chamber. The advantage of a large plenum-cooler is that it affords a large residence time for the gas in the plenum-cooler; this residence time could be made many times larger than the period of one work cycle. The heat fluxes through the cooler could then be made relatively small, and the efficiency of the cooler high, and in certain cases direct gaseous cooling (e.g., ambient air) could be used. With ambient air as the coolant, the coolant flowing from the cooler in line **50** may be fed directly into line **32** of the heater, with or without a certain amount of bleedoff, and used in the combustion process. The flow of coolant air can be sustained with the help of an impeller arrangement (not shown), that could be coupled to the rotating shaft **22** (or preferably geared to a higher rotational speed than provided by shaft **22**). Pumping air coolant could alternatively be achieved by using the harmonically changing volume between pistons **16**a and **16**b in cylinder **20**, and an added check valve.

The hot gas engine is shown with a general 3-port 2-way valve **56**, and a 2-port 1-way valve **58**, for control of the flow of work gas through the engine. These valves could be combined into a single valve assembly, or further divided into e.g., a total of three 2-port 1-way valves of a suitable type (disc valves, sliding valves, rotating valves, etc.). The valve action could be controlled in many different ways, but generally most simply by coupling to the rotating shaft **22**. This

coupling is indicated schematically in FIG. 1. The coupling could be eliminated if valve 58 is a simple check valve, or could include cams and cam followers if valve 56 is two disc valves, etc. The arrangement shown in FIG. 1 will suffice for the description of the operating principle of this type of engine which now follows.

Let us assume that quasi-steady conditions have been established, particularly with regard to the temperature profile in the regenerator. The right hand side of the regenerator 28 then has a temperature close to the temperature of gas in the plenum-cooler 46, and the left hand side of the regenerator has a temperature close to that of the heat exchanger 30, with a continuously changing temperature profile therebetween. Thermal conduction inside the regenerator (from left to right) should preferably be small compared to the heat flux through the heat exchanger 30. The work cycle can be described thermodynamically with the aid of FIG. 2 which is a strongly idealized and schematic temperature-entropy diagram for an average gas element flowing through the closed-cycle engine. The volume of the plenum-cooler is assumed to be much larger than the swept volume of the piston, so that the flow of gas from the plenum-cooler 46 into the primary chamber 12 takes place at approximately constant pressure, as indicated by the straight line 60 (isobaric line) in the diagram which employs a logarithmic temperature scale and a linear entropy scale. The plenum-cooler exit condition is denoted 62.

The flow into chamber 12 is preferably initiated when the volume of that chamber is near its minimum, and is achieved by actuating valve 56 (FIG. 1), to connect the plenum-cooler to chamber 12 through regenerator 28 and heat exchanger 30. When the working gas flows from the plenum chamber through the regenerator, the gas temperature is raised by transfer of heat (with a small temperature difference between the gas and the regenerator working elements) from the regenerator. Working gas leaves the thermal regenerator 28 with a temperature and thermodynamic condition indicated by point 64 in FIG. 2. The flow of gas into the primary chamber 12 can be terminated well before the shaft 22 has rotated to a position at which the primary chamber has maximum volume. Depending upon the geometry and desired characteristics of the engine, the termination can take place, e.g., between 25 percent and 100 percent of the forward stroke, with an optimum termination point e.g. near three-quarters of the forward stroke. Valve 56 is then actuated to break the path from the plenum chamber to the primary chamber 12.

The path between the primary and secondary chambers is preferably connected by means of valve 56 when the secondary chamber 14 has minimum volume, in order to avoid possible free expansions and flow irreversibilities. In the following reverse piston motion, the gas in chamber 12 is forced to flow back through the heat exchanger 30 and thermal regenerator 28, into the secondary chamber 14. During this transfer flow process the gas pressure is lower than the injection pressure represented by line 60. Depending upon the area ratio of the primary and secondary chambers and the temperature ratio across the thermal regenerator the pressure during the transfer process may decrease, stay constant or even increase. FIG. 2 relates to a preferred case with decreasing pressure. The isentropic expansion along line 66 to point 68 for the average gas element is then caused by the transfer process, with the element still located in the primary chamber, but could in part also be caused by an expansion following a terminated injection in the forward stroke.

When the average gas element enters the heater during the reverse stroke an isobaric heating process takes place, provided that the dead-space of the heater can be considered small and negligible (strong idealization). The gas element leaves the heater at point 72 in the temperature-entropy diagram. In the regenerator, the gas transfers heat to the regenerator, as indicated by the isobaric line 74, the cooling process ending in point 76 at the exit of the regenerator.

In the present case with decreasing pressure during the transfer flow, the gas flowing into the secondary chamber mixes with slightly colder gas. This mixing process is generally accompanied by an overall increase in entropy, which is generally small and could be ignored in a first-order analysis. In FIG. 2 we neglect such a mixing process and show an isentropic line 78 for the further expansion of the average gas element in the chamber 14. The minimum pressure of the cycle is reached at point 80, when the volume of the secondary chamber is maximum. It should be noted that if the area ratio of the work chambers were chosen such that the gas pressure was increasing during the transfer flow (this process is not shown in FIG. 2), the minimum pressure of the cycle would have to occur at the end of the forward stroke for gas located in the primary chamber, having experienced an expansion following the terminated injection process.

In the following forward piston motion, the average gas element in the secondary chamber is compressed as indicated along line 82 to point 84, at which point the pressure is the same as in the plenum-cooler 46. Valve 58 is then opened (guided valve or check valve) from its closed position, in order to let gas flow through line 52 into the plenum-cooler 46 during the final portion of that piston stroke.

As shown in FIG. 2, working gas that enters the plenum-cooler has a higher temperature (point 84) than when leaving the plenum-cooler (point 62). The average gas element is cooled along the isobaric line 86 to the original condition 62, and the thermodynamic path is closed. A necessary condition for the average-gas model is that the regenerator has transferred zero net amount of heat to the gas, per cycle. Therefore, in a normal temperature-entropy diagram employing a linear temperature scale, the area a line corresponding to line 60 to the entropy axis must be identical with the area under a line corresponding to line 74 to the entropy axis, according to fundamental thermodynamic theories. In such a diagram (linear temperature axis), the net heat addition per unit mass of the average gas in the heat exchanger 30 would be represented by the area under the corresponding line 70, and is generally smaller than the heat which is temporarily transferred to the gas from the regenerator, represented by the area under a line 60 in the modified temperature-entropy diagram. From this we can see that the regenerator is a significant element in the engine of the invention.

The thermodynamic process described in FIG. 2 is only approximate but can nevertheless be used for evaluating performance characteristics of the hot gas engine. Calculations with such a model, and also with much more accurate models, show that the anticipated thermodynamic performance of the device could indeed be high, and generally close to the Carnot-cycle performance. This is particularly true when the overall pressure ratio of the work cycle is kept moderate or small. For a practical engine the net power output per cycle is of importance, as well as efficiency. Therefore the overall pressure ratio cannot be kept close to unity as efficiency considerations would dictate, but rather between two and eight for a practical design.

It should be obvious with the simplified model of FIG. 2 that the thermodynamic processes involved in the hot gas engine of the invention are practically reversible in all separate steps. Entropy-producing free expansions are avoided, as well as throttled flows, since these have strong adverse effect upon the efficiency. With finite volumes of the regenerator, heater, and the channels from the primary to secondary chambers, and hence non-negligible dead-space, the valve actions could be modified to completely avoid free expansions. For example, valve **56** could be actuated to close the path between the regenerator **28** and the secondary chamber **14** before the piston has reached the left turning point in the reverse stroke; the closing could be such that gas in the regenerator, heater and primary chamber is recompressed to the plenum-cooler level in the very last portion of the reverse stroke, after which valve **56** is actuated to open the path between the plenum-cooler and the primary chamber.

FIG. 3 shows a hot gas engine employing recompression in two steps with the help of a third work chamber **88**, formed by the piston and the cylinder **20**. Besides the first plenum-cooler **46**, a second plenum-cooler **90** could be used, operating at a higher pressure level. Additional valves are used, here shown as check valves **92** and **94**. These could be valves of any suitable type and controlled not by the gas pressure as indicated, but by the shaft angle as for valve **56** (e.g. cam, cam follower and disc valve). In the embodiment of FIG. 3, gas in the secondary chamber **14** is forced into the plenum-cooler **46** through the valve **58** at minimum pressure loss, when the pressure in the secondary chamber is raised to a desired level (equal to the gas pressure in the plenum-cooler **46**) in the forward stroke. During the same stroke, another portion of g as flows from the plenum-cooler **46** through valve **92** into the compression chamber **88**. Final compression takes place in the reverse stroke in chamber **88**, to the pressure level of the plenum-cooler **90**. In the final part of that stroke, the valve **94** is opened and gas transferred from chamber **88** into the plenum-cooler **90**, where the gas is cooled to its original temperature. The steady-state pressure level of the first plenum-cooler **46** will depend upon the relative sizes of chambers **14** and **88**, as well as the cooling in plenum-cooler **46**. Among advantages of two-step or multiple-step recompressions can be mentioned the possible reduction in the total amount of heat that has to be rejected in the coolers, for a given overall pressure ratio of the cycle, as well as a possible increase in the net power output and thermal efficiency due to less compression work necessary. Another advantage could be better balancing and distribution of angular moment on the output shaft **22**. In practice, these features should have to be weighed against the added mechanical complexity with two-step recompression.

FIG. 4 shows another embodiment of my hot gas engine. The rotating output shaft **22** is here placed between the primary chamber **12** and the secondary chamber **14**. For simplicity, we have chosen the same diameter for these chambers and piston portions **16*a*** and **16*b***, making the design attractively simple. A bevelled gear **96** on the shaft **22** drives a pinion **98** and another bevelled gear **100**, which is coaxial with the gear **96**, and counter-rotating. Two connecting rods **24** are used to rotate the piston assembly **16**. The rods are each fastened to a respective one of the counter-rotating gear wheels. Counter-weights **102** and **104** are shown schematically attached to each of the gears. With this arrangement a practically perfect balanced engine can be obtained, and with no significant side forces on the piston assembly. Such an absence of side forces is desirable in solving the sealing problem, and would permit an evenly distributed wear on piston rings, etc.

The valves **56** and **58** are here shown as rotating valves, located directly on the axis of gear **100**. The valve action is the same as described for the engine in FIG. 1. The output power is delivered during the first half of the forward stroke with this type of engine. Therefore, the engine could preferably be equipped with a flywheel arrangement for smooth constant angular speed of shaft **22**. The flywheel could preferably be located on one of the gear shafts, or to another high-speed shaft, geared to any of the original shafts. The plenum-cooler **46** is shown schematically and not in proper size relationship to the primary and secondary chambers. The plenum-cooler is preferably large with respect to the chambers as for prior embodiments; the principles discussed for FIG. 1 as to the size of the cooler also holding here.

FIG. 5 and FIG. 5*a* show a hot gas engine having an oscillating main shaft in a double-acting mode. The rotation of shaft **22** is developed from the oscillating motion of shaft **106** by means of crank **26** and rod **24**. Several other schemes of generating the oscillating motion and limitation of the rotation angle are possible, but they shall not be further discussed here since such configurations are widely known. Two vanes **108** and **110** are fastened to the oscillating shaft **106** and constitute moving walls in the chambers **112**, **114**, **116** and **118**. Chambers **112** and **114** constitute one basic hot gas engine, and chambers **116** and **118** another, with a common plenum-cooler **46**. The A-engine (**112** and **114** chambers) and the B-engine (**116** and **118** chambers) have individual heaters **30*a*** and **30*b*** and regenerators **28*a*** and **28*b***. If desired, the heaters could be partially integrated. The vane **108**, oscillating in the primary chambers **112** and **116**, is shorter than the vane **110**. Therefore, the maximum volumes of the primary chambers are smaller than those for the secondary chambers, as for the engine of the embodiment of FIG. 1. The total angular motion of the oscillating shaft **106** is limited to less than 180°. The four valves **56*a***, **56*b***, **58*a*** and **58*b*** could be of any suitable type and, for example, integrated and located on a rotating shaft in

phase with shaft **22**, and operated in the same basic manner as described for the embodiment of FIG. **1**. This double-acting hot gas engine could be designed to deliver a positive amount of mechanical power to the rotating shaft **22** for typically half the period of a 360° rotation of the shaft. A flywheel could preferably be connected to the output shaft, or to a shaft geared to shaft **22**.

FIG. **6** shows a hot gas engine of the invention with a multiple cylinder arrangement. Four separate work cylinders **20***a*–**20***d* are shown, each of constant cross-sectional area, similar to the engine of FIG. **4**. Each cylinder has an individual regenerator **28***a*–**28***d*, but the heater **30** is partially common for the work cylinders, and the plenum-cooler is completely common for all cylinders. As indicated in the figure, the plenum-cooler could be remotely located without dead-space penalty, an advantage, e.g., for automobile-engine applications. The valves **56** and **58** (of the embodiment of FIG. **1**) are in the embodiment of FIG. **6** integrated into valves **124***a*–**124***d* of the 4-port 3-way type. The valve units **124***a*–**124***d* could again be e.g., cam-guided disc valves or linear valves, or valve of rotating types mounted on a shaft rotating in phase with the main shaft **22**. Bevelled gears **96** and pinions **98** are utilized as in the embodiment of FIG. **4**, in order to achieve good balancing and small side forces on the pistons. Naturally, other known crank mechanisms can be used as well. A single flywheel is attached to the main shaft **22** (two counter-rotating flywheels could preferably be used on two counter-rotating gear wheels in order to eliminate possible strong gyroscopic effects). The work pistons in FIG. **6** are arranged with 90° phase difference, in an individual order that could be changed.

FIGS. **7***a*–**7***d* show schematically three ways of regulating the power output from a hot gas engine by changing the valve opening and closing characteristics. FIG. **7***a* shows a basic hot gas engine of the invention with a schematic 4-port 3-way valve **124**, replacing the two valves **56** and **58** in FIG. **1**. The possible paths of gas through this valve are denoted "I" for injection into the primary chamber, "II" for the transfer flow to the secondary chamber, and "III" for the exit flow from the secondary chamber **14** into the plenum-cooler. The crank angle of the main shaft is denoted "ϕ", with $\phi = 0$ for the piston at the left turning point. FIG. **7***b* shows schematically a preferred way of regulating the power output from the engine, namely by regulating the amount of injection of gas into the primary chamber through control of path I. Maximum power is here obtained with curve **128**, for shutoff of path I for a certain crank angle less than 180°. Delaying shutoff as indicated by curve **130**, the overall pressure ratio of the cycle is decreased and the mechanical power output per cycle is decreased. The delay may continue to crank angles larger than 180° if desired.

This regulating technique can be said to introduce the plenum-cooler as dead-space when the injection is not terminated according to the ideal curve **128** in FIG. **7***b*. The dead-space is only coupled to the work chamber **12** for a fraction of the cycle. The control function can be controlled mechanically, e.g. with an axially moveable three-dimensional cam for the case when cam, cam followers and disc valves are used, or by rotating the outer valve housing for a rotating valve, or in any other known way generate the desired change in phase (with respect to the crank shaft) of the closing of path I.

FIG. **7***b* shows the alternative regulating procedure, whereby path I is closed earlier than normal, represented by line **132**. This method decreases the mass flow through the engine compared to the first scheme, and also increases the overall pressure ratio of the work cycle (and therefore slightly decreases the efficiency compared to the optimum value).

FIG. **7***c* shows a second basic way of regulating the power output, which will generally be less preferred. The normal opening of path III (indicated by curve **134**) takes place at a crank angle for which the recompression in the secondary chamber has proceeded to a stage at which the gas pressure equals the plenum-pressure. By delaying the opening of path III, e.g., as indicated by curve **136**, additional pressure build-up in the secondary chamber will take place, before the pressurized gas will flow irreversibly into the plenum-cooler **46**. The result of such a delay is a lowering in the net mechanical power output per work cycle, but also a lowering in the thermodynamic efficiency due to the flow irreversibility.

The technique of delaying opening of path III can be used even when the path is regulated with help of a check valve. A special type of check valve could be employed using a variable counterpressure to a hydraulic or pneumatic circuit for the control. Such a regulating technique could be made relatively simple, and is to be preferred before a direct mechanical control technique.

The normal closing of path III takes place for $\phi = 180°$, as shown by curve **138** in FIG. **7***c*. If this closing instead is delayed, as represented by curve **140**, and path II is opened in a normal fashion, the lowering of the gas pressure in the work chambers is prevented during the reverse stroke, and gas from the plenum-cooler redrawn into the work chambers through line **52**. This regulating scheme is essentially identical with that described in FIG. **7***b* by curve **130**, and could be a favored scheme for certain applications.

Finally, FIG. **7***d* shows power regulation by changing the opening of path II from the ideal, as represented by the curve **142**, to a delayed opening, as represented by curve **144**. This scheme gives entropy production from irreversible flow into the secondary chamber, but could be effective in rapidly stopping or slowing the engine.

Other power regulating techniques than the mentioned ones could be used. A simple method could make use of a separate dead-space chamber of variable volume, which could be connected to any of the work chambers continuously during the work cycle (previous methods used interrupted connection with the plenum-chamber dead-space). With the dead-space being minimum the power output would be maximum. The dead-space could be continuously variable, e.g. with help of a hydraulically actuated piston in a cylinder dead-space. Alternatively, step-wise changing of the dead-space volume could be used with help of a series of smaller dead-space chambers that could be connected one by one with the work chambers, by help of a valve arrangement (e.g., sliding valve). Still other power control schemes are possible whereby the pressure level of the plenum-cooler (and possible the

plenum-cooler volume) are directly changed. Many variations are possible here, e.g. using separate gas-storage chambers of different pressure than the main plenum chamber, using hydraulically actuated pistons to compress gas in the plenum-cooler, and special devices for limiting the amount of gas in the plenum-cooler that can flow into the primary chamber (without introducing unnecessary irreversibilities). These schemes shall not be further discussed here, since they are known within the art.

While the engine of the invention has been described by reference to preferred embodiments, it should be apparent that numerous changes could be made within the spirit and scope of the invention disclosed herein. Some of the possible features are discussed in the following sections of preferred embodiments for the refrigerator, and many of the discussed engine features can be used for the refrigerator of the invention.

DESCRIPTION OF THE PREFERRED EMBODIMENTS (REFRIGERATOR)

For the refrigerator we shall discuss both closed-cycle and open-cycle devices. The closed-cycle refrigerators have much in common with the previously described closed-cycle hot gas engines. Open-cycle engines have not been discussed in detail or in the drawings of preferred embodiments, mainly due to the relatively low power output per unit swept volume, when operated with ambient air as the working medium.

We first discuss the closed-cycle refrigerator in thermodynamic detail using the average gas model, as for the engine. FIG. **8** shows the closed thermodynamic path with the average gas model (strongly idealized). Basically, the refrigerator is operated similarly to the engine, but with the difference that the heat exchanger **30** is now a simple refrigeration-load heat exchanger **30**, with a temperature lower than the gas in the plenum-cooler **46**, and that mechanical energy has to be delivered to the shaft **22** in FIG. **1**, instead of being extracted therefrom. FIG. **8** relates to a refrigerator with decreasing pressure during the transfer flow process, i.e. to a case when the effective cross section area of the secondary chamber is at least as large as that of the primary chamber. As mentioned earlier, the refrigerator of the invention could be operated in a mode with increasing (or constant) pressure during the transfer flow, provided that an initial expansion in the primary chamber takes place during the later portion of the forward stroke. Such a refrigerator could utilize a secondary chamber with even equal of smaller cross-section area than that of the primary chamber, but will not considered in FIG. **8**.

Referring to FIG. **8**, the average gas element leaves the plenum-cooler **46** (in a configuration as FIG. **1**) with the thermodynamic condition indicated by point **146**. Thereafter, the gas flows through the thermal regenerator (ordinary thermal regenerator or counter-flow heat-exchanger with two one-way legs) along path **148** of constant pressure to point **150**, which ideally has the temperature of the refrigeration-load heat exchanger **30**. In the primary chamber **12** the gas element is expanded adiabatically in the reverse stroke, as indicated by line **152**, to point **154**. The average gas element then returns through the refrigeration-load heat exchanger **30**, receiving the refrigeration load as shown by path **156**, to point **158** (having the same temperature as point **150**). The refrigeration load accepted by the gas element per unit mass is represented by the area under line **156** in a temperature-entropy diagram with linear temperature scale (FIG. **8** employs a logarithmic scale). The gas element returns further through the regenerator **28** and is then reheated by the regenerator matrix to point **160**, which has the temperature of gas leaving the plenum-cooler, in this idealized treatment.

Before completion of the reverse stroke, the average gas element undergoes additional expansion, along an isentropic line **162** to point **164**, which has the lowest pressure of this ideal cycle. The recompression takes place ideally along path **166** to point **168** (with pressure identical to point **146**). Finally, the gas element is cooled in the plenum-cooler along the isobaric line to point **146**, and the thermodynamic path is closed.

For an open-cycle, the gas element under study is simply exhausted at point **168**, provided that employed inlet and exhaust pressures are equal, and the refrigerator not "driven" fully or partially with pressurized gas. Such a device will be discussed later.

With the idealized refrigerator model of FIG. **8**, the refrigeration-load was applied in a separate heat exchanger **30**. Naturally, in a practical situation no such separate exchanger may be required, and the load applied e.g. directly to the walls of the primary chamber, or to other heat-exchanging surfaces that the working gas can be in contact with. The same argument could hold for the heat rejection in the plenum-cooler. Part or all of the heat rejection could take place through a heat exchanger (not shown in any of the figures) located in the secondary chamber or in the gas line to the right of the thermal regenerator, or simply by cooling the walls of the secondary chamber. However, for already discussed reasons, the major portion of the cooler could preferably be located in the plenum chamber for closed-cycle operation.

FIG. **9** shows an open-cycle refrigerator with cold gas bleedoff, without a refrigeration-load heat exchanger. This refrigerator is specially adapted for cooling an enclosed volume of gas (air), and transfers the absorbed heat from this volume to gas located outside the enclosed volume (the ambient air). The enclosed volume is assumed to contain gas which is substantially cooler than the ambient gas. For optimum efficiency, the refrigerator is equipped with two thermal regenerators **28** and **170**, one "cold" gas inlet **172** from the enclosed volume, one "warm" gas inlet **174** from the ambient atmosphere, a "still colder" gas outlet **176** (into the enclosed volume), and a "still warmer" gas outlet **178** (to the ambient atmosphere). The necessary valves are here shown as five 2-port 1-way valves **180**, **182**, **184**, **196** and **190**. These valves could be combined into fewer units, as discussed previously for the engines.

The open-cycle refrigerator in FIG. **9** can be operated in several ways, e.g. as follows: At the first portion of the forward stroke, valve **182** is kept open to permit cold gas to flow through the regenerator **28** into the primary chamber **12**. The right side of the regenerator **28** has a temperature which ideally is close to the inlet temperature in channel **172**. Thereafter, in a later portion of the forward stroke warm gas from line **174** is

fed through valve **184** and through both re-generators into the primary chamber. The right side of the regenerator **170** has a temperature at steady-state condition which ideally is close to the temperature of gas in the warm inlet line **174**. The injection of gas through line **174** could be terminated at an arbitrary point of the forward stroke, e.g. at the end of the stroke, and for which case the effective area of the secondary chamber must be larger than that of the primary chamber (as shown in FIG. **9**). With the primary and secondary chambers made up from a single constant-diameter tube, the injection through line **174** must be terminated before the end piston has completed the forward stroke. In the reverse stroke still colder gas is forced through valve **180** to the exit line **176**, followed by the transfer process of the remainder of the gas in chamber **12** to chamber **14** through valve **190**. In an alternative scheme the transfer process could take place before ejection of gas through line **176**, in the reverse stroke, whereby entropy increase would be less and the overall thermodynamic efficiency ideally better.

The cold gas that is bled through valve **180** has attained the lower temperature from the falling temperature profile in regenerator **28** (going from to left). The amount of bleed is controlled by valve **180** and should be identical with the amount of gas taken in through line **172**.

If the cold gas bleedoff takes place before the transfer flow in the reverse stroke, the valve **190** should not be opened until valve **180** is closed. If no backflow has been permitted through valve **186** into the secondary chamber during the cold-gas bleedoff, a free expansion will take place when valve **190** is opened that have an adverse effect on the overall efficiency of this refrigerator. For this reason such backflow could preferably be used.

In the forward stroke, valve **186** is opened when the gas pressure in the secondary chamber has been raised to the ambient level, and the still warmer gas exhausted through line **178**. Valve **186** could be a simple check valve if no backflow of the mentioned kind is desired.

With this type of refrigerator, bleeding an intermittent flow of cold gas without the use of refrigeration-load heat exchangers, the load is obviously placed upon the regenerators. With the refrigerator of FIG. **9** the load is on regenerator **28**, and gas from the ambient atmosphere used in the work cycle to maintain the desired falling temperature profile in regenerator **28** for cooling of the cold gas, and for transport of the heat (and mechanical power from the shaft converted to heat) to the ambient atmosphere. The thermodynamic characteristics of such systems have been computed using gas models as shown in FIG. **8**, as well as with correct infinite element models. With cold-gas bleedoff a fundamental difficulty exists in the treatment of the energy balance of the regenerators, due to the fact that the regenerator experiences a net mass flow and a small amount of entropy has to be generated from heat transfer in the matrix.

In the device of FIG. **9**, the cold-gas bleed could be replaced with a refrigeration-load heat exchanger, with elimination of valves **180** and **182**. Naturally, the same open-cycle refrigerator could be modified to employ closed-cycle operation for gas that flows through the secondary chamber and establishes the desired temperature profiles in the regenerator matrixes. In this case a cooler and a plenum chamber have to be added in the normal fashion. Cold-gas bleedoff is maintained as previously described, with the cold-gas bleedoff rate equal to the rate of inlet flow for steady-state operation.

FIG. **10** shows a simpler open-cycle refrigerator than FIG. **9**, with only one regenerator **28**, one inlet **174**, and a cold gas outlet **176**, and a hot gas outlet **178**. The valves **184** and **190** of FIG. **9** have been replaced in FIG. **10** with a single 3-port 2-way valve **192**, with an operating function similar to that of valve **56** in FIG. **1**.

According to one operating scheme, permitting an arbitraty area ratio for the work chambers, injection through line **174** is terminated before completion of the forward stroke. In the reverse stroke the transfer process is accomplished through valve **192**, after which recompression of gas in chamber **12** follows, and subsequently cold-gas bleedoff occurs through line **176** when the pressure level in the primary chamber has been raised to the desired level. A device of this kind could be built with the primary and secondary chambers made from a single cylinder and a simple piston, and could be used for simple heat-pump or refrigeration applications.

With the device shown in FIG. **10** recompression takes place in the secondary chamber, with subsequent exhaust through valve **186** (check valve or mechanically controlled valve). The refrigeration load is applied on the thermal regenerator as for the previously discussed open-cycle device.

An alternative operating scheme with a device as shown in FIG. **10** could employ cold-gas bleed before the transfer flow process. Such a device should not use an initial expansion following injection during the forward stroke, and must therefore necessarily have a larger cross section for the secondary chamber to accomplish a decreasing pressure during the transfer process.

In order to avoid split-up in transfer flow and cold-gas bleedoff during the reverse stroke, e.g. with the configuration of FIG. **10**, a separate cold-gas storage chamber can be employed. Such a chamber would be filled during the forward stroke, and emptied during the full reverse stroke. Overflow from this additional chamber to the primary and secondary chambers during the reverse stroke (with decreasing pressure in the last chambers) can be prevented with help of a check-valve arrangement. The additional chamber could be formed by help of an extension of the piston and an additional cylinder. Such devices will be briefly discussed with regard to FIG. **13**, and are within the spirit and scope of the invention.

FIG. **11** shows an open-cycle refrigerator similar to that of FIG. **10**, but with the basic difference that the piston assembly is divided in two parts, with a phase difference in motion. This is accomplished with the two cranks **26** that have individual connecting rods **24**. The part of the piston assembly that reciprocates in the primary chamber **12** is in an earlier phase than the piston in the secondary chamber **14**. With this device, cold-gas bleedoff through valve **180** can take place during the first portion of the reverse stroke of piston **16***a* (in the primary chamber), and the transfer flow can be initiated when the volume of chamber **14** is minimum, i.e.

the flow into chamber **14** can be made in a reversible fashion. Proper initiation of the transfer flow is assured by valve **192**, which in a conventional way could be guided by the angular position of the rotating shaft **22**.

The types of refrigerators shown in FIG. **10** and FIG. **11** could be driven by pressurized working gas, or by a combination of pressurized gas and mechanical power into the rotatable shaft **22**. With no mechanical energy supplied to the rotatable shaft **22**, the working gas is then expanded when flowing through the refrigerator (the recompression could be eliminated), and the exit pressure is smaller than the inlet pressure. The crank mechanism of the rotating shaft is then used to control the length of the piston stroke, to guide the valves (proper opening and closing characteristics), and possibly also to provide a connection with a flywheel device for smooth reciprocating motion of the piston assembly.

FIG. **12** is an example of an open-cycle refrigerator, driven by pressurized working gas, with the refrigeration load applied to a heat exchanger, and employing a two-step expansion process. The inlet pressure in line **194** is higher than the exit pressure inline **196**, here fed to the ambient atmosphere. No mechanical energy is supplied to shaft **22**, and mechanical energy may instead even be extracted through the shaft, or dissipated in an arbitrary fashion. An optional counterflow heat-exchanger **198** is used for an initial cooling process, in which the incoming gas (line **194**) transfers heat to the outgoing gas (line **196**). The refrigeration load is applied to exchanger **30**, thus implementing the refrigeration. A third working chamber **200** is employed for the second expansion step in the transfer from the secondary chamber **14** to this chamber. Therefore, the effective cross-sectional area of the face of the piston **16b** in chamber **200** is larger than in the secondary chamber **14**. An additional valve **202** (e.g. controlled by the angular position of shaft **22**) is used for the control of the second transfer flow into chamber **200**. The geometry of the refrigerator is preferably adjusted such that the gas pressure in chamber **200** is identical with the ambient pressure (the pressure in the exhaust line **196**) after the final transfer flow to chamber **200**.

Similar to the previous embodiments, the matrix of the regenerator has a decreasing temperature profile, going from right to left in FIG. **12**. This temperature profile is established and maintained so that the gas pressure in the transfer flow process is lower than during gas injection into the primary chamber. Basically, the cooling in this refrigerator takes place in two steps, first in the counterflow heat-exchanger **198**, and thereafter in the regenerator **28**. The decreasing temperature profile in the heat exchanger **198** (following the inlet gas) is maintained with help of the gas cooling from adiabatic expansion in the second transfer flow from the secondary chamber **14** to chamber **200**. The function of valve **202** is to provide an open path between chambers **14** and **200** during the forward stroke of the piston. The function of valve **58** is to provide an open path between chamber **200** and the exit channel **196** during the reverse stroke.

The average residence time of gas in the refrigerator of FIG. **12** corresponds to nearly two complete piston cycles: injection into the primary chamber **12** in the first forward stroke, transfer to the secondary chamber **14** in the first reverse stroke, transfer to the additional expansion chamber **200** in the second forward stroke, and finally exhaust from chamber **200** to line **96** in the second reverse stroke.

The refrigerator of FIG. **12** could be modified in a large number of ways. One modification could be elimination of both the heat exchanger **30** and the regenerator **28**, but with addition of one valve. This valve would provide cold-gas bleedoff from the primary chamber. The cooling in such a refrigerator would only be in one step, due to the transfer flow from the secondary chamber **14** to the additional chamber **200**, and the primary chamber **12** used for pumping cold gas. Briefly, the operation of the refrigerator would involve the following steps: filling of the primary chamber **12** during the first forward stroke, filling of the secondary chamber from line **194** with help of valve **56** (this valve now has a different function) during the first reverse stroke, and cold-gas bleedoff from chamber **12** through the mentioned valve addition during the same stroke, transfer flow under decreasing pressure from the secondary chamber to chamber **200** during the secondary forward stroke (by help of valve **202**), new filling of the primary chamber **12** during the same stroke, and finally exhaust from chamber **200** to line **196** and cold-has bleedoff during the second reverse stroke. Since this refrigerator would operate with full gas pressure of the source in chamber **14** after completed reverse stroke, the pressure ratio for the device should be smaller than for the first described device, or alternatively the injection from the source (line **194**) into chamber **14** terminated before completion of the reverse stroke, in analogy with schemes discussed for the engine of the invention.

FIG. **13** shows a closed-cycle refrigerator with two-step cooling and three work chambers. The plenum-cooler is not shown, nor are the rotatable shaft and crank mechanisms, that could be used to provide the proper reciprocating piston motion. Two regenerators **28** and **204** are used, and an additional chamber **206** formed by the reciprocating piston and a cylinder **207** with a larger effective cross-sectional area than cylinder **209** (forming the cold chamber **12**).

Chambers **14** and **206** could be made with the same constant-diameter cylinder and hence with the corresponding piston portions of the same diameter, making the mechanical construction considerably simpler. The operation of such a device necessitates that the injection into chamber **12** be terminated before the completion of the forward stroke, and is also characterized by an increase in pressure during the transfer flow to chamber **14**.

In understanding the operation of this device, the regenerator could be considered to operate in conjunction with the larger primary chamber **206** and provide the first-step cooling, with the refrigeration load being "second" step (heat exchanger **30**, etc.). Hence, the colder primary chamber **12** could be considered to be the primary chamber in a refrigerator with chamber **206** as the secondary chamber.

To adapt this refrigerator for use as an open system with cold-gas bleedoff with expenditure of gas from the gas source in line **54**, the heat exchanger **30** would be eliminated and a bleed valve introduced near the chamber **12**. If the bleed from chamber **12** takes place

during the full reverse stroke, the regenerator **28** could be eliminated, and a valve introduced in its place to avoid backflow from the chamber **12** when transfer flow takes place with gas from the chamber **206** to chamber **14**, in analogy with the scheme discussed for the modified refrigerator of FIG. **12**. In this case the two-step cooling is lost, and cooling of the bleed gas performed in regenerator **204** alone. A scheme of this kind could be of great interest for heat-pump applications, with simultaneous production of cold gas (from a bleed channel) and hot gas through the exhaust line **52**.

FIG. **14** shows schematically a refrigerator with three-step cooling using three "primary" chambers arranged in parallel. This refrigerator while shown as a system with no cold-gas bleedoff can readily be adapted for bleedoff operation by techniques already discussed. The use of a plenum-cooler and closed operation or open-cycle operation without a cooler is not shown. The arrangement of FIG. **14** with the cold chambers in parallel could be advantageous e.g. to facilitate sealing of the chambers; the seals could be made to operate e.g. at near room temperatures.

In FIG. **14** the refrigeration load is applied to exchanger **30** near the smallest chamber **12**. Chamber **208** is larger than chamber **12**, and chamber **210** larger than chamber **208**, in order to achieve the correct staging. Generator **214** should preferably be larger than regenerator **212**, which in turn is larger than regenerator **28**.

This type of refrigerator could naturally also be modified for cold-gas bleedoff, as the previous devices. For example, the primary chamber **12** could be solely used for pumping of cold gas, with elimination of regenerator **28** and introduction of necessary valving here, and a valve for the bleedoff. The cooling of the bleed gas would then be in two steps only; first in the regenerator **214** and then in regenerator **212**. Such a device could alternatively be driven by pressurized gas, with a scheme that preferably allows reversible expansion of the cold gas in chamber **12** before the bleed flow. This could be accomplished by terminating the inflow to that chamber by the added valve (at the regenerator position) at the proper phase of the forward stroke. Such an operation would also be suitable with only two primary chambers.

Most of the open-cycle refrigerators could be used for separating condensable substances from the work gas, e.g. separating water from humid air (air dryer). The condensable substance could be drawn from the regenerator or from a convenient point in the primary chamber. FIG. **15***a* shows an embodiment of an air dryer using centrifugal separation of the condensable from the thermal regenerator. The refrigerator is an open-cycle device without cold-gas bleedoff and with the refrigeration load being the heat of vaporization released in the regenerator or primary chamber.

In FIG. **15** *a* the bevelled gear **98**, which is connected to the main rotating shaft, drives shaft **216** at a high speed, and gives the regenerators **28** (two or several) a large radial acceleration. A "distributor" box is shown schematically, transferring gas from the rotating lines **220** to the stationary line **222**, etc. the liquid condenses in the regenerators **28**, and is forced to leave the regenerators by the centrifugal force, through suitably small orifices, or through separate check valves **224** (shown in the figure).

Among many possible configurations , a preferred embodiment of the regenerators could be circular cylindrical tubes with flat end plates, filled with a large number of wire mesh screens, or metal sheets, perforated with small holes for the reciprocating gas flow. The check valves **224** could preferably be adjusted to open only by centrifugal forces from condensed liquid in the regenerator, even though the gas pressure is varying over the cycle. Other release systems for the condensable are possible, e.g. using capillary forces to pump the condensable.

The open-cycle refrigerators for separation of condensables could be modified to include counterflow heat-exchangers (with valve modifications) instead of the thermal regenerators of conventional kind. The use of counterflow exchangers with one-way legs could circumvent difficulties with re-vaporization of condensed liquid in the common regenerator, e.g. consisting of wire mesh or small rounded pebbles, when the gas is transferred from the primary to secondary chambers. Such schemes are shown and discussed in connection with FIG. **15***b*–FIG. **15***f*.

FIG. **15***b* shows the normal location of the thermal regenerator of suitable kind, in relation to the valve **56**. FIG. **15**c shows one possible modification using the counterflow heat-exchanger **228**, with two valves **230** and **232** replacing the single valve **56** (FIG. **1**). Proper operation of these valves guarantees the flow in the upper leg always to be oriented in the left direction, and in the lower leg in the right direction.

FIG. **15***d* shows how valve **230** can be moved to the left side of the heat exchanger (generalized regenerator). Such an arrangement does not introduce the volume of the upper channel in the heat exchanger as dead-space, but has the drawback that the valve **230** could operate in a low-temperature region.

FIG. **15***e* shows still another scheme with two check valves **234** and **236**, that insure correct one-way flows, and also the valve **56** of the 3-port 2-way type. The check valves could obviously be located on the other side of the exchanger (or one valve on each side of the exchanger).

Finally, FIG. **15***f* shows a combination of a regenerator of common kind and a counterflow heat-exchanger, with the two valves **230** and **232** located one on each side of the heat exchanger.

When applied to devices for condensable separation (e.g. air dryer), the liquid drain could preferably be placed in the upper leg of the counterflow exchangers shown in FIGS. **15***c*–**15***f*. Drain channels could also be placed in the low legs, but would be less efficient.

While the invention has been described by reference to preferred embodiments, it should be apparent that numerous changes could be made within the spirit and scope of the invention disclosed herein. Accordingly, the invention is not intended to be limited except by the language of the following claims.

I claim:

1. In a gas compressing and expanding device for use as an engine or refrigerator,

primary and secondary gas work chambers,

piston assembly means having first and second piston portions mounted for reciprocating motion in said primary and secondary chambers respectively, said piston portions being connected to each other so that with motion thereof, the effective volumes

of said chambers are varied in substantially inverse relationship,

a source of gas,

means for feeding gas from said gas source to the primary chamber during a first period of time corresponding to a predetermined percentage of the forward stroke of said piston assembly means,

means for providing a fluid communications path between said primary and secondary chambers to transfer gas therebetween during a second period of time including at least a portion of the time of a predetermined reverse stroke of said piston assembly means, the average pressure of the gas when transferred being lower than when fed to the primary chamber,

said means for feeding gas to the primary chamber and said means for providing a fluid communications path between the primary and secondary chambers including common channel means comprising a thermal regenerator having a connection for each end thereof, one of said regenerator connections being coupled to said primary chamber, and valve means interposed between the other of the regenerator connections and the source of gas and the other of the regenerator connections and the secondary chamber for connecting said source of gas to said regenerator during said first period of time and said regenerator to said secondary chamber during said second period of time,

means for causing heat exchange with said gas, and

means for releasing gas from the secondary chamber during at least a portion of the forward stroke.

2. The device of claim 1 wherein the first period of time corresponds to substantially 75 percent of the forward stroke.

3. The device of claim 1 wherein the first period of time corresponds to 25–100 percent of the forward stroke.

4. The device of claim 1 wherein said second period of time commences when the forward stroke has been completed.

5. The device of claim 1 wherein said means for releasing gas from said secondary chamber comprises valve means for venting the gas to the ambient atmosphere.

6. The device of claim 1 wherein said means for releasing gas from said secondary chamber comprises a plenum chamber and means for feeding the gas to the plenum chamber, said source of gas comprising the gas in said plenum chamber.

7. The device of claim 1 which is a refrigerator wherein is included means for driving said piston assembly means to cause the expansion and compression of the gas in said chambers, said means for causing a transfer of heat to said gas comprising a refrigeration heat load.

8. The device of claim 1 which is an engine, said means for causing heat exchange with said gas including a heater device.

9. The device of claim 8 wherein said source of gas comprises a gas filled plenum chamber and means for cooling the gas in said chamber.

10. The device of claim 1 wherein said source of gas comprises the ambient atmosphere.

11. The device of claim 9 wherein said gas is air.

12. The device of claim 1 wherein said piston assembly means comprises a central portion interconnecting said first and second piston portions, said piston portions moving reciprocally parallel to a predetermined axis.

13. The device of claim 9 wherein the volume of said plenum chamber is substantially larger than that of said primary and secondary chambers.

14. The device of claim 9 wherein said cooling means comprises ambient air.

15. The device of claim 1 wherein said piston assembly means comprises a rotatably supported central shaft portion, said first and second piston portions comprising vane members extending radially from said central shaft portion, said work chambers being formed by a cylindrical member enclosing said piston portions.

16. The device of claim 1 wherein said means for causing heat exchange with said gas further includes a heat exchanger.

17. The device of claim 7 wherein said primary gas work chamber is divided into a plurality of separate sections of different cross-sectional areas, said first piston portion including a separate piston member for each of said chamber sections.

18. The device of claim 7 and further including a third gas work chamber positioned between said primary and secondary work chambers.

19. The device of claim 7 wherein said heat transferring means includes a regenerator and condenser means attached to the regenerator for separating condensable liquid from the working gas.

20. In a hot-gas engine,

primary and secondary gas work chambers,

piston assembly means having first and second piston portions mounted for reciprocating motion in said primary and secondary chambers respectively, said piston portions being connected to each other so that with motion thereof, the effective volumes of said chambers are varied in substantially inverse relationship,

a source of gas,

external heat source means for applying heat energy to said gas,

means for feeding the gas to the primary chamber during a first period of time corresponding to a predetermined percentage of the forward stroke of said piston assembly means,

means for providing a fluid communications path between said primary and secondary chambers during a second period of time including at least a portion of the time of a predetermined reverse stroke of said piston assembly means, thermal regenerator means forming a common channel interposed between said source of gas and said primary chamber for applying heat to said gas as it flows to said primary chamber and interposed in said path between said primary and secondary chambers for withdrawing heat from the gas as it flows from the primary to the secondary chamber,

valve means interposed between the regenerator and the source of gas and the regenerator and the secondary chamber for connecting said source of gas to said regenerator during said first period of time and said regenerator to said secondary chamber during said second period of time,

means for releasing gas from the secondary chamber during at least a portion of the forward stroke, and

output drive shaft means coupled to said piston assembly means.

21. The device of claim 20 wherein the first period of time corresponds to 25–100 percent of the forward stroke.

22. The device of claim 20 wherein the first period of time corresponds to substantially 75 percent of the forward stroke.

23. The device of claim 20 wherein the second period of time commences when the forward stroke has been completed.

24. The device of claim 20 wherein said piston assembly means comprises a central portion interconnecting said first and second piston portions, said piston portions moving reciprocally parallel to a predetermined axis.

25. The device of claim 20 wherein said means for feeding the gas to the primary chamber and said means for providing a communications path between the chambers includes valves operatively responsive to the shaft means.

26. The device of claim 20 wherein said piston assembly means comprises a rotatably supported central shaft portion, said first and second piston portions comprising vane members extending radially from said central shaft, said work chambers being formed by a cylindrical member enclosing said piston portions and having a separator portion dividing said cylindrical member into said primary and secondary chambers.

27. The device of claim 20 wherein said means for applying heat to said gas further includes a heat exchanger.

28. The device of claim 27 wherein said source of gas includes a plenum chamber filled with gas and cooler means for cooling the gas in said plenum chamber, said means for releasing gas from said secondary chamber comprising means for interconnecting the secondary chamber and the plenum chamber.

29. The device of claim 20 and further including a third gas work chamber positioned between said primary and secondary chambers.

30. The device of claim 20 wherein said piston assembly means comprises a central portion interconnecting said piston portions for reciprocal motion in said chambers, said output drive shaft means being coupled to said central portion.

31. The device of claim 20 and further including a plurality of additional primary and secondary chambers and piston assembly means operating similarly to said first mentioned chambers and piston assembly means, the piston portions of said various piston assembly means all being coupled to said output drive shaft means, said piston portions being phased relative to each other to provide rotatable drive of said drive shaft means.

32. In a refrigerator,

primary and secondary gas work chambers,

piston assembly means having first and second piston portions mounted for reciprocating motion in said primary and secondary chambers respectively, said piston portions being connected to each other so that with motion thereof, the effective volumes of said chambers are varied in substantially inverse relationship,

means for driving said piston portions reciprocally in said chambers,

a source of gas,

means for providing a refrigeration heat load,

means for feeding said gas to said primary chamber during a first period of time corresponding to a predetermined percentage of the forward stroke of the piston assembly means,

means for feeding said gas from said primary chamber to said refrigeration heat load means and to said secondary chamber during at least a portion of the reverse stroke of said piston assembly means,

said means for feeding gas to said primary chamber and said means for feeding gas from said primary chamber to said secondary chamber including common channel means formed by a regenerator having a connection for each end thereof, one of said regenerator connections being coupled to said primary chamber, and valve means interposed between the other of the regenerator connections and the source of gas and the other of the regenerator connections and the secondary chamber for connecting said source of gas to said regenerator during the first period of time and said regenerator to said secondary chamber during the second period of time, and

means for releasing gas from said secondary chamber during at least a portion of the forward stroke.

33. The refrigerator of claim 32 wherein said means for feeding the gas from the primary chamber comprises means for feeding gas to said refrigeration heat load means during a first portion of the reverse stroke and means for feeding the gas to the secondary chamber during a second portion of the reverse stroke.

34. The device of claim 32 wherein said primary gas work chamber is divided into a plurality of separate sections of different effective cross-sectional areas, said first piston portion including a separate piston member for each of said chamber sections.

35. In a refrigerator,

primary and secondary gas work chambers,

piston assembly means having first and second piston portions mounted for reciprocating motion in said primary and secondary chambers respectively, said piston portions being connected to each other so that with the motion thereof, the effective volumes of said chambers are varied in substantially inverse relationship,

means for driving said piston portions reciprocally in said chambers,

a source of gas,

means for feeding said gas to said primary chamber during a first period of time corresponding to a predetermined percentage of the forward stroke of the piston assembly means,

means for feeding a first portion of said gas from said primary chamber to external means to be cooled and a second portion of said gas to said secondary chamber during at least a portion of the reverse stroke of the piston assembly means,

said means for feeding gas to said primary chamber and said means for feeding gas from said primary chamber to said secondary chamber including common channel means formed by a regenerator having a connection for each end thereof, one of

said regenerator connections being coupled to said primary chamber, and valve means interposed between the other of the regenerator connections and the source of gas and the other of the regenerator connections and the secondary chamber for connecting said source of gas to said regenerator during the first period of time and said regenerator to said secondary chamber during the second period of time, and

means for releasing gas from said secondary chamber during at least a portion of the forward stroke.

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