METHOD FOR OBTAINING MECHANICAL ENERGY FROM A THERMAL GAS CYCLE WITH LIQUID PHASE COMPRESSION

Filed Nov. 26, 1965 3 Sheets-Sheet 1

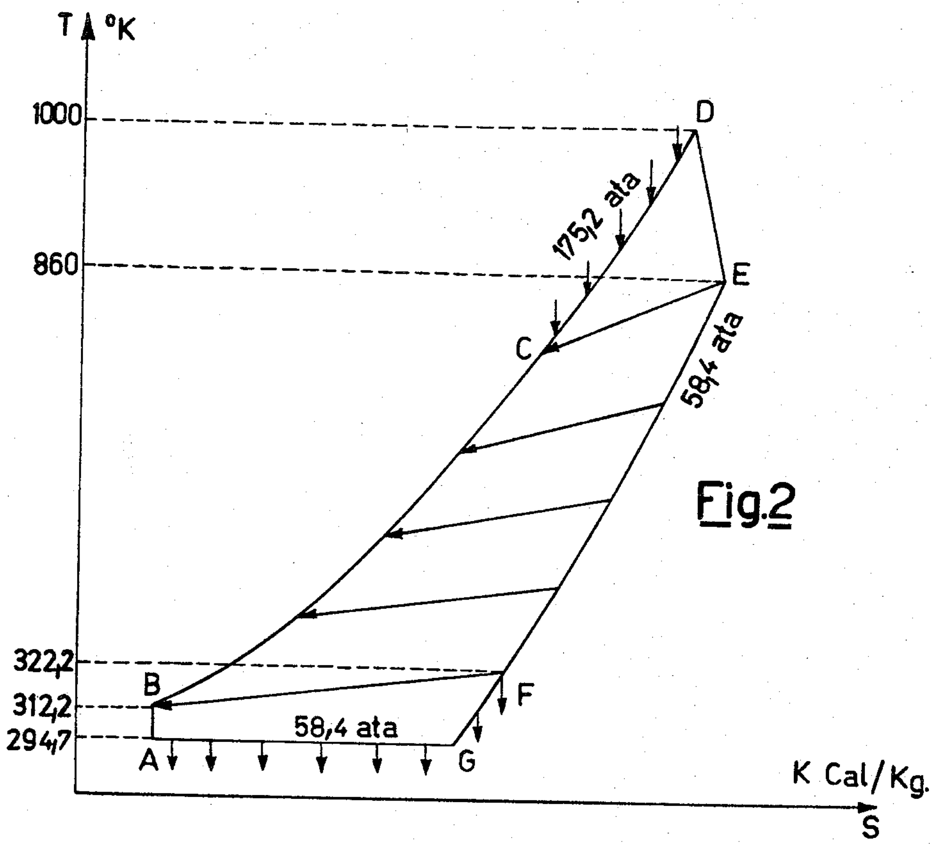


Fig.2

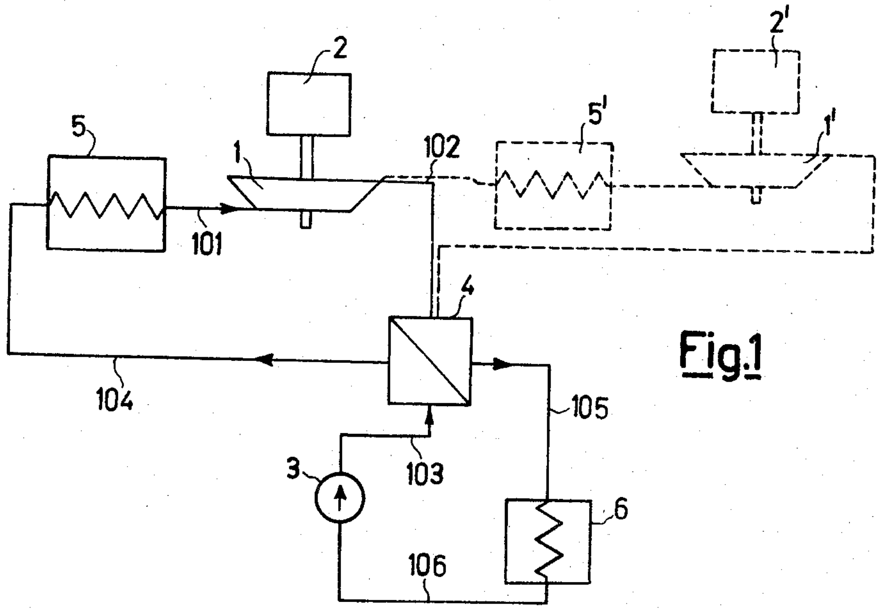
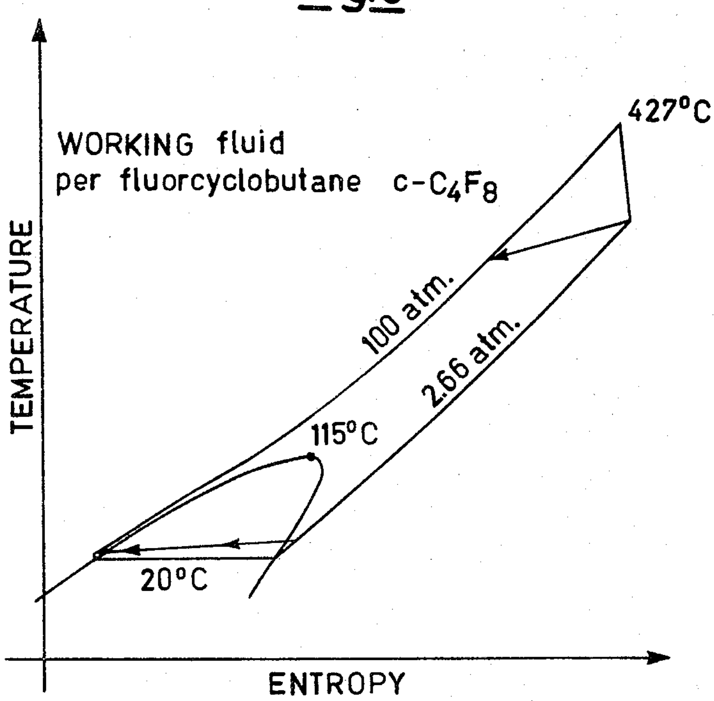


Fig.1

Fig.3


WORKING fluid
per fluorcyclobutane c-C_4F_8
TEMPERATURE
ENTROPY
427°C
100 atm.
2.66 atm.
115°C
20°C

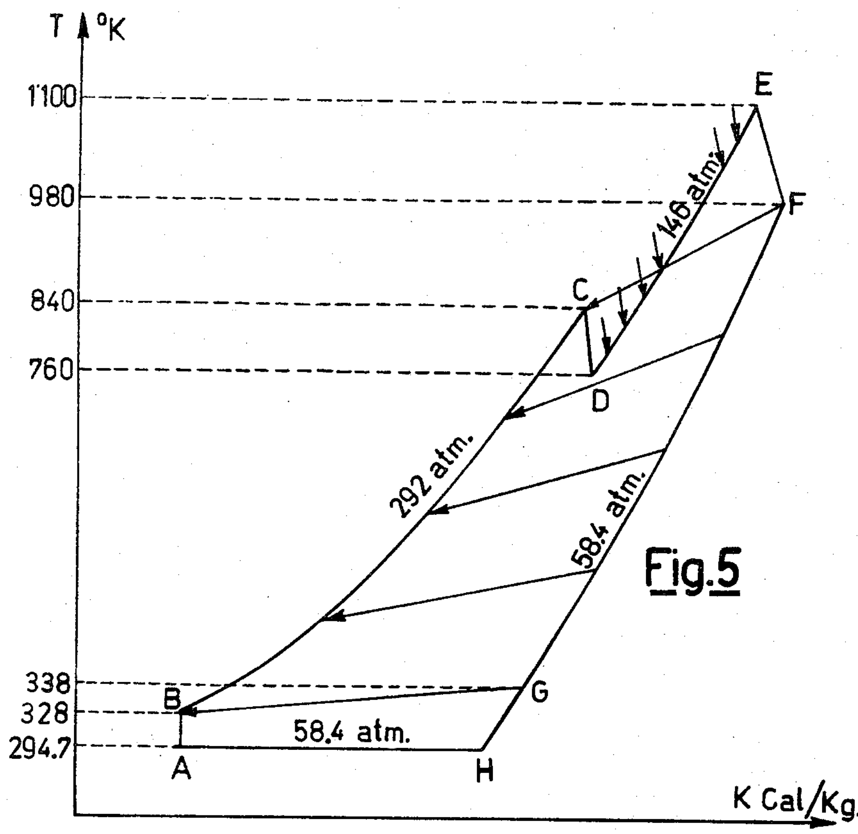

T
°K
1100
980
840
760
338
328
294.7
E
F
C
D
146 atm.
292 atm.
58.4 atm.
B
G
A
H
58.4 atm.
K Cal/Kg.

Fig.5

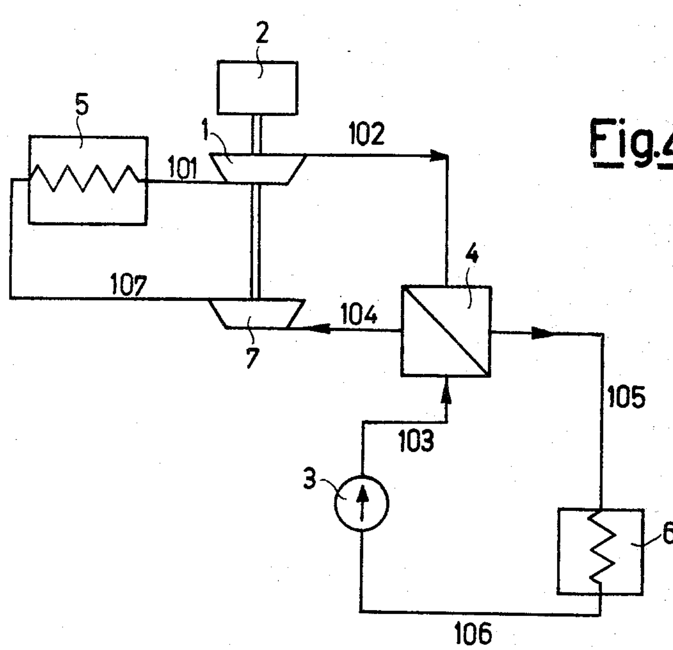

2
5
1
101
102
4
107
104
7
105
103
3
6
106

Fig.4

United States Patent Office 3,376,706
Patented Apr. 9, 1968

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3,376,706
METHOD FOR OBTAINING MECHANICAL ENERGY FROM A THERMAL GAS CYCLE WITH LIQUID PHASE COMPRESSION

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2 Claims. (Cl. 60—73)

ABSTRACT OF THE DISCLOSURE

Fluid is compressed above its critical pressure and then heated above its critical temperature and then expanded as a working gas such that after expansion it is still above its critical temperature. The gas is then used in a heat exchanger to heat further gas and is refrigerated to liquid state and recycled. Additional engines are driven in intermediate phases.

The present invention relates to the obtaining of mechanical energy from an expanding gas which is compressed in liquid phase after a refrigerated condensation and is then expanded after having been heated; the peculiarities of the cycle according to which this device operates yield high efficiencies in the conversion of thermal into mechanical energy.

To illustrate the characteristics of the present invention a detailed description is given of how the same can be carried into practice.

In the drawing:

FIG. 1 is a diagrammatic representation of a power plant;

FIG. 2 represents the cycle according to which the plant of FIG. 1 operates in accordance with the invention;

FIG. 3 is another operating cycle in accordance with the invention;

FIG. 4 illustrates a modified plant; and

FIG. 5 illustrates the cycle for FIG. 4.

With reference to FIG. 1, a turbine **1** moves a device **2** utilizing mechanical energy. A working fluid, in gaseous state, is admitted to the turbine through duct **101** and is exhausted through duct **102**.

The desired level of pressure for the working fluid is higher than its critical pressure and is provided by pump **3** which begins the compression of the fluid in liquid phase, then sending it through duct **103** to the heat exchanger **4** where it is heated by the gas exhausted by the turbine **1** through duct **102**. The quantity of heat exchanged in the exchanger **4** is greater than the equivalent of the mechanical work performed by the turbine **1**. From heat exchanger 4 the fluid is sent through duct **104** to the heater **5** which further raises its temperature by means of thermal energy supplied by an external source.

The gas exhausted by the turbine, after passing through heat exchanger **4**, is sent through duct **105** to the refrigerator **6** where its temperature is further lowered to the condensation temperature and where the total condensation of the fluid takes place. The fluid in liquid phase is then sucked by pump **3** through duct **106** and discharged through duct **103** beginning in this way a new cycle.

If the working fluid is carbon dioxide (CO_2) it is convenient to follow a cycle of the kind represented in FIG. 2 where on the axis of the abscissa is slower the fluid specific entropy and where on the axis of the ordinates is reported the fluid absolute temperature in degrees Kelvin.

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According to this diagram, the working gas which at point A is in the liquid state at a temperature of 294.7° K. and at a pressure of 58.4 absolute atmospheres is compressed by pump **3** to a pressure of 175.2 atm. or to a higher pressure if it is desired to take into account pressure losses along the ducts, the fluid being heated on account of this compression to a temperature of 312.2° K. as represented by point B.

Then the working gas enters the heat exchanger **4** where an isobaric heating takes place the gas being discharged by the heat exchanger under the conditions defined by point C at a temperature of about 720° K.; the gas is then sent to the heater **5** where it is isobarically heated to the temperature defined by point D which in the present example is about 1000° K. The gas is then admitted to the turbine from which it is discharged under the conditions defined by point E; the gas then expands to the condensation pressure of 58.4 atm. while temperature drops to about 860° K. The gas leaving turbine **1** is sent to heat exchanger **4** which in this example is of the counterflow type where it undergoes an isobaric cooling to the temperature of 322.2° K. In this way it is admitted that the minimum temperature difference in the heat exchanger between the hot and the cold fluid flow is 10° C.; the maximum temperature difference is given by the difference of the temperatures at points E and C and turns out to be about 140° C. The gas, when discharged by heat exchanger **4** is under the conditions defined by point F. The gas is further cooled in the refrigerator **6** to dew point which is indicated by point G and then is condensed at the constant pressure of 58.4 atm., its final conditions being represented by point A. An external refrigerant removes the heat rejected by the gas during these transformations.

It is possible to compute the efficiency of the cycle performed by the present device in order to show more clearly the main advantages which it offers. To carry out the computations it is assumed that the efficiency of turbine **1** defined as the ratio of work actually produced to the work produced in an isentropic expansion is 0.88; similarly the efficiency of pump **3** is assumed to be 0.75; it is further admitted that it is possible to take into account pressure drops along the whole circuit by assigning the pump a head 20% higher than required when pressure losses are absent.

Under these assumptions the working gas expands through the turbine from 175.2 to 58.4 atm. producing a theoretical work of 46.7 kcal./kg. which corresponds to an actual work of 41.1 kcal./kg.

Similarly it is possible to compute the actual pump work which turns out to be 5.85 kcal./kg. with a resulting temperature rise of 17.5° K. In this case, pressure at point B is 198 atm.; the exceeding pressure with respect to the ideal case value is supposed to be spent in overcoming all pressure losses of the circuit. Useful work, difference between actual turbine work and the actual pump work is 35.25 kcal./kg.

Heat rejected from point F to point G is 13.3 kcal./kg.; then follows a constant pressure condensation G–A by which the gas rejects to the refrigerant the heat of condensation which is 35.0 kcal./kg. Since the condensation temperature is 21.7° C. the refrigeration may be accomplished by water at a temperature of 10–15° C.

Heat rejected on the whole is 48.3 kcal./kg.; from the first law of thermodynamics head added from external sources to the working gas in heater **5** is given by the sum of the heat rejected from point F to point A and useful work in thermal units as above indicated; it turns out to be 83.55 kcal./kg. The device efficiency, ratio of useful work to heat absorbed from external sources is about 42%.

Carbon dioxide is by no means the only fluid suited for application in the present cycle. As it appears from the above example it requires very high pressures and temperatures to yield elevated efficiencies. Instead, organic fluids like hydrocarbons, fluorocarbons, ethers, fluoroethers substituted hydrocarbons and other fluoro-compounds yield in many cases higher efficiencies with lower turbine inlet temperatures and pressures.

Many of the aforementioned compounds have a sufficient thermal stability for practical application in the energy production industry including nuclear power plants. Besides thermal stability, the fluid selected for the proposed cycle must have a critical temperature higher than the refrigerant temperature by at least 15° C. and lower by at least 80° C. than the maximum temperature of the cycle. The fluid must furthermore be non-corrosive up to the maximum operating temperature. For instance a cycle performed by the present device and employing perfluorocyclobutane as working fluid at a maximum pressure of 100 atm. and a maximum temperature of 427° C. yields an overall efficiency of about 38% which compares favourably with the efficiencies of the best plants working under similar conditions. The aforesaid cycle is shown in FIG. 3 in a temperature-entropy diagram. A typical cycle is shown as may be performed employing high molecular weight organic fluids.

Accounting for the elevated efficiency of the present device are the particular organization of the various components and the configuration of the thermodynamic cycle. In particular a favourable influence on the efficiency results from the fact that heat begins to be added to the working gas only at a very high temperature which, in the above example, is about 720° K. as indicated by point C.

The described cycle substantially differentiates itself from the usual steam cycle (Rankine cycle) because in the first one the turbine expansion takes place at temperatures that are always higher than the critical temperature and because, after the expansion is accomplished, a considerable amount of heat is recovered, whereas in the Rankine cycle the turbine expansion takes place mainly at temperatures lower than the critical value and, after the expansion is accomplished, no heat regeneration is performed.

It differentiates itself also from the regenerative gas (Brayton) cycle because a process of condensation is present in the first one which is absent in the latter. As a consequence, the large size compressor present in the ordinary gas cycles is substituted by a small size pump with considerable benefits for the plant efficiency and capital costs.

The value of the temperature at the end of the expansion which must be fairly high for a good efficiency is dependent on the thermodynamics characteristics of the working fluid; it may be raised by means of a fractionized expansion with reheating after each expansion except the last one. How the organization of the various components is modified by reheating is shown as an example in FIG. 1 (components **5'**, **1'**, **2'** drawn with dotted lines).

It is possible to realize another variant of the previously described device presenting some construction and efficiency advantages under specified working conditions.

Having this in view a new engine capable of producing mechanical work under the action of an expanding gas is introduced in the diagram of FIG. 1. The resulting diagram is represented in FIG. 4 which differs from the diagram of FIG. 1 only on account of the addition of engine **7** and duct **107**. The compressed fluid discharged by pump **3** after being heated through exchanger **4** acts directly on engine **7** in which a first expansion takes place. Through duct **107** it is then transferred to heater **5** where it is heated by means of external energy. It then reaches engine **1** where it completes its expansion to the cycle minimum pressure. Heat is then transferred to the cold fluid discharged by the pump through exchanger **4** and finally the working gas is desuperheated and condensed in refrigerator **6**. The liquid is recirculated by pump **3**. FIG. 5 shows summarily a carbon dioxide cycle which may be performed by the device of FIG. 4. It differs from the cycle of FIG. 2 on account of the fact that the working gas leaving the heat exchanger (point C) before being heated by means of external energy (from point D to point E) undergoes a first expansion which produces useful work. In this way the cycle maximum temperature is not concomitant with the maximum pressure but with a considerably lower pressure with an evident relief for material problems. In the example of FIG. 5 to the maximum temperature of 1100° K. corresponds a pressure of 146 atm. in place of the maximum pressure of 292 atm. The cycle efficiency, computed under assumptions similar to those employed in the computation of the cycle of FIG. 2 turns out to be 46.5%. Removing engine **7** for operation according to a simple cycle with a maximum pressure of 146 atm. and a maximum temperature of 1100° K. efficiency would have been only 42.8%.

What is claimed is:

1. A method of converting thermal energy into mechanical energy by means of an engine capable of producing mechanical work under the action of an expanding gas, a pump, a heat exchanger, a heater and a refrigerator, said method comprising conveying a fluid in liquid phase to said pump and compressing the fluid to a pressure higher than its critical pressure; and conveying the fluid via the said pump to said heat exchanger, and subsequently to said heater, heating said fluid in said heat exchanger and heater to a temperature higher than the critical temperature of the fluid, expanding said fluid in the gaseous state and as a working gas in the aforesaid engine, the first said temperature being such that the temperature of the working gas after the expansion is still higher than its critical temperature, passing the gas exhausted from the aforesaid engine through said heat exchanger to exchange a quantity of heat greater than the equivalent of the work developed by the engine, the gas undergoing a further refrigeration and a condensation in said refrigerator from which it is forwarded in liquid phase to said pump, the gas discharged by the said engine being heated in an additional heater by means of external energy being then fed to an additional engine for producing additional mechanical power and being subsequently forwarded to the first said heat exchanger.

2. A method of converting thermal energy into mechanical energy by means of an engine capable of producing mechanical work under the action of an expanding gas, a pump, a heat exchanger, a heater and a refrigerator, said method comprising conveying a fluid in liquid phase to said pump and compressing the fluid to a pressure higher than its critical pressure, and conveying the fluid via the said pump to said heat exchanger, and subsequently to said heater, heating said fluid in said heat exchanger and heater to a temperature higher than the critical temperature of the fluid, expanding said fluid in the gaseous state and as a working gas in the aforesaid engine, the first said temperature being such that the temperature of the working gas after the expansion is still higher than its critical temperature, passing the gas exhausted from the aforesaid engine through said heat exchanger to exchange a quantity of heat greater than the equivalent of the work developed by the engine, the gas undergoing a further refrigeration and a condensation in said refrigerator from which it is forwarded in liquid phase to said pump, the fluid heated in said heat exchanger being conveyed to a further engine capable of extracting work from the expanding gas before sending the fluid to said heater.

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