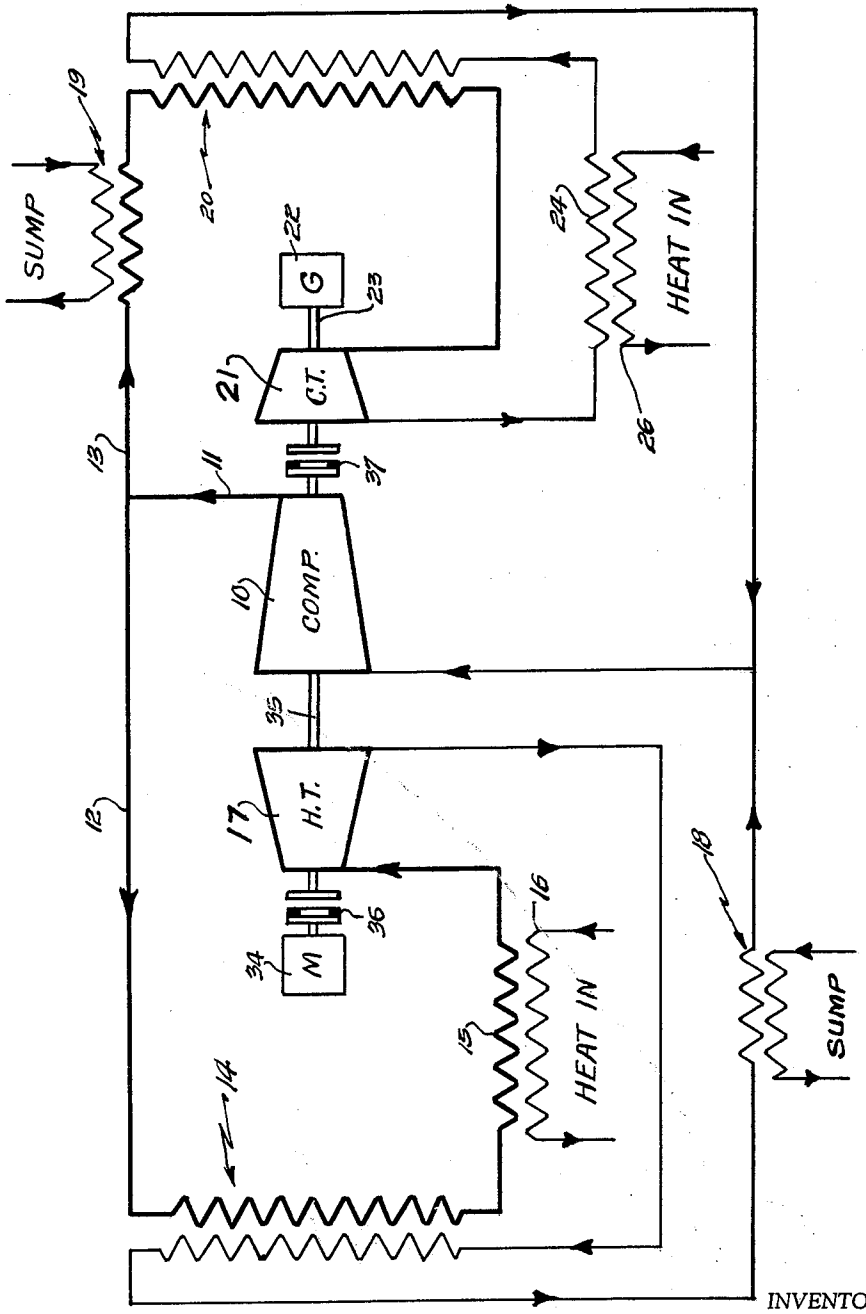
July 13, 1965 J. K. LA FLEUR 3,194,026

POWER-REFRIGERATION SYSTEM

Filed Oct. 24, 1963



INVENTOR:
JAMES K. LA FLEUR
BY Paul Bliven
ATTORNEY.

United States Patent Office 3,194,026 Patented July 13, 1965

3,194,026

POWER-REFRIGERATION SYSTEM

James K. La Fleur, Hermosa Beach, Calif., assignor to The La Fleur Corporation, Los Angeles, Calif., a corporation of California

Filed Oct. 24, 1963, Ser. No. 318,564
37 Claims. (Cl. 62—88)

The present application is a continuation-in-part of my application Serial Number 87,311, filed Feb. 6, 1961, for Method and Apparatus for Separation of Air and Gaseous Mixtures, and the disclosure of such prior application is incorporated herein by reference thereto.

This invention relates to a new closed power-refrigeration system in which a gas is the common working medium in both a power cycle and a refrigeration cycle, that is, such cycles are interconnected so that the common working medium circulates between both cycles of the system. Alternatively, this invention may be viewed as a system comprised of a closed gas turbine cycle having connected thereto a closed refrigeration cycle obtaining its motivation by bleeding a portion of the common working medium from and returning such medium to the power cycle. Further novel features of the present invention are the use of a common compressor for the power and refrigeration cycles; the use of a hot turbine in the power cycle and a cold turbine in the refrigeration cycle, with both such turbines feeding power to the common compressor that provides compressed working medium gas to both cycles of the system; the use of a working medium having a critical point at or below the cryogenic temperature to be achieved by such refrigeration; and the attainment of cryogenic temperatures to the order of 2° Rankine with the employment of a single expansion of the refrigerant working medium which remains a gas thruout both cycles, and to effect such single expansion over a narrow, critical, pressure ratio that provides maximum cycle efficiency.

The present invention has particular application to and involves the liquefaction of air and other gases such as helium and hydrogen, gases having very low critical points; and has particular application in a system for obtaining large heat transfers at cryogenic temperatures. The present system is particularly suitable for economically liquefying many tons of air per day.

The basic elements of the system of the present invention are, in the following order in the power cycle, starting with a compressor for a gaseous working medium, the medium flows from the compressor to and thru a hot regenerator for imparting heat to the medium, thru a heater or furnace for further heating the medium, thru a hot power turbine in which the medium expands, again thru the hot regenerator for removing heat from the medium, and then thru a heat sump to reduce the medium to ambient temperature, and in the power cycle, suitable conduits to link such elements and suitable connecting means so that the hot turbine delivers power to the compressor; and in the following order in the refrigeration cycle, starting with the above same compressor, from such compressor, bleeding some of the working medium from the power cycle to a heat sump to reduce the medium to ambient temperature, then to and thru a cold regenerator for removing heat from the medium, then to and thru a cold turbine for expansion of the medium and further temperature reduction, to and thru a heat source, the refrigeration load, and then back thru the cold regenerator to the compressor common to both power and refrigeration cycles, and the refrigeration cycle having suitable conduits to connect such elements of the refrigeration cycle in such order and suitable connecting means so that the cold turbine delivers power to the compressor.

The prior art processes and apparatuses for the liquefaction of air have been well review by Russell B. Scott in his book "Cryogenic Engineering" published by D. Van Nostrand Company, Inc. Princeton, New Jersey, 1959. This book contains an extensive bibliography. According to present usage, cryogenic engineering is concerned with temperatures below —240° F., (Fahrenheit), or 220° R. (Rankine). Cryogenic gases may be considered as those having critical temperatures below these temperatures.

The early Hampson air liquefier was a simple device in which air was compressed to between 2000 and 3000 pounds per square inch (p.s.i.), its static pressure, cooled to ambient temperature, purified, passed thru a counterflow heat exchanger, or regenerator, expanded into a chamber at atmospheric pressure to obtain the needed refrigeration by the Joule-Thomson effect, passed thru the heat exchanger to cool the incoming air, and then back to the compressor to be again compressed and recycled until liquefaction occurred in the expansion chamber. The major disadvantage of the Hampson process is the requirement for these high pressures and pressure ratios that entail heavy and expensive compressing machinery.

Most of the advances in this field by Linde, Claude, and others have been to reduce the need for such pressures. This has been done by the use of auxiliary or preliminary refrigeration systems to reduce the temperature of the air before it enters the compressor and before it enters the counterflow heat exchanger with the out flowing unliquefied air components; and with the use of multistage expansion as in the Linde system; and with the use of cascades of liquefiers, each using a different refrigerant, as in the cascade air liquefier suggested by Keesom. The use of expansion engines instead of an expansion valve has contributed to further efficiencies in the liquefaction of air. All of such improved systems have added complexity to the simple process and simple equipment of the original Hampson system. Thus, it is one of the objects of the present invention to practice a cryogenic refrigeration process, and machinery therefor, having a large heat capacity and that has the simplicity of a single expansion step with a small pressure ratio as compared with the prior art.

The Hampson simplicity was partially recaptured in the Claude and Kapiza cycles by the use of expansion engines instead of the Joule-Thomson effect. However, all of these systems were open, systems in which the working medium, or refrigerant, was the gas, air, to be liquefied. This is a disadvantage in that contaminants move between the air and the machinery, and air is not the best refrigerant for its own liquefaction.

A closed refrigeration system for the liquefaction of gases has many advantages, such as freedom from atmospheric contamination and in arbitrary choice of static pressures and density of working medium, heat transfer characteristics, and boiling points, or critical temperatures. In a closed refrigeration system, one is not saddled with the characteristics of the gas to be liquefied in determining the working medium for the system, other than that the medium be capable of achieving the desired refrigeration effect.

The Phillips process of refrigeration, while a closed cycle using a regenerator and an expander, has been performed by the use of reciprocating machinery. In that reciprocating machinery is not suited for large plants, the advantages of the Phillips process have never been realized in large scale gas liquefaction where it is desirable to employ a system having a constant volume and a constant pressure ratio. Thus, it is a further object to operate the present process at constant volume and under

constant temperature and pressure ratios. A description of the Phillips system is found in: Handbuch der Physik, Kaltephysik I; Springer 1956, and in United States patent to J. A. L. Ijzer, 2,934,909, May 3, 1960.

Specifically, it is an object of the present invention to devise a simple closed refrigeration cycle particularly adapted for the liquefaction of air, and to use as a working medium in such system one of the gaseous elements having a critical point tempertaure below that of the air components to be liquefied so that such may be carried out at or near atmospheric pressure. Thus, if oxygen and argon are to be liquefied, then nitrogen may be used as the refrigerant working medium. If nitrogen is to be liquefied, then either neon, deuterium, hydrogen, or helium, or combinations thereof, would be used. Further, in view of the use of a refrigerant medium having a critical temperature below that of the gases to be liquefied, it becomes possible to operate the refrigeration cycle so that the working medium is above its condensation point thruout the cycle; and it is an object of the present invention to so operate a refrigeration cycle.

Further, hydrogen and helium have the lowest boiling points of the cryogenic gases. Other desirable features of using these gases are heir greater heat content and their higher heat transfer coefficients as compared with air. The specific heat of helium is about five times and that of hydrogen about thirteen times that of air. Thus, with a closed cycle using either helium or hydrogen as the working fiuid, substantial savings in equipment may be made as compared with the use of air as the working medium. This use of helium or hydrogen as the working medium, and the liquefaction of air at atmospheric pressures, as compared with the use of refrigerants which would require the liquefaction to take place at elevated pressures, effects a very considerable saving in plant and opertaing costs.

Having adopted the closed cycle using one of the gases specified, and gotten away from the limitations imposed by the use of the gas to be liquefied as the working medium, as set forth in the prior art, applicant is in a position to carry out his process in the most efficient manner possible. Hence, it is another object of the present invention to devise a simple closed cycle refrigeration system for cryogenic temperatures that is more efficient than that found in the prior art.

For large refrigeration loads, such as the liquefying of many tons of air per day, it is found that the most inexpensive, cost per horsepower hour, compressors and expanders are of the turbine type. Further, turbines are particularly well suited to moving large quantities of gas with low pressure differentials. Also the gas turbine is a favorable source of power in the range of the power needs for such refrigeration loads.

If the lowest temperature in a single expansion refrigeration cycle is required to be reduced to lower and lower temperatures, the effectiveness of the cold regenerator must, of necessity, be raised to higher and higher values in order to make the cycle work at all. For example, if the lowest temperature required is something in the order of zero degrees F., probably, no cold regenerator would be used. As the temperature is lowered, the imperativeness of efficient regeneration is increased. As the temperatures to be achieved in the present invention are at least somewhat of the order of liquid nitrogen and less, colder, or less than 140° R., it is necessary that the cold regenerator effectiveness be in the order of 99% for the system to reach and effectively operate at 140° R., regardless of the other components' efficiency. For the liquefaction of air, and using helium as the refrigerant, the cold regenerator effectiveness would have to be above 97% to obtain refrigeration load above losses. If methane were being liquefied with nitrogen as the refrigerant, the effectiveness of the cold regenerator could be as low as 95%, but this is the low limit for work in the cryogenic range. Within the acceptable limits of a cold regenerator effectiveness for a given temperature, there is a reasonable freedom of choice based purely on economics. However, in order to minimize the entropy gain in the refrigeration side of the system and because of the extremely high effectiveness required of the cold regenerator for other considerations, as mentioned above, it is necessary to make the other components in the system, other than the cold regenerator, either extremely efficient or else minimize their effect on the over-all cycle. As the primary component affecting the entropy gain in the system, other than the cold regenerator, is the expander turbine, the cold turbine, and as expansion turbine efficiencies higher than 90% are probably not obtainable in a real system, the only way of further diminishing the effect of the entropy gain in the expander turbine is to decrease the pressure ratio to the point where the entropy gain of the turbine is at least not disproportionate as compared to the cold regenerator. This results in a rather low pressure ratio for the expander tubine, as set forth herein. However, within the acceptable range of low pressure ratios in the low temperature refrigeration area forming the present subject matter, as the refrigeration temperature to be achieved is lowered toward absolute zero, there is a small increase in pressure ratios for optimum performance. Regarding the system as a whole, including the hot side, the power side, it will be necessary, because the cost of producing the cold in terms of inefficiencies is so much greater than producing shaft horsepower from the hot side of the system, to let the cold side control the design of the system. This being so, it will be necessary for the efficiency of the hot regenerator to be higher than would normally be economically feasible in a power generating gas turbine cycle. While the hot regenerator efficiency may be lower than that for the cold regenerator because the loss is easily made up by the supply of more heat to the power cycle, the pressure drop thru the hot loop, the power cycle, must be substantially that in the cold loop. These considerations set the hot regenerator efficiency above 90%, and preferably much higher.

In a refrigeration cycle that includes a compressor turbine, regenerator, an expander such as a turbine, and a liquefaction load, certain operating conditions therefor may be assumed such as the kind of gas to be used in the system, the ambient temperature for the compressed working gas entering the regenerator, the regenerator performance, the compressor efficiency, the after cooler efficiency, and the expander efficiency. These assumptions are those inherent in the machines and devices to be used and can be known in advance. Such assumptions include the desired temperature of the working gas leaving the refrigeration load. With the ambient and refrigerant load outlet temperatures and the regenerator performance, the expander inlet and outlet temperatures may be calculated. These values may be used to calculate the actual heat pumped from the load per unit of working medium. Also, these values may be used to calculate the curves of the actual temperature-entropy diagram. These actual conditions may be compared to ideal Carnot cycle conditions by extrapolating the actual diagram to the Carnot diagram and conditions. This comparison gives an index of performance. This index is the heat pumped, or transferred, from the low temperature source by the actual cycle of the actual device, the refrigeration load, divided by the heat that would be transferred by the ideal Carnot cycle extrapolated from the actual cycle. Further, by assuming various actual pressure ratios for the refrigeration cycle, with the other conditions constant, there may be obtained various indices of performance. These indices may be plotted against the various pressure ratios to give a performance curve that will indicate the optimum pressure ratio. Further, such performance curves may be constructed for various working mediums such as nitrogen, neon, helium and hydrogen, at various refrigerate temperatures. Applicant has found that for all of these gases as working mediums there is a limitation defining an optimum pressure ratio range for a cycle or a

system for satisfactory performance. Inherent in this satisfactory performance from an engineering and cost point of view is the requirement of high regenerative efficiency and high expander efficiency as well as high efficiencies in the other heat transfers in the cycle. This defining limitation for optimum pressure ratio range for satisfactory performance, includes these required minimum apparatus efficiencies. The values of these apparatus efficiencies has been previously discussed.

The optimum pressure ratio is that ratio having the highest, the maximum (M), index of performance, the peak of the performance curve. Applicant has found that using nitrogen or, but in particular, neon, deuterium, hydrogen, or helium, as a gas, or combinations of such gases, for the working medium, the refrigerant, that the range of performance indices variation, that is the limits of pressure ratio variation, are: (*a*) the low pressure ratio limit is (the optimum pressure ratio value plus one) divided by (two); and (*b*) the high pressure ratio limit is (twice the optimum pressure ratio value less one). This may be symbolized by designating the optimum pressure value as M, then the limits are: (M+1)/2 and 2M—1. The use of ratios outside of this range make it uneconomical or impossible to accomplish useful work by means of the mechanisms and processes of the present invention in the temperature range required for the liquefaction of air and its components, and such refrigerant; by a single expansion step of the refrigerant and for such liquefaction at substantially atmospheric pressure. For the liquefaction of helium by the use of helium as the refrigerant, the helium to be liquefied must be held above the cycle pressure but the needed refrigerative effect can be achieved by a single expansion of the refrigerant. The limits of pressure ratio variation set forth above are independent of the absolute pressure at which a cycle or system operates.

As the outlet temperature of the working medium from the refrigeration load is lowered, the temperature drop across the load, and the expander, is decreased, and such decreases are exponential functions of the expander inlet temperature. This means that as absolute zero is approached, the pressure and temperature ratios, expander inlet to outlet, become small as set forth above and as compared to prior practices. To obtain an expander outlet temperature of 128° R. and an outlet temperature of 139° R. on the load outlet system, an optimum pressure ratio of about 1.5 must be achieved, using helium, by means of the present invention. If gaseous helium is used to liquefy hydrogen, the temperature ratio needed is still obtainable as their boiling points differ by 29°, and the present process will operate at a much smaller temperature ratio at such temperatures. A pressure ratio of substantially 2.4 would be sufficient, with the same plant efficiency. Certainly, with the present process and reasonable efficiencies, the optimum pressure ratio with helium for the liquefaction of hydrogen, should be less than 2.5. Thruout the cryogenic range from 220° R. to 2° R., the optimum pressure ratio should be between 1.25 and 3.0, that is with reasonable efficiencies in plant operation and according to the present invention using helium, hydrogen, deuterium, or neon as the working medium in turbine machinery. If nitrogen were to be used as a refrigerant down to 220°, the optimum pressure ratio would increase to 2.425. It is an object of the present invention to device a plant and process therefor in which refrigeration is obtainable in such cryogenic range with such pressure ratios using such gases in turbine machinery.

If the value of the pressure ratio is greatly increased above the design pressure range, the relative increase in expansion turbine losses reduces the cycle performance and if the ratio is slightly decreased below the range, the losses associated with the cold regenerator become large relative to the temperature drop in the expansion turbine, again reducing cycle performance. Applicant has found

that outside of the above described range such losses become unbearable in the face of good practice within the present invention. Thus it is an object of the present invention to construct and operate a closed cycle constant volume system for cryogenic refrigeration using such refrigerant working gases and pressure ratio ranges.

In summary, an object of the present invention is the construction of machinery for and the performance of a process for obtaining cryogenic refrigeration temperatures by means of a closed system operating at constant volume, at constant low temperature and in such low pressure expansion ratio ranges, and at low static pressure. Further, it is an object to operate such a system with a single expander or expansion step. Specifically, it is an object of the present invention to operate such a closed system at pressure ratios between 1.15 and 3.85. Another specific object of the invention is to devise such machinery and process of operation at static pressures less than five hundred pounds per square inch. A further specific object is to operate such a system for the liquefaction of air components, and to use in the system helium, hydrogen, deuterium, or neon as the refrigerant working medium, and, particularly to use either helium or hydrogen, those elements having only K-shell electrons, as they have the best specific heat coefficients and the lowest critical temperatures. Yet another specific object is to carry out such air liquefaction and separation at atmospheric pressure.

The adoption in a refrigeration cycle of the above described pressure ratio range for the expansion turbine sets the expansion ratio for a compressor turbine in such cycle. Also, if the compressor is powered by a gas turbine and if all three, gas turbine, compressor, and expander, use the same working gas in a single closed system, the expansion ratio of the power turbine is the same as that of the compressor and the expander, excepting small differences due to differences in the parasitic losses of the two cycles, the power cycle and the refrigeration cycle. Further, as previously explained, the cold regenerator efficiencies found to be required, require the regenerator in the power cycle to have high efficiency. This uniformity of pressure ratios and regenerator efficiencies thruout a power-refrigeration system is another object of the present invention. Thus it is another object of the invention to operate a gas turbine as a power source for the refrigeration load, and to do so at such optimum pressure ratios. Further, such paralleling of optimum operating conditions means that the two systems, power and refrigeration, can be combined into a single system using a working medium in common and a common compressor, with a power turbine, an expansion turbine, and a compressor turbine all mounted on a common shaft for the interchange of power between the turbines. Thus, it is another object of the invention to devise a refrigeration-power system in which is circulated a common gas working medium, and, further, that there be a direct exchange of power in such system.

The above mentioned defects of the prior art devices are remedied and the aforementioned objects achieved by the use of suitable equipment having, briefly, in operation therein closed gas turbine and refrigeration cycles operating with a common compressor turbine directly connected to a power turbine and a refrigeration expansion turbine, and with a common working gas medium, and operating such cycles at constant volume and with the pressure ratios described.

A system such as described briefly above is schematically illustrated in the accompanying drawing.

For the purpose of the following description and by way of example, the system will be described as using helium as the working gas medium for both the power and refrigeration cycles. The temperatures and pressures hereinafter are by way of example, and are variable within the above mentioned limits. All pressures are pounds per square inch absolute (p.s.i.) and tempera-

tures are degrees Rankine (° R.). Assuming the whole system of power and refrigeration has been in operation for a sufficient time to reach the intended operating conditions of temperature and pressure, helium enters a compressor turbine 10 at a pressure of 181 p.s.i. and an ambient temperature of 530°. Helium is discharged from the high pressure side of the compressor at 268 p.s.i. and 618°. The flow from the compressor outlet conduit 11 is divided into two high pressure side streams, namely a power stream which flows thru one branch 12, or power loop, and a refrigeration stream which flows thru another branch 13, or refrigeration loop, of the outlet conduit 11. The high side of the power stream, or hot stream, first passes thru one side of a regenerator 14, the power or hot regenerator, where it is heated to 1493°. From the regenerator the high side power stream passes thru a combustion chamber heat exchanger coil 15 which serves to heat the gas to 1660°. Any suitable fuel or source of heat, as the coil 16, may be used. The power stream is led to and used to drive a hot turbine 17 which provides a large part of the power for the compressor 10. The gas expands and cools in the turbine, the pressure dropping to 190 p.s.i. and the temperature dropping to 1498°, and then passes thru the other side, the low pressure side, of the hot regenerator 14 where it is cooled while heating the high side power stream in counter current flow thereto, to approximately the compressor discharge temperature. While, generically, this is called regeneration of the gas, or working medium, applicant uses the terms, "generation" and "degeneration" to indicate the gain or loss of heat, respectively, in a regenerator. Finally the gas passes thru a precooler 18 from which it is returned to the compressor 10. The precooler may be water or air cooled, and serves as a heat sump for the power cycle to bring the working medium down to ambient temperature. The above described circuit constitutes the power cycle of the system. In the drawing, the high side conduit lines are heavier than the low side lines.

The cold, or refrigeration, stream that flows thru conduit 13, passes first thru a heat sump 19, or after-cooler, where it is cooled to 530°, the ambient temperature, the pressure drop being slight, about 5 p.s.i. The stream then passes thru a regenerator 20, the cold regenerator, where it is cooled, or degenerated, to 141°. The gas emerging from the regenerator drives a turbine 21, called the cold turbine, wherein the gas expands with a drop in temperature to 128°. The cold turbine 21 serves as a source of power to a generator 22 connected thereto by a shaft 23. The cooled low pressure stream of helium then passes thru heat exchanger coil 24 or other conduit means of high heat conductivity which acts as the refrigeration load for the system and the cycle. From the load coil 24, the low pressure helium returns to the cold regenerator 20 where it serves to cool the high side helium. The helium then completes its refrigeration loop by returning to the compressor at ambient temperature and compressor inlet pressure of 181 p.s.i. The material giving heat to the refrigeration load is passed thru a coil 26 in heat exchange relationship with the refrigerant coil 24, the coils 24, 26 constituting a heat exchanger. This material to be cooled may be a gas such as air to be liquefied and its components separated as by rectification. Such refrigerate gas is moved counter current to the flow of the helium refrigerant. This is an important aspect of the use of a closed cycle, that for the liquefaction and separation of gas components the flow of refrigerant and refrigerate may be in counter current heat exchange relationship.

Initially, in placing the system in operation, a starting electric motor 34 or other external source of motive power is used to spin the compressor and the other turbines until the system reaches a point where it is thereafter self-sustaining. All of the turbines 10, 17, 21 and the starting motor may be mounted on a single shaft 35 or they may be coupled by mechanical or hydraulic means, or by electrical or magnetic means. Particularly, the starting motor may be clutched to the shaft by a clutch 36 so that when its work of starting the system is over it may be disconnected therefrom. Also, the cold turbine 21 may be directly coupled to the compressor 10 by a clutch 37. Thus, the cold turbine may deliver power directly by shaft means to the compressor or it may furnish electrical power by means of the generator 22 for any desired use.

By the above described process and equipment, applicant has devised simple means for the economical condensation and liquefaction of the so-called cryogenic, or permanent gases. This process includes a power cycle and a cryogenic refrigeration cycle, which cycles are interdependent, integrated, in that the same working medium is circulated in both cycles of the system, and such working medium is intermingled for both systems by the use of a common compressor 10. The defects of the prior art inherent in the use of open cycles is remedied by the use of a closed system, or cycles, for both power and refrigeration. This allows the use of the one gas medium best suited to a particular problem, both power and refrigeration, the best as to specific heat characteristic and the best as to critical temperature. Also, the selectivity of working medium, allows the refrigeration and the power, each to be had by a single expansion step, by the use of the single expansion cold turbine 21 and the hot turbine 17.

The system is started in operation by first supplying cooling water or air to the heat sumps 18, 19, by supplying gas or air to be condensed to the refrigerate coil 26, and then by spinning the compressor 10 and turbines 17, 21 by means of the motor 34 to start the gas working medium circulating in both loops 12, 13 of the cycles and the system. Once the turbines are up to speed, heat is supplied to the furnace, or heat input exchanger 15, so that the hot turbine 17 will take on the compressor's work of circulating the working medium, and power to the starting motor 34 may be discontinued and the motor disconnected from the turbines' common shaft 35. When the refrigeration coils 24, 26 are reduced in temperature to the proper degree, air components will be liquefied by passage thru the coil 26.

If helium, hydrogen, or neon, or a combination thereof, is used as the working medium, such of the permanent gases as methane, oxygen, argon, and nitrogen may be liquefied at atmospheric pressure. This is a decided advantage as compared with the prior art processes where air components have been liquefied by the refrigerative effect of air expanded from high pressure in an open system. Helium may be liquefied under low pressures in the present system.

The system will be operated on the temperature and pressure differentials of working medium previously described, once it has reached equilibrium. These differentials, or operating conditions, are those that will allow the system to operate with a minimum of power input. Further, these operating conditions are new in the closed cycle power and refrigeration field, and in the operation of turbines. The fact that changes in the refrigeration load directly effect a change in the power production system, by the use of a common working medium, and are thereby compensated for by changes in density of such medium resulting from changes in the load, is an important factor in system stability.

No disclosure is made herein of the special relationships of the presently disclosed invention to the problems of and processes of gas liquefaction and separation, other than those found in connection with the discussion of the prior art, and the statement that air may be liquefied in the refrigerate load coil 26 at atmospheric pressure when helium, hydrogen, or neon is used as the refrigerant.

Having thus described my invention, the operation of its process and machinery for the performance of such

process, the products obtainable therefrom, and a specific example of the temperatures and pressures usable within the defined critical range, I claim:

1. A closed cycle power-refrigeration process in which a cryogenic gas is the working medium in both a power cycle and a refrigeration cycle of a system, comprising: establishing a closed system and, in said system, effecting a power cycle by circulating and acting on said gas thru the following seriatum steps, compression, generation in a first regenerator, heating, expansion to accomplish work, degeneration in such regenerator, cooling, and again compressing to complete such power cycle, and in which power cycle such work is used for such compression; and, in such system, effecting a refrigeration cycle by circulating and acting on said gas thru the following seriatum steps, bleeding gas from said power cycle immediately after said compression, cooling, degenerating in a second regenerator, expanding to reduce the temperature of said gas, adding heat from a refrigeration load, generating in such second regenerator, and returning such bled gas to said power cycle for compression.

2. The process of claim 1 in which the expanding of said gas in said refrigeration cycle is accompanied by the derivation of power from such expansion, and in which process such power is used for the compression of said gas.

3. The process of claim 1 in which the pressure ratio in such system is held between the following pressure ratio limits: the low limit is (the optimum pressure ratio value plus one) divided by (two) and the high pressure ratio limit is (twice the optimum pressure ratio value) less (one).

4. The combination of claim 3 in which said optimum pressure ratio is between 1.25 and 3.0.

5. The combination of claim 1 in which each of said regenerators has an effectiveness better than 90%.

6. The combination of claim 1 in which said second regenerator has an effevtiveness better than 95%.

7. The combination of claim 1 in which said first regenerator has an effectiveness better than 90%, and said second regenerator has an effectiveness better than 95%.

8. The process of claim 1 in which said gas in said cycles remains a gas during the whole of said cycles.

9. The combination of claim 8 in which said gas is helium, hydrogen, deuterium, or neon.

10. The combination of claim 1 in which said compression, expansion to accomplish work, and expansion to reduce temperature, each has a pressure ratio held between the following pressure ratio limits: the low limit is half the sum of (the optimum pressure ratio value plus one) divided by (two) and the high pressure ratio limit is (twice the optimum pressure ratio value) less (one).

11. The combination of claim 10 in which said gas is helium, hydrogen, deuterium, or neon.

12. The combination of claim 11 in which the temperature achieved by said expansion is less than 220° R.

13. The combination of claim 11 in which the temuerature achieved by said exp.ansion is less than 140° R.

14. In a method of providing very low temperature refrigeration utilizing a very low boiling point gaseous refrigerant medium in a closed system that comprises the steps of confining the medium at ambient temperature at a pressure of several atmospheres, compressing said medium substantially, and dividing the compressed medium into a first stream and a second stream; heating the first stream to raise the temperature thereof substantially, allowing the heated first stream to expand and deriving power from such expansion, and returning such expanded first stream to such compression step; and cooling the second stream, allowing the second stream to expand and deriving power from such expansion, allowing the expanded second stream to absorb heat at a very low temperature, and utilizing the resulting heated second stream for said aforementioned cooling of the second stream to provide a further heated second stream, and returning such further heated second stream to such compression step.

15. In a closed cycle very low temperature refrigeration system adapted to contain a very low boiling point gas under pressure as the refrigerant medium, a compressor having an exhaust side and an intake side, a first turbine mechanically connected to drive said compressor, a second turbine mechanically connected to drive said compressor, conduit means extending from the exhaust side to the intake side of the compressor providing a hot circuit for flow of a part of the medium leaving the compressor, conduit means extending from the exhaust side to the intake side of the compressor providing a cold circuit for the flow of the remainder of the medium leaving the compressor, said first turbine being connected into said hot circuit and said second turbine being connected into said cold circuit, means for supplying heat to the medium in said hot circuit at a region between the compressor exhaust side and said first turbine, means between said compressor exhaust side and said second turbine in said cold circuit for cooling the medium therein, and a load heat exchange means between said second turbine and the compressor intake side in said cold circuit for extracting heat from a load.

16. A refrigeration system as set forth in claim 15 in which the means between the compressor and the second turbine in the cold circuit for cooling the medium includes a regenerator heat exchanger through which the medium flows before reaching said second turbine and after leaving the load heat exchange means.

17. A power-refrigeration process in which a gas is the working medium in both a closed power cycle and a closed refrigeration cycle, such cycles forming a closed system for the practice of such process, comprising: establishing a closed system wherein there is circulated a gas and, in said system, effecting a power cycle by circulating and acting on said gas by means of the following seriatum steps, compression, a first generation heat transfer, heating, expansion with the accomplishment of work, degeneration to said first generation heat transfer, cooling to ambient temperature, and again compressing to complete such power cycle, and in such power cycle using such power for such compression; and in such system, effecting a refrigeration cycle by circulating and acting on said gas by means of the following seriatum steps, bleeding gas from said power cycle immediately after said compression, cooling to ambient temperature, a second degeneration heat transfer, expanding with the accomplishment of temperature reduction, adding heat from a refrigeration load, generation to said second degeneration heat transfer, and returning such bleed gas to said power cycle for compression.

18. The process of claim 17 in which said gas in said system remains a gas at all time.

19. The process of claim 17 in which the expanding of gas in said refrigeration cycle is accompanied by the production of work that is used for the compression of said gas.

20. The process of claim 18 in which said gas is helium, hydrogen, or neon.

21. The process of claim 17 in which said refrigeration cycle, the expansion of said gas is accomplished with the production of both work and said temperature reduction.

22. The process of claim 18 in which said gas is a gas having a critical temperature equal to or lower than the critical temperature of nitrogen.

23. The combination of claim 18 in which said gas is from the group having only K-shell electrons.

24. A power system having closed but interconnected hot and cold loops for the conduct of a gas working medium in such loops, a compressor having an inlet and an outlet, said compressor being common to such loops and each loop having a connection to the inlet and the

outlet thereof to provide such loops' interconnection, said hot loop having therein, in addition to said compressor, in series from said compressor outlet to said compressor inlet: a hot regenerator, a high temperature heat source, the inlet of a hot turbine, such hot turbine, the outlet of such turbine, said hot regenerator, and a hot loop heat sump; said cold loop having therein, in addition to said compressor, in series from said compressor outlet to said compressor inlet; a cold loop heat sump, a cold regenerator, the inlet of a cold turbine, such cold turbine, the outlet of such cold turbine, a low temperature heat source, and said cold regenerator; both of said turbines being operated for the production of power, and said compressor being operated for the consumption of power.

25. A power system having closed but interconnected first and second loops for the conduct therein of a gas working medium, a compressor interconnecting said loops, a separate turbine connected in each of said loops and dividing it into a high pressure side and a low pressure side, each loop having its high pressure side and its low pressure side arranged in regenerative relationship, and a regenerator in each loop providing such relationship; the first of said loops having a high temperature heat source in its high pressure side between its regenerator and turbine, and a heat sump in its low pressure side between its regenerator and the compressor; and the second of said loops having a low pressure temperature heat source in its low side between its regenerator and turbine, and a heat sump in its high pressure side between its regenerator and the compressor; both of said turbines being operated for the production of power, and said compressor being operated for the consumption of power.

26. The combination of claim 25 in which the working medium is a gas having a critical temperature equal to or lower than that of nitrogen.

27. In a method of providing very low temperature refrigeration utilizing a very low boiling point gaseous refrigerant medium in a closed cycle which comprises the steps of confining the medium at ambient temperature at a pressure of several atmospheres, compressing said medium substantially, dividing after compression and before other steps the compressed medium into a first stream and a second stream, heating the first stream to raise the temperature thereof substantially, allowing the heated first stream to expand and deriving mechanical energy therefrom to provide part of the power required to compress the medium, cooling the second stream, allowing the second stream to expand and deriving mechanical energy therefrom to provide the remainder of the power required to compress the medium, allowing the expanded second stream to absorb heat at a very low temperature, and utilizing the resulting heated second stream for said aforementioned cooling of the second stream before returning the heated second stream for recompression.

28. The method set forth in claim 27 in which the refrigerant medium is selected from the class consisting of air, argon, helium, hydrogen, nitrogen, and neon.

29. A power-refrigeration system having closed but interconnected hot and cold loops for the conduct of a gas working medium thru such loops, a compressor having an inlet and an outlet, said compressor being common to such loops and each loop having a connection to the inlet and the outlet thereof to provide such loops' interconnection, said hot loop having therein, in addition to said compressor, in series from said compressor outlet to said compressor inlet: a hot regenerator, a heater having a high temperature heat source, the inlet of a hot turbine, such hot turbine, the outlet of such turbine, said hot regenerator, and a hot loop heat sump, the temperature of said gas from said compressor in said hot loop being seriatum progressively increased in said hot regenerator and said heater, and then progressively decreased in said hot turbine, said hot regenerator, and thru said hot loop sump to said compressor; said cold loop having therein, in addition to said compressor, in series from said compressor outlet to said compressor inlet: a cold loop heat sump, a cold regenerator, the inlet of a cold turbine, such cold turbine, the outlet of such cold turbine, a low temperature heat source, and said cold regenerator, the temperature of said gas from said compressor in said cold loop being seriatum progressively decreased in said cold loop sump, cold regenerator, and cold turbine, and then progressively increased thru said heat source and said cold regenerator to said compressor; the gases in each of said loops being at substantially the same temperature as they leave said hot loop sump and said cold regenerator for said compressor; and both of said turbines being operated for the production of power, and said compressor being operated for the consumption of power.

30. The combination of claim 29 in which the volume of said system, said compressor, turbines, and loops, is constant.

31. A power-refrigeration system having closed but interconnected hot and cold loops for the conduct of a gas working medium thru such loops, a compressor having an inlet and an outlet, said compressor being common to such loops and each loop having a connection to the inlet and the outlet thereof to provide such loops' interconnection, said hot loop having therein, in addition to said compressor, in series from said compressor outlet to said compressor inlet: a hot regenerator, a heater having a high temperature heat source, the inlet of a hot turbine, such hot turbine, the outlet of such turbine, said hot regenerator, and a hot loop heat sump, the temperature of said gas from said compressor in said hot loop being seriatum progressively increased in said hot regenerator and said heater, and then progressively decreased in said hot turbine, said hot regenerator, and thru said hot loop sump to said compressor; said cold loop having therein, in addition to said compressor, in series from said compressor outlet to said compressor inlet: a cold loop heat sump, a cold regenerator, the inlet of a cold turbine, such cold turbine, the outlet of such cold turbine, a low temperature heat source, and said cold regenerator, the temperature of said gas from said compressor in said cold loop being seriatum progressively decreased in said cold loop sump, cold regenerator, and cold turbine, and then progressively increased through said heat source and said cold regenerator to said compressor.

32. A power-refrigeration system in which a gas is the working medium in both a closed power cycle and a closed refrigeration cycle, said cycles being combined in a unitary system, which comprises circulating said gas in a power cycle and including therein the steps of compressing said gas, passing said compressed gas in heat exchange relation with hot expanded gas to raise the temperature of said compressed gas, further heating said hot compressed gas, expanding said hot compressed gas by passage thereof through a first prime mover, passing said hot expanded gas in heat exchange relation with said initially compressed gas to reduce the temperature of said expanded gas, and again compressing said gas to complete said power cycle; circulating said gas in a refrigeration cycle and including the steps of taking a portion of said compressed gas from said power cycle, cooling said compressed gas portion, passing said cooled compressed gas in heat exchange relation with a cold expanded gas and further lowering the temperature of said cooled compressed gas, expanding the exiting cold compressed gas by passage thereof through a second prime mover, passing said cold expanded gas in heat exchange relation with a refrigeration load, passing the exiting gas in heat exchange relation with said cooled compressed gas to add heat to said gas leaving said refrigeration load, and returning said last mentioned gas to said power cycle for compression.

33. The process of claim 32, wherein the power resulting from the expansion of said hot compressed gas in said first prime mover is employed for the compression of said gas.

34. The process of claim 32, wherein the power resulting from the expansion of said hot compressed gas in said first prime mover and from the expansion of said cold compressed gas in said second prime mover are employed for the compression of said gas.

35. The process of claim 34, wherein said gas is helium and wherein the pressure ratio of said system is maintained such that the low pressure ratio limit is (the optimum pressure ratio value plus one) divided by two, and the high pressure ratio limit is (twice the optimum pressure ratio value) less one.

36. The process of claim 32, in which the pressure ratio of said system is maintained such that the low pressure ratio limit is (the optimum pressure ratio value plus one) divided by two, and the high pressure ratio limit is (twice the optimum pressure ratio value) less one.

37. The process of claim 36, wherein the pressure ratio in said system is between 1.15 and 3.85.

References Cited by the Examiner

UNITED STATES PATENTS

1,264,807	4/18	Jefferies	62—88
1,440,000	12/22	Bonine	62—402
2,929,217	3/60	Collman	62—59

ROBERT A. O'LEARY, *Primary Examiner.*