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CRYOGENIC REFRIGERATING METHOD AND APPARATUS

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FIG.2

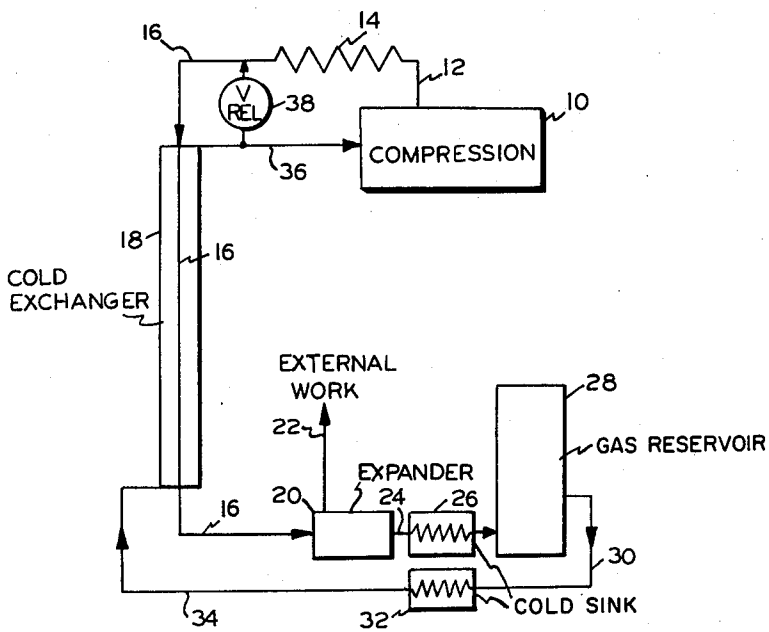
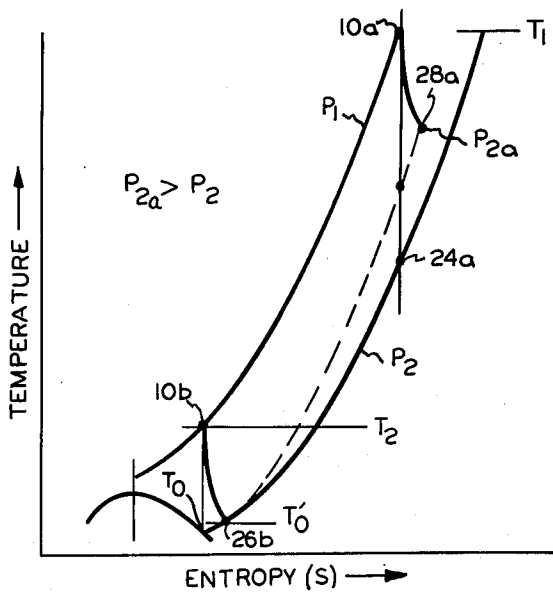


FIG. 1

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CRYOGENIC REFRIGERATING METHOD AND APPARATUS

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This invention relates to extremely low temperature refrigeration processes and apparatus and, more specifically, it relates to improvements in a very low temperature closed cryogenic refrigeration system, operatively associating the compression, pre-cooling and expansion of a gaseous cryogen. In carrying out this invention a pressure control arrangement embodying a pressure responsive receptacle for gas within the coolest portion of a cryogenic refrigerating system is utilized to provide for the efficient and continuous production of temperatures well below those obtained by conventional refrigerating devices.

The development of industrial applications to very low temperature physics, particularly as evidenced in the fields of superconductivity and the like, has posed somewhat complex and involved requirements for creating environments of extremely low temperatures. Generally, the temperatures deemed acceptable for such use reside only in the helium range, often closely approaching absolute zero. In view of the wide range of industrial demands existing and continually coming to bear, these cold environments must be produced as efficiently and reliably as possible, while the apparatus deriving them must be as light and compact as possible. Further, the refrigerating apparatus must be adaptable to a relatively wide range of fluctuating heat loads and must remain operable in a broad variety of installations often remote from immediate human attendance.

The cryogenic arts for the most part have been devoted to systems designed for the mass production of conventional, liquefied gases. The efficiencies of systems producing these liquefied gases have been improved considerably since their inception, however, liquification of conventional gases is obtained at temperatures well above those required by the present very low temperature environmental demands. Accordingly, efficient cold production has been only inconveniently realized and then at temperatures usually substantially higher than those now sought. Efforts have been made to accommodate conventional liquification schemes to meet the new demands for the continuous production of extremely low or undercooling environments, however, the attempts have met with but marginal success, especially in view of a wide variance in desired operational characteristics. Undercooling temperatures are considered as those existing below the conventional liquification of a cryogen.

Conventional liquid gas production is considered to occur in an "open" cryogenic system in which gas is fed into the warm input end of the cooling apparatus and is removed in time as a liquid, having been accumulated in a collecting apparatus located at the operational termination of the system. The span of time required to liquefy the gas customarily is referred to as the "cooling-down" period.

For the present purposes of providing undercooled environments, it is desirable to utilize a closed system, whereby cryogen is neither added to nor taken from the system and there is substantially no liquification of gas cryogen. By merely modifying or rearranging conventional open system components to become closed systems in order to meet the instant needs, a multitude of operational shortcomings occur.

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For example, during the course of the cooling-down period of conventional liquification schemes modified to become closed systems, the working cryogen, by the nature of the cooling process, will contract, reducing system pressure and thereby disrupt the very important and critical pressure balance across the system. The somewhat delicate corrective procedure of introducing makeup cryogen into the compression stage is used throughout cooling-down to compensate volume contraction and re-establish the volume-temperature balance. This corrective requirement derogates the system's reliability, lengthens the cooling-down period and, additionally, precludes remote operation of the apparatus. These deficiencies occur since the introduction of supplementary warm gas serves to add an additional warm media into the system which must utilize cooling energy and consequently, lower thermodynamic efficiency. Precise and vigilant valving control must be established during the introduction so as to avoid a disruption of the pressure balance of the overall system. Since an auxiliary input of cryogen is necessitated at each start-up of the system, independent operation removed from human assistance has been unrealistic. A variation in the heat load upon undercooling systems will further hinder the establishing of a stable pressure balance, since a relatively small alteration of cryogen temperature will evolve pressure alterations. This unbalance generally requires similar correction amounting to a continuing and undesirable surveillance of the control features of the system.

Very low temperature cryogenic units find their operational basis in the use of expanders which transform the energy of a compressed gas into cold through the media of adiabatic expansion. The efficiency of temperature reduction evolved from this expansion is critically dependent upon the system pressure at the point of expansion. An ideal system design would introduce high input pressures into a single expanding machine to effect a wide temperature drop in a single and efficient operation. However, to the present, accepted design criteria show that the use of but one expander engenders extreme difficulties in obtaining efficient expansion since the input pressures required at the commencement of the cooling-down period must be inefficiently high and expander speeds excessive. It is the common practice, therefore, to utilize several stages of intermediate expansion within the basic system, each serving to pre-cool the cryogenic media in a progressive or step-wise fashion. This progressive cooling demands intricate valving arrangements, auxiliary gas input as above described and liquid gas accumulation means. The lack of simplicity resulting from the accommodation of valving and similar control evokes further derogation of total efficiency. In consequence, the overall size of the cryogenic unit must increase, the system becomes over-sensitive and, more important, the goal of attaining temperatures approaching absolute zero becomes most difficult to attain.

The problems of system efficiency are especially apparent in systems using turbine expanders. The expansion efficiency of turbines is critical to system efficiency and is dependent upon operation within a design range of speed, losses tending to increase with high pressure input and resulting speed increases. Speed reduction and consequent upgrading of turbine efficiency gradually may be realized by the progressive densification of gaseous cryogen during the period of system cooling-down. However, until such densification is effected, the inefficiencies resulting from excessive rotational speed will be unacceptable. Typically, turbine expander units utilizing helium as the working medium would require relatively high ratios of expansion in the order of ten to fifteen and above in order to induce low temperatures. However, even at expansion ratios in excess of fifteen the efficiency of cold

production is compromised at temperature development below about 20 degrees K.

As before described, a design utilizing but one turbine expander operating with a gas compressed through a given pressure ratio is unrealistic, since at the start of the cooling down period the turbine must operate at a speed greatly in excess of that considered most efficient at the termination of the cooling-down period. To date, the problem has been approached by utilizing a system having multiple expanders and/or auxiliary pre-cooling arrangements and has led to inconvenient designs and correspondingly impractical and expensive refrigeration arrangements. These arrangements are unacceptable to modern industrial demand.

The invention as now presented provides a continuous, simple and highly efficient method for controlling overall pressure balance within a low temperature refrigerating system having substantially no liquid gas phase. It provides further advantage in avoiding requirements for additional external gas feed-in for pressure fluctuation compensation.

The system as provided by the invention allows a shortened cooling-down period, consequent greater efficiencies and greater cooling capacities due to operation with higher cryogen densities. The invention provides further efficiencies by significantly reducing the need for valving control of expander components and the like. The invention is particularly characterized in providing a substantial reservoir of cool gases serving as a back pressure cold sink within the coolest portion of the system and as a continuous, temperature responsive speed control for turbine expander rotation.

The provision of a cold sink reservoir at the coolest portion of the system additionally establishes a high cooling capacity, thereby accommodating substantial fluctuations of heat load. It is a further object of the invention to provide a pressure control component which is reliable and is adaptable to a large variety of compact packaging arrangements, providing advantage in portability, low bulk and minimized weight of an overall closed system cryogenic refrigerating apparatus. Further, the system is compatible with a wide variety of industrial applications and is operable in remote areas under minimal surveillance.

These and other objects of the invention are further described and illustrated by the following discussion and related drawings in which:

FIG. 1 is a schematic representation of the basic components within the inventive cryogenic refrigerative system;

FIG. 2 presents a graphical Temperature-Entropy analysis of the refrigerating system of the invention.

Referring to the drawings, FIG. 1 is illustrative of a schematic and generalized embodiment of the instant invention showing the arrangement of components for developing the inventive method for controlling overall pressure balance within a low temperature refrigerating system having substantially no liquid gas phase. To operate successfully at temperatures within the under-cooling region approaching absolute zero, gas cryogens available for use within the system must be selected having appropriate physical characteristics. In particular the gases utilized within the system must be capable of sustaining a pressure-temperature relationship such that at no point within the refrigerative cycle is there any substantial liquefaction of the gas. Illustrative of such gases are helium, xenon and hydrogen.

Looking to the drawing, gaseous cryogen is compressed within a compressor shown generally at 10. Any of a wide variety of compressing means and complementary auxiliary equipment such as purifiers and the like familiar to the art are operable in the system, the choice of components being dependent upon such design considerations as desired pressure ratio, power demands, packaging and like characteristics.

Gas leaving the compressor stage 10 is led through a conduit 12 into a cooling device or "cooler" 14 which serves to remove heat of compression from the compressed gas. The cooled, compressed gas is now directed along a conduit 16 through a cold exchanger 18 where it is cooled preferably in out-of-contact, counter-current thermal exchange with gas returning from later stages of the cooling cycle. The use of out-of-contact, counter-current heat exchange is suggested as the prevailing most efficient mode of heat transfer, however a variety of such means is available to the art, the choice of component being dependent upon the overall characteristics of the system and cold environment desired.

Upon leaving the cold exchanger 18, the cooled gas is introduced into an expander 20, for example a turbine, wherein work of expansion is extracted adiabatically to be disposed of in an environment exterior to the system. Such disposal is indicated by an arrow 22.

Typical of expanding devices operable with the arrangement is that described in my co-pending application, Ser. No. 107,148, entitled "Cryogenic Expander," filed May 2, 1961.

Expanded gas exhausts from expander 20 through conduit 24 into a cold sink which is denoted simply as a region 26. The cold sink is devoted to the ultimate function of the refrigerating system. It is here that the cooling environment is established for such uses as the system may be designated. The cold sink region may assume any of a wide variety of available configurations dependent upon the uses intended for the cold production.

Cold sink 26 is in pressure responsive connection with a gas contraction chamber 28. Chamber 28 is located as close as possible to the coolest portion of the system and may assume a myriad of configurations. However, it must be capable of containing a relatively substantial volume of the total gas retained within the closed system. The chamber may be highly insulated so as to establish a transient pressure dependent upon the temperature of gas contained within it.

During portions of the operating cycle of the system wherein the temperatures are transient, pressures as are prevailing within the contraction chamber 28 will provide a back pressure to restrict the turbine speed of the expander 20. Exemplary treatment of considerations for establishing design parameters of the contraction chamber are presented in the operational discussion provided later.

Gas exit means are provided from contraction chamber 28 along a conduit 30 to an alternate cold sink arrangement 32. The configuration of cold sink 32 is dependent upon the intended use of the system and the design parameters attendant with such use similar to cold sink 26. Either cold sink 32 of 26 may be used with the refrigerating system, or both may be used conjointly depending upon design desiderata pertinent to particular refrigerating requirements.

From cold sink 32 the gas enters cold exchanger 18 through a conduit 34. Within the cold exchanger the gas acts to establish counter-current thermal exchange with earlier compressed gas entering the exchanger from conduit 16. Upon leaving cold exchanger 18, the gas is directed along a conduit 36 to the suction side of compressor 10 where it is recompressed to continue an uninterrupted closed system cycle.

A relief valve 38 is shown in connection between conduits 16 and 36. This valve interconnects the suction and compression sides of compressor 10 and serves to provide suitable over-pressure protection for the compressor. Although the valve is shown separately on the drawing, similar means may be included as an integral part of the compression arrangement.

Operation

The generalized and schematic embodiment of the system as above outlined is further described when con-

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sidered in conjunction with the elemental Temperature-Entropy diagram of FIG. 2, wherein the operational aspect of the system is more clearly portrayed. For clarity, certain of the component numbers shown in FIG. 1 are reproduced in the T-S diagram having suffix "a" or "b."

During its operation, the system maintains the cryogenic media in a constant gaseous state; maintains operability while a closed system and automatically establishes a temperature responsive control of the expansion phase of the cooling process. To observe this control aspect of the invention, a first consideration is that of the initial status of the cryogen gas pressure and temperature T_1 at start-up. At this situation or point of time there exists an equilibrium of temperature throughout the system, gas density is relatively low and the overall volume of gas confined within the apparatus resides at an average pressure somewhat higher than that existing at later stages of operation.

As the warm gas is compressed, the turbine expander is subjected to an initially high input pressure $P_1(10a)$, thereby inducing excessive speeds and a debilitating drop of expansion efficiency. Since the system must operate to effect an extreme change of temperature from start-up to design on cold operation, it is necessary at initial compression to establish a large pressure ratio. The pressure drop associated with this large ratio is depicted on the diagram as the difference between curves P_1 and P_2 (theoretically, $10a$ to $24a$). This high pressure requirement follows from considering that as the temperature at expansion lowers, a diminished enthalpy drop ensues and consequently, the further lowering of temperature become increasingly difficult to attain. For example, when operating with He or Xe as the cryogen, pressure ratios from about 10 to 30 are necessitated at compression. Should pressure control valving for reducing rotational speed be established at the expander input, a most undesirable efficiency loss would ensue as well as a demand for constant adjustment of the valving as the cryogen gradually densifies, thereby slowing turbine speed.

At the initial stage, however, contraction chamber 28 serves to create a back pressure against the outflow of expander 20. This back pressure opposes turbine rotation and consequently provides an efficiency gaining speed control of the expander. The back pressure effect is seen in conjunction with curve P_{2a} on the diagram, which curve is representative of the final pressure following a total or final contraction of gases within chamber 28. Its effect at the initial stage is shown at position $28a$ of the curve. As the closed system continues to recycle and progress to its final stage of cold attainment, the increasingly cooling expander exhaust gradually lowers the average internal temperature of the contraction chamber 28, thereby establishing a densification of gases contained therein and reducing the buffer back pressure against the expander. The gradual reduction of back pressure is realized concurrently with an overall densification of gases within the system. As the cryogen within the system densifies, expander turbine rotational speed will decrease in complement therewith to continue operation at efficient design speeds. The progressive diminishment of back pressure may be observed in noting the gradual convergence of curves P_2 and P_{2a} of the drawing as they near the ultimately approached saturation temperature T_0 . It will be understood that temperature level T_0 represents a theoretical level of low temperature attainment. Practically, a certain higher equilibrium temperature T'_0 will be reached due to obvious deviations from the theoretical isentropic condition in view of unavoidable irreversibilities inherent in the delicate systems at hand. At final operating stage, the refrigerating system may be operated or cycled continuously between T_2 and T'_0 and at the design pressures shown as the separation of points $10b$ and $26b$.

The inclusion of a relief valve 38 between the suction and output portion of the compression unit protects the

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compressor from excessive input pressures. Further, the inclusion of valving means at this position seeks to isolate the volume of gas contained in the colder region of the unit from that volume within the general compression area, thereby allowing the fully automatic pressure and speed control evolved from the contraction chamber to operate without further, hand controlled valving means. In effect, the above noted identification of two working volumes of gas operates to buffer one against the other and allow a transient but continuously established volume of gas within the contraction chamber.

A further advantage of the inclusion of the contraction chamber arises from the provision of a substantial reservoir of cooled gases within the coldest portion of the system, thereby allowing a facile and immediate compensation of any erratic or unforeseen heat loads generated within the heat sink. The chamber advantageously establishes a high cooling capacity while the system remains operable with a minimal cryogen flow.

As an example of the considerations and parameters present in establishing the size of a contraction chamber, the following idealized general computations associated with the use of helium are presented:

In a helium cycle operating at the maximum pressure limits between one and 25 atmospheres ($R=25$), the specific volume of the gas is about 250 cms.³/gram before expansion and about 1750 cms.³ should the expansion be complete, ($R=25$ beginning at room temperature of 300° K). In such case, the isentropic speed after expansion would be about 1720 meters/sec. Should the system be intended to reach 4° K., as an ultimate limit while using a possibly efficient cold exchanger for precooling the gas before expansion to about 15° K., the ultimate speed of isentropic expansion would be only about 290 meters/sec. Practically no efficient turbine expander could be built for this extreme speed.

However, if in accordance with the invention a contraction chamber is provided, the picture changes considerably. The size and other proportions in design derive from the fact that using this contraction chamber at the cold end, from room temperature, enough back pressure is secured in order to cut down the expansion speed at the room temperature. To that effect, in the case of helium, consideration is given to the ratio of contraction R_c , between the beginning and the end of the cooling down period, as specified in the thermodynamic diagrams for the desired pressure conditions: namely, if an expansion ratio of i.e. only 2.5 is desired instead of 25, as it shall be at the end of the cooling down, the following contraction factors define the proper design parameters:

Specific volume at $R=2.5=490$ cms.³/gram

Specific volume at the coldest point after idealized expansion=60 cm.³/gram at 4° K.

Therefore, considered the volume of the contraction chamber V_c to stay in relation to the total volume enclosed in the recirculating parts (pipings, cold exchanger, etc.) at

$$\frac{490}{60} V_p \approx V_p + V_c$$

which leads to the condition

$$V_c = \left(\frac{49}{6} - 1 \right) V_p$$

Obviously this is associated with isentropic assumptions. The skilled cryogenist may translate inefficiencies occurring practically and compute more accurately the condition in realistic systems. At any rate, the desired effect, that is the reduction of the speed at the beginning of the operation is achieved insofar as the isentropic speed of the turbine expander at the beginning would be only about 960 meter/sec., instead of 1750 meter/sec. In view of the fact that the squares of speeds are impor-

tant in structural dynamics, the considerable speed reduction associated with the invention makes obvious the practical value of it.

While the invention has been disclosed in a preferred embodiment, other and further modifications within the scope of the concept herein disclosed may occur to those skilled in the art. Accordingly, it is intended that the invention may be limited only by the scope of the appended claims wherein:

What is claimed is:

1. A closed cryogenic refrigerating system comprising in combination:

- (a) a fixed mass of cryogen contained within the said closed system and having an exclusively gaseous phase;
- (b) a compressor for compressing said cryogen, having a low pressure side and a high pressure side;
- (c) at least one turbine expander for extracting energy from said cryogen and operable in response to the pressure of cryogen compressed at the high pressure side of the said compressor and exhausting expanded cryogen within the system to establish the coldest point within said system;
- (d) conduit means, including at least one cold exchanger, in connection between said compressor high pressure side and said expander for delivering a cooled compressed cryogen to said expander;
- (e) a pressure modulating reservoir in connection with said expander and adapted to receive said expanded cryogen and retain under pressure an amount of said cryogen sufficient to restrict the rate of expansion of point within said system;
- (f) return loop conduit means in connection between said expander and said compressor low pressure side, said return loop conduit means returning expanded cryogen to recompression and providing thermal exchange within said cold exchanger; and
- (g) cold sink means in connection with said return loop conduit means and situate intermediate said expander and said cold exchanger, for utilizing the cold production of the system.

2. A closed cryogenic refrigerating system operating with a cryogen having an exclusively gaseous phase comprising in combination:

- (a) a compressor for compressing the gaseous cryogen, having a low pressure side and a high pressure side;
- (b) at least one turbine expander for extracting energy from the cryogen and operable in response to the pressure of gas cryogen compressed at the high pressure side of the compressor and adapted to exhaust expanded gases within the said system, said gases being at the lowest temperature within the system;
- (c) cold exchanger means in connection between said compressor high pressure side and said expander for delivery of a cooled compressed cryogen to said expander through thermal exchange between portions of the system;
- (d) a pressure modulating reservoir in connection with the turbine expander, said reservoir being connected to receive gases exhausting from the expander and locating a substantial volume of the cryogen within the system sufficient to affect the rate of expansion of said cryogen, said volume being under a pressure dependent upon the lowest temperature of said exhausted expanded gases;
- (e) return loop conduit means in gas transfer connection between said reservoir and said compressor low pressure side, said conduit means returning expanded cryogen to recompression and providing thermal exchange within said cold exchanger means; and
- (f) cold sink means in connection with said return loop conduit means and situate intermediate said reservoir and said cold exchanger, for utilizing the cold production of the system.

3. A closed loop cryogenic refrigerating system comprising in combination:

- (a) a fixed mass and volume of cryogen contained within the closed system and having an exclusively gaseous phase;
- (b) a gas compressor having a low pressure side and a high pressure side for compressing said cryogen;
- (c) at least one cold exchanger in gas transfer connection with said compressor and adapted to receive cryogen from said high pressure side for cooling said compressed cryogen;
- (d) a turbine expander for extracting energy from compressed gas cryogen and having an input port connected to receive cooled compressed gas cryogen from said cold exchanger and an exhaust port for expelling expanded cryogen at the lowest temperature within said system;
- (e) a reservoir connected to receive said cryogen expelled from said exhaust port and adapted to retain under temperature-dependent pressures a substantial volume of cryogen, said volume providing a back pressure sufficient to restrict the rate of speed of said expander turbine;
- (f) return loop conduit means in connection between said reservoir and said compressor low pressure side and adapted to provide thermal exchange within said cold exchanger; and
- (g) cold sink means in connection between said expander exhaust port and said reservoir, for utilizing the cold production of the system.

4. A closed cryogenic refrigerating system comprising:

- (a) a fixed mass of cryogen contained within the closed system having a substantially gaseous phase;
- (b) a compressor for compressing said cryogen having high and low pressure sides interconnected by pressure responsive adjustment means;
- (c) a turbine expander for extracting energy from said compressed cryogen and having an input port connected to receive compressed cryogen and an exhaust port for expelling expanded cryogen at the lowest temperature within said system;
- (d) conduit means, in thermal bond with at least one cold exchanger, in connection between said compressor high pressure side and said expander input port for delivering cooled compressed cryogen to said expander;
- (e) a pressure modulating reservoir connected to receive cryogen expelled from said exhaust port and adapted to retain under temperature-dependent pressures a substantial volume of gaseous cryogen, whereby said volume provides a variable back pressure inhibiting expander exhaust port output so as to restrict the rate of rotation of said expander turbine;
- (f) return loop conduit means in connection between said reservoir and said compressor low pressure side and adapted to provide thermal exchange within said cold exchanger; and
- (g) cold sink means in connection with said return loop conduit means and situated intermediate said reservoir and said cold exchanger, for utilizing the cold production of the system.

5. A closed cryogenic refrigerating system operating with a cryogen having a substantially gaseous phase comprising in combination:

- (a) a compressor for compressing the cryogen, having a low pressure side and a high pressure side;
- (b) valving means in connection between the low and high pressure sides of the compressor for preventing the overcompression of the cryogen;
- (c) cold exchanger means adapted to receive compressed cryogen from said compressor high pressure side for cooling said cryogen through out-of-contact, counter-current heat exchange between said compressed cryogen and cooled gases within said system;
- (d) at least one turbine expander connected to receive

cooled, compressed cryogen from said cold exchanger for extracting energy from said cryogen and adapted to exhaust expanded cryogen gases within the said system, said gases being at the lowest temperatures within the system;

- (e) a pressure modulating reservoir connected to receive exhaust cryogen expelled from said expander and adapted to retain under temperature-dependent pressures a substantial volume of expanded gaseous cryogen, whereby said volume provides a variable back pressure inhibiting expander exhaust output so as to efficiently restrict the rate of rotation of said expander turbine;
 - (f) return loop conduit means in connection between said reservoir and said compressor low pressure side, and adapted to provide thermal bond within said cold exchanger, for returning expanded cryogen to recompression; and
 - (g) cold sink means in connection with said return loop conduit means and situated intermediate said reservoir and said cold exchanger, for utilizing the cold production of the system.
6. A closed cryogenic refrigerating system operating with a cryogen having a substantially gaseous phase comprising:
- (a) a compressor for compressing the cryogen, having a low pressure side and a high pressure side;
 - (b) valving means in connection between the low and high pressure sides of the compressor for preventing the overcompression of the cryogen;
 - (c) cold exchanger means adapted to receive compressed cryogen from said compressor high pressure side for cooling said cryogen through out-of-contact, counter-current heat exchange between said compressed cryogen and cooled gases within said system;
 - (d) at least one turbine expander connected to receive cooled, compressed cryogen from said cold exchanger for extracting energy from said cryogen and adapted to exhaust expanded cryogen gases within the said system, said gases being at the lowest temperature within the system;
 - (e) a pressure modulating reservoir connected to receive exhaust cryogen expelled from said expander and adapted to retain under temperature-dependent

pressures a substantial volume of expanded gaseous cryogen, whereby said volume provides a variable back pressure inhibiting expander exhaust output so as to efficiently restrict the rate of rotation of said expander turbine;

- (f) cold sink means for utilizing the cold production of the system, in connection intermediate the turbine expander and said pressure modulating reservoir; and
 - (g) return loop conduit means in connection between said reservoir and said compressor low pressure side for returning expanded cryogen to recompression and providing thermal exchange with said cold exchanger.
7. A closed loop cryogenic refrigerating process comprising:
- (a) compressing a gaseous cryogen;
 - (b) circulating the said cryogen in a closed loop;
 - (c) precooling the cryogen;
 - (d) removing energy from the cold and compressed cryogen by expansion so as to develop a coolest temperature within the said loop;
 - (e) retaining relatively substantial volume of expanded cryogen within a portion of the loop sufficient to induce a back pressure for inhibiting the rate of expansion, said back pressure being dependent upon the degree of coolest temperatures prevailing in said system; and
 - (f) returning said expanded cryogen to compression within said system.

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