Sept. 27, 1966 W. H. HOGAN ETAL 3,274,786
CRYOGENIC REFRIGERATION METHOD AND APPARATUS OPERATING ON AN EXPANSIBLE FLUID
Filed July 27, 1964 7 Sheets-Sheet 1

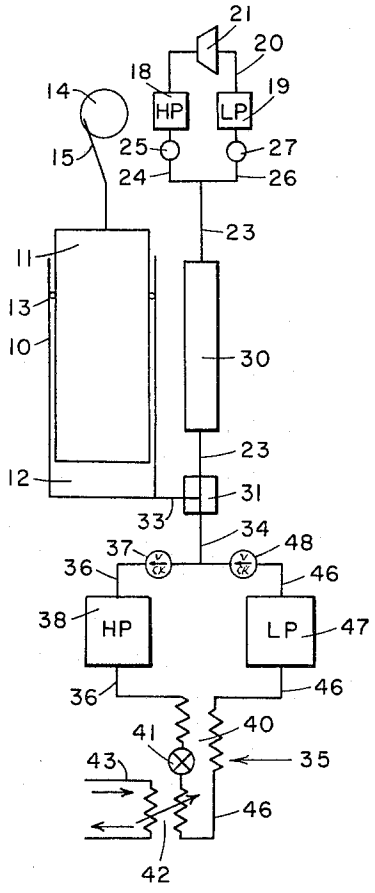

Fig. 1

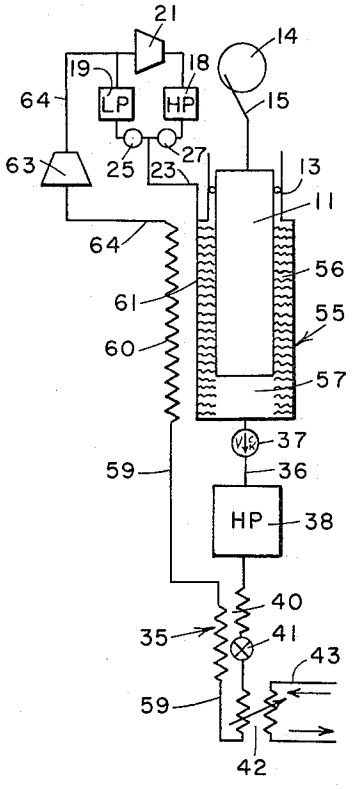

Fig. 2

INVENTORS
Walter H. Hogan
Robert W. Stuart
BY
Bruce A. Lyon
Attorney

Sept. 27, 1966 W. H. HOGAN ETAL 3,274,786

CRYOGENIC REFRIGERATION METHOD AND APPARATUS OPERATING ON AN EXPANSIBLE FLUID

Filed July 27, 1964 7 Sheets-Sheet 2

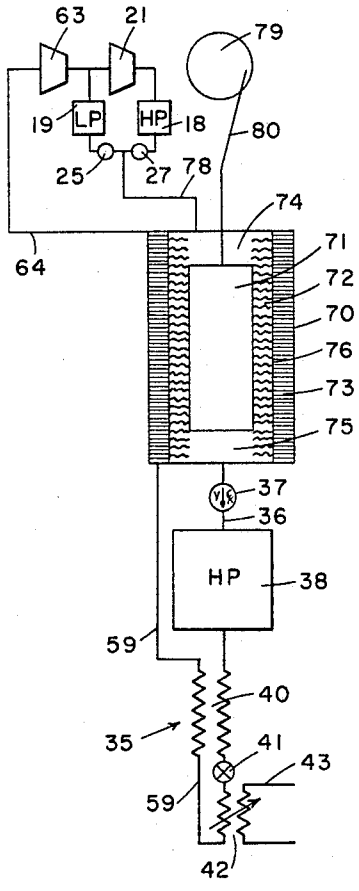

Fig. 3

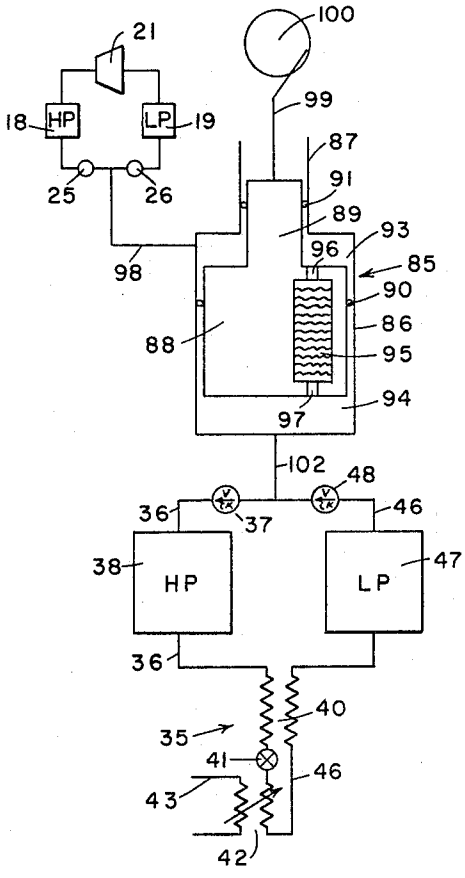

Fig. 4

INVENTORS
Walter H. Hogan
Robert W. Stuart

BY Bruce G. Lyons
Attorney

Sept. 27, 1966 W. H. HOGAN ETAL 3,274,786
CRYOGENIC REFRIGERATION METHOD AND APPARATUS OPERATING ON AN EXPANSIBLE FLUID
Filed July 27, 1964 7 Sheets-Sheet 3

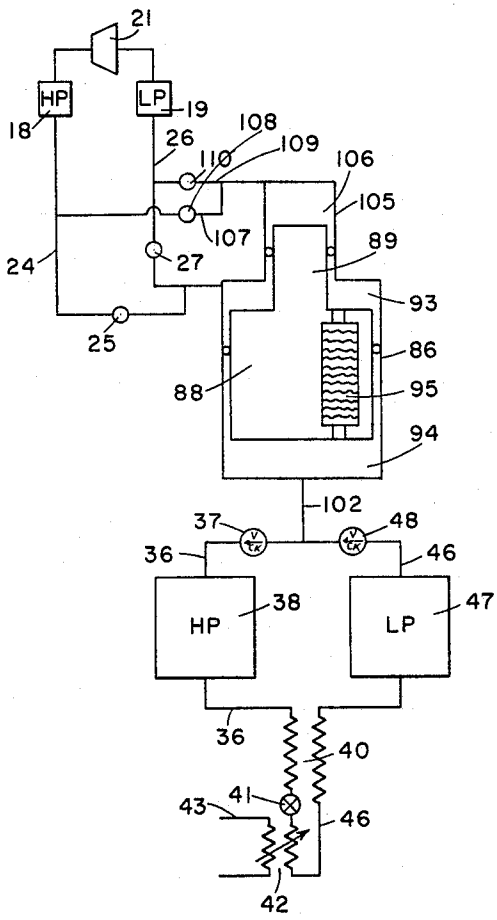

Fig. 5

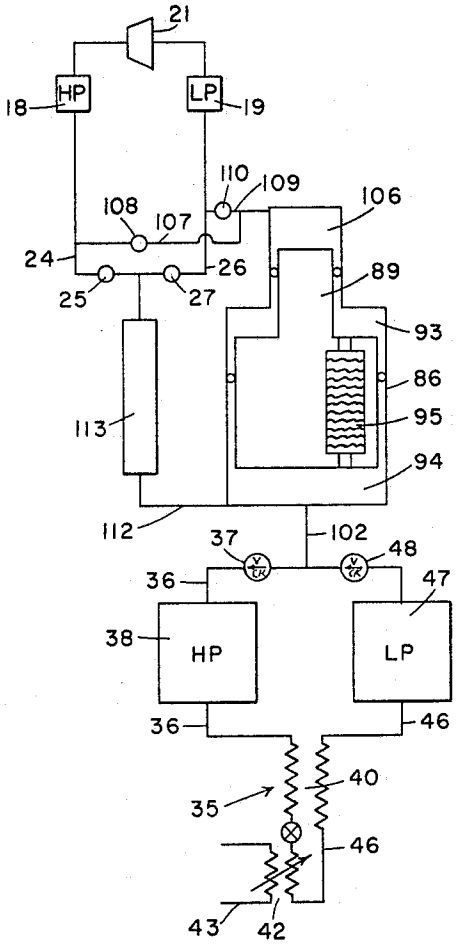

Fig. 6

INVENTORS
Walter H. Hogan
Robert W. Stuart
BY
Attorney

Sept. 27, 1966 W. H. HOGAN ETAL 3,274,786
CRYOGENIC REFRIGERATION METHOD AND APPARATUS OPERATING ON AN EXPANSIBLE FLUID
Filed July 27, 1964 7 Sheets-Sheet 4

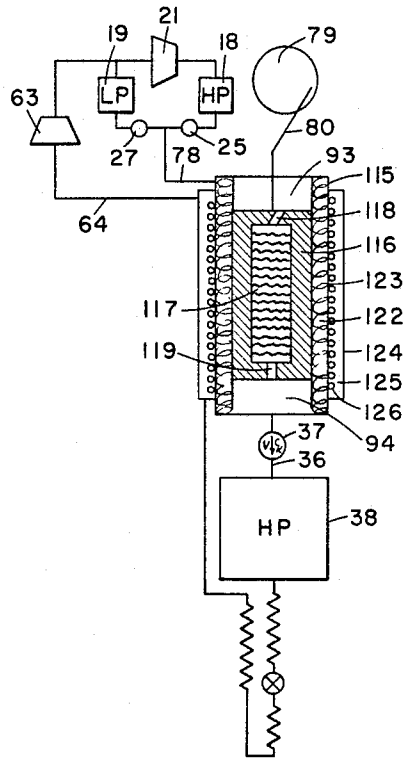

Fig. 7

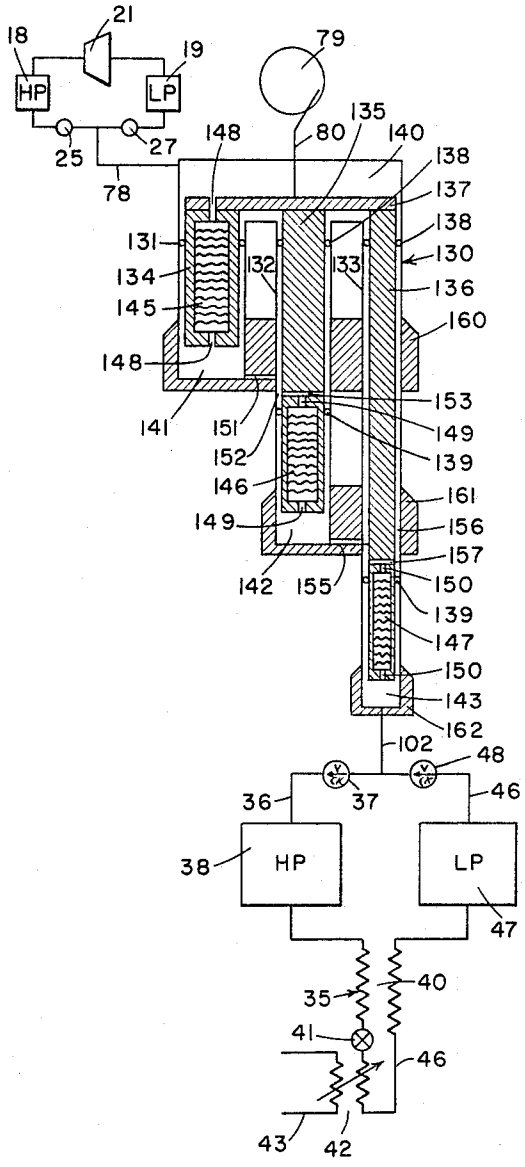

Fig. 8

INVENTORS
Walter H. Hogan
Robert W. Stuart
BY
Bruce A. Lyppee
Attorney

Sept. 27, 1966 W. H. HOGAN ETAL 3,274,786
CRYOGENIC REFRIGERATION METHOD AND APPARATUS OPERATING ON AN EXPANSIBLE FLUID
Filed July 27, 1964 7 Sheets-Sheet 5

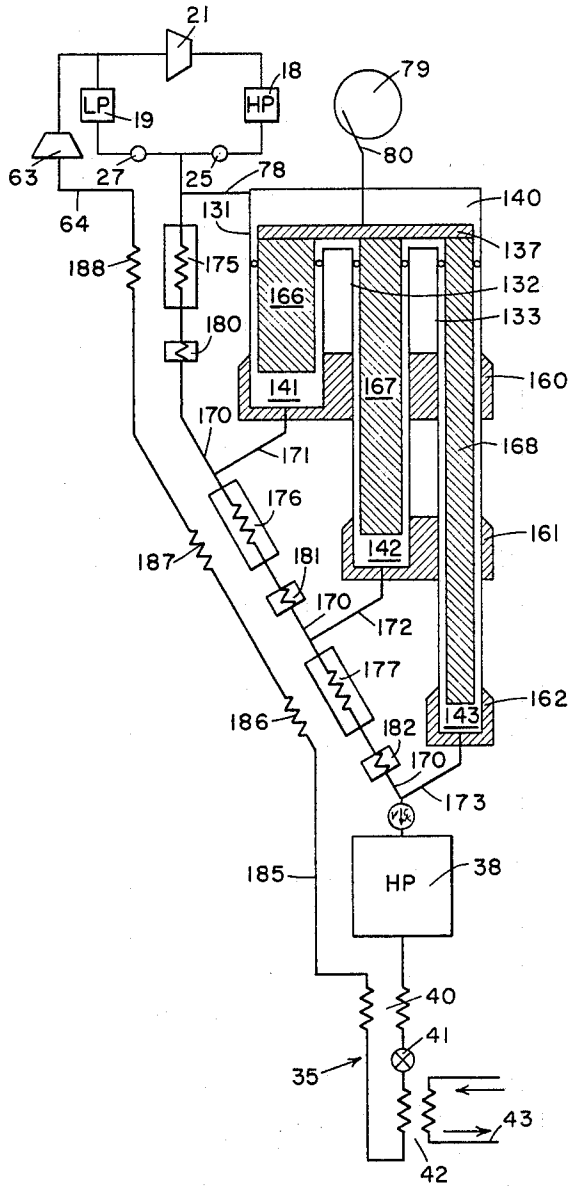

Fig. 9

INVENTORS
Walter H. Hogan
Robert W. Stuart
BY
Bruce A. Lyppe
Attorney

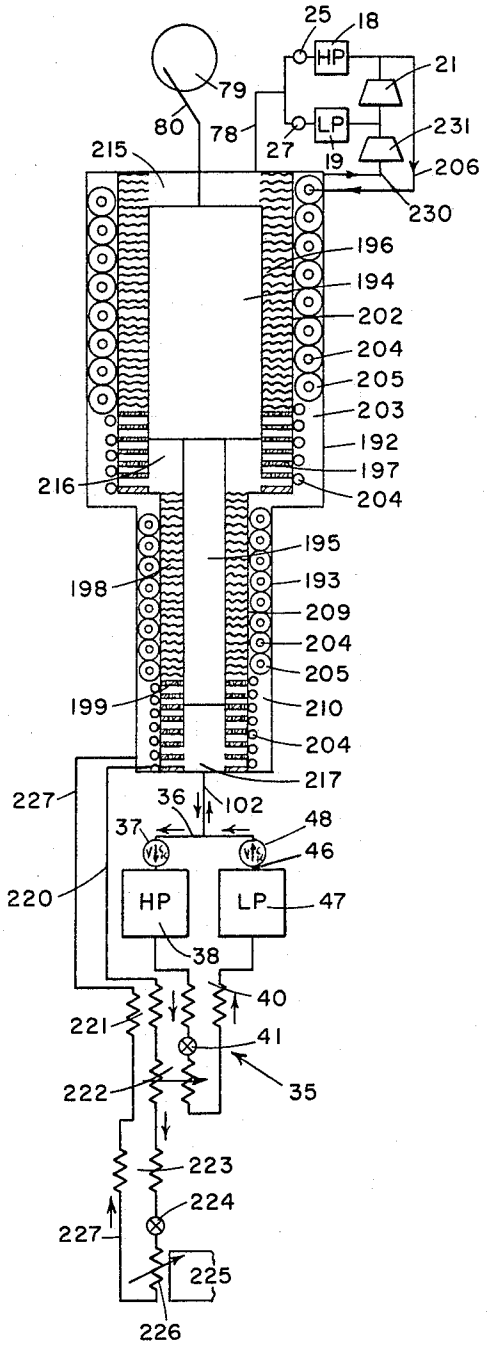

Fig. 10

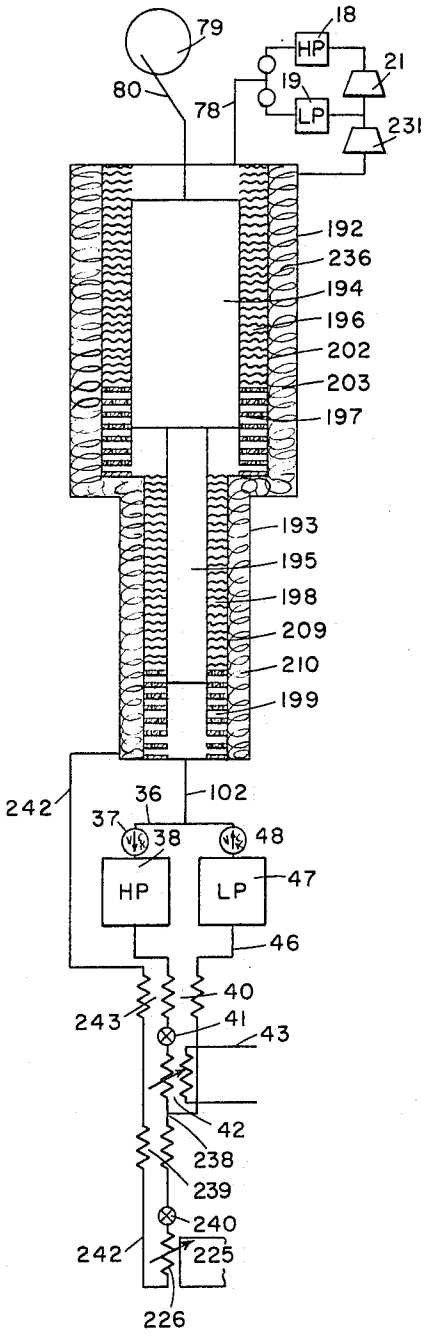

Fig. 11

INVENTORS
Walter H. Hogan
Robert W. Stuart
BY
Bruce A. Lyons
Attorney

Sept. 27, 1966 W. H. HOGAN ETAL 3,274,786
CRYOGENIC REFRIGERATION METHOD AND APPARATUS OPERATING ON AN EXPANSIBLE FLUID
Filed July 27, 1964 7 Sheets-Sheet 7

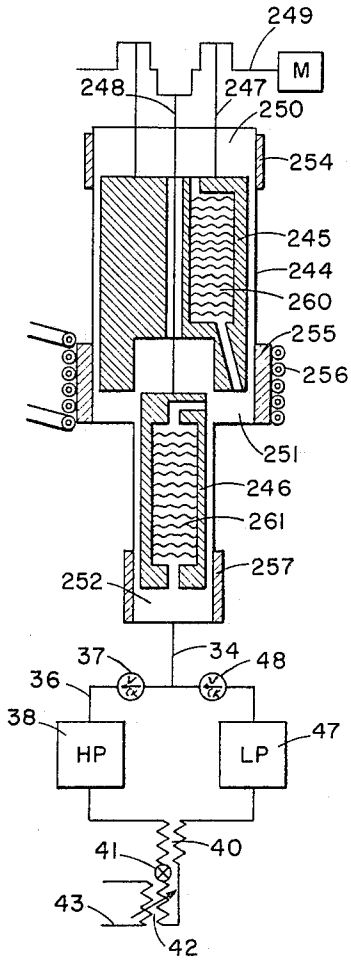

Fig. 12

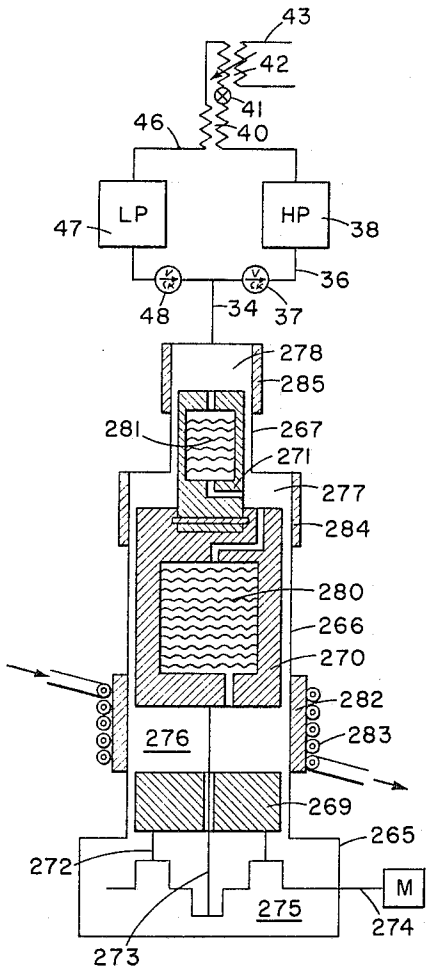

Fig. 13

Walter H. Hogan
Robert W. Stuart
INVENTORS

BY Bruce A. Lyons
Attorney

United States Patent Office 3,274,786 Patented Sept. 27, 1966

3,274,786

CRYOGENIC REFRIGERATION METHOD AND APPARATUS OPERATING ON AN EXPANSIBLE FLUID

Walter H. Hogan, Wayland, and Robert W. Stuart, Wakefield, Mass., assignors to Arthur D. Little, Inc., Cambridge, Mass., a corporation of Massachusetts

Filed July 27, 1964, Ser. No. 385,436
33 Claims. (Cl. 62—6)

This invention relates to a novel refrigeration method and apparatus, and more particularly to one which is based upon the expansion of a refrigerating fluid and incorporates a pressure controlling ballast system.

The need for highly reliable, long-term, continuous-duty cryogenic refrigeration apparatus continues to grow. This need stems from the increasing use of masers and parametric amplifiers in communication systems, such as Telstar, satellite, or missile tracking systems; superconducting computer circuitry; and high-field-strength superconducting magnets—to cite but a few examples of their applications. These applications, which were only a few years ago confined to the laboratory or to experimental field use, are now the concern of the systems engineer and demand performances approaching one hundred percent reliability. For example, the retention of a billion-bit memory or the continuous operation of a satellite communication system depends on the reliable function of liquid-helium-temperature refrigerators.

A number of unique refrigeration cycles and apparatus have been recently developed. There are based upon the controlled cycling of an expansible fluid with suitable heat exchange to obtain refrigeration. As an example of the new class of refrigerators and refrigeration cycles there may be cited U.S.P. 2,906,101, U.S.P. 2,966,035, U.S.P. 3,045,436, U.S.P. 3,115,015 and U.S.P. 3,115,016. In addition, in copending applications Serial Nos. 280,557, now U.S.P. 3,148,512; 322,781, 322,782 and 322,790, now U.S.P. 3,188,818, 3,818,819 and 3,188,820, respectively, filed in the names of one or both of us as sole or joint inventors and assigned to the same assignee as this application, there are disclosed modifications and improvements over the basic cycles and apparatus described in U.S.P. 2,906,101 (hereinafter designated the basic "work" cycle) and in U.S.P. 2,966,035 (hereinafter designated the basic "no-work" cycle). Refrigerators built in accordance with the teachings of these patents and these copending applications have exhibited continuous operation for more than one thousand hours between routine maintenance cycles. However, the demands for even greater reliability and longer continuous operation between routine maintenance have led to the development of the improved refrigeration cycle and apparatus described herein.

In the basic work and no-work cycles and their various modifications and improvements described in the above-identified patents and patent applications, refrigeration is developed through controlled fluid expansion and heat exchange; and energy is delivered external of the system substantially equivalent to the refrigeration developed. More particularly, in the work cycle the energy is delivered as mechanical energy while in the no-work cycle the energy is thermal energy and refrigeration is obtained by removing more sensible heat from the system than is taken into the system by the refrigerating fluid used. The remaining above-identified patents and copending applications disclosed modifications and combinations of these basic cycles. Common to all of these cycles is the supplying of high-pressure fluid from an external source, the initial cooling of the high-pressure fluid by means of regenerators prior to expansion while maintaining the high pressure level, and the final cooling of the

initially cooled high-pressure fluid through expansion and discharge from the refrigerator system. The lowest temperature practically attainable with a regenerator system, such as the basic cycles described, is about 10° to 12° K. This limitation is due to the thermal characteristics inherent in the materials from which regenerators can be constructed. However, these lower temperatures are well below the inversion temperature of helium, thus making the application of Joule-Thomson expansion, say from 20 atmosphere (i.e., about 300 p.s.i.a.) to 4 atmosphere (i.e., about 60 p.s.i.a) feasible to obtain refrigeration down to 6° to 8° K. Providing of refrigeration at these lower temperatures (in contrast to the otherwise limiting temperatures between 10° and 12° K.) is required to extend these unique cycles to uses such as cooling superconducting magnets and operating circuits depending upon components being at superconducting temperatures. In addition to the use of a Joule-Thomson loop the apparatus of this invention includes an essential component, one or more countercurrent heat exchangers serving as recuperators and a ballasting system to provide for substantially continuous fluid flow through the Joule-Thomson loop and heat exchangers. The ballast system may be associated with only the high-pressure side of the refrigerator or with both high-pressure and low-pressure sides. As will be apparent from the detailed description, the finally cooled low-pressure fluid may be returned and discharged either through the refrigerator and regenerator, or external of the refrigerator, in either case serving to precool the incoming high-pressure fluid.

The use of the ballast system overcomes undesirable pulsating inherent in the type of refrigerators disclosed herein. The pressure in such refrigerators pulsates between a high inlet pressure level and a low exhaust pressure level. This means that the fluid enters the high-pressure ballast chamber only when the pressure is high and exhausts from the low-pressure ballast chamber, where one is used, only when the pressure is low. However, the flow through the Joule-Thomson expansion valve is in all cases continuous and so the upstream pressure tends to drop during that period when the high-pressure ballast chamber is not being supplied with fluid, and similarly where a low-pressure ballast chamber is used the pressure in it increases because of the continuous flow during that period when the low pressure is not able to discharge into the regenerator. The extent of fluctuation is of course a function of the volume of the ballast chambers and these chambers should be sized to reduce the pressure variation on both the supply and discharge sides of the Joule-Thomson valve to a minimum. For example a fluctuation less than 5% of the absolute pressure is desirable.

When only a high-pressure ballast is used, the regenerator-refrigerator system operates best with supply-to-exhaust pressure ratios of about 5:1—i.e., between about 250 p.s.i.a. and 50 p.s.i.a. For example, using helium and these pressure ratios, cooling in the refrigerator reduces the temperature of the helium to about 15° K., and by the method and apparatus of this invention this can be further reduced to 7° K. For further reduction to 4.2° K. and liquefaction of some of the helium it is necessary to expand to one atmosphere or 14.7 p.s.i.a. Thus it will be seen that the apparatus of this invention offers the freedom of choosing pressure levels which in turn determines the temperature levels at which refrigeration can be delivered.

In an alternative arrangement the high-pressure supply to the Joule-Thomson expansion valve is provided in a similar manner but the low-pressure fluid after passing through the Joule-Thomson expansion valve returns to room temperature in thermal contact with the periodical-flow regenerator of the expansion engine rather than

through the refrigerator itself. In this way the refrigeration is recovered from the exhaust fluid. Thus a packed bed is used both as a recuperator and a regenerator. This in turn offers a significant advantage in resistance to contamination. In all low-temperature refrigeration systems there are always major problems involving seals, leakage, bearings, gas contamination, and the like. The apparatus and method of this invention reduces these problems and particularly the problem of contamination from condensable vapors.

In a recuperator heat-exchanger, wherein the flow is unidirectional, contaminants, usually in the form of the higher boiling fluids (e.g., CO_2 water vapor and methane), which enter the system tend to collect and plug the heat exchangers, expansion engines, and eventually the Joule-Thomson expansion valve. Regenerators, on the other hand, wherein the flow is reversed during each cycle, have a greatly increased tolerance for contamination, partly because the heat exchanger surface area in a regenerator can conveniently be greater than in a practical recuperator and partly because of the self-purging characteristic of the periodic flow. For example, continuous runs of several weeks' duration have been made using a single-stage regenerator-expansion engine using humid shop compressed air, filtered only for particulate matter, in an open cycle discharging to atmosphere. Temperatures down to —200° F. were maintained, although during this period many pints of water in the compressed air entered and left the regenerator which had a volume of one cubic inch. A recuperator and conventional expansion engine system would have plugged in a few minutes' running time under these conditions. Thus, the apparatus and method of this invention offer improved performance in terms of lower refrigeration temperatures, and reduction in operational problems which in turn increases reliability.

It is therefore a primary object of this invention to provide improved refrigeration cycles and apparatus which are capable of delivering refrigeration at lower temperatures. It is another object of this invention to provide apparatus of the character described which incorporates the use of a heat exchanger system serving in the role of a recuperator and regenerator and which thereby minimizes the problem of contaminant plugging. It is yet another object of this invention to provide improved refrigeration apparatus which is capable of unattended continuous operation for longer periods of time than have heretofore been attainable. Other objects of the invention will in part be obvious and will in part be apparent hereinafter.

The invention accordingly comprises the several steps and the relation of one or more of such steps with respect to each of the others, and the apparatus embodying features of construction, combinations of elements and arrangement of parts which are adapted to effect such steps, all as exemplified in the following detailed disclosure, and the scope of the invention will be indicated in the claims.

For a fuller understanding of the nature and purposes of the invention reference should be had to the following detailed description taken in connection with the accompanying drawings in which

FIGS. 1 and 2 illustrate the application of the two modifications of this invention to refrigeration apparatus of the "work" type; such as is described in detail in U.S.P. 2,906,101;

FIG. 3 illustrates the application of this invention to a no-work refrigerator such as described in U.S.P. 2,966,035;

FIG. 4 illustrates the application of this invention to a combination of the work and no-work type refrigerator as described in copending application Serial No. 322,790;

FIG. 5 illustrates the application of this invention to a pneumatically driven no-work refrigerator as described in copending application Serial No. 322,782;

FIG. 6 illustrates the application of this invention to a modified no-work refrigerator as described in copending application Serial No. 322,781;

FIG. 7 illustrates the application of this invention to a modification of the apparatus of FIG. 6 which is mechanically driven;

FIGS. 8 and 9 illustrate multistage embodiments of the no-work refrigerator with the improvements of this invention;

FIGS. 10 and 11 illustrate the application of this invention to no-work refrigerators which are multistaged and have a stepped configuration as described in copending application Serial No. 280,557; and

FIGS. 12 and 13 illustrate the application of this invention to modified Taconis and typical Sterling apparatus.

In FIG. 1 a typical closed cycle, work type refrigerator constructed in accordance with this invention is illustrated in a somewhat diagrammatic fashion. The basic refrigerator and the cycle on which it operates is described in detail in U.S.P. 2,906,101. It will be seen to consist of an enclosure **10** in which a piston **11** is caused to move to define within the enclosure an expandable-volume chamber **12** in which refrigeration is normally accomplished. Sealing rings **13** associated with the piston **11** provide a fluid-tight seal within enclosure **10** and the work output of the piston is absorbed by suitable means such as a brake **14** which is connected to the piston by a shaft **15**. High-pressure expansible fluid is provided from a source **18**, while the low-pressure fluid discharged from the refrigerator is delivered to a low-pressure reservoir **19** which in turn is connected though conduit **20** and compressor **21** to the high-pressure source **18**. Such connection may, of course, be eliminated in an open cycle system. A main fluid conduit **23** leads to the refrigerator and is connected to the high-pressure source **18** through conduit **24**, controlled by valve **25**, and to the low pressure reservoir **19** through conduit **26**, controlled by valve **27**. Located in the main fluid conduit **23** is a heat storage means, typically a regenerator **30**, and a heat station **31**. Conduit **23** leads into branch conduit **33** which is in fluid communication with chamber **12**, and into branch conduit **34** which in turn is connected to the Joule-Thomson loop generally designated at **35**. High-pressure conduit **36**, which is controlled by a one-way check valve **37**, leads into a high-pressure ballast **38** and thence into a heat exchanger **40**. At the cold end of heat exchanger **40** is the Joule-Thomson expansion valve **41** which in turn is connected to heat exchanger **42** which in this case represents the external refrigeration load shown to effect out-of-contact heat exchange with a fluid flowing in a conduit **43**. It will of course be appreciated that this refrigeration load may be a solid body in thermal contact with heat exchanger **42** or in any other suitable form for transferring refrigeration from the cold expanded fluid leaving the Joule-Thomson valve **41**. After delivering refrigeration, the cold low-pressure fluid is returned to the system by means of low-pressure conduit **46**, through heat exchanger **40**, into low-pressure ballast **47** and finally back into the refrigeration system through one way check valve **48**.

It will be helpful in explaining the operation of the Joule-Thomson loop, the ballast system and the heat exchangers to first very briefly review the basic work cycle. High-pressure fluid is supplied to refrigerator chamber **12** (FIG. 1) from high-pressure reservoir **18** through valve **25** and conduits **24** and **23**, during that portion of the cycle when the piston **11** is moving upward and chamber **12** is attaining maximum volume. Valve **27** is closed during this time when high-pressure fluid is being supplied. In the supplying of high-pressure fluid to chamber **12** the fluid is cooled in regenerator **30** by virtue of the refrigeration stored there in the previous cycle. The high-pressure valve may be closed only when chamber **12** has reached maximum volume or it may be closed shortly before the

piston has reached top dead center. The passage of cold fluid in and out of heat station 31 serves to stabilize the fluid temperature at that point, i.e., just before discharge or entry into regenerator 30 and hence to optimize regenerator performance.

At this point in the basic cycle there is cold high-pressure fluid in chamber 12 and the next step is to close valve 25 and open valve 27 to permit the expansion and further cooling of the fluid and its discharge into low-pressure reservoir 19 with the cooling of regenerator 30. The piston 11 also moves downwardly and chamber 12 reaches its minimum volume. The cycle is then ready to begin again.

Returning now to FIG. 1 it may be shown how this basic work cycle is modified to obtain refrigeration at a lower temperature through the use of Joule-Thomson expansion without encountering the usual operational problems of contaminant plugging inherent in the normal heat exchange systems associated with Joule-Thomson valves and without carrying over into the Joule-Thomson loop the pulsating operation characteristic of the basic cycle. During the supplying of high-pressure fluid to chamber 12, the pressure within the system builds up, for example to 20 atmospheres. Check valve 37 allows the flow of cold high-pressure fluid from chamber 12 by way of heat station 31 to flow into high-pressure ballast 38 to raise the pressure in this ballast chamber to 20 atmospheres. When the refrigerator exhaust valve 27 is open and the refrigerator pressure drops to 4 atmospheres for example, any excess pressure in low-pressure ballast 47 is allowed to pass check valve 48 into the regenerator and back through the regenerator 30. The net result is to maintain fluid in ballast 38 at approximately 20 atmospheres and at approximately 4 atmospheres in ballast 47. Of course, other fluid pressures may be used depending upon the operational conditions chosen. By using ballast chambers of sufficient size it is possible to provide a substantially continuous flow of fluid through the countercurrent heat exchanger 40, the Joule-Thomson expansion valve 41 and the refrigeration load 42. Prior to its entry into the Joule-Thomson valve 41, the cold high-pressure fluid is further cooled by out-of-contact heat exchange with the colder low-pressure fluid which has been expanded in the Joule-Thomson valve 41 and delivered some refrigeration to the load 42. Subsequently, the low-pressure fluid enters low-pressure ballast 47 and is returned to the refrigerator to cool regenerator 30 during that part of the basic cycle in which low-pressure fluid is discharged into low-pressure reservoir 19. The check valve 48 is so adjusted to coordinate the flow of fluid from ballast 47 to coincide with the exhausting step of the refrigerator.

Using helium as the refrigeration fluid and the pressure ratio given in this example it is possible to reduce the temperature at which refrigeration is delivered to an external load from 24° K. to 8° K. using the simple one stage apparatus shown. By precooling the fluid in regenerator 30, contaminants such as water vapor, CO_2, methane, and the like are frozen out of the fluid stream and deposited on the regenerator packing, thus providing to the heat exchanger 40 and the Joule-Thomson valve 41 a fluid which is free of the contaminants which would otherwise rapidly plug their passages. Moreover, the returning fluid causes the solidified contaminants in the regenerator to revaporize and these are swept out of the system, thus preventing any accumulation of solids in the regenerator over any extended period of operation.

FIG. 2 is a modification of the apparatus of FIG. 1 in that no low-pressure ballast is provided and that the low-pressure cold fluid is not returned through the refrigeration system directly, but is used to effect heat exchange between the low pressure cold fluid and the regenerator. This modification also has the regenerator surrounding the piston and located within the refrigerator itself. In FIG. 2 identical elements bear the same reference numbers as those in FIG. 1 and the basic cycle is the work cycle.

It will be seen in FIG. 2 that the enclosure 55 which defines the refrigeration system contains within it, in the form of an annular ring, the regenerator 56 which surrounds the piston 11. This in turn eliminates the external passages in FIG. 1. Operation of the apparatus of FIG. 2 is substantially the same as that of the apparatus of FIG. 1 to the point where the cold expanded fluid delivers refrigeration to load 42. Since there is no low-pressure ballast corresponding to ballast 47 of FIG. 1 provision must be made to return the low-pressure fluid into the refrigeration system. For this purpose there is provided low-pressure conduit 59 which is in fluid communication with a heat exchanger 60 in thermal contact with the regenerator 56 through the walls 61 of the enclosure 55. The heat exchanger 60 is in turn connected to a booster compressor 63 through a suitable conduit 64 which in turn connects into the main compressor 21, thus returning the refrigeration fluid to the system for recycling.

In the case of the apparatus of FIG. 2 only the high-pressure stream is supplied from the refrigerator system at the chosen high-pressure level of the system. The fluid in ballast 38 is allowed to flow through heat exchanger 40 into the Joule-Thomson valve 41 to expand to a low-pressure level which is determined by the booster compressor 63. The low-pressure fluid returns outside the regenerator through out-of-contact heat exchanger 60 to exchange heat between the low-pressure stream and the incoming high-pressure stream travelling down through regenerator 56. Thus this modification also provides for refrigeration at lower temperatures and the removal of contaminants in the self-purging regenerator prior to the entrance of the fluid into the heat exchangers and Joule-Thomson valve.

FIG. 3 illustrates the application of the ballast and the Joule-Thomson loop to refrigeration apparatus capable of operating on the so-called "no-work" cycle, as described in detail in U.S. Patent No. 2,966,035. This cycle differs from that of the apparatus of FIG. 1 in that there are two chambers (warm and cold) the volumes of which are controlled by the movement of a displacer driven by suitable external means. Heat of compression is developed in the warm chamber and the high-pressure fluid enters the regenerator and leaves the system at a temperature somewhat higher than that at which it was supplied. Thus the discharged fluid carries with it out of the system a quantity of heat which is substantially equivalent to the refrigeration developed. The embodiment of FIG. 3 employs only a high-pressure ballast and returns the low pressure-cooled fluid through a heat exchange system which is an integral part of the refrigerator. In FIG. 3 like numbers refer to like elements in FIGS. 1 and 2. The refrigerator enclosure 70 contains within it a displacer 71 which is surrounded by an annular regenerator 72 and an annular recuperator 73 which are in thermal contact with each other such as through a suitable metal wall 76. In its movement within this enclosure the displacer 71 defines two variable-volume chambers, i.e., warm chamber 74 and cold chamber 75. The high-pressure fluid in this no-work apparatus is introduced through conduit 78 into chamber 74 and motion is imparted to the displacer 71 by means of a suitable driving mechanism such as wheel 79 which is connected to the displacer through shaft 80.

The modification of FIG. 3 offers the opportunity of more efficient heat transfer between the high-pressure fluid passing through regenerator 72 and the cold low-pressure fluid entering recuperator 73 from conduit 59 for the recuperator may be so constructed as to provide an increased surface, such as through the use of fins, for heat transfer. It should be noted that since the flow of high-pressure fluid through regenerator 72 is periodic and of low-pressure fluid through recuperator 73 is continuous the heat transfer is not that which normally takes place in a countercurrent heat exchanger.

The use of a low-pressure ballast in conjunction with

the high-pressure ballast is, of course, also applicable to the no-work cycle, in which case the refrigerator enclosure **10** of FIG. 1 would be completely enclosed to define a second warm chamber, the piston would be a displacer and the conduit **23** would be connected to the upper chamber, comparable to the arrangement of FIG. 3. The cold low-pressure fluid would then be returned through the refrigerator as in FIG. 1.

FIG. 4 illustrates the application of this invention to the refrigeration apparatus and method described in copending application Serial No. 322,790. It will be seen that this apparatus combines the features of the work and no-work apparatus and cycle. The apparatus embodies a modified configuration of the enclosure **85** which is seen to be formed of a lower section **86** and an upper section **87**. The volume defining means within the enclosure **85** has a configuration essentially corresponding to that of the enclosure, namely a lower portion **88** and an upper portion **89**, each having associated with them sealing rings **90** and **91**. This movable member is in fact a combination of a piston and displacer and defines within the enclosure **85** an upper chamber **93** and a lower chamber **94** which are connected by a fluid path being in this case a regenerator **95** located within the bottom portion **88** of the piston-displacer and connected to the upper and lower chambers through conduits **96** and **97**, respectively.

Communication between the refrigerator and the high-pressure fluid supply **18** and low-pressure reservoir **19** is by way of passage **98**. The piston-displacer is mechanically connected to a shaft **99** and to a wheel **100**, which by virtue of the operation of the refrigerator serves partly as a driving means and partly as a means for absorbing mechanical energy.

The operation of the apparatus of FIG. 4 as far as the Joule-Thomson loop **35**, the ballast, and the heat exchange system are concerned, is the same as that of the apparatus of FIG. 1.

FIG. 5 is a modification of the apparatus of FIG. 4 showing the use of pneumatic drive in place of a mechanical driving means. This requires that the upper section **105** of the enclosure be fluid tight to define a driving chamber **106** into which high-pressure fluid is introduced through conduit **107**, controlled by valve **108**, and released through conduit **109**, which is controlled by valve **110**. Through the proper timing of valves **25**, **27**, **108**, and **110**, it is possible to introduce and release fluid from the system to drive the piston displacer to furnish the required low temperature high pressure fluid into chamber **94**. The manner in which this is accomplished is described in detail in copending application Serial No. 322,782.

FIG. 6 illustrates the application of this invention to a pneumatically driven modified no-work refrigeration apparatus and cycle. In addition to the apparatus components shown in FIG. 5, in which like numbers refer to like elements, this refrigerator incorporates an additional fluid path communicating between the cold chamber **94** and the high-pressure source and low-pressure reservoir. Within this path is located a second regenerator **113**. The basic apparatus and the cycle on which it operates is described in detail in copending application Serial No. 322,781. It is of course possible to use a pneumatic means as shown in FIGS. 5 and 6 to drive any of the other apparatus.

FIG. 7 embodies the use of the same general apparatus and cycle for the refrigeration portion as the apparatus of FIG. 6. However, it will be seen that the ballast and Joule-Thomson loop are those associated with the apparatus of FIGS. 2 and 3, and that both of the regenerators are located within the refrigerator enclosures **115**. The displacer **116** has the primary regenerator **117** located in its central interior and the fluid path between upper chamber **93** and lower chamber **94** is completed through passages **118** and **119**. A secondary regenerator **122** is in the form of an annular ring around the displacer **116** and is in thermal contact with the central wall **123** of the refrigerator. Around this is a second heat exchanger **124** which defines a fluid path **125** containing surfaces **126** designed to materially increase the heat transfer surface and hence to improve the thermal contact between the regenerator **122** and the cold fluid path **125** through enclosure wall **123**. As in the case of the apparatus in FIGS. 2 and 3 a secondary booster **63** and its attendant fluid conduit **64** are provided to return the low-pressure fluid passing through heat exchanger **124** into the main system.

FIG. 8 illustrates this invention as applied to a multistaged no-work refrigeration apparatus and cycle. The refrigerator enclosure **130** will be seen to be formed in three sections **131**, **132**, and **133**. Within each of these sections is a displacer **134**, **135**, and **136**, respectively. In this modification all of those displacers depend from and are supported by a common support head **137**. Sealing rings **138** are provided to form a fluid tight seal between the upper portion of each displacer and the enclosure section in which it moves; and sealing rings **139** are provided to isolate the lower portions of enclosures **132** and **133**. There is in this refrigerator a single upper warm chamber **140** and three successively colder lower chambers **141**, **142**, and **143** the sum of which correspond to lower cold chamber **94** in FIG. 4.

Each of the displacers has an internal regenerator **145**, **146**, and **147** and with each of these is associated passageways **148**, **149**, and **150** internal of the displacers **134**, **135**, and **136**, respectively. The fluid path between the upper warm chamber **140** and the cold chambers may be traced as follows. From upper chamber **140** it leads through internal passageway **148** which incorporates the internal regenerator **145** into the first cold chamber **141**. This is in turn connected by means of passage **151** to an annular passage **152** within section **132** which is defined by the internal walls of section **132** and the wall of the displacer **135** moving therein and is terminated by sealing rings **138** and **139**. Displacer **135** is also provided with a transverse fluid passage **153** which leads from the annular passage **152** into internal passage **149** and regenerator **146** into cold chamber **142**. In like manner passages **155**, annular passage **156**, transverse passage **157**, internal passage **150**, and regenerator **147**, complete the fluid path into cold chamber **143**.

Three heat stations are thermally bonded to the cold chambers **141**, **142**, and **143**. Heat station **160** serves as a thermal contact between the cold fluid in chamber **141** and sections **132** and **133** of the refrigerator, thus providing some precooling of these sections. In a like manner heat station **161** thermally bonds the cold chamber **142** with section **133**. The purpose of heat station **162** is to maintain the coldest chamber **143** at a more consistently low temperature than would be possible if it were not provided. Heat stations **160** and **161** also serve as temperature equalizers and hence increase the efficiency of the regenerators.

The ballasts **38** and **47** operate in the same manner as those in FIG. 1, and all of the cold low-pressure fluid is returned within the refrigerator by way of chambers **143**, **142**, and **141** and the fluid path described. This multistage arrangement makes possible the attainment of lower refrigeration temperatures, down to 4.2° K. at the outlet of the Joule-Thomson valve. Since contaminants are removed in stages in regenerators **145**, **146**, and **147**, the heat exchanger **40** and Joule-Thomson valve **41**, even though operating in this very low temperature range are not subject to plugging. Thus there is provided a refrigeration system capable of delivering refrigeration at very low temperatures (e.g., liquid helium) and operating continuously over an extended period of time.

The apparatus of FIG. 9 also represents the application of this invention to a three-stage, no-work refrigerator and cycle. However, as in the case of FIG.3, only

one ballast (that is tne high-pressure ballast) is used and the cold low-pressure fluid resulting from the expansion in the Joule-Thomson valve **41**, after giving up refrigeration to the external load **42**, is returned via heat exchanger **40** by way of out-of-contact heat exchange with a series of regenerators as in the case of FIG. 3. In this apparatus the displacers **166**, **167**, and **168** do not contain regenerators. This in turn requires that an external fluid path **170** be supplied to furnish the necessary fluid communication between the upper warm chamber **140** and the lower chambers **141**, **142** and **143** by way of branch conduits **171**, **172**, and **173**, respectively. In the fluid path **170** are the three regenerators **175**, **176** and **177**, and three heat stations **180**, **181**, and **182**, the latter being designed to stabilize temperature and to optimize regenerator efficiency. The low-pressure fluid after passing through heat exchanger **40** is returned to the system by means of conduit **185** and out-of-contact heat exchangers **186**, **187**, and **188** designed to effect heat exchange with regenerators **177**, **176**, and **175**, respectively.

It would of course be possible to use both a high-pressure and low-pressure ballast with the apparatus of FIG. 9 as shown in FIG. 8. In such an arrangement, the heat exchanges **186**, **187**, and **188** would be eliminated and all of the cold low-pressure fluid would be returned via a low-pressure ballast, and one-way valve (as ballast **47** and valve **48** of FIG. 8) into the coldest refrigeration chamber **143**.

Finally, FIGS. 10 and 11 illustrate the application of this invention to "no-work" type apparatus which are multistaged and formed in a concentric, stepped configuration. This basic refrigeration apparatus is described in detail in copending application Serial No. 280,557.
change with regenerators **177**, **176**, and **175**, respectively.

The refrigeration system of FIG. 10 incorporates a multi-stage, stepped no-work refrigerator with the Joule-Thomson loop and ballasts and a second separated Joule-Thomson loop associated with a separate heat exchange system which in turn is associated with the regenerator-heat station system of the refrigerator. The use of the refrigeration developed in the first Joule-Thomson expansion to precool fluid for the second Joule-Thomson expansion in a separate loop permits the final attainment of temperatures down to 2.5° K. provided the expansion in the second colder loop is down to about 0.1 atmosphere pressure. The use of the two loops also makes it possible to use flow rates optimized for the two temperature and pressure levels attained. Finally, since only a portion of the fluid is expanded to about 0.1 atmosphere the compressor displacement for boosting the fluid in the system is less than if the entire quantity of fluid were expanded.

The stepped configuration of the refrigerators in FIGS. 10 and 11 are described in detail in copending application Serial No. 280,557. It will therefore only be necessary to identify the elements which go to make up this refrigerator structure. The main refrigerator enclosure is formed of an upper section **192** and a lower section **193** which make up a fluid-tight enclosure. Within this enclosure a displacer, likewise formed of an upper section **194** and a lower section **195**, moves vertically. In the upper section there is an annular regenerator **196** and below it an annular heat station **197**, the former being typically constructed of copper screen and the latter of spaced perforated lead discs. The detail of these constructions is described in copending application Serial No. 280,557. In like manner there is an annular regenerator **198** and an annular heat station **199** in the lower section **193** of the refrigerator.

It will be apparent that such a refrigerator is not limited to two sections, i.e., two stages, and that a number of such stages may be used, each successive chamber being smaller to accommodate the decrease in fluid mass due to the lower temperatures. This decrease in volume may be one which is achieved by a conically shaped enclosure as illustrated in FIG. 11 of copending application Serial No. 280,557 and it is meant to include such a configuration within the meaning of the term "stepped" configuration.

Surrounding the annular regenerator **196** (FIG. 10) is a fluid tight inner wall **202** which with the outer enclosure wall **192** forms a fluid passage **203** in which heat transfer tubing **204** having fins **205** is wound helically. Into tubing **204** is introduced a separate stream of high-pressure fluid by way of conduit **206**. That portion of tubing **204** which corresponds to the section of the refrigerator occupied by heat station **197** does not have finning but makes a direct thermal bond to the internal wall **202** and hence to the heat station **197**. In like manner internal wall **209** in the bottom section defines a fluid passage **210** which is in communication with passage **203**, and in this fluid passage is tubing **204** finned down to the position of heat station **199**. The unfinned portion is thermally bonded through wall **209** to heat station **199**. Within the refrigerator enclosure are a warm chamber **215** and two cold chambers **216** and **217**. Communication between these two chambers is of course by way of the regenerators and heat stations which are open to the passage of a fluid.

As in the case of FIG. 8 previously described, this apparatus has two ballast chambers and a first Joule-Thomson loop which returns all of the fluid through the main refrigerator. It also has a second Joule-Thomson loop which consists of a high-pressure fluid conduit **220** in fluid communication with the heat transfer tubing **204** which in turn is thermally bonded to heat station **199** and hence to the coldest end of the refrigerator itself. This fluid conduit **220** leads to heat exchanger **221** which in turn is in out-of-contact heat exchange relationship with heat exchanger **40** of the first loop and hence is designed to effect a first precooling of the separate fluid stream in the second Joule-Thomson loop. There is also provided a second heat exchanger **222** which in effect is the external refrigeration load to which fluid in its coldest condition after passing through the first Joule-Thomson loop supplies refrigeration prior to its direct return into the refrigerator. A third heat exchanger **223** serves to effect a third and last precooling prior to the introduction of the separate fluid stream into a second Joule-Thomson valve **224** for expansion and final cooling. The external refrigeration load **225** is represented here as a body **225** thermally bonded to heat exchanger **226** which contains the coldest fluid in the entire system. The cold expanded fluid return conduit **227** is in fluid communication with heat exchangers **223** and **221** and fluid passage **210** surrounding tubing **204**. The low-pressure fluid is removed from this separate flow path by conduit **230** which takes the low-pressure warmed return fluid by way of compressor **231** into the main fluid supply system.

FIG. 11 again shows the concentric design of displacer-regenerator system but with a combination of the arrangement shown in FIG. 3. There is then a primary loop which operates between the two pressure levels of the engine to provide a level of refrigeration down to 6 or 8° K. However, instead of using the refrigeration produced through the first Joule-Thomson valve **41** to refrigerate a separate stream as shown in FIG. 10, some of the gas which has been expanded from 20 atmospheres down to 4 atmospheres, for example, is now allowed to pass through conduit **238** to a further countercurrent heat exchanger **239** to expand through a second Joule-Thomson valve **240** to provide refrigeration to an external load **225** in the heat exchanger **226** at a temperature, dictated by the return pressure at that point, which may be as low as 2.5° K. The low-pressure return **242** now passes countercurrent through heat exchanges **239** and **243** and finally in heat exchange relation with the regenerators all the way up to room temperature. The advantages of this arrangement are that the mass of gas required to be handled at the lowest pressure is greatly reduced because the last stage of refrigeration is at a lower temperature, say 5° K. instead of 12° to 15° K., and also all of the gas

in this case that goes into the Joule-Thomson loop has first passed through the regenerator system and has been purified of any contaminants that were originally in the gas stream.

The apparatus of FIG. 11 employs the same basic refrigerator as that of FIG. 10. However, it will be noted that the finned tubing and the tubing associated with the two heat stations in the apparatus of FIG. 10 is replaced with the use of suitable means to extend the heat transfer surface available in the fluid passages **203** and **210**. Such a means may constitute fins **236** or flat wires.

In addition to the unique refrigeration cycles and apparatus which have been identified and to which the Joule-Thomson loop has been applied and described with reference to FIGS. 1 through 11, the improvement of this invention is also applicable to two other general types of cycles and apparatus which may be referred to as a modified Taconis cycle and a Sterling cycle. The application of the Joule-Thomson loop to these apparatus is illustrated in FIGS. 12 and 13.

FIG. 12 shows in diagrammatic fashion a typical modified Taconis apparatus. Inasmuch as the apparatus and cycle are described in full in U.S.P. 2,567,454, and in the "Proceedings of the 1956 Cryogenic Engineering Conference of the University of Colorado," Boulder, Colorado, February 1957, pp. 188–196, it is not necessary to go into any great detail or to present the apparatus in its most complete form. For this reason in FIG. 12 it is shown diagrammatically and sealing rings etc. are not included inasmuch as these design refinements are not part of this invention.

The modified Taconis apparatus illustrated in FIG. 12 is comprised of a stepped enclosure **244** having operable therein an upper piston **245** and a lower piston **246**, the former being connected to rod **247** and the latter to rod **248** for independent driving from a suitable drive shaft **249**. (The terms "upper" and "lower" are used only for convenience and indicate their relative positions.) By their movement within the enclosure **244**, pistons **245** and **246** define what may be conveniently referred to as a warm chamber **250**, an intermediate-temperature chamber **251** and a cold chamber **252**. These chambers are of variable volume, the actual volume of which depends upon the movement of the pistons. Inasmuch as this cycle requires variable volume chambers to be maintained at three different temperature levels, the apparatus of FIG. 12 is shown to include a heat station **254** associated with the warm chamber, a heat station **255** having cooling coils **256** associated with the intermediate temperature chamber **251**, and a heat station **257** associated with the cold chamber **252**. As in the case of the other apparatus described above, there are located within the two pistons two regenerators **260** and **261**, their purpose being the same as the regenerators in the other apparatus described. Inasmuch as these regenerators are located internal of the apparatus and provide no means for out-of-contact heat exchange with returning low-pressure cold fluid, the Joule-Thomson loop includes both the high-pressure and low-pressure ballast along with the heat exchanger **40**, the Joule-Thomson expansion valve **41** and means for delivering refrigeration to an external load such as represented at **43**.

FIG. 13 illustrates the application of the external Joule-Thomson loop to a typical Sterling engine as it is embodied in a Philips engine. Like the apparatus of FIG. 12, this apparatus is represented in diagrammatic fashion inasmuch as the actual Philips-Sterling apparatus is known and described (see for example U.S.P. 2,657,553). This apparatus like the modified Taconis is a completely closed cycle in contrast to the apparatus illustrated in FIGS. 1 through 11 in which compressed fluid is supplied from an external source. It will be appreciated from an examination of FIGS. 12 and 13 that in these apparatus the fluid compression actually takes place within the refrigerator enclosure.

The typical Philips-Sterling apparatus in FIG. 13 is seen to consist of an enclosure made up of a lower section **265**, an intermediate or middle section **266**, and an upper or refrigeration section **267**. Within this enclosure there are operable a compressor piston **269** and two refrigerator pistons **270** and **271**. These pistons are driven by means of rods **272** associated with the compressor piston and rod **273** associated with the refrigerator piston, both of them being connected mechanically to a drive shaft **274**. Within the enclosure are defined a drive mechanism chamber **275**, a compressor chamber **276**, an intermediate temperature chamber **277** and a cold or refrigeration chamber **278**. Within the two refrigerator pistons are regenerators **280** and **281**, the purpose of which is the same as has already been described above for the other apparatus. As in the case of the apparatus of FIG. 12, the chambers **276**, **277** and **278** are to be maintained at three different temperature levels and for this purpose heat station **282** with cooling coils **283** is associated with the compressor chamber **276**, heat station **284** with chamber **277** and heat station **285** with chamber **278**. The Joule-Thomson loop is associated with the coldest chamber **278** and consists of both a high-pressure and a low-pressure ballast along with the heat exchange system illustrated. As in the case of the modified Taconis apparatus, the regenerators are internal of the piston and hence the cold low-pressure fluid returned from the Joule-Thomson loop must be reintroduced into the cold expansible chamber **278** for heat exchange with the regenerator **281** and **280**.

It will, of course, be understood that the apparatus of FIG. 13 represents but one type of Sterling engine and that the use of the Joule-Thomson loop in accordance with this invention is applicable to the other well-known Sterling apparatus so long as they embody the feature of an expansible-volume chamber to which initially cooled high-pressure fluid is delivered for subsequent expansion.

The apparatus and method of this invention can be seen to provide refrigeration at lower temperature levels than has heretofore been possible using a cycle which develops refrigeration through a combination of fluid expansion and regeneration. Where the expansible fluid is cooled first in a regenerator prior to being finally expanded in a Joule-Thomson valve, any contaminants present in the fluid are removed in the regenerator portion of the heat exchange system. Since the regenerator is in effect self-purging, the refrigeration system may continue to operate at these low temperatures over long periods of time. The combination of low temperatures and reliability over extended periods of time makes it possible to incorporate the refrigerators of this invention in many new devices which require such performance characteristics.

It will thus be seen that the objects set forth above, among those made apparent from the preceding description, are efficiently attained and, since certain changes may be made in carrying out the above method and in the constructions set forth without departing from the scope of the invention, it is intended that all matter contained in the above description or shown in the accompanying drawings shall be interpreted as illustrative and not in a limiting sense.

We claim:

1. A cryogenic refrigeration apparatus in which a movable member defines within an enclosure at least one chamber of variable volume and in which a high-pressure expansible fluid is introduced through a fluid flow path, incorporating at least one heat storage means, into said chamber and is subsequently expanded and then discharged through said fluid flow path to a low-pressure reservoir, characterized by having a Joule-Thomson loop associated with said chamber, said Joule-Thomson loop comprising

(a) a high-pressure ballast in fluid communication with said fluid chamber and adapted periodically to receive cold high-pressure fluid from said chamber,

(b) an expansion valve adapted to expand to further cool said high-pressure fluid delivered thereto from said high-pressure ballast at an essentially constant rate,

(c) a heat exchanger adapted to effect out-of-contact heat exchange between said cold high-pressure fluid entering said expansion valve and the expanded and further cooled fluid resulting from passage through said expansion valve; and

(d) means for returning said expanded and further cooled fluid in heat exchange relationship with said heat storage means to said low-pressure reservoir over at least that period of time corresponding to that part of each refrigeration cycle when said fluid is being expanded and discharged to said low-pressure reservoir.

2. A cryogenic refrigeration apparatus in accordance with claim 1 wherein said means for returning said expanded and further cooled fluid to said low-pressure reservoir includes a low-pressure ballast adapted to return low-pressure fluid to said chamber and through said heat storage means during that period of each refrigeration cycle when fluid is being exhausted into said low-pressure reservoir, whereby said returning low-pressure fluid provides refrigeration to said heat storage means for cooling incoming high-pressure fluid prior to its introduction into said chamber.

3. A cryogenic refrigeration apparatus in accordance with claim 2 wherein said heat storage means are internal of said movable member.

4. A cryogenic refrigeration apparatus in accordance with claim 1 wherein said means for returning said expanded and further cooled fluid includes heat exchange means adapted to effect out-of-contact heat exchange with said heat storage means whereby said returning low-pressure fluid continuously provides refrigeration to said heat storage means for cooling incoming high-pressure fluid prior to its introduction into said chamber.

5. A cryogenic refrigeration apparatus comprising in combination

(a) an enclosure;

(b) a member movable within said enclosure and defining therein at least one fluid chamber of variable volume;

(c) a high-pressure fluid reservoir;

(d) a low-pressure fluid reservoir;

(e) a fluid flow path communicating between said fluid chamber and said reservoirs;

(f) heat storage means in said fluid flow path; and

(g) a Joule-Thomson loop associated with said chamber, said Joule-Thomson loop comprising

(1) a high-pressure ballast in fluid communication with said chamber and adapted periodically to receive cold high-pressure fluid from said chamber,

(2) an expansion valve adapted to expand and further cool said high-pressure fluid delivered thereto from said high-pressure ballast at an essentially constant rate,

(3) a heat exchanger adapted to effect out-of-contact heat exchange between said cold high-pressure fluid entering said expansion valve and the expanded and further cooled fluid resulting from passage through said expansion valve, and

(4) means for returning said expanded and further cooled fluid in heat exchange relationship with said heat storage means to said low-pressure reservoir over at least that period of time corresponding to that part of each refrigeration cycle when said fluid is being expanded and discharged to said low-pressure reservoir.

6. A cryogenic refrigeration apparatus in accordance with claim 5 further characterized by the fact that said member movable within said enclosure defines in addition to said fluid chamber of variable volume a second chamber of variable volume in fluid communication therewith through said heat storage means in said fluid flow path.

7. A cryogenic refrigeration apparatus in accordance with claim 5 wherein said means for returning said expanded and further cooled fluid to said low-pressure reservoir includes a low-pressure ballast adapted to return low-pressure fluid to said chamber and through said heat storage means during that period of each refrigeration cycle when fluid is being exhausted into said low-pressure reservoir, whereby said returning low-pressure fluid provides refrigeration to said heat storage means for cooling incoming high-pressure fluid prior to its introduction into said chamber.

8. A cryogenic refrigeration apparatus in accordance with claim 7 wherein said heat storage means is internal of said movable member.

9. A cryogenic refrigeration apparatus in accordance with claim 5 wherein said means for returning said expanded and further cooled fluid includes heat exchange means adapted to effect out-of-contact heat exchange with said heat storage means whereby said returning low-pressure fluid continuously provides refrigeration to said heat storage means for cooling incoming high-pressure fluid prior to its introduction into said chambers.

10. A cryogenic refrigeration apparatus in accordance with claim 9 further characterized in that said heat storage means is of an annular configuration within said enclosure and defines the volume in which said movable member moves.

11. A cryogenic refrigeration apparatus in accordance with claim 5 including fluid flow control means in said fluid flow path adapted to control the introduction of high-pressure fluid into and discharge of low-pressure from said enclosure.

12. A cryogenic refrigeration apparatus in accordance with claim 5 including driving means for imparting motion to said member movable within said enclosure.

13. A cryogenic refrigeration apparatus in accordance with claim 12 wherein said driving means comprises a separate fluid chamber within said enclosure adapted to contain fluid in force-applying relationship with said movable member, and means for controlling the flow of fluid into and out of said separate fluid chamber.

14. A cryogenic refrigeration apparatus, comprising in combination

(a) an enclosure;

(b) a member movable within said enclosure and defining therein a plurality of successively colder fluid chambers of variable volumes;

(c) a high-pressure fluid reservoir;

(d) a low-pressure fluid reservoir;

(e) a fluid flow path communicating between said fluid chambers and said reservoirs;

(f) a heat storage means for each of said successively colder fluid chambers in said fluid flow path; and

(g) a Joule-Thomson loop associated with said chambers, said Joule-Thomson loop comprising

(1) a high-pressure ballast in fluid communication with the coldest of said fluid chambers and adapted periodically to receive cold high-pressure fluid from said coldest chamber,

(2) an expansion valve adapted to expand and further cool said high-pressure fluid delivered thereto from said high-pressure ballast at an essentially constant rate,

(3) a heat exchanger adapted to effect out-of-contact heat exchange between said cold high-pressure fluid entering said expansion valve and the expanded and further cooled fluid resulting from passage through said expansion valve; and

(4) means for returning said expanded and further cooled fluid in heat exchange relationship with said heat storage means to said low-pressure reservoir over at least that period of time corresponding to that part of each refrigeration cycle when said fluid is being expanded and discharged to said low-pressure reservoir.

15. A cryogenic refrigeration apparatus in accordance with claim **14** wherein said means for returning said expanded and further cooled fluid to said low-pressure reservoir includes a low-pressure ballast adapted to return low-pressure fluid to said coldest chamber and through said heat storage means during that period of each refrigeration cycle when fluid is being exhausted into said low-pressure reservoir, whereby said returning low-pressure fluid provides refrigeration to all of said heat storage means for cooling incoming high-pressure fluid prior to its introduction into said chamber.

16. A cryogenic refrigeration apparatus in accordance with claim **15** wherein said heat storage means are internal of said movable member.

17. A cryogenic refrigeration apparatus in accordance with claim **14** wherein said means for returning said expanded and further cooled fluid includes heat exchange means adapted to effect out-of-contact heat exchange with each of said heat storage means whereby said returning low-pressure fluid continuously provides refrigeration to said heat storage means for cooling incoming high-pressure fluid prior to its introduction into said chambers.

18. A cryogenic refrigeration apparatus in accordance with claim **14** further characterized by the fact that said member movable within said enclosure defines in addition to said plurality of successively colder fluid chambers a warm chamber of variable volume in fluid communication therewith through said heat storage means in said fluid flow path.

19. A cryogenic refrigeration apparatus in accordance with claim **14** wherein said movable member comprises a plurality of cylindrical bodies depending from a single head means and movable within individual cylindrical sections of said enclosure, each of said cylindrical bodies and its associated cylindrical sections defining one of said successively colder chambers of variable volume.

20. A cryogenic refrigeration apparatus in accordance with claim **14** wherein said movable member and said enclosure are of a stepped configuration, each step of which defines one of said successively colder chambers of variable volume.

21. A cryogenic refrigeration apparatus, comprising in combination

(a) an enclosure;

(b) a member movable within said enclosure and defining therein a first warm chamber, a second cold chamber and a third driving chamber, all of said chambers being of variable volume;

(c) a high-pressure fluid reservoir;

(d) a low-pressure fluid reservoir;

(e) a first fluid flow path communicating between said second chamber and said reservoirs including therein a first heat storage means;

(f) a second fluid flow path communicating between said first and second chambers including therein a second heat storage means;

(g) first fluid flow control means associated with said first fluid flow path adapted to control the introduction of high-pressure fluid from said high-pressure fluid reservoir into said second chamber and the discharge of low-pressure fluid from said second chamber into said low-pressure reservoir;

(h) second fluid conduit means communicating between said third chamber and said reservoirs;

(i) second fluid flow control means associated with said second fluid conduit means adapted to control the introduction of fluid into and discharge of fluid from said third chamber;

(j) a Joule-Thomson loop associated with said second chamber, said Joule-Thomson loop comprising

(1) a high-pressure ballast in fluid communication with said second chamber and adapted periodically to receive cold high-pressure fluid from said chamber,

(2) an expansion valve adapted to expand and further cool said high-pressure fluid delivered thereto from said high-pressure ballast at an essentially constant rate,

(3) a heat exchanger adapted to effect out-of-contact heat exchange between said cold high-pressure fluid entering said expansion valve and the expanded and further cooled fluid resulting from passage through said expansion valve,

(4) fluid return means including a low-pressure ballast adapted to return low-pressure fluid from said out-of-contact heat exchanger to said second chamber during that period of each refrigeration cycle when fluid is being exhausted into said low-pressure reservoir, whereby said returning fluid provides refrigeration to said first heat storage means for cooling incoming high-pressure fluid prior to its introduction into said second chamber.

22. A cryogenic refrigeration apparatus, comprising in combination

(a) an enclosure;

(b) an annular first heat storage means within said enclosure;

(c) a member movable within the volume defined by said annular first heat storage means and defining within said enclosure a first warm chamber of variable volume and a second cold chamber of variable volume, said annular first heat storage means providing a first fluid flow path between said first and second chambers;

(d) a second heat storage means internal of said movable member providing a second fluid flow path between said first and second chambers;

(e) a high-pressure fluid reservoir and a low-pressure fluid reservoir in fluid communication with said first chamber;

(f) fluid flow control means associated with said reservoirs adapted to control the introduction of high-pressure fluid from said high-pressure fluid reservoir into said first chamber and the discharge of low-pressure fluid from said first chamber into said low-pressure reservoir;

(g) a Joule-Thomson loop associated with said second chamber, said Joule-Thomson loop comprising

(1) a high-pressure ballast in fluid communication with said second chamber and adapted periodically to receive cold high-pressure fluid from said chamber,

(2) an expansion valve adapted to expand and further cool said high-pressure fluid delivered thereto from said high-pressure ballast at an essentially constant rate,

(3) a first out-of-contact heat exchanger adapted to effect out-of-contact heat exchange between said cold high-pressure fluid entering said expansion valve and the expanded and further cooled fluid resulting from passage through said expansion valve,

(4) fluid return conduit adapted to return fluid from said Joule-Thomson loop to said reservoirs, and

(5) a second out-of-contact heat exchanger in said fluid return conduit in out-of-contact heat exchange relationship with said first heat storage means.

23. A cryogenic refrigeration apparatus, comprising in combination

(a) an enclosure having a plurality of cylindrical enclosed sections depending from a common section;

(b) a member within said enclosure having a plurality of cylindrical bodies depending from a head and movable within said cylindrical enclosed sections to define within said sections a plurality of successively colder chambers of variable volume and within said common section a warm chamber of variable volume, said cylindrical bodies defining with the inner walls of said cylindrical enclosed sections annular fluid passages;

(c) fluid sealing means between said cylindrical bodies and said inner walls adapted to confine fluid in said successively colder chambers;

(d) heat storage means within each of said cylindrical bodies;

(e) heat station means in thermal contact with the fluid in said successively colder chambers through the walls of said cylindrical bodies, all of said heat stations except that associated with the coldest of said chambers being also in thermal contact with the colder cylindrical sections;

(f) a fluid flow path between said warm chamber and said successively colder chambers including said heat storage means and said annular fluid passages;

(g) a high-pressure fluid reservoir and a low-pressure fluid reservoir in communication with said warm chamber and through said fluid path with said cold chambers;

(h) fluid flow control means adapted to control the introduction of high-pressure fluid from said high-pressure fluid reservoir into low-pressure fluid from said warm chamber into said low-pressure reservoir; and

(i) a Joule-Thomson loop associated with the coldest chamber, said Joule-Thomson loop comprising

(1) a high-pressure ballast in fluid communication with said coldest chamber and adapted periodically to receive cold high-pressure fluid from said chamber,

(2) an expansion valve adapted to expand and further cool said high-pressure ballast at an essentially constant rate,

(3) a heat exchanger adapted to effect out-of-contact heat exchange between said cold high-pressure fluid entering said expansion valve and the expanded and further cooled fluid resulting from passage through said expansion valve, and

(4) fluid return means including a low-pressure ballast adapted to return low-pressure fluid from said heat exchanger to said coldest chamber during that period of each refrigeration cycle when fluid is being exhausted into said low-pressure reservoir, whereby said returning fluid provides refrigeration to said heat storage means for cooling incoming high-pressure fluid prior to its introduction into each of said cold chambers.

24. A cryogenic refrigeration apparatus, comprising in combination

(a) an enclosure having a plurality of cylindrical enclosed sections depending from a common section;

(b) a member within said enclosure having a plurality of cylindrical bodies depending from a head and movable within said cylindrical enclosed sections to define within said sections a plurality of successively colder chambers of variable volume and within said common section a warm chamber of variable volume;

(c) heat station means in thermal contact with the fluid in said successively colder chambers through the walls of said cylindrical bodies, all of said heat stations except that associated with the coldest of said chambers being also in thermal contact with the colder cylindrical sections;

(d) a high-pressure fluid reservoir;

(e) a low-pressure fluid reservoir;

(f) a fluid flow path communicating between said warm chamber and each of said successively colder chambers and with said reservoirs;

(g) a plurality of heat storage means in said fluid flow path;

(h) a Joule-Thomson loop associated with the coldest chamber, said Joule-Thomson loop comprising

(1) a high-pressure ballast in fluid communication with said coldest chamber and adapted periodically to receive cold high-pressure fluid from said chamber,

(2) an expansion valve adapted to expand and further cool said high-pressure fluid delivered thereto from said high-pressure ballast at an essentially constant rate,

(3) a heat exchanger adapted to effect out-of-contact heat exchange between said cold high-pressure fluid entering said expansion valve and the expanded and further cooled fluid resulting from passage through said expansion valve, and

(4) conduit means for returning said expanded and further cooled fluid in out-of-contact heat exchange relationship with said heat storage means to said reservoirs.

25. A cryogenic refrigeration apparatus, comprising in combination

(a) an inner fluid-tight enclosure of stepped configuration defining a plurality of enclosure sections of successively decreasing diameter;

(b) annular heat storage and heat station means within each of said enclosure sections enclosing a volume;

(c) a member of stepped configuration movable within said volumes enclosed by said heat storage and heat station means and defining in its movement a warm chamber and a plurality of successively colder chambers in said enclosure sections, said chambers being in fluid communication through said heat storage and said heat station means;

(d) an outer enclosure surrounding said inner enclosure and defining therewith a fluid-tight passage;

(e) tubing within said fluid-tight passage and in heat exchange relationship with fluid in said heat storage means and said heat station means through the wall of said inner enclosure;

(f) a high-pressure fluid reservoir;

(g) a low-pressure fluid reservoir;

(h) a first fluid conduit communicating between said reservoirs and said warm chamber and having fluid flow control means;

(i) a second fluid conduit communicating between said fluid-tight passage and said low-pressure reservoir;

(j) a third fluid conduit communicating between said tubing and said high-pressure reservoir;

(k) a first Joule-Thomson loop associated with the coldest chamber, said Joule-Thomson loop comprising

(1) a high-pressure ballast in fluid communication with said coldest chamber and adapted periodically to receive cold high-pressure fluid from said coldest chamber,

(2) an expansion valve adapted to expand and further cool said high-pressure fluid delivered thereto from said high-pressure ballast at an essentially constant rate,

(3) a first heat exchanger adapted to effect out-of-contact heat exchange between said cold high-pressure fluid entering said expansion valve and the expanded and further cooled fluid resulting from passage through said expansion valve,

(4) a second heat exchanger adapted to deliver refrigeration to a first load by out-of-contact

heat exchange with said expanded and further cooled fluid prior to its passage through said first heat exchanger, and

(5) fluid return means, including a low-pressure ballast, adapted to return low-pressure fluid from said first heat exchanger to said coldest chamber during that period of each refrigeration cycle when fluid is being exhausted into said low-pressure reservoir, whereby said returning fluid provides a first portion of refrigeration to said heat storage means and said heat station means for cooling incoming high-pressure fluid prior to its introduction into each of said cold chambers;

(l) a second Joule-Thomson loop providing a fluid connection between said tubing and said fluid tight passage, comprising

(1) a third heat exchanger in heat exchange relationship with said first heat exchanger of said first Joule-Thomson loop and adapted to effect out-of-contact heat exchange between high-pressure fluid delivered to it from said tubing and low-pressure fluid being returned to said fluid-tight passage,

(2) said second heat exchanger of said first Joule-Thomson loop whereby high-pressure cold fluid in said second Joule-Thomson loop is said first refrigeration load of said first Joule-Thomson loop,

(3) a fourth heat exchanger adapted to effect additional out-of-contact heat exchange as in said third heat exchanger,

(4) an expansion valve adapted to expand and further cool said high-pressure fluid subsequent to its passage through said fourth heat exchanger, and

(5) a fifth heat exchanger adapted to deliver refrigeration to an external load.

26. A cyrogenic refrigeration apparatus, comprising in combination

(a) an inner fluid-tight enclosure of stepped configuration defining a plurality of enclosure sections of successively decreasing diameter;

(b) annular heat storage and heat station means within each of said enclosure sections enclosing a volume;

(c) a member of stepped configuration movable within said volumes enclosed by said heat storage and heat station means and defining in its movement a warm chamber and a plurality of successively colder chambers in said enclosure sections, said chambers being in fluid communication through said heat storage and said heat station means;

(d) an outer enclosure surrounding said inner enclosure and defining therewith a fluid-tight passage;

(e) a surface-extending means within said fluid-tight passage;

(f) a high-pressure fluid reservoir;

(g) a low-pressure fluid reservoir;

(h) a first fluid conduit communicating between said reservoirs and said warm chamber and including fluid flow control means;

(i) a second fluid conduit communicating between said fluid-tight passage and said reservoirs;

(j) a Joule-Thomson loop associated with the coldest chamber, said Joule-Thomson loop comprising

(1) a high-pressure ballast in fluid communication with said coldest chamber and adapted periodically to receive cold high-pressure fluid from said coldest chamber,

(2) a first expansion valve adapted initially to expand and initially further cool said high-pressure fluid delivered thereto from said high-pressure ballast at an essentially constant rate,

(3) a first heat exchanger adapted to effect out-of-contact heat exchange between said high-pressure fluid entering said first expansion valve and a first portion of the initially expanded and initially further cooled fluid resulting from passages through said first expansion valve,

(4) first conduit means, including a low-pressure ballast, for returning said first portion of fluid from said first heat exchanger to said coldest chamber during that period of each refrigeration cycle when fluid is being exhausted into said low-pressure reservoir,

(5) second conduit means adapted to return a second remaining portion of said initially expanded and initially further cooled fluid to said reservoirs by way of said fluid-tight passage and in out-of-contact heat exchange relationship through the wall of said inner enclosure with said heat storage and said heat station means;

(6) a second expansion valve in said second conduit means adapted finally to expand and finally cool said second portion of fluid,

(7) a second heat exchanger in said second conduit adapted to effect out-of-contact heat exchange between said second portion of initially expanded and initially further cooled fluid entering said second expansion valve and the finally expanded and cooled fluid resulting from passage through said second expansion valve,

(8) heat exchange means adapted to deliver refrigeration from said finally expanded and cooled second portion of fluid to an external load, and

(9) third heat exchange means in said second conduit associated with said first heat exchange means whereby said second portion of fluid contributes to the refrigeration of said high-pressure cold fluid entering said first expansion valve.

27. A completely closed cycle cryogenic refrigeration apparatus, comprising in combination

(a) an enclosure;

(b) two separately movable members within said enclosure which in their movement define

(c) at least three chambers of variable volume, the coldest of which is adapted to receive initially cooled high-pressure fluid for subsequent expansion and further cooling;

(d) a fluid flow path incorporating heat storage means adapted to effect fluid communication among said three chambers;

(e) a Joule-Thomson loop associated with said coldest chamber, said Joule-Thomson loop comprising

(1) a high-pressure ballast in fluid communication with said coldest chamber and adapted periodically to receive cold high-pressure fluid from said chamber;

(2) an expansion valve adapted to expand and further cool said high-pressure fluid delivered thereto from said high-pressure ballast at an essentially constant rate,

(3) a heat exchanger adapted to effect out-of-contact heat exchange between said cold high-pressure fluid entering said expansion valve and the expanded and further cooled fluid resulting from passage through said expansion valve,

(4) fluid return means including a low-pressure ballast adapted to return low-pressure fluid from said out-of-contact heat exchanger to said coldest chamber during that period of each refrigeration cycle when fluid is being exhausted from said coldest chamber.

28. A cryogenic refrigeration method, comprising the steps of

(a) delivering to an enclosed expansible chamber a high-pressure fluid which has been initially cooled by contact with precooled regenerative surfaces prior to expansion and final cooling;

(b) periodically transferring at least a portion of said

initially cooled high-pressure fluid to a high-pressure ballast thereby to provide a supply of initially cooled high-pressure fluid at a temperature below its Joule-Thomson inversion temperature;

(c) expanding said intially cooled high-pressure fluid at a substantially continuous rate thereby to further cool it; and

(d) returning the resulting further cooled and expanded low-pressure fluid to a low-pressure fluid zone during at least each expanding step in the refrigeration cycle, said returning being first in out-of-contact heat exchange relationship with said initially cooled high-pressure fluid prior to its expansion and then in heat exchange relationship with said regenerative surfaces thereby providing refrigeration for the precooling of said high-pressure fluid entering said chamber.

29. A cryogenic method in accordance with claim 28 including the step of accumulating said cooled low-pressure fluid subsequent to its exchange of heat with said initially cooled high-pressure fluid and returning said cooled low-pressure fluid during the period in the refrigeration cycle when the fluid is being exhausted from said chamber whereby it contacts directly said regenerative surfaces to provide said refrigeration.

30. A cryogenic method in accordance with claim 28 wherein the step of returning said further cooled and expanded low-pressure fluid in heat exchange relationship with said regenerative surfaces comprises passing said fluid in out-of-contact heat exchange relationship with said regenerative surfaces.

31. A cryogenic method, comprising the steps of

(a) transferring from enclosed expansible chambers at successively colder temperatures a high-pressure fluid;

(b) initially and successively cooling said high-pressure fluid during said transferring by contact with precooled regenerative surfaces prior to its entering the coldest of said chambers;

(c) periodically transferring at least a portion of said initially cooled high-pressure fluid from said coldest chamber to a high-pressure ballast thereby to provide a supply of initially cooled high-pressure fluid at a temperature below its Joule-Thomson inversion temperature;

(d) expanding said initially cooled high-pressure fluid at a substantially continuous rate thereby to further cool it; and

(e) returning the resulting further cooled and expanded low-pressure fluid to a low-pressure fluid zone during at least each expanding step in the refrigeration cycle, said returning being first in out-of-contact heat exchange relationship with said initially cooled high-pressure fluid prior to its expansion and then in heat exchange relationship with said regenerative surfaces thereby providing refrigeration for the precooling of said high-pressure fluid entering said coldest chamber.

32. A cryogenic refrigeration method in accordance with claim 31 further characterized by the steps of employing said further cooled and expanded low-pressure fluid to deliver refrigeration to a separate quantity of said high-pressure fluid, expanding said separate quantity of high-pressure fluid thereby to provide a stream of cold low-pressure fluid, and returning said stream of cold low-pressure fluid in out-of-contact heat exchange relationship with said high-pressure fluid and with said regenerative surfaces.

33. A cryogenic refrigeration method in accordance with claim 31 further characterized by steps of diverting a quantity of said cooled and expanded low-pressure fluid from the returning stream, further expanding said quantity to provide a finally cooled and expanded stream of low-pressure fluid, and returning said finally cooled and expanded fluid first in out-of-contact heat exchange with said quantity of fluid thus diverted, then with said initially cooled high-pressure fluid prior to expansion, and finally with said regenerative surfaces.

References Cited by the Examiner

UNITED STATES PATENTS

2,966,035	12/1960	Gifford	62—6
3,045,436	7/1962	Gifford	62—6
3,101,596	8/1963	Rinia	62—6
3,101,597	8/1963	Dros	62—6
3,115,014	12/1963	Hogan	62—6
3,115,015	12/1963	Hogan	62—6
3,115,016	12/1963	Hogan	62—6
3,148,512	8/1964	Hoffman	62—6
3,221,509	12/1965	Garwin	62—6

References Cited by the Applicant

UNITED STATES PATENTS

2,567,454	9/1951	Taconis.
2,657,553	11/1953	Jonkers.
2,906,101	9/1959	McMahon et al.

WILLIAM J. WYE, *Primary Examiner.*