United States Patent [19]

Dehne

[11] **3,788,088**

[45] **Jan. 29, 1974**

[54] **DOUBLE ACTING EXPANDER ENDING AND CRYOSTAT**

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[73] Assignee: **Hughes Aircraft Company,** Culver City, Calif.

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[21] Appl. No.: **310,301**

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[52] **U.S. Cl.**..... **62/6,** 62/467, 60/24

[51] **Int. Cl.**..... **F25b 9/00**

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

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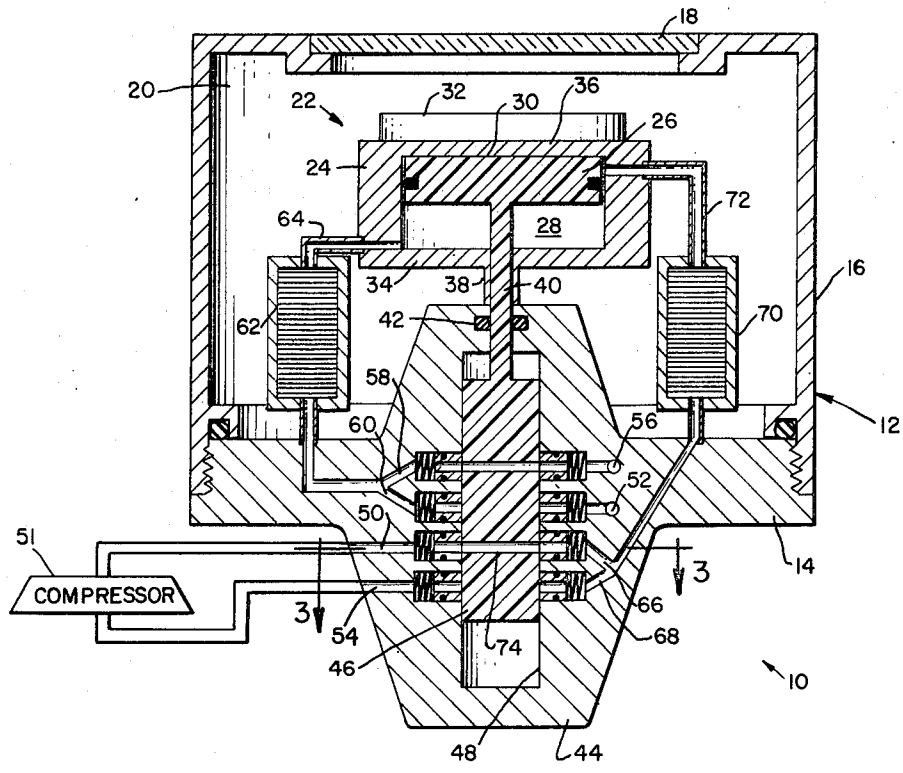
Primary Examiner—William J. Wye

Attorney, Agent, or Firm—W. H. MacAllister; Allen A. Dicke, Jr.

[57] **ABSTRACT**

Double-acting expander engine minimizes shuttle losses by having a cylinder at low temperature along its entire length, and minimizes pumping losses by requiring only a thin piston and minimizes conduction losses by providing a thin piston rod. Refrigeration capacity is approximately doubled by the double-acting configuration.

8 Claims, 12 Drawing Figures



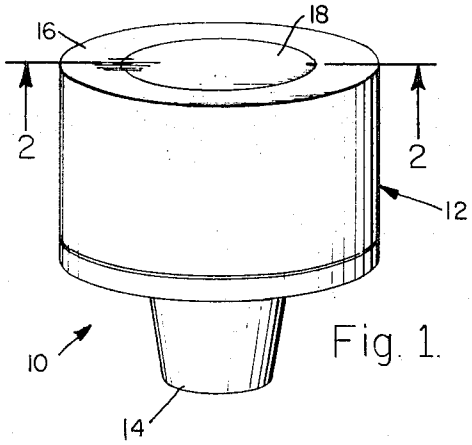


Fig. 1.

Fig. 4.

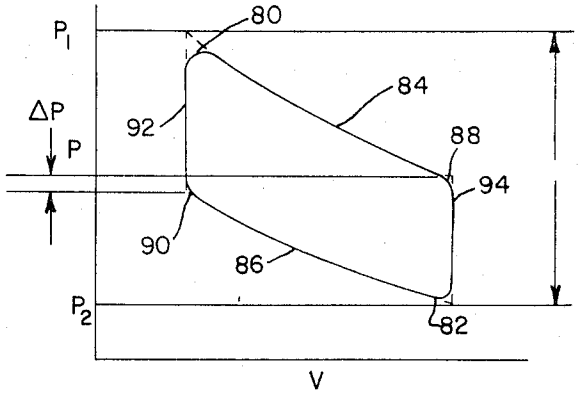


Fig. 2.

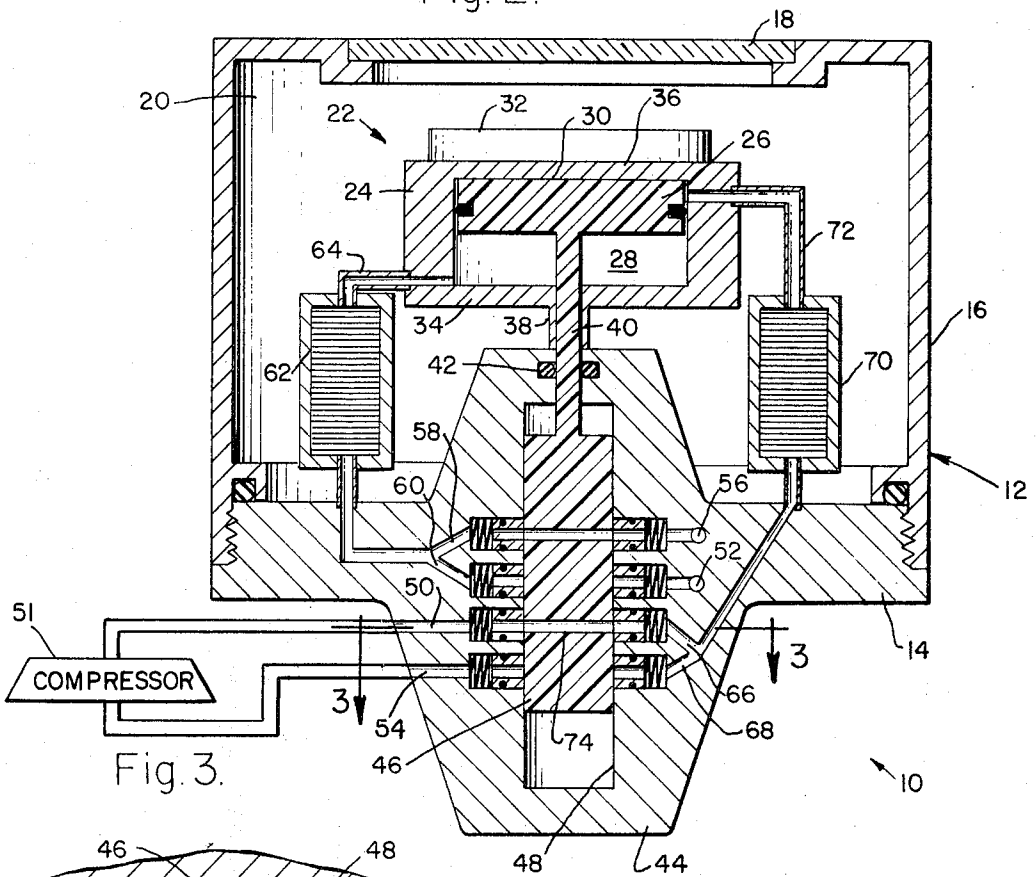
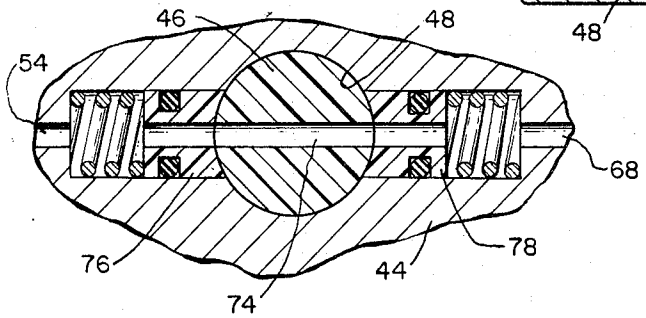


Fig. 3.



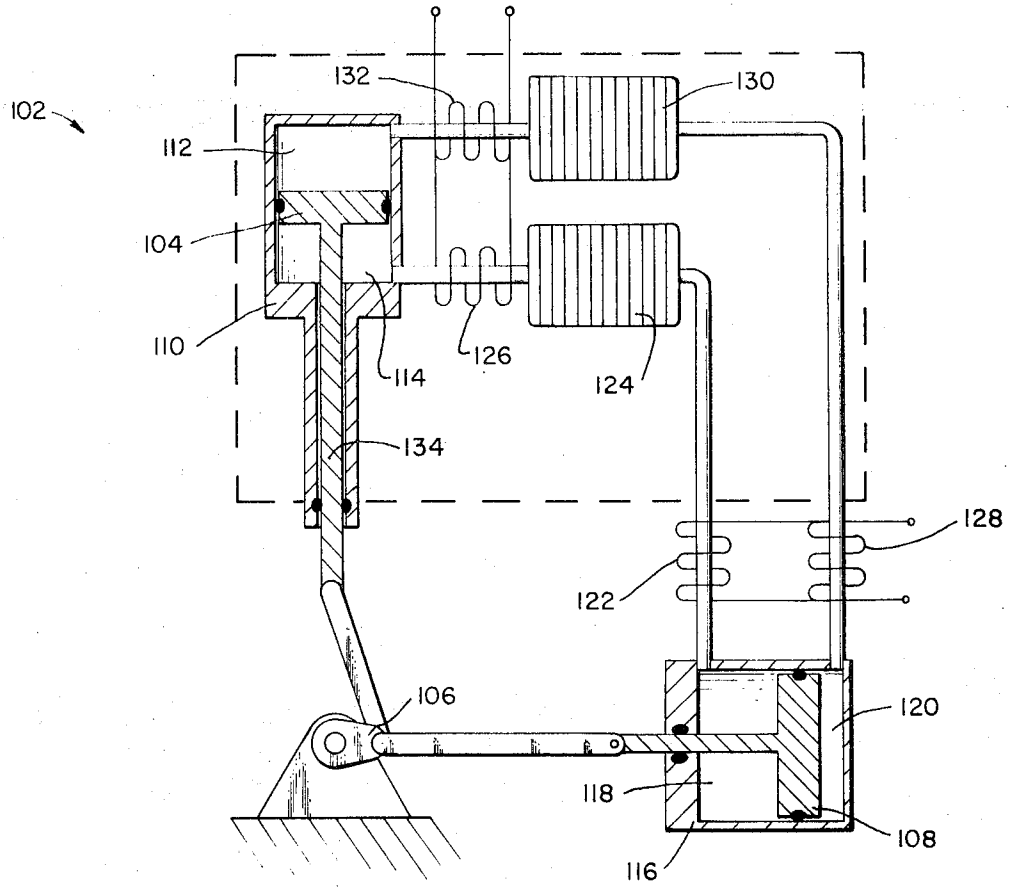

102
132
130
112
104
110
114
126
124
134
128
122
106
120
118
108
116

Fig. 5.

Fig. 6.

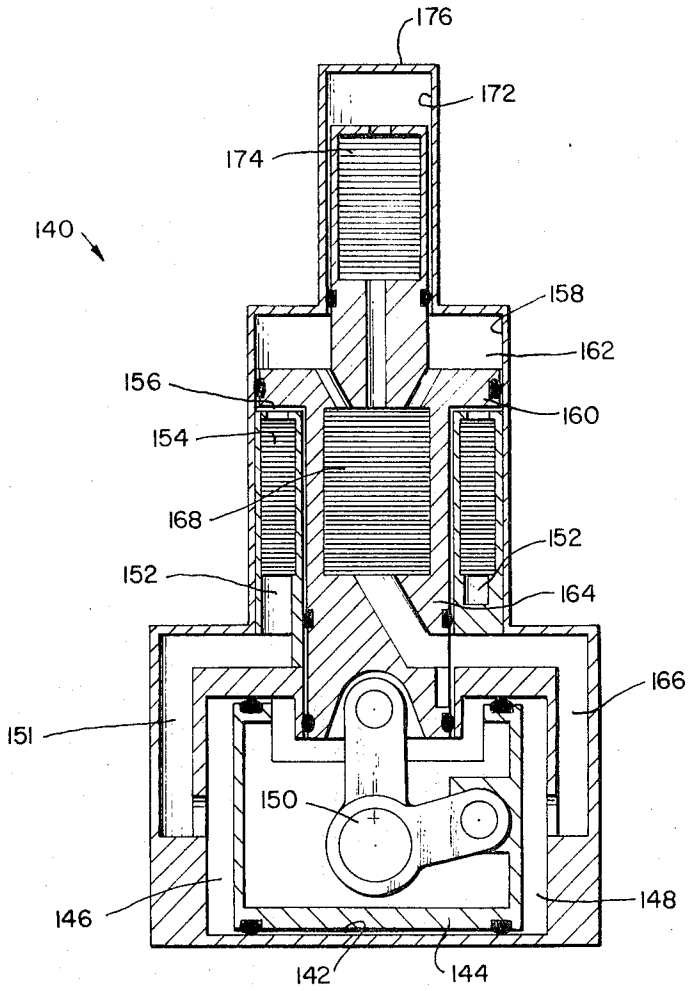

176
172
174
140
158
162
156
160
154
168
152
152
164
166
151
150
146
148
142
144

Fig. 7.

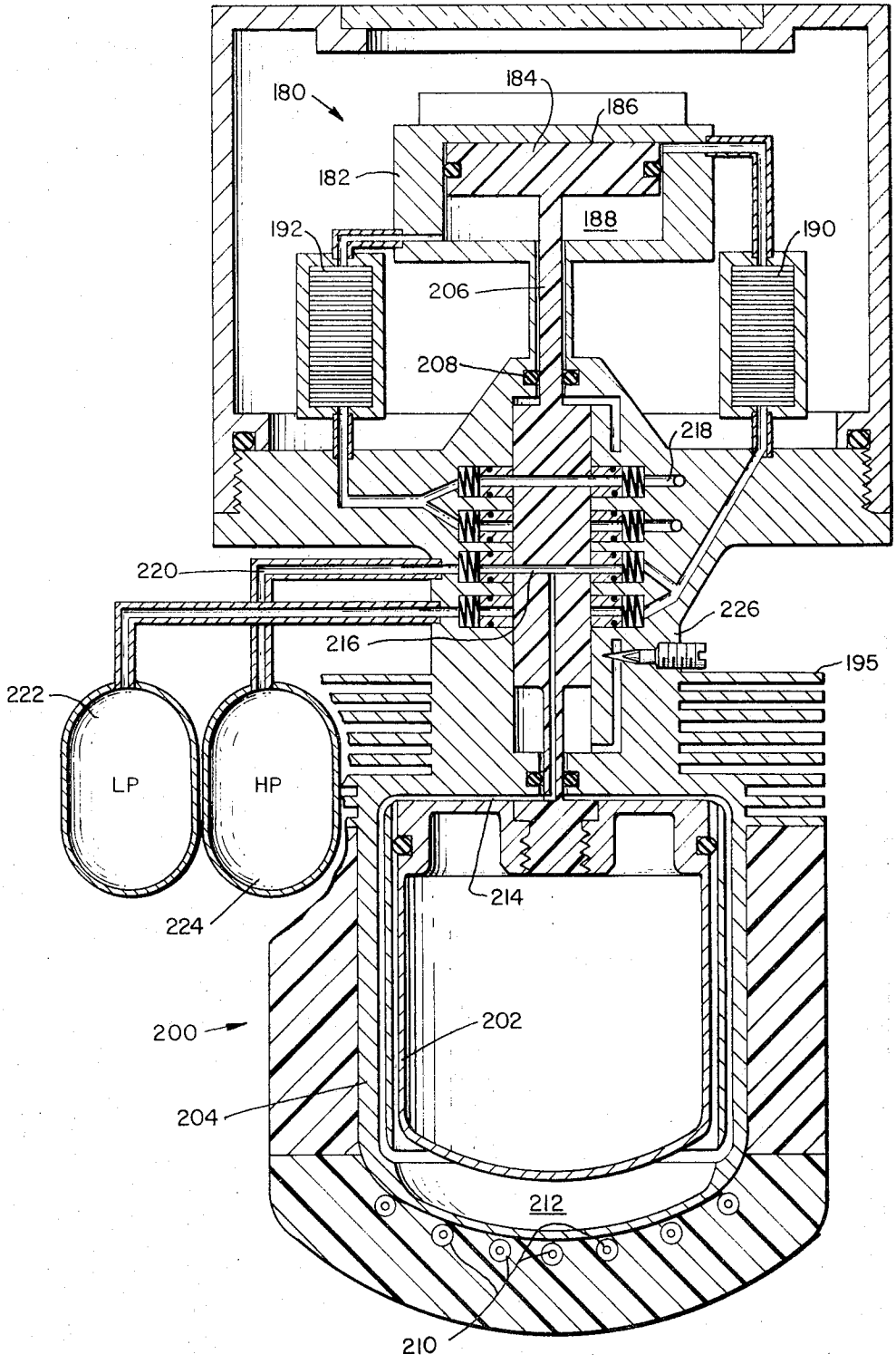

180
184
186
182
188
192
190
206
208
218
220
226
216
195
222
LP
HP
224
214
200
202
204
212
210

Fig. 8.

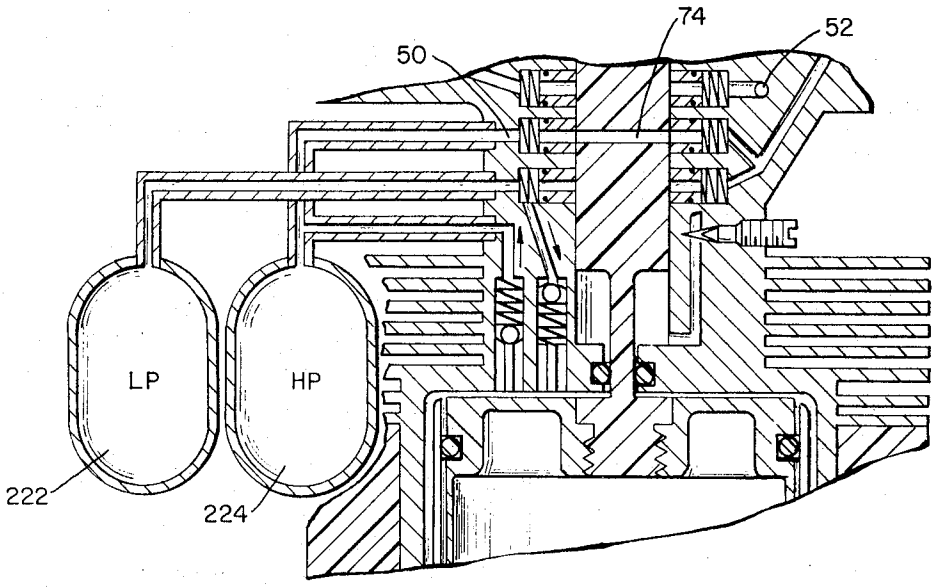
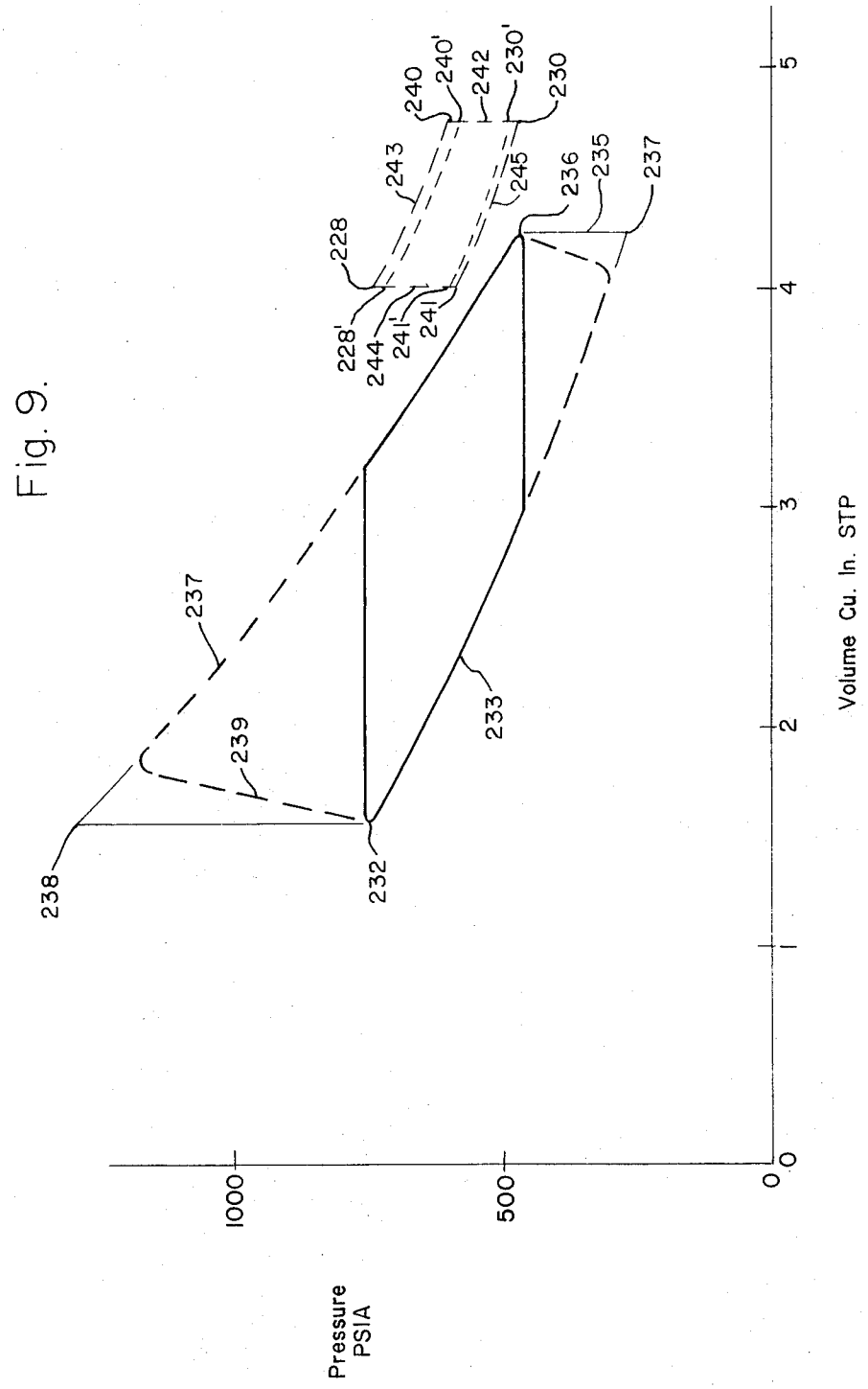


Fig. 9.


Pressure PSIA
1000
500
0
Volume Cu. In. STP
0
1
2
3
4
5
238
239
237
232
233
228
228'
244
241'
241
243
240
240'
242
230'
230
245
236
235
237

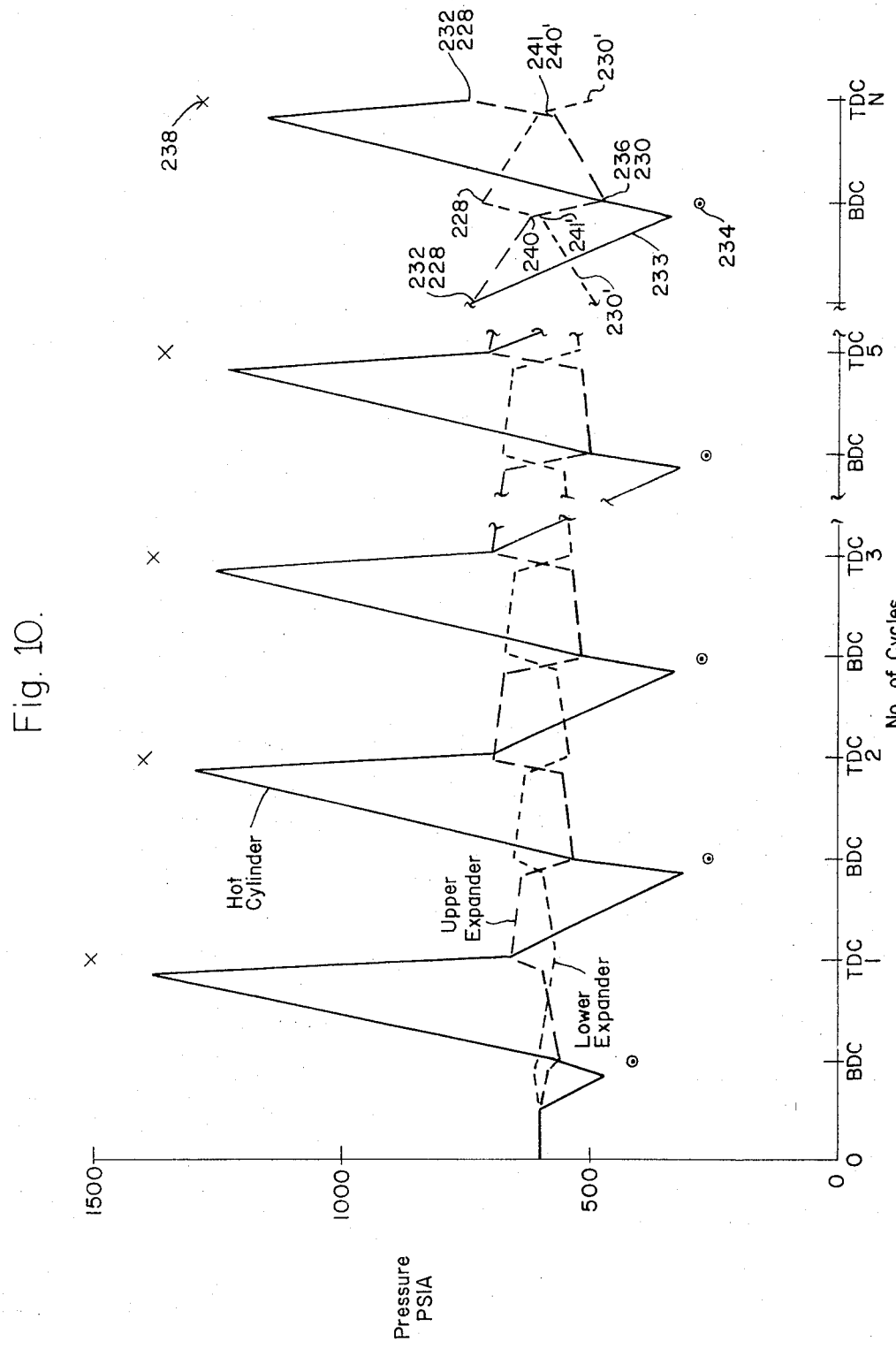

Pressure PSIA
1500
1000
500
0
Hot Cylinder
Upper Expander
Lower Expander
238
232
228
241
240'
230'
236
230
228
240
241'
232
228
230'
233
234
BDC
TDC
1
2
3
5
N
No. of Cycles

Fig. 10.

Fig. 11.

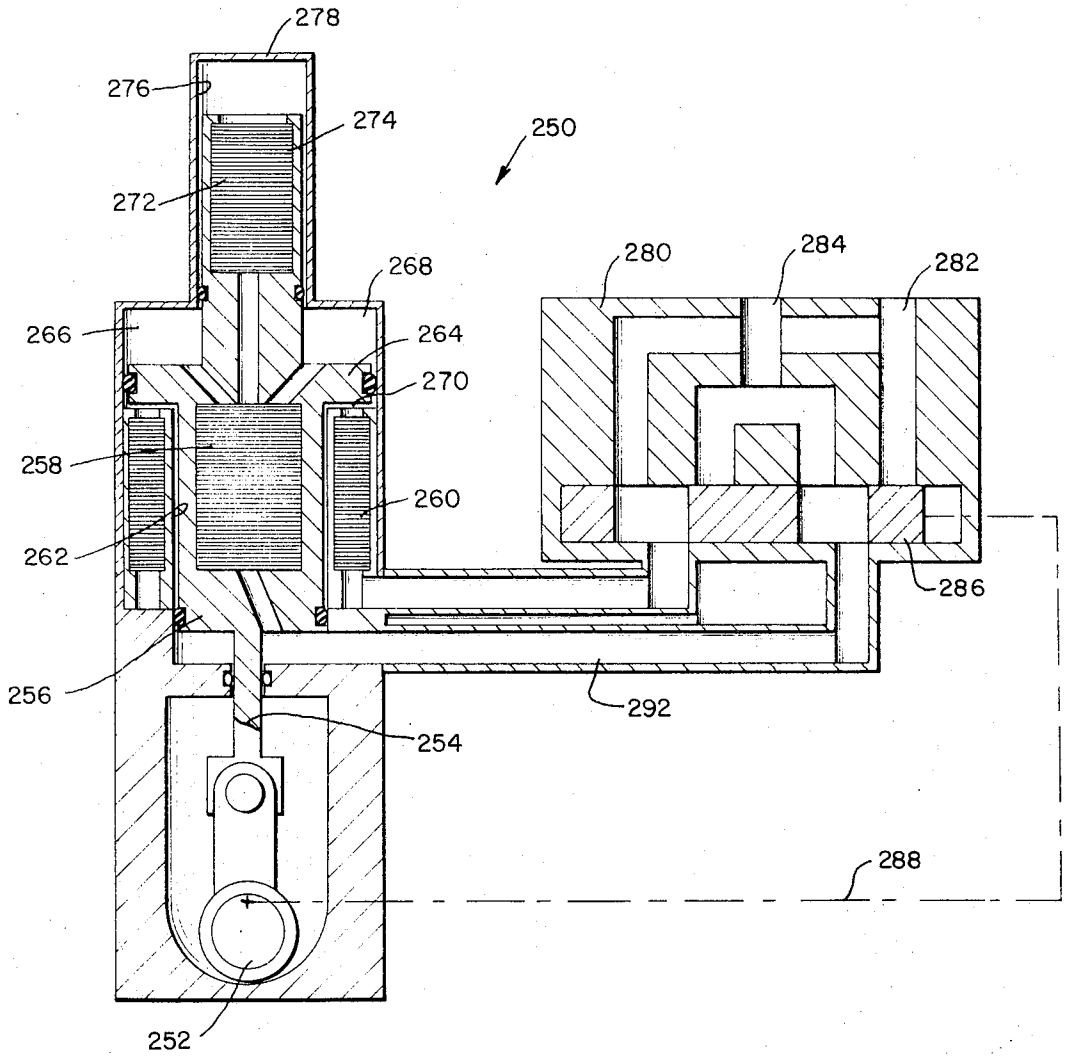
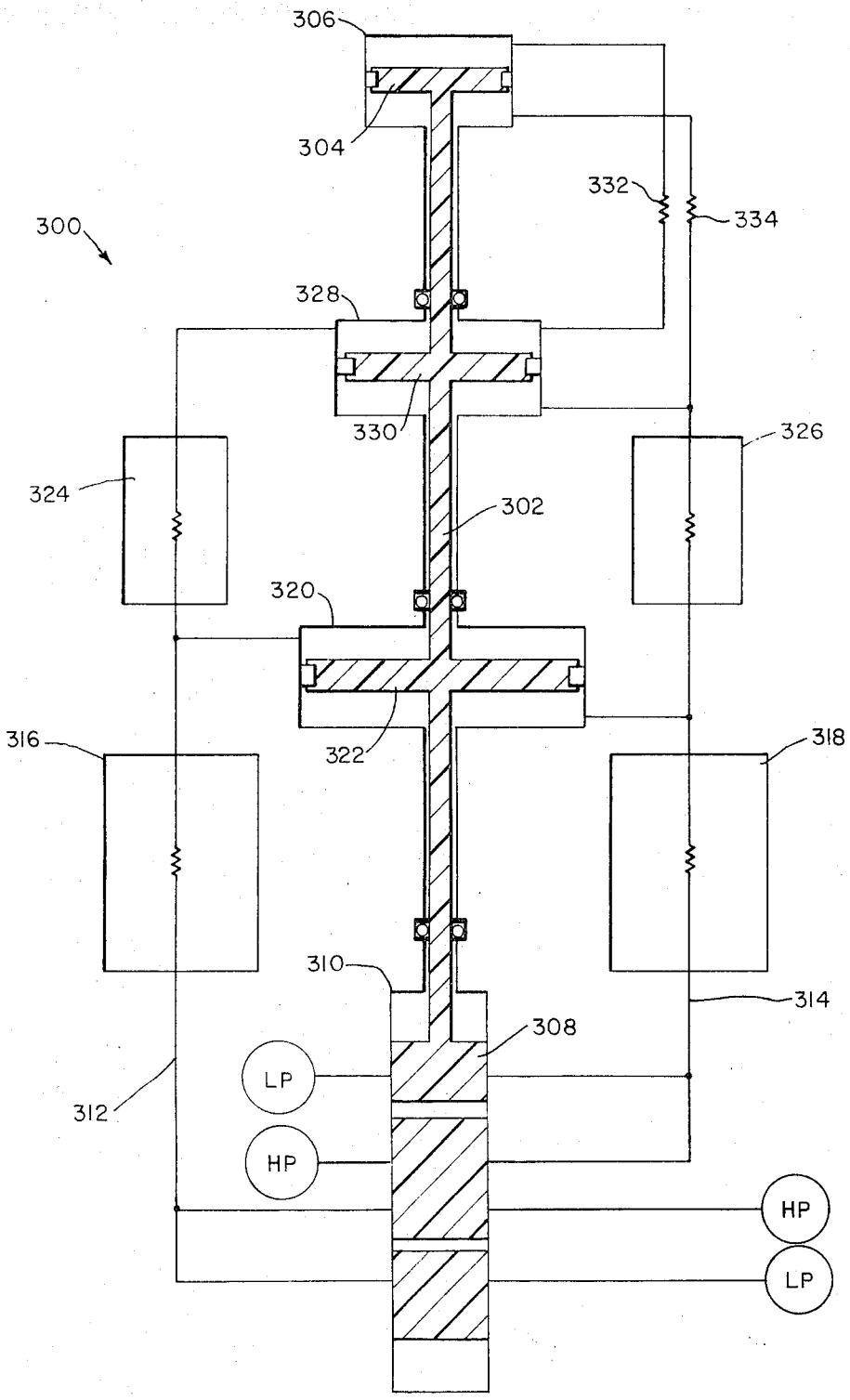

278
276
274
250
272
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284
282
268
266
264
270
258
260
262
286
256
292
254
288
252

Fig. 12.


306
304
332
334
300
328
330
326
324
302
320
322
316
318
310
314
308
LP
312
HP
HP
LP

DOUBLE ACTING EXPANDER ENDING AND CRYOSTAT

BACKGROUND

This invention is directed to a double acting cryogenic expander engine.

The simpler of the prior art structures for producing refrigeration employs throttling through an orifice to expand a refrigerant gas into a lower pressure zone. Under the proper conditions depending on the nature of the gas and the pressure, such causes refrigeration. The type of expansion is commonly called Joule-Thompson refrigeration. In view of the fact that gas expansion does not always cause cooling, Joule-Thompson refrigeration can only be employed in the correct circumstances.

Gas expansion with work output produces more refrigeration than the Joule-Thompson effect. A greater amount of refrigeration can be produced by more closely approaching the isentropic line of expansion with work out, than the approach to the isenthalpic line of Joule-Thompson expansion.

S. F. Malaker et al. Pat. No. 3,074,244 illustrates the Stirling cycle. FIGS. 1 and 2 illustrate the theoretical processes in the cycle, and FIG. 3 illustrates the rounded corners in the P-V diagram which result from the fact that the two pistons are crank-coupled and operate on a sinusoidal curve. It is equally clear that the Stirling cycle can be operated by an expander piston when connected to sources of high pressure and low pressure refrigerant gas which are suitably valved to connect a refrigerant gas pressure and exhaust to the expansion chamber at appropriate times during its cycle. This is referred to as a modified Stirling cycle. By eliminating the crank interconnection between the compressor and expander cylinders, to eliminate the required sinusoidal motion interconnection, an improvement in efficiency can be achieved. Furthermore, the expansion engine is relatively compact compared to the integral Stirling refrigerators of current design since it permits separation of the expander from the compressor and the motor. This feature is particularly desirable on applications where the refrigeration is for the purpose of cooling infrared sensors that are gimbaled or when large volumes or weights are not acceptable. There are several losses in the normal construction of cryogenic expander engines. These are thermodynamic losses, including pumping loss and shuttle loss. Pumping loss is that loss of refrigeration capacity caused by cold gas traveling into the annulus between cold cylinder and displacer or piston, during the high pressure portion of the cycle, warming at the ambient end of the annulus, and then returning in a warmer state to the cold end of the piston and the swept volume.

The fact that a smaller annulus exists also reduces the mass flow rate of gas passing through one regenerator (lower expander), and reduces that volume to zero in case of the upper expander, thus the regenerator becomes more efficient once it need not cool and heat that mass flow normally flowing into the annulus, i.e., the regenerator loss is reduced.

Shuttle loss is that loss of refrigeration capacity caused by heat flow from the cylinder wall to the refrigerant gas in the annular space between piston and cylinder, the heat then flowing to the piston when the piston is axially displaced. When the piston now strokes to the cold end it gives this heat up to the colder gas and

colder end of the cylinder. The heat flow thus shuttles from the warmer end of the cylinder through the gas to the colder piston, and vice versa when the piston strokes to the cold end of the cylinder. The loss is a function of the surface areas of the cylinder and piston, as well as the gap between these two members. The pumping loss and the shuttle loss are inversely related through the annular clearance between piston and cylinder and in practice must be balanced to yield a minimum heat loss.

SUMMARY

In order to aid in the understanding of this invention it can be stated in essentially summary form that it is directed to a double-acting cryogenic expander enging wherein both sides of a thin piston are directed toward a space which acts as a refrigerant expander space, to minimize shuttle and pumping losses.

Accordingly, it is an object of this invention to provide a double-acting expander engine wherein thermodynamic losses are minimized at cryogenic temperatures to improve refrigerator efficiency. It is another object to provide a double-acting expander engine which has a push rod so that pumping losses into the annular space around the push rod are minimized since this volume is reduced by the smaller circumference. It is yet another object to provide a double-acting expander engine which has minimized shuttle losses because the surfaces of the push rod and the bore are kept to a minimum by the smaller push rod and cylinder circumferences. It is a further object to provide a double-acting expander engine which has minimum structural support for the piston and cylinder to minimize thermal conductivity losses between the cylinder and the ambient temperature structure on which it is supported.

Other objects and advantages to this invention will become apparent from a study of the following portion of this specification, the claims and the attached drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a perspective view of a cryostat having therein the double-acting expander engine of this invention operating on the modified Stirling cycle.

FIG. 2 is an enlarged vertical section taken generally along the center line of FIG. 1, along line 2—2 thereof.

FIG. 3 is a further enlarged horizontal section, with parts broken away, taken generally along line 3—3 of FIG. 2.

FIG. 4 is a pressure-volume diagram showing the thermodynamic cycle of the expander engine and the closeness of its configuration to the ideal Stirling cycle diagram.

FIG. 5 is a schematic mechanical diagram of a double-acting Stirling cycle engine.

FIG. 6 is a vertical section through a two-stage double-acting Stirling cycle expander engine.

FIG. 7 is a schematic diagram of another embodiment of a double-acting modified Stirling cycle expander engine, together with a coupled compressor.

FIG. 8 is a broken-away showing, partly in section, of a modified form of a portion of the structure shown in FIG. 7.

FIG. 9 is a P-V diagram of the engines shown in FIG. 7 and 8.

FIG. 10 is a pressure-cycle diagram of the cryostat shown in FIG. 7.

FIG. 11 is a vertical section through a double-acting expander engine operating on the Gifford-MacMahon cycle.

FIG. 12 is a schematic diagram of a multiple stage Stirling cycle refrigerator, similar to the expander engine of FIG. 1.

DESCRIPTION

FIG. 1 illustrates a cryostat 10 which has a body 12 composed of base 14 and cover 16. Front cover 16 carries window 18 of appropriate transparency to permit the passage of radiation of interest. As seen in FIG. 2, cover 16 is screwed upon base 14 with a suitable seal to provide a vacuum space 20 which is of insulation significance. Preferably, the interior of cover 16, and other appropriate structures, are plated with highly reflective surfaces to minimize radiation. In view of the small size of the preferred embodiment of cryostat 10, the vacuum insulation is the most practical insulation that can be provided. In larger size equipment, super insulation can be packed inside cover 16.

Expander engine 22 is provided within the cryostat. It is built as part of the cryostat, but can just as conveniently be a separately constructed machine mounted within an insulated envelope to form the cryostat. Expander engine 22 comprises cylinder 24 in which piston 26 is reciprocably mounted. Piston 26 divides cylinder 24 into lower expansion space 28 and upper expansion space 30. Device 32 is the heat load which is to be cooled and is directly mounted upon the upper axial end of cylinder 24. Device 32 can be an infrared detector or other electronic device which needs to be cooled to cryogenic temperatures for effective operation, or can be any other type of heat load. When device 32 is an infrared detector, window 18 passes a substantial part of the infrared spectrum to which the detector is sensitive. Since both the lower and upper spaces 28 and 30 provide refrigeration by the expansion of cryogen, cylinder 24 has thick walls to conduct the head load from device 32 down the walls adjacent lower expansion space 28, so that the lower expansion space can also participate in supplying the refrigeration. Lower head 34 is substantially at the same temperature as upper head 36, except for the head load device 32 and particular local losses.

One of the local heat losses of lower head 34 is the support of cylinder 24. Thin walled rod housing 38 supports cylinder 24 in place. The lower end of rod housing 38 is mounted upon base 14 which is at ambient temperature, and thus there is a severe temperature gradient along rod housing 38 from lower head 34 to the ambient temperature of base 14. Therefore, rod housing 38 is made with thin walls and is made as long as possible to minimize conduction losses. Rod 40 is secured to piston 26 and is slideably mounted through rod housing 38. Rod 40 extends downwardly into the base 14, so that the lower end is at ambient temperature. Rod 40 is as thin as structurally possible to minimize conduction loss. At the ambient temperature end, within base 14, seal 42 engages around rod 40 to seal lower expansion space 28.

Valve body 44 is integrally formed as part of the center of base 14, for convenience in construction as well as minimum size and weight. If desired, it can be separately constructed and secured to base 14. Valve spool 46 is reciprocably mounted in valve bore 48 in valve body 44. Rod 40 is connected to valve spool 46 to move the spool in accordance with the position of piston 26 within its cylinder. In view of the fact that rod 40 does not transmit appreciable force rod 40 can be very small diameter to minimize thermal conduction losses. This in turn permits a minimum diameter rod housing 38.

There are a number of ways in which the desired valving can be accomplished. The following structure is one convenient way of accomplishing the valving, but it is clear that there are alternative structures which can accomplish the same function. Ports 50 and 52 are connected to a high pressure source of refrigerant gas, for example at the pressure P1 shown in P-V diagram of FIG. 4, typically at 300 PSIA. These ports can be connected together by inner drilling or can be separately connected to the source of high pressure refrigerant gas. The source of high pressure refrigerant gas is preferably compressor 51, when a long running time is desired. The gas applied is at ambient temperature, and when the cryostat is designed to refrigerate a heat load at 10°K, the gas is helium. Low pressure ports 54 and 56 can be connected together and are for connection to a point where low pressure, ambient temperature refrigerant gas can be delivered, such as to the suction of the compressor delivering high pressure refrigerant gas. The low pressure ports are maintained at a relatively low pressure, such as the pressure P2 in FIG. 4, typically at 150 PSIA. Ports 58 and 60 are connected together and are respectively positioned across valve bore 48 from ports 56 and 52. Ports 58 and 60 are connected to regenerator 62, which is in turn connected through line 64 to lower expansion space 28. Similarly, prots 66 and 68 are connected together into regenerator 70. On its other end regenerator 70 is connected by line 72 to the upper expansion space 30. Valve spool 46 contains diametric holes 73 and 74 therein across the spool. The ports and holes are arranged so that when piston 26 and spool 46 are in the upper-most position, shown in FIG. 2, low pressure port 56 is connected through hole 73 to port 58 so that low pressure gas communication is provided through regenerator 62 to lower expansion space 28. In the same position, high pressure port 50 is connected through hole 74 and port 66, through regenerator 70 to upper expansion space 30. Means is provided to prevent rotation of valve spool 46 within the valve body.

Furthermore, when the piston 26 and valve spool 46 are in their lower-most position, now shown in the drawings, communication is provided from high pressure port 52 through hole 73, port 60, regenerator 62 to the lower expansion space 28. In the same lower-most position, communication is provided from low pressure port 54 through hole 74, port 68, regenerator 70 to the upper expansion space 30. The spaces above and below valve spool 46 within valve bore 48 perform no function and can be connected together and such connection can be adjustably throttled to dampen motion, i.e., to retard or increase operating frequency, thereby controlling refrigeration capacity or regulating refrigeration temperature. Regenerators 62 and 70 are of conventional miniature regenerator design which may be the type discussed in F. F. Chellis et al. Pat. No. 3,218,815.

FIG. 3 illustrates one manner in which seal is accomplished between the valve ports and the valve spool. Plugs 76 and 78 are slideably mounted in and are sealed with respect to plug bores in body 44 which in-

tersect with the valve bore **48.** Springs load the seal plugs against the valve spool so that face-to-face sealing is accomplished. Seal plugs are preferably made of polymer composition material, such as teflon, to minimize wear and friction. Through-holes through the seal plugs connect the cross-bore in the valve spool with the ports in valve body **44.** Other types of sealing mechanism can be provided, such as a spring loaded rectangular block which carries all of the corresponding ports therein.

Considering the operation of the double-acting expander engine and viewing the cycle as represented by FIG. **4,** it is appreciated that the conditions in the two expander chambers **28** and **30** can be individually traced around the P-V diagram of FIG. **4,** and that the conditions in the two expander chambers are diagonally opposite on the P-V diagram. Considering the position of the piston and valve spool as shown in FIG. **2,** high pressure port **50** has been opened through regenerator **70** to upper expansion chamber **30.** This is at point **80** in FIG. **4.** Expansion chamber **28** has been exhausted through regenerator **62** to low pressure port **56,** so that it is under the conditions represented at point **82.** Under the pressure differential piston **26** moves downward, and as soon as it moves it cuts off the porting connections to both cylinder chambers. In downward motion, the expanding cryogen gas in chamber **30,** the regenerator **62** and the associated flow passages **64, 60** and **58,** which expands down line **84** causes compression in the cryogen gas in chamber **28,** causing the conditions in this chamber to follow line **86** of FIG. **4.** Thus, work is expended by the expanding gas on the compression of the gas in the other chamber, as well as overcoming the mechanical work of moving the piston. When the piston approaches its lower-most position, the conditions in the upper expansion chamber **30** are approaching point **88** and the conditions in the lower expansion chamber are approaching point **90.** The pressure differential ΔP prevents piston stoppage before end of stroke valving takes place. Now, the ports are opened with high pressure port **52** being open through regenerator **62** to lower expansion space **28,** to move the conditions in that space up line **92** while the low pressure port **54** is connected to the upper expansion space through regenerator **70** to move the conditions in that space down line **94.** Valve action causes rounding of the corners in the P-V diagram. The cycle is now ready to repeat itself, with corresponding expansion with resultant refrigeration in lower chamber **28,** with the work of expansion going into compression of the refrigerant gas in the upper expansion space **30,** repeating the operations. Thus, expansion with work out is accomplished in both chambers and the resultant refrigeration is provided to the heat load. The materials employed in the construction are preferrably such that long life sliding and sealing is possible without oil lubrication and without excessive wear. Therefore, many of the parts can be made of synthetic polymer composition material. Cylinder **24** is preferably made of metal for helium retention reasons, but the piston **26** moving therein, together with its rod **40** can be made of epoxy impregnated fiberglass and other low conductive, low friction materials such as polytetraflouroethylene or nylon. Similarly, the plugs **76** and **78** are preferably made of a non-stick synthetic polymer.

FIG. **5** is the schematic mechanical diagram of a double-acting modified Stirling cycle refrigerator generally indicated at **102.** It is a conventional Stirling cycle refrigerator modified to the double acting principle because expander piston **104** is a mechanically connected through crank **106** to compressor piston **108.**

Expander piston **104** divides expander cylinder **110** into volumes **112** and **114.** Similarly, compressor piston **108** divides compressor cylinder **116** into compressor volumes **118** and **120.** Compressor volume **118** is connected through heat exchanger **122,** which rejects heat to the atmosphere, regenerator **124** and heat exchanger **126** which receives heat from the heat load to expander volume **114.** Similarly, compressor volume **120** is connected through heat exchanger **128** which rejects heats of the atmosphere, regenerator **130** and heat exchanger **132** which receives heat from the heat load to expander volume **112.**

The mechanical connections between the pistons is such that they operate at substantially 90° apart on their cycle, and each half works substantially on its own Stirling cycle. The cycle is not purely Stirling, because both pistons are in motion at the same time, while in the theoretical Stirling such is not the case. Both expander spaces **112** and **114** produce refrigeration for a minimized shuttle loss, and piston **104** can be made short in its direction of motion to reduce pumping loss into the annular volume between its periphery and the cylinder wall. Furthermore, the cold piston rod **134** can be of small diameter to minimize pumping loss into the annular space therearound. Thus, in addition to producing twice as much refrigeration by employment of a double-acting expander structure, losses are also minimized.

FIG. **6** shows a vertical section through a structural physical embodiment of a double-acting modified Stirling cycle refrigerator **140** having a super mounted Stirling cycle stage to provide a two-stage refrigerator. Refrigerator **140** has a compressor cylinder **142** in which is located compressor piston **144** which divides the cylinder into compressor volumes **146** and **148.** Piston **144** is reciprocated in its cylinder **142** by means of crank **150,** which is shown in bottom dead center position so that compressor piston **144** is located midway between the ends of its stroke. As the compressor piston reciprocates, gas moving up out of the compressor volume **146** moves through head exchanger passage **151** so that heat therefrom is rejected to the atmosphere surrounding the lower part of refrigeration **140.** Passage **151** is connected to annular passage **152** which forms the lower entrance to annular regenerator **154.** The upper side of regenerator **154** is connected to expander chamber **156** is expander cylinder **158.** Expander piston **160** divides expander cylinder **158** into expander chamber **156** and expander chamber **162.** Expander piston **160** is mounted upon hollow piston rod **164.** Piston rod **164** in turn is connected by a suitable connecting rod to be reciprocated by crank **150** from the lower-most position shown, to an upper-most position wherein expander chamber **162** is minimized.

Compressor volume **148** is connected through heat exchanger passage **166** through a sliding joint between the non-moving structure and piston rod **164** and through regenerator **168** to the upper expander chamber **162.** Heat is absorbed from the heat loaded into the refrigerant gas in the system through the walls of expander cylinder **158.** Thus, the described portion of the refrigerator **140** is the same in operation as the refrigerator in FIG. **5.**

In addition, however, double-acting modified Stirling cycle refrigerator **140** has a displacer **170** operating in cold cylinder **172.** Displacer **172** contains regenerator **174** which is connected on its lower side to the cold end of regenerator **168** and is connected on its upper end into the cold cylinder **172.** Thus, the warm end of regenerator **174** is equivalent in temperature to the expander chamber **160,** and by the Stirling cycle expansion the temperature at the cold end of regenerator **174** and in cold cylinder **172** is lower. Head **176** of cold cylinder **172** is the point of heat load, so that a device to be cooled can be mounted upon that head or in thermally conductive relation to that head. The efficiency of this two-stage refrigerator is enhanced by the improvement in efficiency of the first stage, by minimizing shuttle losses and pumping losses.

FIG. **7** illustrates refrigerator **180** which is a double-acting Stirling cycle refrigerator driven by a thermocompressor. The entire cryostat expander valving and regenerator arrangement is the same as cryostat **10** in FIG. **2.** Cold cylinder **182** has piston **184** therein which divides the cold cylinder into expander volumes **186** and **188.** These volumes are respectively supplied with alternate high and low pressure refrigerant gas respectively through regenerators **190** and **192.** Effective refrigeration is delivered to the heads of cylinder **182** at which the useful thermal load is delivered. The entire cold end, comprising the cylinder, its contents, regenerators **190** and **192** are preferably enclosed in an insulated environment. The entire upper end of the refrigerator **180** is the same as the expander end of refrigerator **10.**

The compressor of refrigerator **180** is generally indicated at **200.** Compressor **200** comprises a displacer **202** which is reciprocably mounted within a cylinder **204.** Displacer **202** is mounted upon push rod **206,** which is also connected to piston **184.** Thus, the piston and the displacer move together, and together with push rod **206.** Push rod **206** is provided with seal **208** to prevent movement of refrigerant gas along the push rod. The center portion of the push rod, below seal **208** is provided with valving to control the flow of refrigerant gas to and from the cold cylinder. The position of the push rod controls the valving. The valving is the same as that described with respect to FIG. **2.**

Valve port **216** is in continuous communication with upper volume **214.** It is in communication with high pressure line **220** when rod **206** is in the raised position shown, and is in communication with low pressure line **218** when rod **206** is in its lower position. Accumulator vessels **222** and **224** are respectively connected to lines **218** and **220.**

The valve ports through push rod **206,** and the valve ports through valve body **226** which forms the intermediate structure between the low temperature and high temperature cylinders provides the valving between the accumulator and the expansion volumes in the low temperature expander cylinder. With the moving parts in the raised position shown in FIG. **7,** high pressure accumulator **224** is connected through porting to the lower end of regenerator **190** and thus to expander volume **186.** Similarly, the valving connects low pressure accumulator **222** through inner drilling to regenerator **192** to expander volume **188.** It must be noted that these two volumes remain 180° apart in the cycle and the conditions therein move around the P-V diagram in the direction of the arrows shown in FIG. **9.**

Displacer **202** has a sliding seal between it and the interior wall of cylinder **204.** Heat is added by heater or combustion means **194** attached to lower cylinder wall **210** so that the refrigerant gas in lower volume **212** is heated when displacer **202** is in the upper position shown in FIG. **7.** Furthermore, tubes **196** are arranged in annular order and connected to spaces **212** and **214** to act as a regenerator so that gas cools as it passes from lower volume **212** to upper volume **214.**

The hot cylinder **210** is heated by heater **194** at the lower end, and is cooled to near ambient temperature at the upper end, near the volume **214** by cooling fins **195.** In startup, compression is accomplished by physically, or magnetically displacing the combination of displacer **202** and piston **184,** together with the push rod **206.** By moving this assembly from top to bottom, pressure in the cylinder **204** drops since the average temperature is decreased as the gas is regeneratively cooled while being displaced from volume **212** to **214.** The P-V diagram of FIG. **9** depicts this process from point **232** to point **234** along the line **233,** where the volume change corresponds to the reduced, temperature compensated volumes in the hot cylinder **204.** The corner is actually rounded due to valving. Upon completion of the stroke the valve port **216** lines up with the low pressure line **218** and thus allows gas from the low pressure tank **222** to flow into the cylinder **210.** The tank pressure is that point of FIG. **8** corresponding to point **236.** The cylinder pressure was lower at point **234** due to the higher compression ratio of the hot cylinder **210.**

If the annular clearance volume in the cylinder is neglected, in an exemplary structure the compression ratio is 3:1. The compression ratio of the expander, on the other hand, should not be designed to be as large. For Vuilleumier refrigerators this ratio is generally near 1.5:1 while that of Stirling refrigerators is generally 2:1, but these ratios are for the entire cycle. The compression ratios applicable to this invention are calculated for each cylinder and are not the aggregate of the two. The graph of FIG. **10** depicts the pressure fluctuations of the gas in the various portion of the refrigerator during cooldown. Assuming the device to be charged to 600 PSI (which is common for VM refrigerators), that the displacer assembly is in the half-way position, and that the volumes of the device represent a given relationship thereby fixing the compression ratios of the device, the pressures can be seen to fluctuate in such a manner that they become motion sustaining after only one cycle. It should be noted that compression ratio for the hot cylinder is 2.68:1 since it includes the dead volume of the regenerator, not previously included. The volume relationship is such that a 0.5 engine dead volume is included with the 4 cm in. swept volume in the hot cylinder. Similarly, the expander expansion ratio is only 1.225:1 when the swept volume is 0.15 cm in. and the regenerator volume is 2 cm in. each. Each of the two accumulator tanks was fixed at a volume of 20 cu in. Self-sustained motion occurs when the upper and lower pressures do not reverse the differential. This is the case in cycle 2; during the TDC-BDC stroke the lower expander pressure always exceeds the upper expander pressure, similarly the reverse happens on the return stroke. The first few cycles serve to stabilize pressures in each of the two gas reservoirs. If the expansion ratio were based on the ratio of accumulator pressures than it would be 1.55 : 1, which

compares well with VM refrigerators. The vertical line just prior to each BDC or TDC represents valve (overlap) action and is extended in scale to permit observation of pressure equalization during the valve action. The last cycle has numbered processes corresponding to those of the PV diagram, FIG. 9. Both upper and lower expanders are numerically represented by FIG. 9, but the lower expander exhibits pressures slightly different because of the phase lag, since the upper expander always exchanges gas with the tanks as well as the hot cylinder, while the lower expander exchanges gas with the tanks only.

Continuing from point 236, FIG. 9, the pressures having equalized, the displacer assembly now moves from hot to cold (BDC to TDC). Since the pressure differential is initially too small this differential reverses before completion of the first cycle and an exterior force is required for completion of the stroke. Upon completion of this stroke (path 237 for the hot cylinder, path 242 for the upper expander 244, and path 244 for the lower expander) pressures equalize again; the hot cylinder discharges gas along line 239 to the high pressure tank and to the upper expander whose pressure thus rises along line 244, while the lower expander discharges gas into the low pressure tank along line 242. The low pressure tank will thus experience a slight rise in pressure, depending on the tank size relative to the expander size, and it is for this reason, together with a lowered high pressure tank, that the lower expander has a slightly lower work output. The the overall expansion ratio, after equalization is 1.44 : 1. This reduced work output, together with the smaller expansion volume, due to the push rod, is not necessarily detrimental since it is more removed from the locale of the heat load (detector) and thus acts similar to staging by assuming a warmer temperature.

Two other features become apparent when FIG. 10 is examined: a warm expander aids start up, due to lower expansion ratio, while a cold expander tends to reduce the driving pressure differential, due to the higher expansion ratio, thus limiting or controlling the lower end temperature of the refrigerator by slowing the frequency of the cycles. Low end temperature control is desirable since infrared detectors experience a maximum signal to noise ratio at only one temperature and this temperature has heretofore been controlled by electrical heaters at the detector, by controlling the speed of multispeed motor, or by lowering the hot cylinder temperature of VM refrigerators. The invention described herein controls the expander temperature automatically by design. Reference characters 228 through 244 indicate cycle points and processes in FIG. 9 and 10.

The description of the cycle has up to nowncluded only one type of compressor, i.e., a spool valved arrangement that pumps or draws gas only at the end of each stroke. This approach facilitates the lowest number of moving parts, namely one, but it entails a significant drawback insofar as this cycle requires more work than one that employs the use of check valves at each tank. Such an arrangement is depicted in FIG. 8 and the solid lines of PV diagram in FIG. 9. The lower work input into the compressor stems from the absence of the high and low pressure spikes since the compressor either delivers or draws gas respectively once high pressure tank pressure is exceeded or the low pressure and pressure is higher.

FIG. 11 illustrates a double-acting expander engine, with an additional expander stage superposed thereon, operating in the Gifford-McMahon cycle. The expander engine is generally indicated at 250. The expander engine includes an eccentric 252 which drives piston rod 254. Rod 254 in turn drives displacer 256 in which is located a regenerator 258. Additionally, regenerator 260 is located in the annular space around cylinder 262 in which the displacer moves. A seal is located on the lower end of the displacer 256 within cylinder 262.

Piston 264 divides cylinder 266 into upper expansion volume 268 and lower expander volume 270. The upper end of regenerator 258 is connected to expander volume 268, while the upper end of regenerator 260 is connected with expander volume 270. Thus, this structure is a double-acting expander engine. Furthermore, a second stage regenerator 272 is located in second stage displacer 274 which is mounted upon the top of piston 264 and reciprocates in second stage expander cylinder 276. The effective refrigeration is produced on cylinder head 278 thereof. The lower end of regenerator 272 is connected to the upper end of regenerator 258, while the upper end of regenerator 272 is connected into the expander space above displacer 274 in second stage expander cylinder 276, just below second stage cylinder head 278. Thus, the double-acting expander serves as a first stage for producing refrigeration and precooling the refrigerant gas supplied into the second stage expander cylinder.

Valve 280 is supplied with high pressure refrigerant gas at port 282 and low pressure refrigerant gas is exhausted from port 284. Valve member 286 is mechanically connected to the crank shaft, which carries eccentrics 252 by mechanical means 288. The mechanical means 288 is a valve drive mechanism which drives valve member 286 in accordance with the position of the eccentric and crank shaft. The valve drive mechanism has just moved valve member 286 so that high pressure refrigerant is connected through regenerator 260 to space 270 while low pressure exhaust has been connected to space 268 through regenerator 258, and also to the lower end of regenerator 272. After a small amount of rotation, permitting pressures to substantially equalize, valve member 286 is moved to a closed position where communication from the ports to the regenerator is cut off, so that expansion takes place in volume 270 with compression taking place in the volume 268. At almost 180° from the position shown, the valve has been reversed from the position shown so that volume 268 is connected to the high pressure and volume 270 is connected to low pressure for a short portion of the angular motion, to equalize pressures, and then through the subsequent nearly 180° the high pressure refrigerant in expander space 268 is allowed to expand with compression of the refrigerant in space 270. Thus, refrigeration is produced.

It should be noted that in the Gifford-McMahon refrigerator 250, the pressure at the lower end of regenerator 258 acts upon the bottom displacer 256. The Solvay refrigerator is similar to the refrigerator 250, except that the lower end of regenerator 258 is ported out the side of displacer 256 to line 292. Thus, the double-acting expander concept is also applicable to Solvay cycle refrigerators.

Referring to FIG. 12, refrigerator 300 is similar to the refrigerator of cryostat 10. It has a central piston rod 302 which carries its top end double-acting expander

piston **304** in expander cylinder **306.** The bottom end of rod **302** carries valve spool **308** in valve body **310,** similar to spool **46** in body **44.** Valve body **310** is connected to high and low pressure refrigerant gas sources, and the valve body and valve spool are provided with suitable ports and seals. Lines **312** and **314** are alternately connected to the high and low pressure refrigerant gas sources. Lines **312** and **314** have regenerators **316** and **318** therein and above these regenerators the lines are connected to opposite sides of expander cylinder **320.** Expander piston **322** is mounted on rod **302** and divides cylinder **320** into a double-acting expander cylinder. With helium as a refrigerant gas, supply pressure of about 750 PSI and a low pressure of about 450 PSI, the first expansion stage temperature resulting from the expansion end first stage expansion cylinder **320** is typically 80–120° K.

Regenerators **324** and **426** are respectively connected in lines **312** and **314** between the first stage expansion cylinder **320** and the second stage expansion cylinder **328.** Cylinder **328** is divided into two expansion chambers by piston **330** which is also mounted on a rod **302.** The second stage expansion temperature, at the top of regenerators **324** and **326** is typically 25–50° K. For a small size refrigerator and cryostat, regenerators **324** and **326** are preferably filled with lead balls, while regenerators **316** and **318** are preferably filled with steel balls.

The cold end of regenerators **324** and **326** are connected through the coils **332** and **334** of a counterflow heat exchanger, and the cold end of these coils are connected to the two expansion chambers in the third stage expansion cylinder **306.** The temperatures typically reached at the heads of third stage expansion cylinder **306** are 5–7° K, depending upon heat load. It should be noted that the upper chambers of each of the expansion cylinders are connected together and the lower chambers are each connected together through heat exchange devices so stagewise cooling is achieved. Thus, the advantages of the double-acting structure to minimize pumping losses and shuttle losses is also achieved in a multiple stage expander engine.

This invention having been described as preferred embodiment, it is clear that it is susceptible to numerous modifications and embodiments within the ability of those skilled in the art and without the exercise of the inventive faculty. Accordingly, the scope of this invention is defined by the scope of the following claims.

What is claimed is:

1. A cryogenic refrigerator comprising:

a cylinder to receive a heat load;

a piston reciprocably mounted within said cylinder to divide said cylinder into a first expander space and a second expander space;

a first regenerator having a warm end and a cold end, said cold end of said first regenerator being connected to said first expander space so that gas flow into and out of said first expander space flows through said first regenerator;

a second regenerator having a cold end and a warm end, said cold end of said second regenerator being connected to said second expansion space so that gas flow into and out of said second expander space flows through said second regenerator;

means for supplying refrigerant gas at high pressure;

means for supplying refrigerant gas at low pressure;

valve means connected to both said refrigerant gas means and to both said regenerators for alternately supplying low pressure and high pressure refrigerant gas to said warm end of said second regenerator and for oppositely alternately supplying high pressure and low pressure refrigerant gas to said warm end of said first regenerator, the improvement comprising:

said valve means being coupled to said piston to be directly actuated by piston motion;

said piston and said valve means being influenced only by gas pressure and friction.

2. The expander of claim **1** wherein said valve means is operated so that gas conditions in said first and second expander spaces are substantially on the Stirling cycle.

3. The expander of claim **1** wherein said valve means is connected so that when high pressure gas is connected to said warm end of said first regenerator, low pressure gas is connected to said warm end of said second regenerator.

4. The cryogenic expander of claim **3** wherein said valve means comprises valve ports connected to said high and low pressure refrigerant gas supplies and a valve spool coupled to said piston and acting in conjunction with ports connected to both of said regenerators.

5. The cryogenic expander of claim **1** wherein said valve means is a valve spool connected to said piston for control by said piston and is operated by said piston as said piston moves in said cylinder while permitting expansion of refrigerant gas in said spaces in said cylinder, said valve being included in said means for supplying refrigerant gas to said warm end of said first regenerator and said means for supplying refrigerant gas to said warm end of said second regenerator.

6. The cryogenic expander of claim **5** wherein said means for alternately supplying high and low pressure refrigerant gas to said warm end of said first regenerator is a valve port and said means for alternately supplying low pressure and high pressure refrigerant gas to said warm end of said second regenerator is a valve port.

7. The cryogenic expander of claim **6** wherein a cover encloses said cylinder, a window in said cover, a device to be cryogenically refrigerated mounted on said cylinder and having a view out of said window, said cover providing a vacuum space between said cylinder and said cover.

8. The cryogenic expander of claim **1** wherein a cover encloses said cylinder, a window in said cover, a device to be cryogenically refrigerated mounted on said cylinder and having a view out of said window, said cover providing a vacuum space between said cylinder and said cover.

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