United States Patent [19]

Lagodmos

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[45] **July 3, 1973**

[54] **CRYOGENIC REFRIGERATOR**

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

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[21] Appl. No.: **235,275**

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

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

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

[56] **References Cited**

UNITED STATES PATENTS

2,508,315	5/1950	VanWeenan	60/24
3,315,465	4/1967	Wallis	60/24
3,416,307	12/1968	Wallis	62/6
3,416,308	12/1968	Livezey	62/6
3,457,722	7/1969	Bush	62/6
3,482,457	12/1969	Wallis	60/24

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Attorney—W. H. MacAllister, Jr. and Allen A. Dicke, Jr. et al.

[57] **ABSTRACT**

A phase angle adjustment mechanism for adjusting the phase angle of cryogen gas displacers in Vuilleumier cycle refrigerators is provided. By this adjustment, the phase angle is adjusted to accommodate for non-ideal conditions in cycle operation to obtain maximum refrigeration.

11 Claims, 12 Drawing Figures

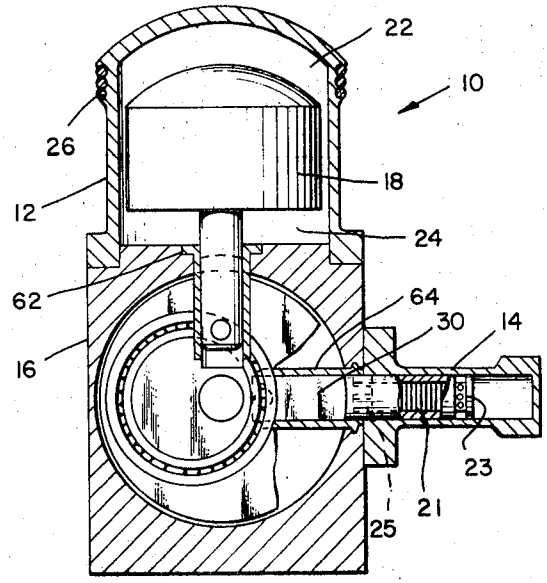


Fig. 1.

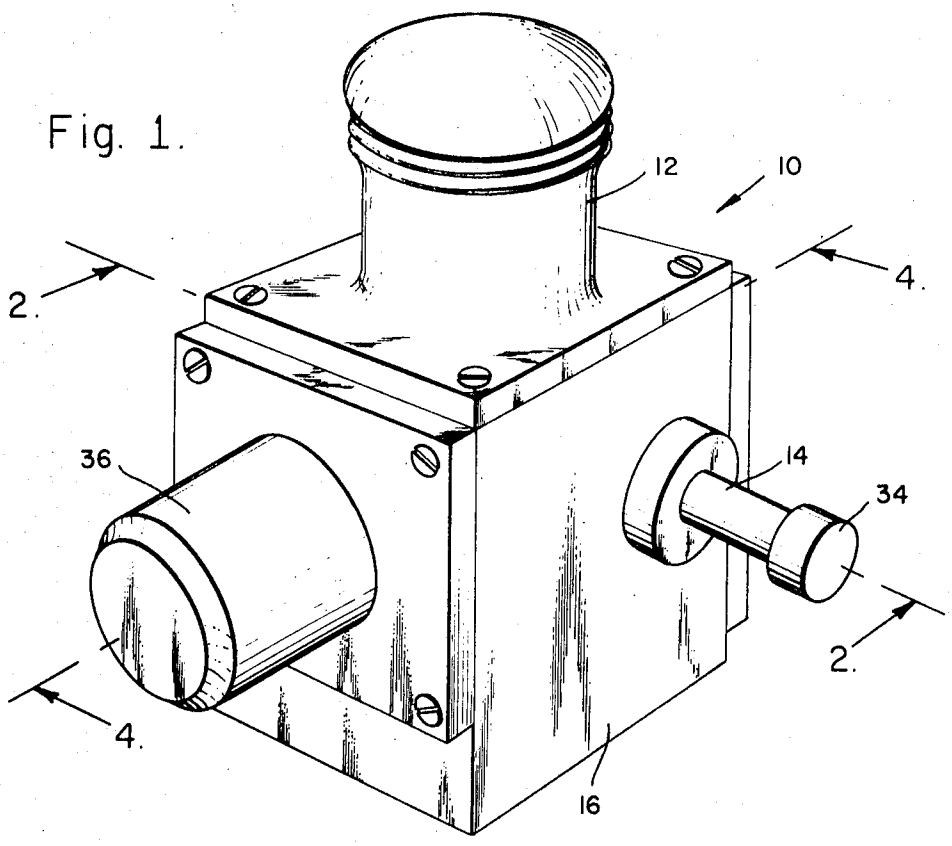
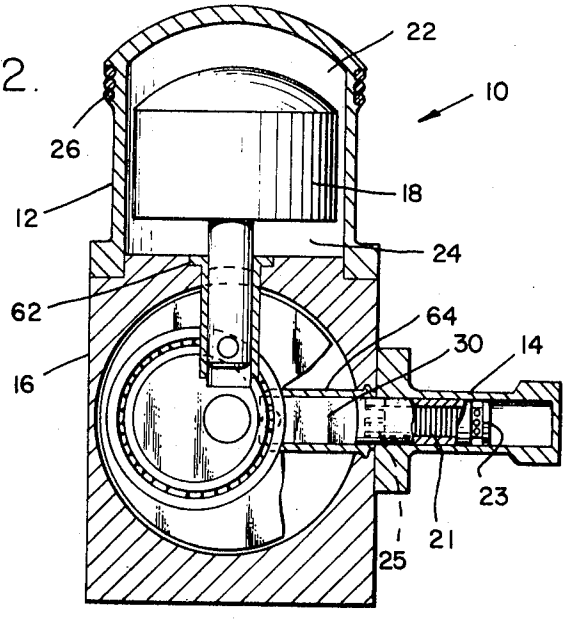


Fig. 2.



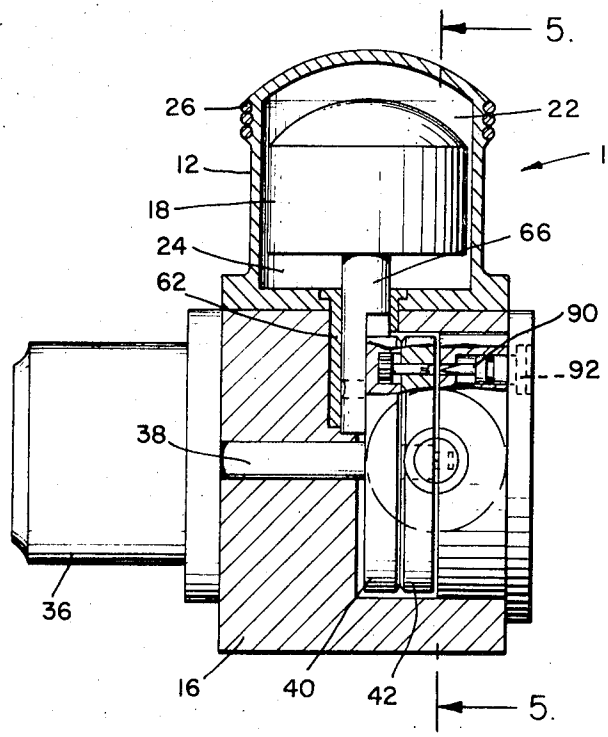


Fig. 4.

Fig. 5.

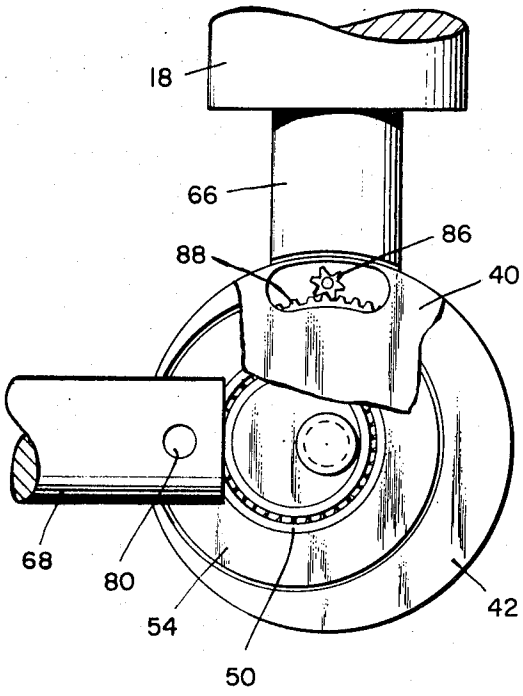


Fig. 3.

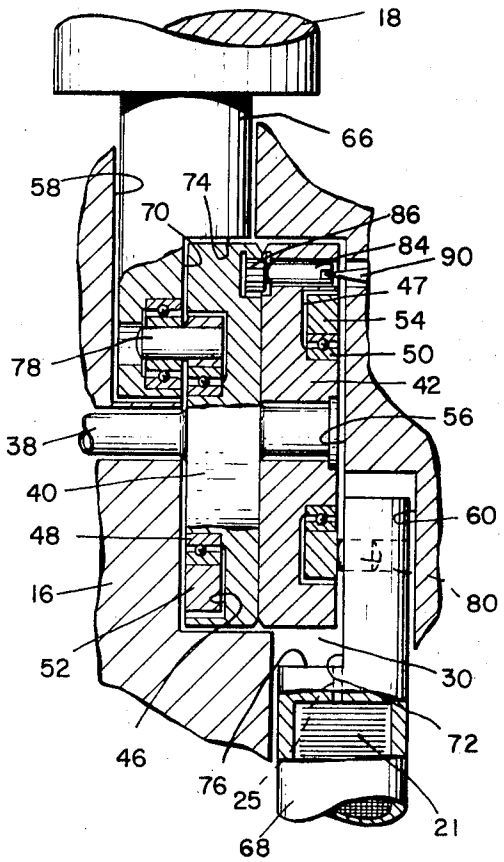


Fig. 6.

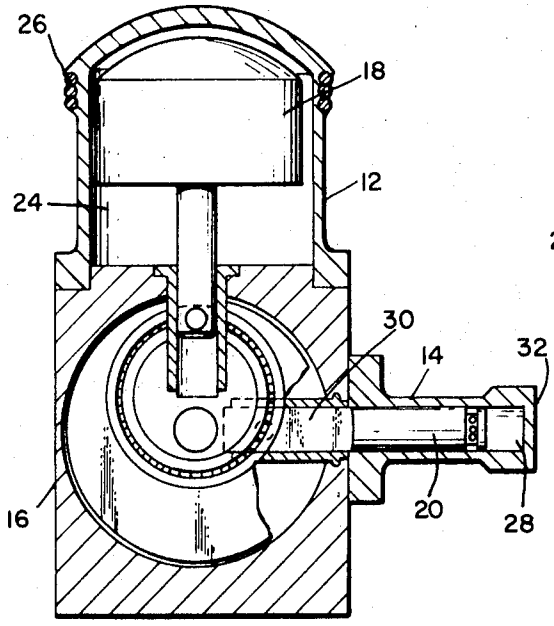


Fig. 7.

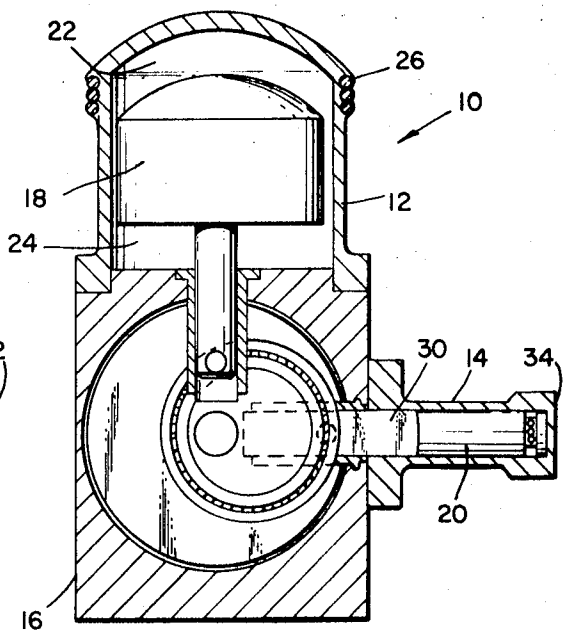


Fig. 8.

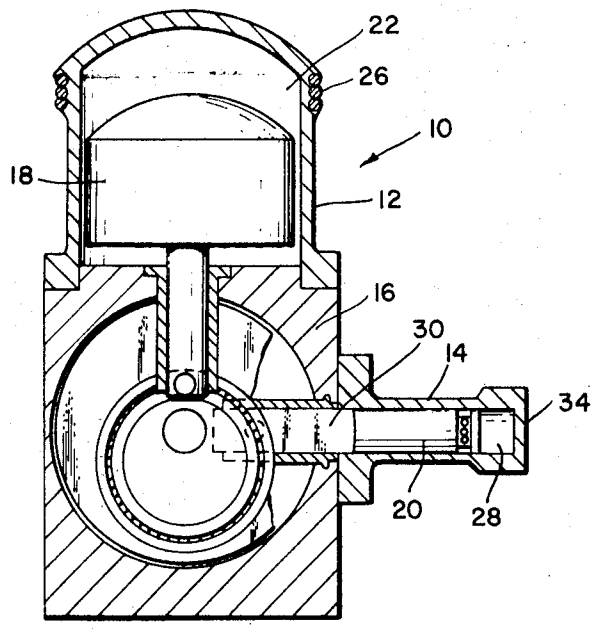
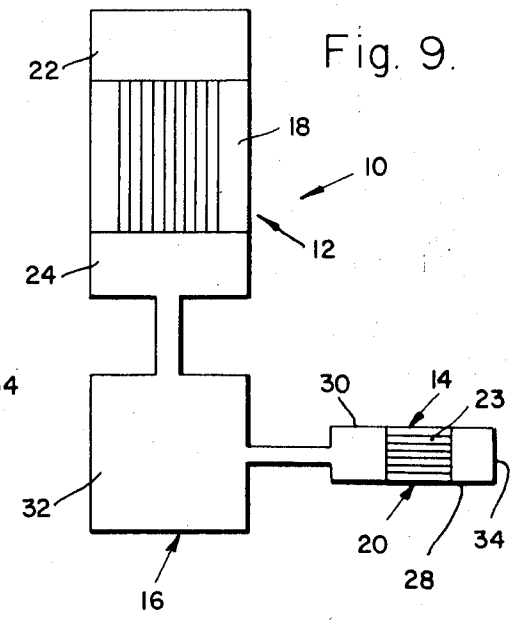


Fig. 9.



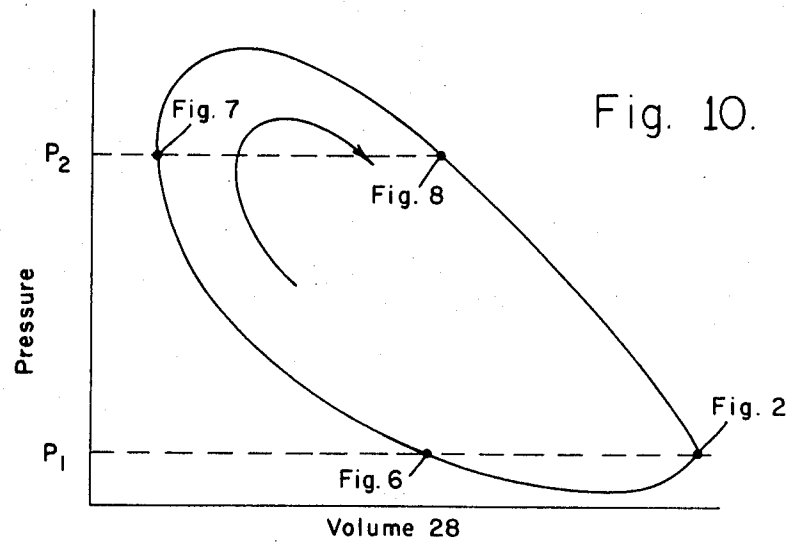

Fig. 7
Fig. 8
Fig. 2
Fig. 6
P2
P1
Pressure
Volume 28

Fig. 10.

Fig. 11.

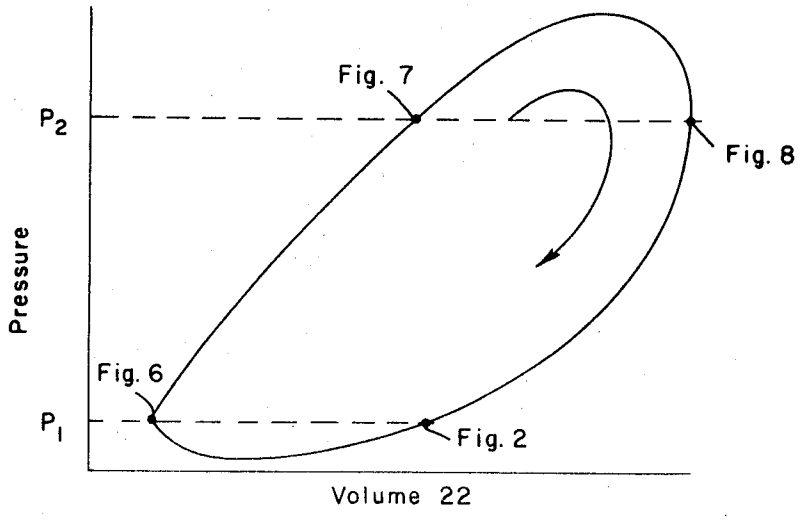
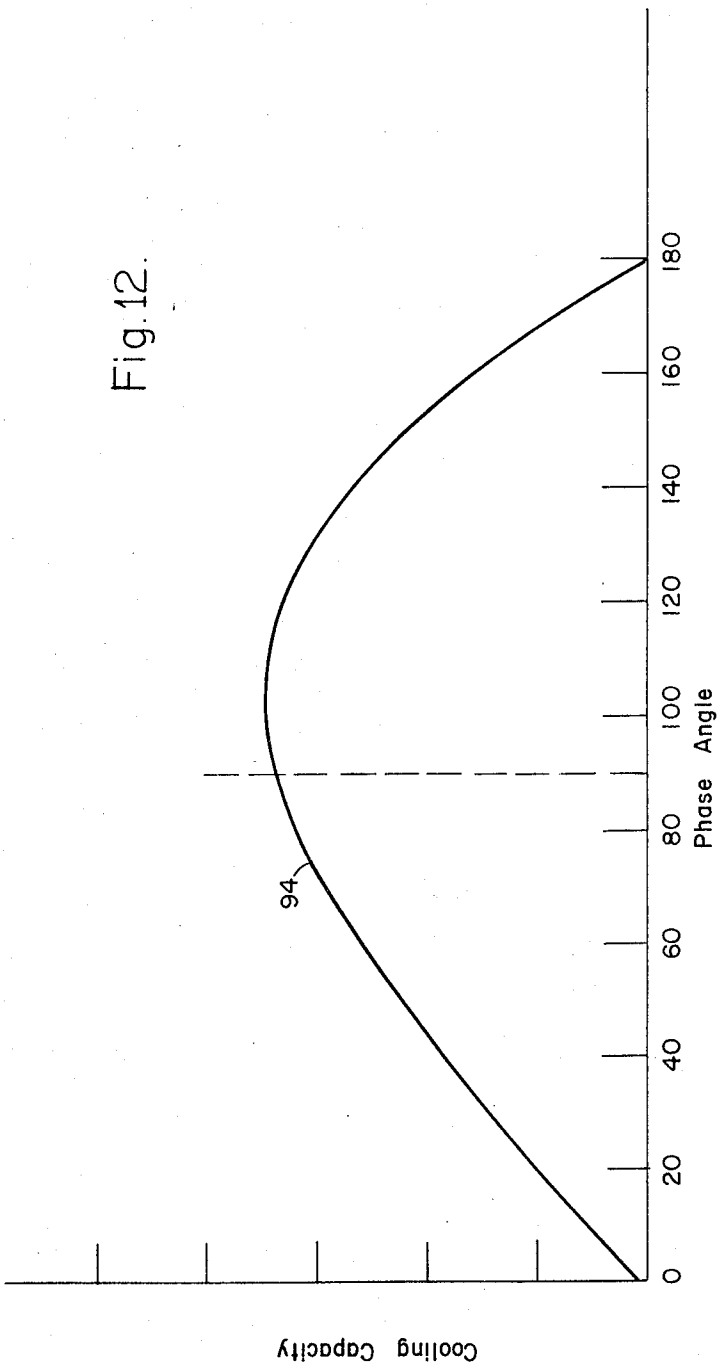

Fig. 7
Fig. 8
Fig. 6
Fig. 2
P2
P1
Pressure
Volume 22

Fig. 12.


Cooling Capacity
0
20
40
60
80
100
120
140
160
180
Phase Angle
94

CRYOGENIC REFRIGERATOR

BACKGROUND

This invention is directed to an apparatus for the adjustment of the phase angle of the gas displacers in Vuilleumier cycle cryogenic refrigerators.

U.S. Pat. No. 1,275,507 describes a three chamber refrigerator cycle commonly known as the Vuilleumier cycle, or the VM cycle, being named after the inventor-patentee. A more modern refrigeration machine employing this cycle is shown in Cowans U.S. Pat. No. 3,423,948 (reissue application Ser. No. 48,755). The equipment described in these patents for performing the cycle is subject to redesign for improvement of cycle efficiency. Further background on this cycle is found in the publication "Miniature Vuilleumier-Cycle Refrigerator," by G. K. Pitcher and F. K. DuPré, published in *ADVANCES IN CRYOGENIC ENGINEERING*, Vol. 15, Plenum Press, N.Y., 1970, at pages 447 through 451.

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 cryogenic refrigerator apparatus wherein two cryogen gas displacers cyclically move with respect to each other, and the cyclic phase angle thereof is adjustable to increase refrigeration efficiency.

Accordingly, it is an object of this invention to provide a cryogenic refrigerator apparatus of improved design. It is a further object to provide a refrigerator apparatus which has two cyclically-moving displacers which move at an optimized cyclic phase angle. It is another object to provide a cryogenic refrigerator apparatus specifically designed for employing the Vuilleumier cycle and wherein the cryogen gas displacers operate at a cyclic phase angle other than 90 degrees, whereby refrigeration is optimized to increase cycle efficiency. It is a further object to provide a refrigerator apparatus which is economic to produce and of long, trouble-free life. Other objects and advantages of this invention will become apparent from a study of the following portion of the specification, the claims, and the attached drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a perspective view of a cryogenic refrigerator apparatus in accordance with this invention.

FIG. 2 is a section taken generally along the line 2—2 of FIG. 1.

FIG. 3 is an enlarged partial section, with parts broken away, taken generally along the line 3—3 of FIG. 2.

FIG. 4 is a partial section, with parts broken away, taken generally along the line 4—4 of FIG. 1.

FIG. 5 is an enlarged partial elevational view, with parts broken away, as seen generally along line 5—5 of FIG. 4.

FIGS. 6, 7, and 8 are sections similar to FIG. 2 showing the apparatus in progressive positions.

FIG. 9 is a schematic diagram of the refrigerator showing the several volumes and the interconnection thereof.

FIGS. 10 and 11 are P-V diagrams showing the conditions in the cold and hot volumes, respectively.

FIG. 12 is a graph showing the relationship between cooling capacity and displacer phase angle.

DESCRIPTION OF THE PREFERRED EMBODIMENT

In order to fully understand the significance of the design, the Vuilleumier cycle should be understood so that the improvement in efficiency by operating at an optimum phase angle can be fully appreciated. Referring principally to FIGS. 1, 2, 4, 6, 7, and 8, the cryogenic refrigerator apparatus 10 is shown therein. The apparatus 10 has a hot cylinder 12 and a cold cylinder 14. These are both mounted upon a crankcase, or crank housing 16. Hot displacer 18 is mounted in hot cylinder 12, while cold displacer 20 is mounted in cold cylinder 4. As the hot displacer 18 moves through the hot cylinder 12, it divides the interior of the cylinder into a hot volume 22 and an ambient volume 24. While it is called an ambient volume, the temperature of volume 24 is slightly above the ambient temperature of the environment, because heat is rejected from the cryogen gas at the temperature of the ambient volume to the exterior environment. However, this volume is called an ambient volume, because it is neither purposely elevated nor at the point of refrigeration.

Heater 26 is provided to heat the cryogen gas in the hot volume 22. Hot displacer 18 contains a regenerator therein, as indicated schematically in FIG. 9, so that as the hot displacer sweeps through the cylinder 12, heat is exchanged between the cryogen gas and the regenerator mass therein.

Similarly, cold displacer 20 divides the cold cylinder 18 into a cold volume 28 and an ambient volume 30. A regenerator 21 is built into cold displacer 20, with access through top and bottom ports 23 and 25, see FIGS. 2 and 3, so that heat is exchanged between the cryogen gas and the regenerator mass as the cryogen moves through the displacer and the displacer sweeps the cold cylinder. The cold spot at which refrigeration is developed is shown at 34.

In the manner which is subsequently described, the hot and cold displacers are mechanically connected so that they operate at substantially a 90° phase angle with respect to each other. This is illustrated by the mechanical configuration of the exterior of the device, where the cylinders are positioned at 90° with respect to each other. On the other hand, the cylinders may be positioned in other orientations with the phase angle regulated by the relative position of the cranks.

The refrigerator is a constant volume device. The total volume of the cold chamber 28, the hot chamber 22, and the total ambient volume comprised of chambers 24 and 30 and the interconnecting passageways, including the ambient volume 32 in crank housing 16 is constant. However, the volume is divided in different fractions, depending upon the positions of the hot and cold displacers 18 and 20. In spite of the fact that the motion of the two pistons cannot vary the total volume, pressure varies as the displacers move because of changes in the average temperature of the gas that fills the total volume. Considering the displacers individually, as the hot displacer 18 moves toward the crankshaft, it forces gas from the ambient chamber 24 to the hot side of hot displacer 18, which is hot chamber 22. Since this movement causes an increase in the average temperature of the gas in these two portions of the total volume, the average pressure in the entire refrigerator increases. Likewise, as the cold displacer 20 moves toward the crankshaft, it forces some of the gas from the

ambient volume 30 to the cold end of displacer 20 into cold volume 28. As the gas is thus cooled, the pressure decreases.

In operation, the crankshaft 30 rotates in the clockwise direction, as seen in FIGS. 2, 6, 7, and 8. The relative volume sizes of cold volume 28 and hot volume 22 are shown in FIGS. 10 and 11 as the displacers move through their cycles. As the crank moves in the clockwise direction, the pressure-volume characteristics of these two volumes change, as indicated. Starting from the position of FIG. 7, the cold displacer 20 is in its extended position in cold cylinder 14, while the hot displacer 18 is in an intermediate position within its hot cylinder 12, and moving toward the crankshaft. As the crank moves 90 degrees to the position of FIG. 8, the cold displacer 20 moves to its central position, and the hot displacer 18 moves to its bottom dead center position. While moving through this quadrant, both pistons are moving toward the center line of the crankshaft. As previously discussed, motion in this direction by the hot displacer 18 tends to cause an increase in pressure, and motion in this direction by cold displacer 22 tends to cause decrease in pressure. These actions are essentially balanced, and the average pressure does not substantially change, as indicated in FIGS. 10 and 11. The net effect is that a certain amount of the working cryogen gas is transferred to the cold volume 28 without undergoing any substantial change in pressure.

Further movement of the crank in the clockwise direction to the position of FIG. 2 brings the cold displacer 20 to its bottom dead center, and hot displacer 18 to an intermediate position. During crank motion through this quadrant, the hot displacer 18 is moving away from the crank, while cold displacer 20 is moving toward the crank. As previously discussed, motion of the pistons in this direction both cause cooling of the gas and the pressure falls from a substantially constant high value, P_2 in FIGS. 10 and 11, to a new, substantially constant low value of P_1 in FIGS. 10 and 11. This is the refrigeration portion of the cycle by gas expansion in cold volume 28. As the crank moves from the position of FIG. 2 to the position of FIG. 6, the hot displacer 18 moves to its top dead center position, while the cold displacer 20 moves to an intermediate position. During this motion, the cold displacer 20 moves gas to a warmer region, and the hot displacer 18 moves gas to a cooler region, substantially without affecting the pressure, as indicated in FIGS. 10 and 11. Thus, the pressure remains substantially at the low value P_1, and gas is transferred from hot chamber 22 to cold chamber 28.

In the next quadrant of motion, from the position of FIG. 6 to the position of FIG. 7, cold displacer 20 is moved to the top dead center position, reducing cold volume 28, and the hot displacer is moved to an intermediate position toward the crank to thus arrive at the initiation point of the refrigeration phase of the cycle. This motion causes both pistons to cause increase in pressure of the gas to raise the pressure from P_1 to P_2.

The pressure-volume curves of FIGS. 10 and 11 show that the amount of work that has been done by the gas in the cold volume exactly matches the amount of work that has been supplied to the gas in the hot volume. The mechanical P-V work of the cryogen gases in the volumes act upon each other so that they are coupled and are equal. The work done in the expansion of the cold cryogen is equal to the work done in compression of the hot cryogen. Therefore, thermodynamic heat addition by heater 26 causes an equal thermodynamic extraction or cooling at cold point 34.

It must be noted, however, that the change in pressure is strongly influenced by the size of the ambient volume. With the same temperature change, a lesser pressure change will occur when the ambient volume is larger. Therefore, the ambient volume should be minimized to achieve higher pressure ratios on the cycle and thus achieve more refrigeration per unit equipment size.

Since the total of volumes 24, 30, and 32 comprises the volume at ambient temperature which is just sufficiently higher than the temperature of the environment so that heat can be rejected from the ambient volume to the environment, this volume does not contribute to pressure changes resulting from the average temperature of the entire cryogen mass. Instead, the crank volume reduces the pressure ratio and thus is detrimental to refrigerator efficiency. With reduction of crank volume 32 to zero, all of the volumes are swept volumes so that all volumes contribute to the cycle operation. Thus, zero volume of the crank volume 32 would produce the maximum pressure changes in the system with the movement of the cryogen gas from one volume to another, in the manner previously described. With the highest compression ratio, the greatest amount of refrigeration is obtainable from a cryogenic refrigerator of a particular size. While a zero ambient volume 32 is not possible, because there must be a connection between the volumes 24 and 30, the ambient volume 32 of the crank chamber can be minimized. FIGS. 3, 4, and 5 show the structure of this invention which produces a minimum crankcase volume.

Crank housing 16 carries motor 36 thereon. Motor 36 is principally a speed control device which regulates the refrigerator cycle speed. While it has a power input thereto, the power input is small and principally overcomes friction. The power input is not the principal power input of this system. Motor shaft 38 extends into crank housing 16 and is rotatably mounted therein on the usual bearings and is equipped with the usual seals to prevent the escape of cryogen gas. Discs 40 and 42 are mounted on the end of motor shaft 38 and are rotated thereby. Eccentric groove 46 is machined in disc 40, while eccentric groove 47 is machined in disc 42. Bearings 48 and 50 are respectively pressed into these grooves at the interior thereof. Eccentric drive rings 52 and 54 are respectively mounted on bearings 48 and 50, and substantially fill the grooves 46 and 47, respectively. The outer race of the bearings and the eccentric drive rings have a small running clearance thereabout.

The entire structure of discs 40 and 42 is mounted within cavity 56 within crank housing 16. Cavity 56 is cylindrical and is in a close running fit with respect to the discs 40 and 42. Thus, minimum clearance volume is provided. Guide bores 58 and 60 are formed in a suitable bearing material, such as bearing sleeves 62 and 64, see FIG. 2. Piston rods 66 and 68 are respectively slidably located in these bearing sleeves. The guide bores intersect with the cavity 56 so that the piston rods can interconnect with the eccentric drive rings. The relative positioning is such that the center line of guidebore 58 is substantially in a line with the left side of the cavity 56, as seen in FIG. 3, while the center line of the guidebore 60 is substantially in line with the right side of the cavity 56, as seen in FIG. 3. In this way, the pis-

ton rods have a flat machined surface on one side thereof so that the surface is a diameter. These flats are shown at **70** and **72.** The end of the machining which results in these flats produces surfaces **74** and **76** which are portions of cylindrical surfaces having substantially the same radius as the discs. The surfaces **74** and **76** are positioned along the piston rods so that, when the piston rod is in its fully retracted position, such as rod **66** in FIG. **3,** there is a close running fit between the surface **74** and the outer surface of disc **40.** Similarly, the terminal ends of the piston rods closely fit the bottom of the guidebores, when in retracted position.

Interconnection between the piston rods and the eccentric drive rings is accomplished by drivepins **78** and **80,** respectively. They are mounted in suitable bearing in the piston rods. Since there is oscillatory motion between the drivepins and their bores in the piston rods, ball bearings, as illustrated, can be employed to reduce friction. The running clearances illustrated are exaggerated to show the spacing between parts to show which were movable with respect to others without touching. Clearances are minimized, particularly between bores **58** and **60,** in order to minimize the crankcase volume **32.** Gas interconnection between the volumes **24** and **32** is accomplished by providing small flats along the sides of piston rod **66** to permit gas passage from the swept volume under piston **18** to the crank chamber cavity **56.** Running spaces are adequate to permit passage of the cryogen gas.

Cold displacer **20** can either be in the form of a larger piston displacer on a smaller piston rod, or the displacer can be of the same diameter as the piston rod, as illustrated in FIG. **3.** In this case, the entire swept volume **30** is in the guidebore **60** and is displaced by the inner end of piston rod **68,** as is best seen in FIG. **3.** Passageway from the swept volume **30** into the regenerator **21** is through ports **25.**

It can be seen that this structure produces the minimum crank housing volume **32,** because most of the space is filled with metal. Minimum dimensioned running fits provide what crank volume **32** remains. Even the volumes at the inner ends of the piston rods are swept by piston rod motion so that minimum fixed volume remains. This improves the efficiency of the cryogenic refrigerator, as compared to one with a greater crankcase volume **32.**

As an example of particular dimensions and fits, in a small VM cryogenic refrigerator, the stroke of both the pistons is 0.25 inch. With this stroke, convenient diameter for the discs **40** and **42** of 1.50 inches and a convenient total axial length thereof is 0.25 inch, the spacing between the discs and the cavity can be made as little as 0.002 inch. Similarly, the spacing between the ends of the piston rods and the bottoms of their guide bores can be as little as 0.002 inch, and the spacing between the surfaces **74** and **76** and the outer diameter of the discs can be as little as 0.002 inch. With small flats provided on the piston rod **66** to communicate with the cavity **56,** this spacing within the cavity **56** is sufficient to permit cryogen gas flow between the two cylinders.

With the efficiency of the system thus improved by minimizing as nearly as possible the unswept ambient volume, the differences between theoretical performance and actual performance are sufficient to warrant the adjustment of the phase angle from the theoretical 90° displacement to a phase angle which produces greater refrigeration capacity. Non-optimum conditions which produces losses include mechanical friction losses, pressure drops due to flow and conductive heat transfer. These losses, even in a well-designed machine, change the operating conditions sufficiently from the ideal that greater capacity is achieved by operating at other than the theoretically correct 90° phase angle. In fact, it is only in a well-designed machine where other losses are minimized that use of a phase angle other than the theoretical 90° makes enough difference to make it worthwhile.

Referring principally to FIGS. **3, 4,** and **5,** disc **42** is rotatably mounted on stub shaft **82** extending from disc **40.** By this construction, the discs are rotatable with respect to each other. Pinion shaft **84** is rotatably mounted in discs **42** and carries pinion **86** thereon. Pinion **86,** in turn, mates on ring gear **88** secured in an annular groove in disc **40.** If desired to maintain minimized ambient volume, the ring gear **88** and its corresponding groove can be segmental, as indicated in FIGS. **3** and **5.**

Rotation of the pinion **86** causes relative rotation of the discs **40** and **42.** The structure is stiff enough that, when no torque is applied to the pinion shaft **84,** the discs **40** and **42** remain in their relative angular position. Other interlocking thereof can optionally be provided by conventional means.

Rotation of pinion shaft **84** can be by any convenient means. Normally, phase angle adjustment during operation of the refrigerator **10** is not necessary, but rotation to effect the phase angle need only be accomplished whn the refrigerator is stopped. The end of pinion shaft **84** is slotted. A sliding and rotatable key **90** is provided in the crankcase housing **16.** When the key is pressed forward, it can enter the slot in the end of the pinion shaft, as illustrated in FIG. **3.** From this position, it can be turned so that pinion **86** operates in ring gear **88** to relatively rotate the discs to change the phase angle. The structure is preferably arranged so that a spring or pressure within the refrigerator holds the key outward when it is not in use. As illustrated in FIG. **4,** a cap **92** can be closed over the keyhole to provide sealing for the cryogen gas.

In operation, the refrigerator is assembled and first operated with a nominal phase angle. Testing thereof privides cooling capacity data. A small adjustment can be made, followed by further testing, to show whether or not there is an increase in cooling capacity. FIG. **12** shows a particular refrigerator structure and illustrates its cooling capacity at the theoretical 90° operating phase angle, as well as its cooling capacity for other relative phase angles. Curve **94** illustrates the cooling capacity of the refrigerator for phase angles varying from zero to 180°. In test of the equipment, it was found that a phase angle of about 105° produced more cooling capacity for the particular refrigerator than the theoretically-desirable 90° angle.

This invention having been described in its 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 apparatus comprising:

a crank housing;

first and second cylinders carried on said crank housing, cryogen gas interconnection means intercon-

necting said first and second cylinders for permitting flow of cryogen gas between said first and second cylinders;

first and second pistons mounted for reciprocation respectively within said first and second cylinders for acting on cryogen gas within said cylinders to cause flow of cryogen gas between said cylinders;

drive means interconnecting said pistons for operating said first and second pistons in cyclic relationship with respect to each other, the improvement comprising:

said drive means being adjustable so that the relative phase of operation of said first and second pistons with respect to each other is adjustable to optimize refrigerator capacity,

said drive means having first and second surfaces eccentrically rotatable about an axis, and positioned within said crank housing, said eccentric surfaces being connected to drive said pistons, said eccentric surfaces being relatively rotatably adjustable about said axis to adjust operating phase angle between said first and second pistons.

2. The cryogenic refrigerator apparatus of claim **1** wherein said eccentric surfaces are mounted on first and second discs, said first and second discs being rotatable on said axis, and being rotatable with respect to each other to obtain the phase angle adjustment.

3. The cryogenic refrigerator apparatus of claim **2** wherein said crankcase has a substantially cylindrical cavity therein, and said discs are rotatably mounted in said cylindrical crankcase cavity.

4. The cryogenic refrigerator apparatus of claim **3** wherein said disc has an eccentric groove in an axial face thereof, said disc carrying a laterally-extending drivepin thereon, said drive pin extending into said groove.

5. The cryogenic refrigerator apparatus of claim **4** wherein an eccentric drive ring is positioned in said eccentric groove is said disc, and said drivepin is mounted in said eccentric drive ring, said eccentric drive ring being rotatably mounted with respect to said disc within said eccentric groove.

6. The cryogenic refrigerator apparatus of claim **5** wherein said eccentric drive ring is mounted on an antifriction bearing with respect to said disc within said eccentric groove.

7. The cryogenic refrigerator apparatus of claim **2** wherein a piston rod is connected to said piston, said piston rod extending into a guidebore within said crank housing, said guidebore intersecting said cylindrical cavity so that said piston rod can interengage with said actuating means so that piston position is controlled by said disc.

8. The cryogenic refrigerator apparatus of claim **7** wherein said piston rod is cylindrical and its axis substantially intersects with the axis of rotation of said disc at substantially right angles, said axis of said piston rod lying substantially on an axial face of said disc.

9. The cryogenic refrigerator apparatus of claim **8** wherein said piston rod is axially positioned along its guidebore in accordance with said eccentric groove, and when said piston rod is positioned closest to the axis of said disc, the portion of said piston rod closest to said disc is in a non-engaging close-running fit with respect to said disc.

10. The cryogenic refrigerator apparatus of claim **9** wherein the end of said piston rod toward the axis of rotation of said disc has an axially-directed flat position thereon, said flat being substantially formed along a diameter of said piston rod and terminating in a surface which has a curvature substantially equal to the cylindrical curvature of said disc and said cavity in said crank housing.

11. The cryogenic refrigerator of claim **2** wherein said first disc carries a gear, and said second disc carries a pinion, said pinion being engaged in said gear so that, when said pinion is rotated with respect to said second disc, said second disc is rotated with respect to said first disc on said axis.

* * * * *