

[54] **FREE PISTON CRYOGENIC REFRIGERATOR WITH PHASE ANGLE CONTROL**

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[52] U.S. Cl. **62/6; 62/86**

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

[58] Field of Search **62/6, 86**

[56] **References Cited**

UNITED STATES PATENTS

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[57] **ABSTRACT**

Cryogenic refrigerator free piston displacer reciprocates as a function of cyclic operating pressure, seal drag and bounce cylinder pressure and volume. Bounce cylinder volume is established for a particular pressure pulse frequency to optimize phase angle.

4 Claims, 3 Drawing Figures

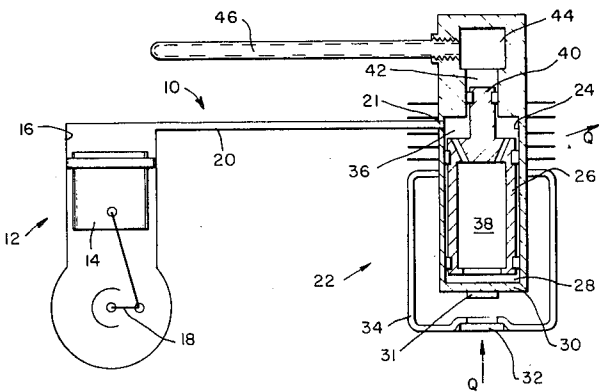


Fig. 1.

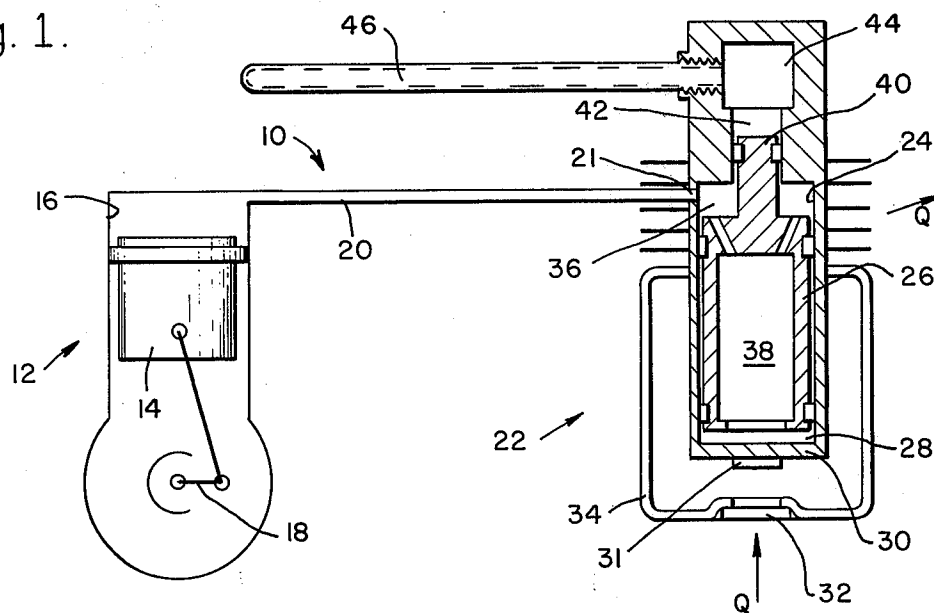


Fig. 3.

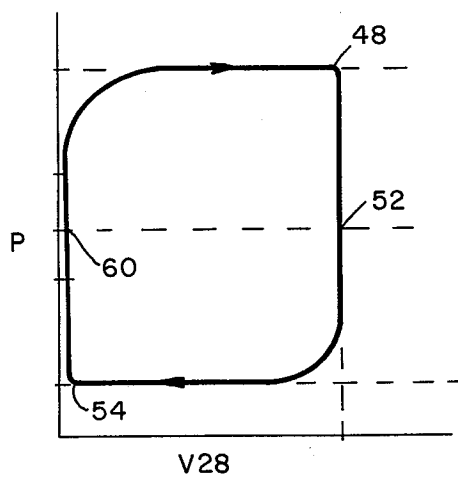
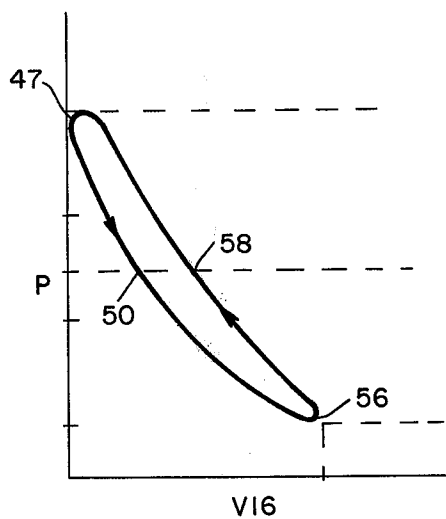


Fig. 2.



FREE PISTON CRYOGENIC REFRIGERATOR WITH PHASE ANGLE CONTROL

BACKGROUND OF THE INVENTION

This invention is directed to a free piston cryogenic refrigerator, and particularly one where the spring constant for returning the cold displacer is conditioned upon structural and operating criteria to cause the cold displacer to operate in an optimum cyclic condition.

Free piston cryogenic refrigerators have a pulse generator which generates cyclic pressure pulses of refrigerant gas. A reciprocating cold cylinder displacer acts as an expander and responds to the pulses by reciprocation within the cold cylinder. Response to the pulses is accomplished by having a larger area on the cold end of the cold displacer than on the ambient end, and have a balance or bounce spring on the warm end.

Higa patent U.S. Pat. No. 3,367,121 is an example of such structure which employs a Stirling pulse generator. Furthermore, the Robert Berry and Axel Dehne application Ser. No. 447,417, filed Mar. 1, 1974 entitled VUILLEUMIER REFRIGERATOR WITH SEPARATE PNEUMATICALLY OPERATED COLD DISPLACER is an example of such a structure when a VM pulse cycle pressure source is used.

SUMMARY OF THE INVENTION

In order to aid in the understanding of this invention it can be stated in essentially summary form that it is directed to a free piston refrigerator which has control of its phase angle so that the phase angle between piston stroke and input pressure cycles is substantially 90° and is preferably at optimum refrigeration efficiency. This is accomplished by controlling the spring rate of the cold displacer bounce spring so that the cold displacer operates at the proper phase angle when the correct pressure pulse input frequency is applied to the warm end of the cold cylinder.

It is thus an object of this invention to define and provide the proper return force on a free piston cold displacer so that the piston operates at the optimum phase angle with respect to the power pulse.

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

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a partly sectional and partly schematic diagram of the free piston refrigerator with phase angle control of this invention.

FIG. 2 is a PV diagram of the pulse generator of this invention.

FIG. 3 is a PV diagram of the cold end of the cold cylinder.

DESCRIPTION

This invention is directed to a free piston cryogenic refrigerator apparatus with phase angle control of the free piston with respect to the pressure pulses from the refrigerant gas pulse generator.

It is applicable to those reciprocating machines where the motions of the compressor, whether it be a mechanical or thermal compressor, and the expander are not mechanically linked. Modified Stirling refrigerators are shown in the invention by W. H. Higa, U.S. Pat. No. 3,421,331 particularly FIG. 5, and in G. Prast,

U.S. Pat. No. 3,487,635. An example of the modified VM refrigerator is shown in Robert Berry and Axel Dehne patent application Ser. No. 447,417 filed Mar. 1, 1974 entitled "Vuilleumier Refrigerator with Separate Pneumatically Operated Cold Displacer", and Bruno S. Leo application Ser. No. 449,182, filed Mar. 7, 1974 for "VM Cryogenic Refrigerator Hot Cylinder Burner Head".

In general, modified Stirling and modified VM refrigerators have compressor and expander sections which are separated, except for a gas passage between the two sections. Motion of the cold displacer piston is controlled by input pulse frequency, pressure, free piston mass, piston area, size and length of the connecting line between the warm end of the free piston and the compressor cylinder and the spring constant of the return spring. In a practical design, quite a number of these factors are fixed by other criteria. For example, the input frequency, pressure, free piston mass and the connecting line size are determined by other factors. It is the spring constant of the return spring which is most convenient of adjustment.

FIG. 1 illustrates a modified Stirling refrigerator 10. Refrigerator 10 comprises a compressor 12 which has a piston 14 reciprocating in compressor cylinder 16. Reciprocation is caused by crank 18 which is driven by an appropriate motor. This produces pressure pulses of refrigerant gas in line 20. For purposes of phase angle considerations there is no phase delay between operation of piston 14 and the pressure pulses at inlet 21 to warm volume 36.

The expander is generally indicated at 22. The expander comprises an expander cylinder 24 in which is mounted a reciprocating expander piston 26. Volume 28 is the expansion volume in which the refrigerant gas is expanded to produce refrigeration. Cold cylinder head 30 is the point of refrigeration and is the location upon which refrigerated devices such as IR device 31 are mounted. Window 32 in insulator housing 34 permits the refrigerated device, such as infrared sensitive device 31, to have a field of view external to the insulator housing. The insulator housing can be in the nature of a dewar or the like. Volume 36 is an ambient volume to which the line 20 is connected. Volume 36 is connected through the interior of cold piston 26 through regenerator 38 to cold volume 28. Sliders or guides on the exterior of the expander piston permit the piston to slide within the expander cylinder. There is no pressure drop along the length of the expander piston, except for the pressure drop through the regenerator, so that the sliders do not have a substantial seal duty.

Balance piston 40 is mounted on the expander piston and is sealed in balance cylinder 42. The upper end of the balance cylinder 42 is spring volume 44.

Calculations show:

$$\tan \theta = 2d \omega \omega_o^{-1} / 1 - (\omega \omega_o^{-1})^2$$

where:

θ = phase angle between the free piston and the input pressure pulse to the warm volume

$d = f / (2 \sqrt{km})$; f = free piston friction factor

ω = compressor piston frequency

ω_o = natural frequency of the free piston = $(k/m)^{1/2}$

$k = k_1 + k_2$

k_1 = spring constant of piston return spring-chamber 44

k_2 = spring constant of gas line 20 and of free piston cylinder 24

m = mass of free piston.

In designing free piston refrigerators having a free piston expander such as indicated at 22, several of the parameters or criteria come out as parts that are fixed by other design criteria. For example, input frequency, pressure, free piston mass and the size and length of the connecting line between the free piston and the pulse generator are quantities which are fairly fixed by the physical conditions of the installation, the net refrigeration required and the temperature of the cold point are similar parameters. The optimum phase angle is about 90°, but this angle is seldom attained in the final design unless particular care is taken in the critical design of the spring 44 to attain the desired spring constant. The spring constant in the bouncing chamber 44 is determined in accordance with the mathematical expression above, and the chamber is then designed to achieve that spring constant. In order to permit convenient and empirical adjustment of the spring constant, both to reduce the criticality of the original manufacture of this spring bounce chamber 44 and to permit critical adjustment of refrigerator operation to maximum operating efficiency even if it is not exactly at a 90° phase angle relationship, tube 46 is attached to the spring bounce chamber 44 to act as a part thereof. The volume of the tube is adjusted in size until the refrigerator is tuned to maximum performance. This adjustment may be means of a variable volume tube by a telescoping structure, or it may be removed and cut at each test, or it may be simply pinched closed for a short length along the closed end to reduce the volume until the maximum performance is achieved.

Referring to the operation of the cycle, FIG. 2 is a PV diagram of the compressor. With a compressor piston at top dead center point 47, the cold displacer is assumed to be at bottom dead center, as at point 48 which is the maximum cold volume. Since the pressure in the cold cylinder is now higher than the mean pressure in the pneumatic spring volume 44, a force acts on the cold displacer tending to hold it in the bottom dead center position at maximum cold volume. Thus, the cold displacer 26 is in the stable position at point 48.

During the first quarter rotation of the crank as compressor piston 14 moves toward bottom dead center, the system pressure decreases steadily and reaches the mean pressure at point 50 approximately at the end of this quarter revolution. The corresponding point in FIG. 3 is at point 52 where the cold displacer remains at the bottom dead center position. The pressure which maintains the cold displacer in this position has dropped to zero at point 52. During the next quarter revolution, the compressor piston continues to move toward bottom dead center, and the system pressure continues to drop. Since the pressure in the expansion volume has decreased below the mean pressure in the pneumatic spring volume 44, an activation force has developed. When this force exceeds the frictional drag of the seals and the pressure drop through the regenerator, the cold displacer will move towards its top dead position 54, as seen in FIG. 3. Below the point 52, the piston starts to move, and, as the pressure in line 20 decreases to its condition of bottom dead center point 56, the cold displacer 26 moves to its top dead center point 54, as illustrated in FIG. 3. At the end of the half cycle, the compressor piston is at bottom dead center and the cold displacer is at top dead center. In this position, pressure in the system is near minimum. The cold dis-

placer is again in a stable position, as being held in its top dead center position.

During the third quarter rotation of the crank, the compressor piston again moves toward top dead center from point 56 past point 58 to the top dead center point 47. During this motion, the system pressure increases steadily and reaches approximately the mean pressure at the quarter cycle point 58. The corresponding pressure is shown at point 60 in FIG. 3. During the final quarter cycle, the compressor piston continues to move toward top dead center and the system pressure continues to rise. When the pressure rises above the mean cyclic value; that is, above points 58 and 60, a resultant force is developed on the cold displacer 26. When this force exceeds frictional drag on the seals, the cold displacer will move toward bottom dead center, as illustrated at the curve above point 60 in FIG. 3. At the end of this cycle, the cold displacer is at bottom dead center and the compressor piston is at top dead center; thus, the cycle is complete. The area enclosed by the indicated PV diagram of the expander, FIG. 3, is the work performed by the gas on the cold displacer and is equal to the gross refrigeration developed at the expander.

This invention having been described in its preferred embodiment, it is clearly susceptible to numerous modifications and embodiments within the ability of those skilled in the art and without the exercise of the inventor 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 pulse tube, means for cyclically generating a refrigerant gas pressure pulse in said pulse tube;
 - a free piston expander comprising a cylinder having a cold end and a warm end, a free piston mounted in said expander cylinder to separate said cylinder into a cold volume and a warm volume, the area of said piston facing said warm volume being of smaller area than the cold end of said free piston facing said cold volume, said pulse tube being connected to said warm volume, the improvement comprising:
 - a spring urging said piston toward said cold end to reduce said cold volume, said spring having a spring constant such that the reciprocation of said free piston in said expander cylinder has an optimum substantially 90° phase angle relationship with respect to refrigerant gas pressure pulses delivered to said warm volume by said pressure pulse tube.
2. The cryogenic refrigerator of claim 1 wherein said spring urging said free piston to reduce said cold volume is a pneumatic spring.
3. The cryogenic refrigerator of claim 2 wherein said free piston has a bounce piston formed on the warm end thereof, said bounce piston engaging in a bounce cylinder, said bounce piston having a smaller area than the area of said free piston at its cold end, said bounce cylinder being connected to a pneumatic bounce chamber, the spring constant of said bounce chamber causing said free piston to operate at an optimum substantially 90° phase angle with respect to incoming pressure pulse cycles.
4. The cryogenic refrigerator of claim 3 wherein said bounce chamber has a spring constant to produce the physical relationship:

$$\tan \theta = 2d \omega \omega_o^{-1} / 1 - (\omega \omega_o^{-1})^2$$
 where:

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θ = phase angle between the free piston and the input pressure pulse to the warm volume
 $d = f/(2 \sqrt{k m})$; f = free piston friction factor
 ω = compressor piston frequency
 ω_o = natural frequency of the free piston = $(k/m)^{1/2}$
 $k = k_1 + k_2$

k_1 = spring constant of piston return spring-chamber 44
 k_2 = spring constant of gas line 20 and of free piston cylinder 24
 m = mass of free piston.
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