

[54] **THERMAL REFRIGERATION PROCESS  
AND APPARATUS**  
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94529  
[22] Filed: **Apr. 19, 1971**  
[21] Appl. No.: **135,490**

**Related U.S. Application Data**

[63] Continuation-in-part of Ser. No. 850,439, Aug. 15,  
1969, abandoned.  
[52] U.S. Cl. .... **62/6, 62/467**  
[51] Int. Cl. .... **F25b 9/00**  
[58] Field of Search ..... **62/6, 467**

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[57] **ABSTRACT**

Process and apparatus for producing refrigeration, or  
refrigeration and work directly from thermal energy.  
The thermal energy input can also be supplemented  
by work input to enhance the refrigeration effect.

**38 Claims, 28 Drawing Figures**

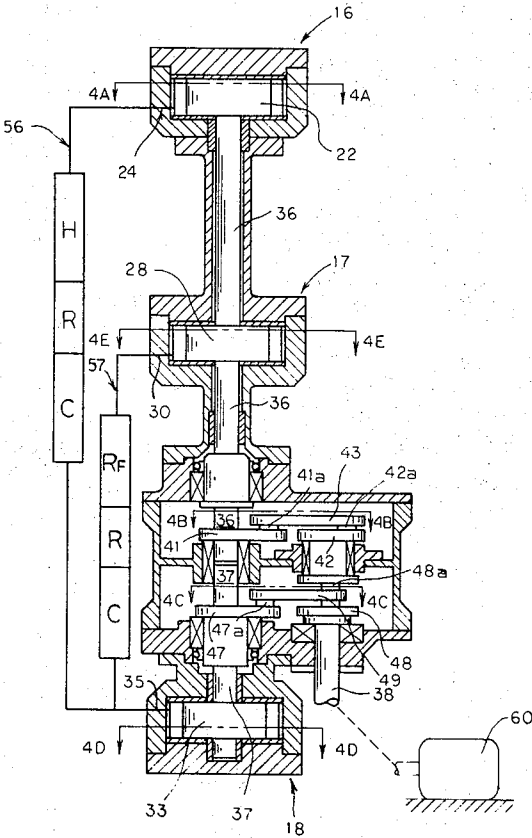


FIG 1

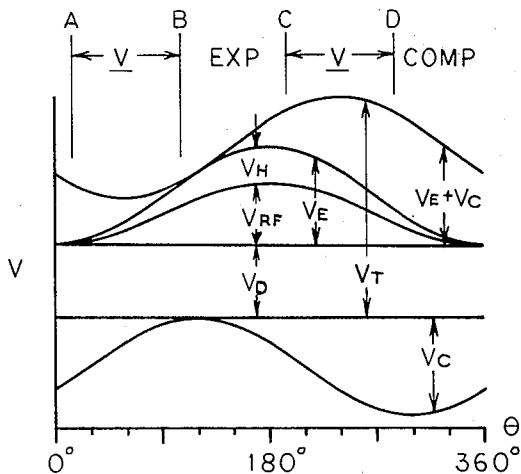


FIG 1A

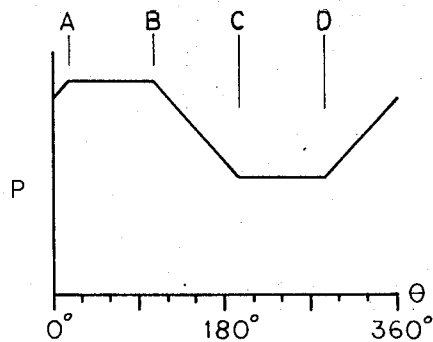
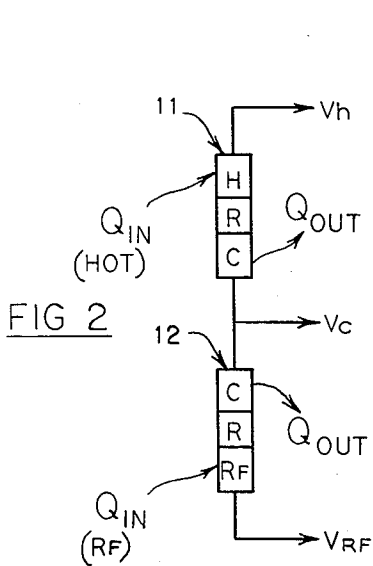
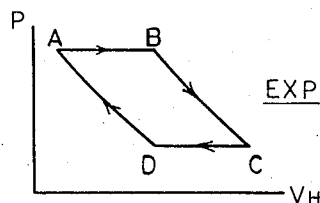


FIG 2A

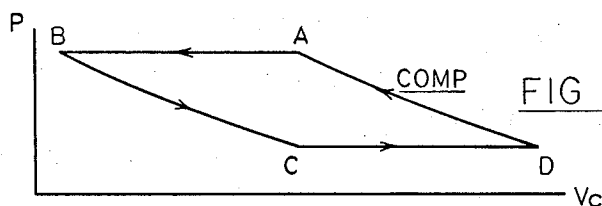


FIG 2B

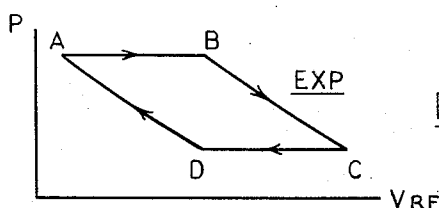


FIG 2C

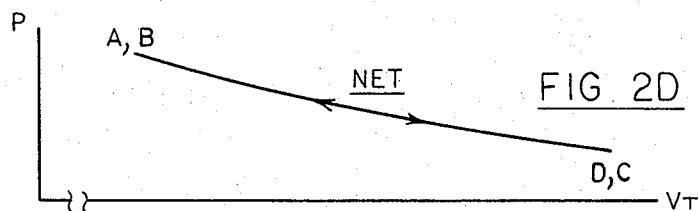
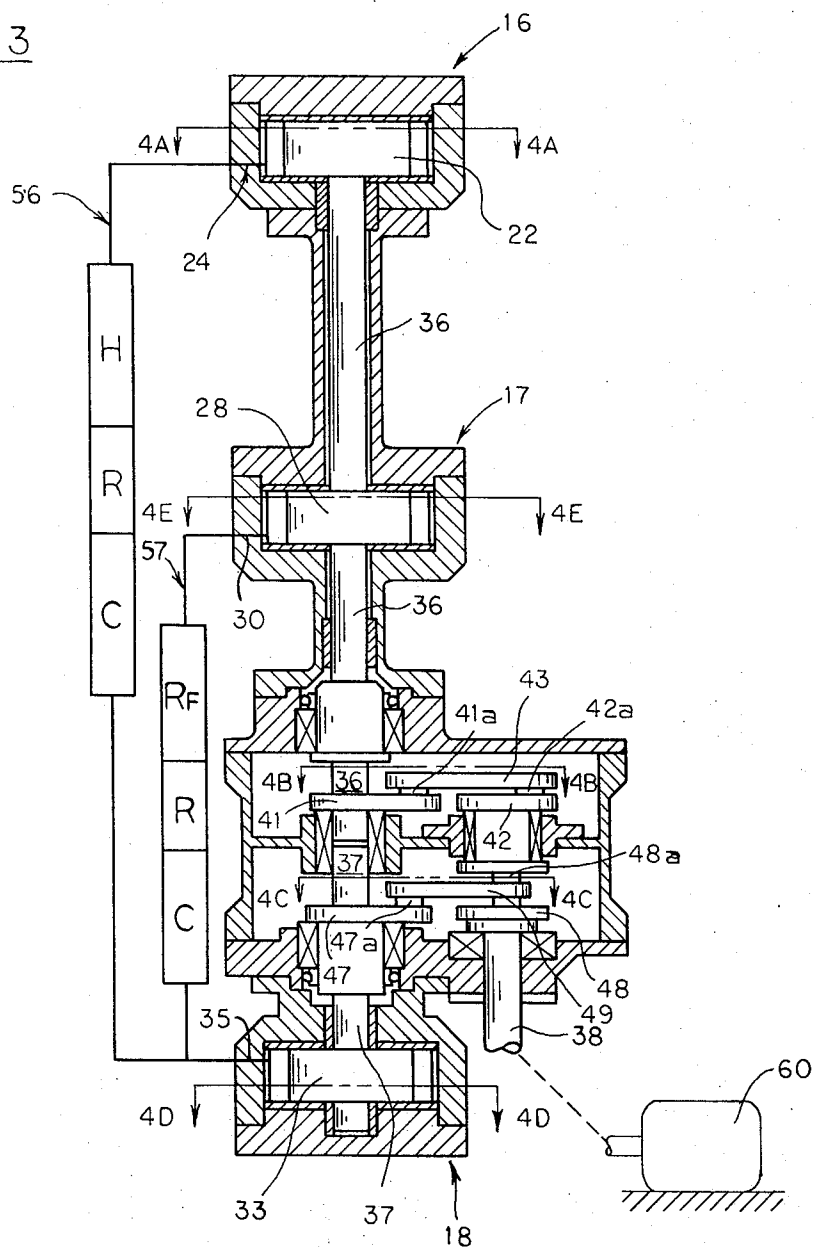


FIG 2D

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FIG 3

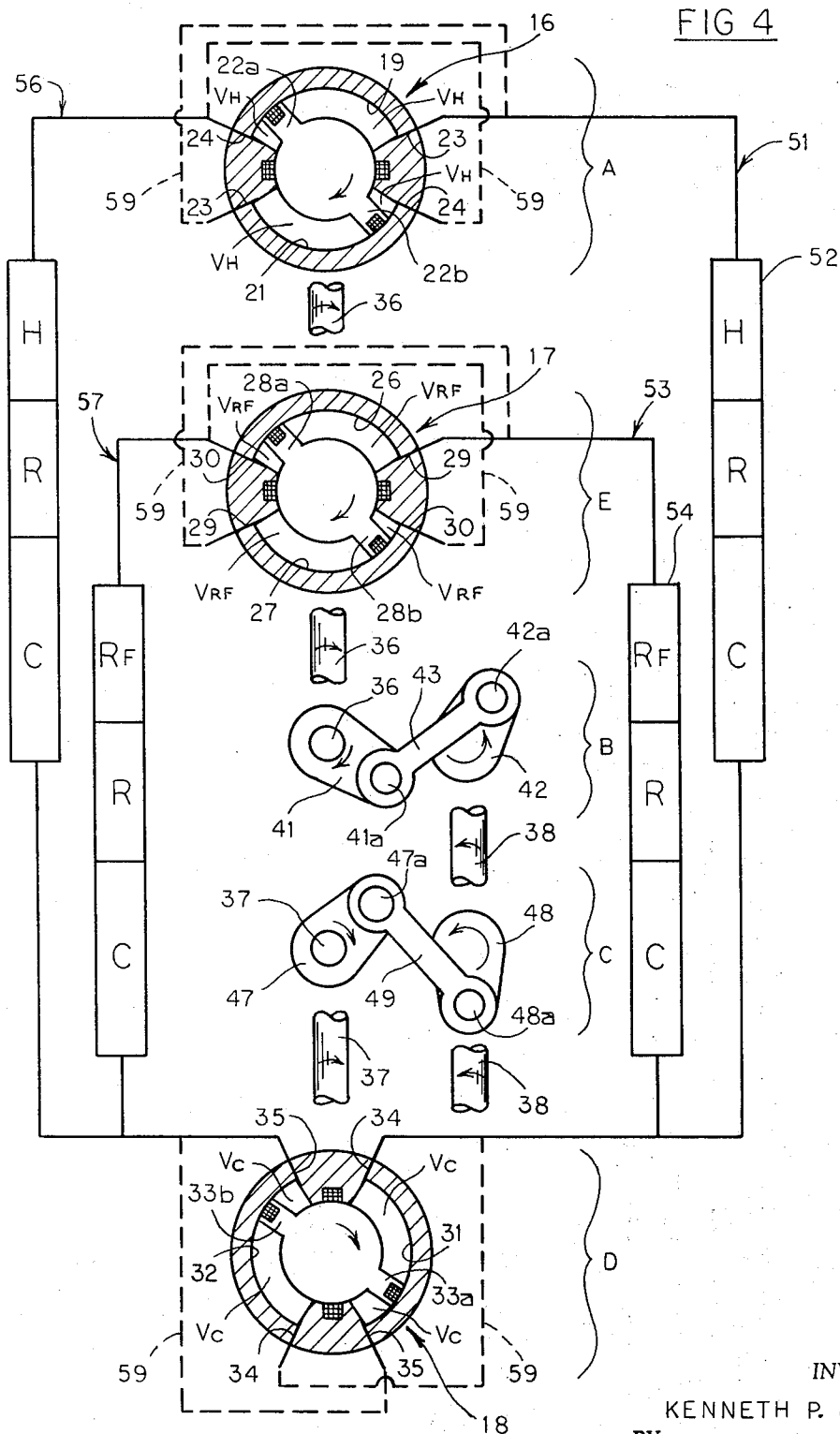


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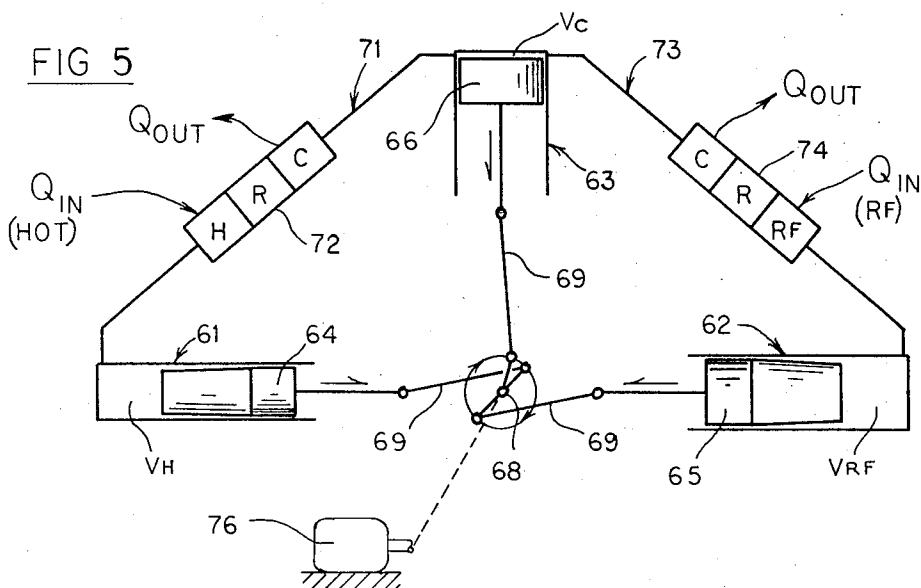
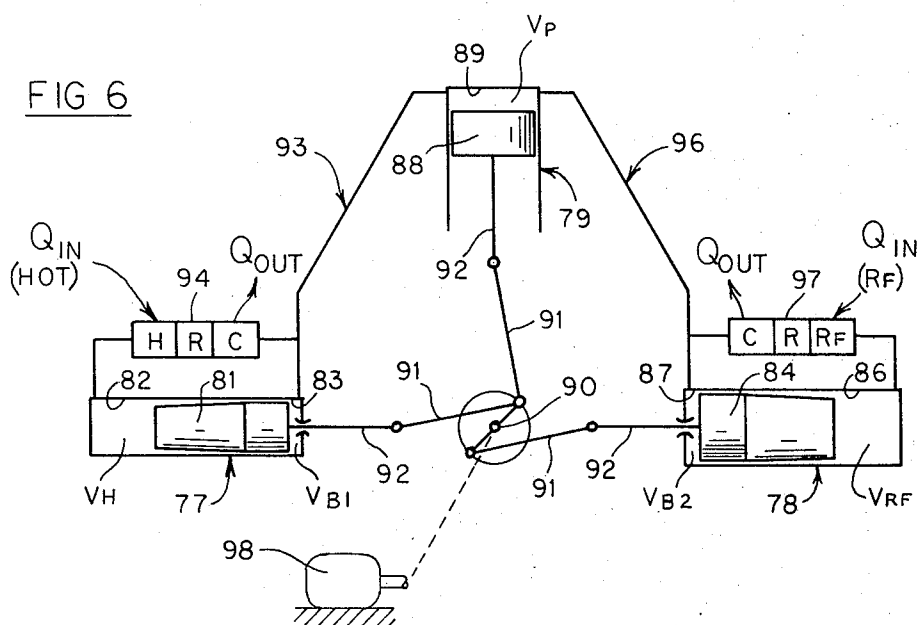
FIG 4



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FIG 7

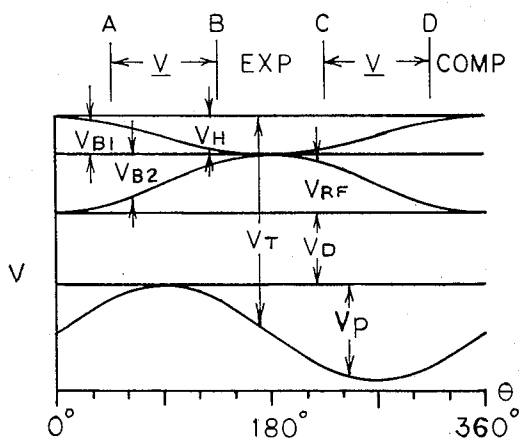


FIG 7A

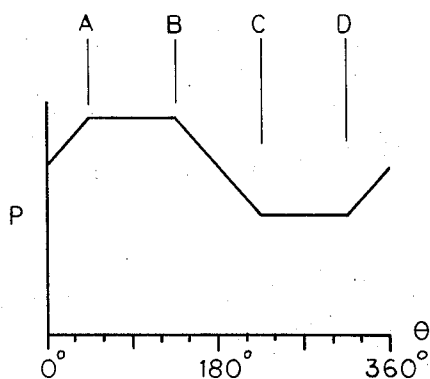
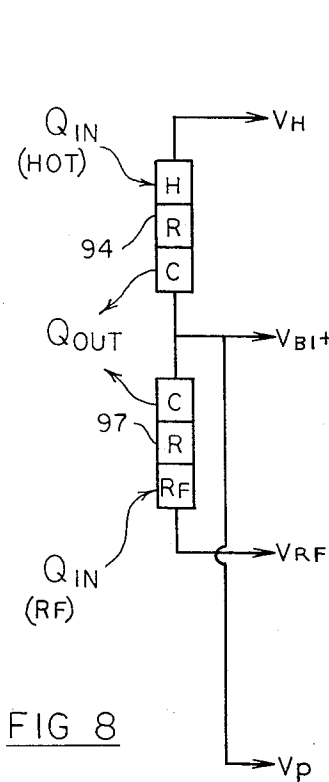


FIG 8

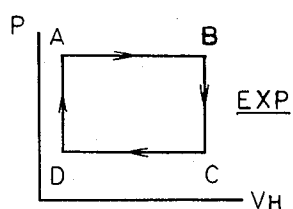


FIG 8A

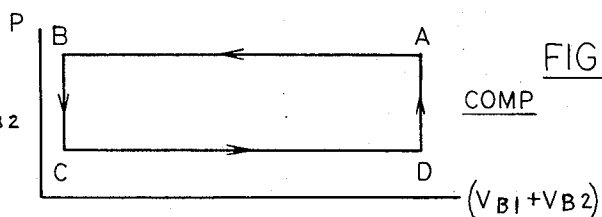


FIG 8B

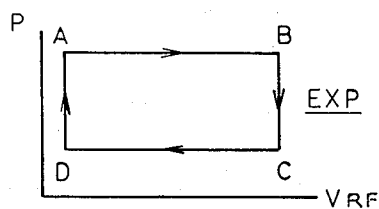


FIG 8C

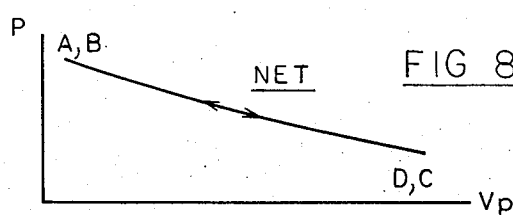


FIG 8D

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FIG 9

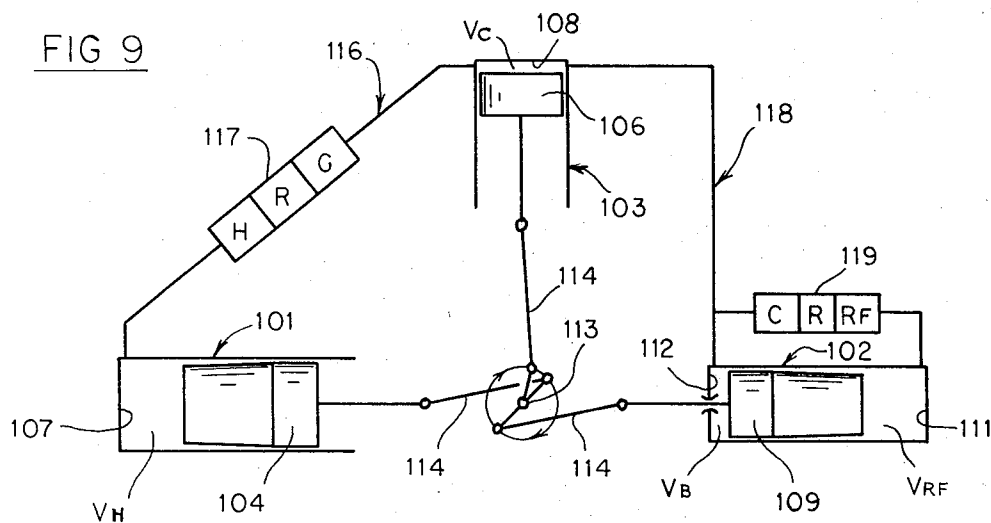
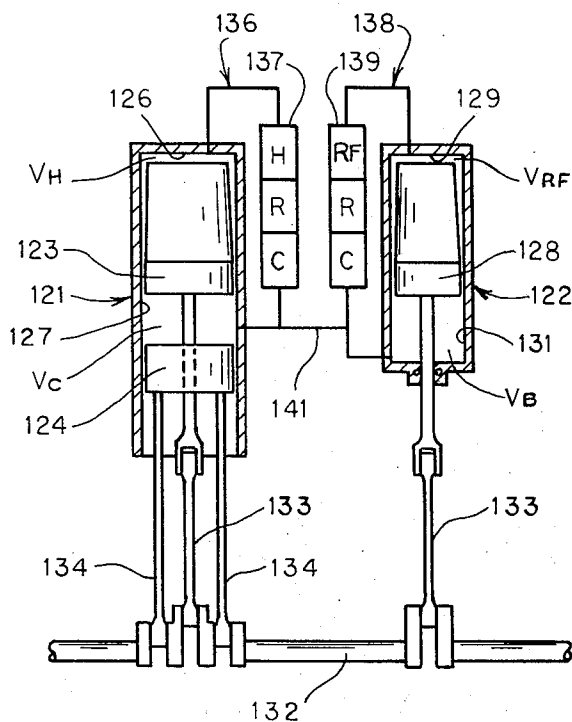


FIG 10



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FIG 11

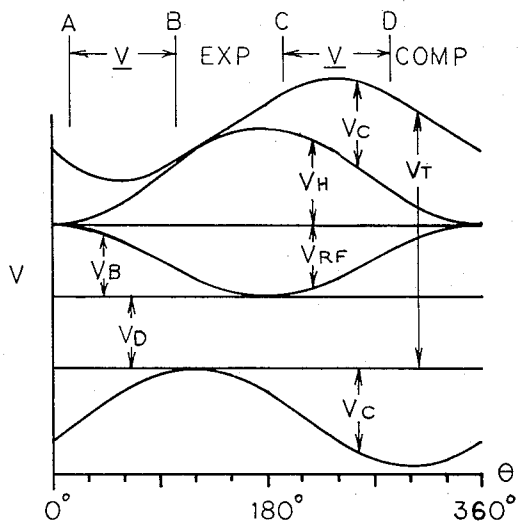


FIG 11A

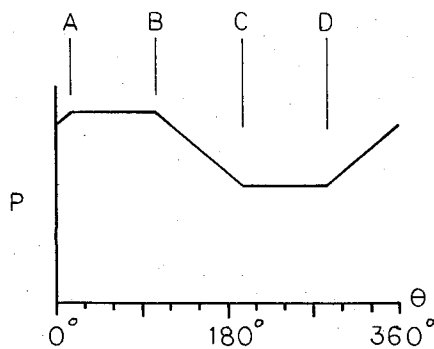
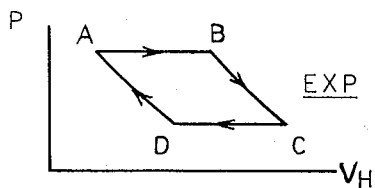


FIG 12A

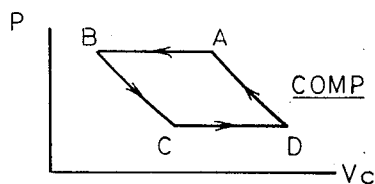


FIG 12B

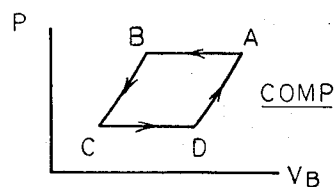


FIG 12C

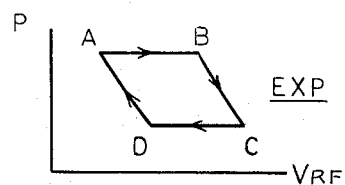


FIG 12D

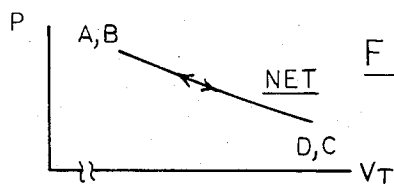


FIG 12E

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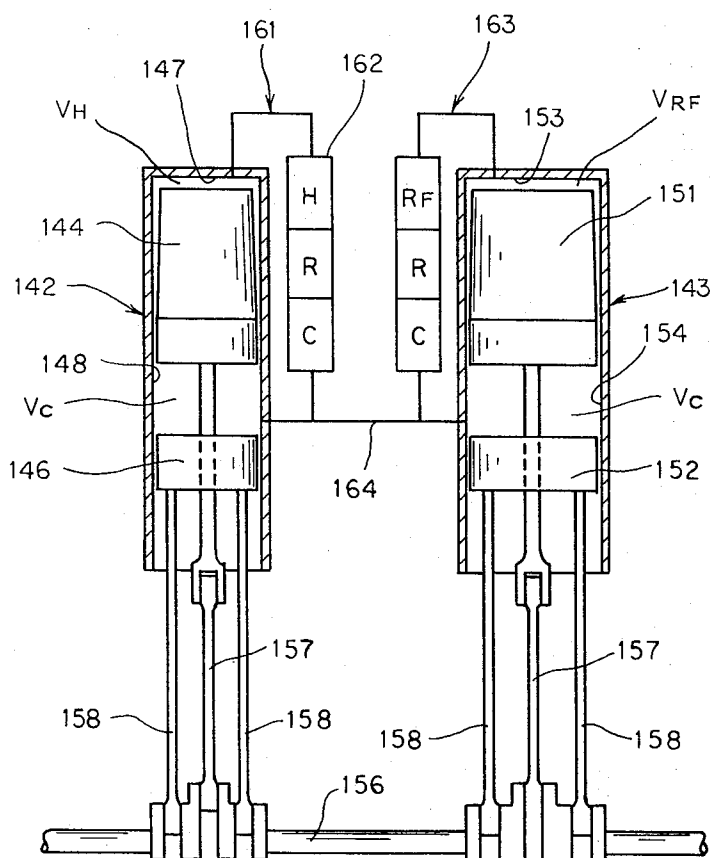


FIG 13

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## THERMAL REFRIGERATION PROCESS AND APPARATUS

### CROSS-REFERENCE TO RELATED APPLICATION

This is a continuation-in-part of copending application Ser. No. 850,439, filed Aug. 15, 1969, now abandoned.

### BACKGROUND OF THE INVENTION

This invention pertains generally to thermodynamic processes and machines and more particularly to a process and apparatus capable of producing either refrigeration or refrigeration and work output from thermal energy. The thermal energy may also be supplemented by work energy to produce refrigeration.

Refrigeration systems heretofore provided have included absorption type refrigeration systems, electric motor driven compressor systems, and certain thermal refrigeration systems.

Absorption type refrigeration systems generally include a refrigeration loop and a heat engine loop in a closed thermodynamic cycle. Limited by the natural limits of the absorption process, the heat engine loop operates over only a narrow temperature range. Heat rejection requirements are high in absorption type systems, with the heat rejected from the engine loop typically being three times greater than that which is rejected from the refrigeration loop. This poor thermodynamic performance leads to high equipment first cost as well as high operating costs.

Conventional electric motor driven compressor systems have inherently high performance, resulting in a minimum of heat rejection components and low first cost. However, because of the high cost of electricity, as compared with the cost of natural gas, the operating costs of electrically motor driven compressor systems are relatively high. While the refrigeration operating temperature range can be extended by cascading compressor stages, the coefficient of performance of the system is decreased significantly by the compressor losses per stage.

In thermal refrigeration systems heretofore provided, such as disclosed in U.S. Pat. No. 2,657,553, coefficient of performance is low because it is necessary to produce work output in order to also produce refrigeration.

### SUMMARY AND OBJECTS OF THE INVENTION

The present invention provides a thermodynamic process and apparatus combining high performance hot gas engine components and gas refrigeration components in a single machine. Either refrigeration alone or refrigeration and work can be produced directly from thermal energy. Also, refrigeration can be produced from thermal energy supplemented by work input. Heat rejection from the engine and refrigeration processes is accomplished through a single heat rejection loop and exchanger which is common to both the engine and refrigerator functions of the machine. The process may be applied over a broad temperature range, i.e., air conditioning through cryogenic applications, with high performance and low cost over the entire range.

It is in general an object of the invention to provide a new and improved thermal refrigeration process and apparatus.

Another object of the invention is to provide a process and apparatus of the above character in which either refrigeration alone or refrigeration and work can be produced directly from thermal energy.

Another object of the invention is to provide a process and apparatus of the above character in which the thermal energy input can be supplemented by work energy input.

Another object of the invention is to provide a process and apparatus of the above character which can be utilized over a broad temperature range.

Another object of the invention is to provide a process and apparatus of the above character which can be economically constructed and operated with high power density.

Additional objects and features of the invention will appear from the following description in which the preferred embodiments are set forth in detail in conjunction with the accompanying drawing.

### BRIEF DESCRIPTION OF THE DRAWING

FIG. 1 is a diagram showing volumetric relationships during one cycle of a thermodynamic process embodying the present invention.

FIG. 1A is a diagram illustrating the relationship between working fluid pressure and crankshaft rotation for the cycle illustrated in FIG. 1.

FIG. 2 is a schematic diagram illustrating the interconnection of the volumes shown in FIG. 1.

FIGS. 2A through 2D are diagrams showing the relationship between pressure and volume during the cycle of operation illustrated in FIG. 1.

FIG. 3 is a sectional view, partly in diagrammatic form, of one embodiment of apparatus incorporating the present invention.

FIG. 4 is a diagrammatic, exploded view with portions indicated as A through E corresponding to section lines 4A through 4E of FIG. 3, respectively.

FIG. 5 shows diagrammatically another embodiment of apparatus incorporating the present invention.

FIG. 6 shows diagrammatically another embodiment of apparatus incorporating the present invention.

FIG. 7 is a diagram showing the volumetric relationships during an operating cycle of the apparatus shown in FIG. 6.

FIG. 7A is a diagram showing the relationship between working fluid pressure and crankshaft rotation for the apparatus of FIG. 6.

FIG. 8 is a schematic diagram illustrating the interconnection of the volumes shown in FIG. 7.

FIGS. 8A through 8D show the relationships between pressure and volume in the apparatus of FIG. 6.

FIG. 9 diagrammatically shows another embodiment of apparatus incorporating the present invention.

FIG. 10 diagrammatically illustrates another embodiment of apparatus incorporating the present invention.

FIG. 11 is a diagram illustrating the volumetric relationships during an operating cycle of the apparatus illustrated in FIGS. 9 and 10.

FIG. 11A is a diagram showing the relationship between working fluid pressure and crankshaft rotation in the apparatus of FIGS. 9 and 10.

FIGS. 12A through 12E illustrate the relationships between pressure and volume in the apparatus of FIGS. 9 and 10.

FIG. 13 shows diagrammatically another embodiment of apparatus incorporating the present invention.

### DESCRIPTION OF THE PREFERRED EMBODIMENTS

The thermal refrigeration process, as illustrated in FIGS. 1 through 2D, comprises generally the steps of compressing a gas working fluid and removing thermal energy therefrom at an intermediate temperature; adding high temperature thermal energy to the working fluid and simultaneously subtracting low temperature thermal energy therefrom while maintaining the volume and pressure of the fluid substantially constant; expanding the working fluid; and adding low temperature thermal energy to and simultaneously subtracting high temperature thermal energy from the working fluid while again maintaining the volume and pressure of the fluid substantially constant. It has been found that these steps can be advantageously performed in a closed system comprising a hot expander chamber of variable volume  $V_H$ , a refrigerator expander chamber of variable volume  $V_{RF}$ , a compressor chamber of variable volume  $V_C$ , a first heat exchanger assembly 11, connected intermediate the hot expander chamber and the compressor chamber, and a second heat exchanger assembly 12 connected intermediate the refrigerator expander chamber and the compressor chamber. The interconnection of the expander and compressor chambers and the heat exchangers is illustrated in FIG. 2. In the preferred embodiments, an ideal gas is used as the working fluid.

The heat exchanger assemblies 11 and 12 are made up of conventional units. The exchanger assembly 11 includes a heater H which is connected to the hot expander volume  $V_H$ , a cooler C connected to the compressor volume  $V_C$ , and a regenerator R connected intermediate the heater and cooler. The exchanger assembly 12 includes a refrigerator heat exchanger  $R_F$  connected to the refrigerator expander volume  $V_{RF}$ , a cooler C connected to the compressor volume  $V_C$ , and a regenerator R intermediate the refrigerator and cooler. The function of the heater H and refrigerator heat exchanger  $R_F$  is to add thermal energy to the working fluid. The coolers C function as a heat sink for removing thermal energy from the working fluid. The regenerators provide transitory storage and release of thermal energy to the working fluid, thus establishing thermal gradients between the heater and cooler and between the refrigerator heat exchanger and cooler.

FIG. 1 illustrates the relationships among the volumes in the system during one complete cycle of operation, corresponding to  $360^\circ$  of crankshaft rotation. The crankshaft rotation  $\theta$  is plotted in degrees on the abscissa, and the volumetric variations are plotted on the ordinate, where

$V_{RF}$  = Refrigerator expander volume

$V_H$  = Hot expander volume

$V_E$  = Total expander volume ( $V_{RF} + V_H$ )

$V_C$  = Compressor volume

$V_D$  = Dead volume (volume of the heat exchangers and flow passageways)

$V_T$  = Total system volume ( $V_{RF} + V_H + V_C + V_D$ )

As illustrated, the hot expander volume  $V_H$  and the refrigerator expander volume  $V_{RF}$  vary substantially harmonically and substantially in phase with each other. Thus, in  $360^\circ$  of crankshaft rotation, they each

vary from a minimum value to a maximum value and back to the minimum value. The difference between the maximum and minimum volumes of a chamber is hereinafter referred to as the working volume of that chamber. The volume  $V_C$  of the compressor chamber also varies substantially harmonically from a minimum value to a maximum value and back to the minimum value in  $360^\circ$  of crankshaft rotation. The variation of the compressor volume  $V_C$  is not, however, in phase with that of the expander chambers. For reasons that will appear hereinafter, it has been found that the best results are obtained when the changes in the expander volumes lead the changes in the compressor volume by an angle on the order of  $110^\circ$  to  $140^\circ$ .

From FIG. 1 it can also be seen that when the total expander working volume  $V_E$  is made approximately equal to the working volume  $V_C$  of the compression chamber, the total volume  $V_T$  is substantially constant during the phases of the cycle between A and B and between C and D. Between B and C the total volume undergoes an expansion phase, and between D and A it undergoes a compression phase.

During the first substantially constant volume phase, that is from A to B, the expander volumes are increasing and the compressor volume is decreasing. Hence, during this phase, working fluid is transferred from the compressor volume to the expander volumes. The fluid passing from the compressor volume to the hot expander volume flows through the first heat exchanger assembly 11 wherein high temperature thermal energy is added to the fluid by the regenerator R and heater H. Also, during this constant volume phase, thermal energy is extracted from the working fluid by the cooler C and second regenerator R as it passes through the second heat exchanger assembly 12. Additional heat is added at low temperature by the refrigerator heat exchanger  $R_F$ . The amount of thermal energy removed from the working fluid in the second heat exchanger assembly is substantially equal to that added to the working fluid in the first heat exchanger assembly, with the result that the pressure of the working fluid remains substantially constant during the first constant volume phase.

During the expansion phase, that is from B to C, the working fluid is expanded substantially isentropically in the expander chambers and in a mixed isothermal and isentropic process in the compressor chamber.

During the second substantially constant volume phase, that is from C to D, working fluid is shifted from the expander chambers to the compressor chamber. As the fluid passes through the second heat exchanger assembly 12, it receives low temperature thermal energy  $Q_{IN}$  from the refrigerator heat exchanger  $R_F$ . Additional thermal energy is added to this fluid by the regenerator R in the heat exchanger assembly 12. The fluid passing from the hot expander chamber to the compressor chamber is cooled initially by the regenerator R in first heat exchanger assembly 11. Additional thermal energy is extracted from the fluid by the cooler in this first exchanger. The thermal energy extracted from the fluid in the first heat exchanger assembly 11 is substantially equal to the thermal energy added in the second exchanger assembly 12, and the pressure of the working fluid remains substantially constant during the second constant volume phase.

During the compression phase, the working fluid is compressed isothermally and isentropically in the expander and compressor chambers.

FIG. 1A shows the pressure changes in the system during the operating cycle. For clarity, the pressure changes shown reflect ideal constant volume shifts. As is more fully discussed hereinafter, the coefficient of performance of the process is significantly enhanced by maintaining the pressure of the working fluid substantially constant during the constant volume phases since the working fluid  $\Delta T$  in each working chamber is thereby minimized.

FIGS. 2A, 2B, and 2C are P-V diagrams which describe the thermodynamic processes occurring in the volumes  $V_H$ ,  $V_C$  and  $V_{RF}$ , respectively. As can be seen in these figures, the working fluid in each of the two expander volumes passes through an expansion cycle, and the working fluid in the compressor volume passes through a compression cycle. Energy is extracted from the working fluid in the hot expansion chamber and resupplied thereto at high temperature as it passes through the heater in the first heat exchanger assembly 11. Energy is extracted from the working fluid in the refrigerator expander volume and is resupplied thereto at a low temperature as it passes through the refrigerator heat exchanger in the heat exchanger assembly 12. Energy is added to the working fluid in the compressor chamber and removed therefrom as it passes through the coolers.

As will be apparent to those familiar with the art, the conditions required for a closed cycle heat engine and a closed cycle refrigerator have been combined into a single process. A thermal machine operating according to this process may produce refrigeration alone, refrigeration and work, or it may be designed to produce refrigeration from thermal energy with a supplemental input of work.

As can be seen in FIGS. 2C and 2D, the expansion cycle of the refrigerator chamber is one quarter cycle out of phase with the net power cycle, and with no pressure change during the constant volume strokes, the process can produce refrigeration without producing net work. Thermal energy input may be supplemented by work energy input to the system. This is to be contrasted with thermal refrigeration systems heretofore provided in which refrigeration is produced in a refrigerator expander volume arranged in phase with the net power cycle of engine components. With the net power cycle and refrigeration volume in phase, the engine cycle must be an expansion cycle and thus produce net work in order to "drive" the refrigerator working fluid in an expansion cycle to produce refrigeration. Further, with the net power cycle in phase with the refrigerator volume, supplemental work input will generate a net engine compression cycle and thus a compression cycle in the refrigerator volume which is the opposite of what is required to produce refrigeration.

The refrigeration produced is equal to the expansion cycle work of the working fluid in the refrigerator expander chamber, and the high temperature heat input to the system is equal to the expansion cycle work of the working fluid in the hot expander chamber. Therefore, the coefficient of performance of a machine operating according to the process can be expressed by the relationship.

C.P. = Refr./ $Q_{IN}$   
Where:

C.P. = Coefficient of performance,

Refr. = Work done by gas in the refrigerator chamber,

$Q_{IN}$  = Work done by gas in the hot expander chamber.

Since the volumes of the expander chambers vary in phase, the working fluid pressure is equal in both chambers for the same percentage of volumetric displacement in each chamber. Therefore, the average  $\Delta P$  in the hot expander chamber is equal to the average  $\Delta P$  in the refrigerator expander chamber. Since the work done in each chamber is equal to the volume of the chamber multiplied by the average  $\Delta P$  in the chamber, it follows that C.P. =  $V_{RF}/V_H$ .

Based on Carnot cycle limits, where no work is produced, the coefficient of performance can be expressed

$$C.P. = (T_{RF}/T_H) [(T_H - T_C)/(T_C - T_{RF})]$$

Where:

$T_{RF}$  = Average absolute temperature of gas in refrigerator expander volume,

$T_H$  = Average absolute temperature of gas in hot expander volume,

$T_C$  = Average absolute temperature of gas in compressor volume.

Therefore, in the ideal machine where there is no friction or net work produced,

$$V_{RF}/V_H = (T_{RF}/T_H) [(T_H - T_C)/(T_C - T_{RF})].$$

In order to produce a constant pressure during the constant volume strokes, as defined in this process, the volumetric ratio ( $V_{RF}/V_H$ ) is dependent on the specific working fluid temperature entering or leaving each chamber. Refrigeration and work are produced when the total expansion net work is greater than the compression work, and the expansion work can be increased relative to the compression work simply by decreasing the ratio  $V_{RF}/V_H$  while holding temperature conditions fixed. Input work is required in addition to thermal energy to produce refrigeration when the expansion work is less than the compression work. With temperature conditions fixed, expansion work can be decreased relative to compression work by increasing the ratio  $V_{RF}/V_H$ . Thus, by adjusting the relative volumes of the expander chambers, the process can be adapted for producing refrigeration alone, refrigeration and work, or for producing refrigeration from thermal energy supplemented by work input. In practical systems, volumetric ratios must be selected which will result in sufficient work to overcome internal friction and leakage losses in the system.

FIG. 2D is a P-V diagram of the process occurring in the total working volume  $V_T$  of an ideal system where expansion work is equal to compression work and no net work is being produced. If the ratio  $V_{RF}/V_H$  is decreased, while holding temperature conditions fixed, the P-V process in FIG. 2D will become an expansion cycle producing work. If the ratio is increased, the process will become a compression cycle, and work input will be required as well as thermal energy to produce refrigeration.

The constant pressure process occurring during the volume changes in each working chamber results in maximum chamber power density combined with a minimum working fluid  $\Delta T$ . This is particularly significant when consideration is given to the temperature relationships which define overall coefficient performance, i.e., C.P. =  $(T_{RF}/T_H) [(T_H - T_C)/(T_C - T_{RF})]$ . Be-

cause the temperature in the foregoing relationship are average chamber working fluid temperatures, the coefficient of performance is significantly enhanced by minimizing the working fluid  $\Delta T$  in each chamber.

This process is applicable at refrigeration temperatures ranging from standard air conditioning temperatures down into the cryogenic range. For the standard air conditioning temperature range of  $T_{\text{Chiller}} = 45^\circ\text{F}$ . and  $T_{\text{Condenser}} = 120^\circ\text{F}$ ., a practical system (utilizing the machine geometry shown in FIG. 6) operating on this process could have a volumetric ratio  $V_{\text{RF}}/V_H = 1.5$  and average chamber working fluid temperatures of  $T_{\text{RF}} = 475^\circ\text{R}$ ,  $T_C = 640^\circ\text{R}$  and  $T_H = 1,595^\circ\text{R}$ . An ideal or process C.P. of 1.72 is indicated and the ideal machine (no friction or thermal losses) will also produce net work. When mechanical and thermal losses are considered it is estimated that the C.P. will be about 1.1. This compares with domestic thermal absorption type air conditioning systems which exhibit a C.P. ranging from 0.3 to 0.4.

This process has been compared, from the standpoint of power density, with the Stirling cycle engine and refrigerator process and machines. The hot expander power density is about equal to the power density in the hot expander of a Stirling engine. The refrigerator expander volume power density in this process however is greater than the Stirling refrigerator expander process by about 85 percent at  $+40^\circ\text{F}$ . and a factor of ten at cryogenic temperatures ( $-320^\circ\text{F}$ .) The increase in refrigerator chamber power density is due to the optimum shape of the P-V diagrams and a more pronounced isentropic expansion, i.e., a constant pressure admission, followed by an isentropic expansion, followed by a constant pressure and then mixed isothermal and isentropic exhaust stroke. This process will thus result in producing more compact machines in this class than have been heretofore possible. The economic advantage afforded by using thermal energy as a source of power will apply over the entire temperature range.

In a practical system embodying the process, the thermal energy for driving the system is supplied to the working fluid through the heater in the heat exchanger assembly 11. The actual heat removal is performed by the refrigerator heat exchanger  $R_F$  in the heat exchanger assembly 12. Although this refrigerator heat exchanger has heretofore been described only as adding low temperature thermal energy to the fluid, it will be appreciated that the extraction of thermal energy from the environment of the refrigerator reduces the temperature of that environment. Heat is removed from the system by the coolers C, both of which can conveniently be connected to a single heat sink.

The thermal refrigeration process described hereinbefore can be practiced in a variety of machine configurations. The necessary volumetric relationships can be provided in both oscillating vane and reciprocating piston machines, and several embodiments of apparatus incorporating the invention are described hereinafter.

FIGS. 3 and 4 illustrate one embodiment of an oscillating vane machine incorporating the present invention. The oscillating vane type of machine is described in detail in by co-pending application Ser. No. 691,054, filed Dec. 15, 1967, now U.S. Pat. No. 3,460,344. Briefly, an oscillating vane machine includes means forming one or more cylinders each containing a pair

of arcuate chambers, an oscillating shaft coaxially disposed within the cylinders, and a pair of diametrically disposed vanes carried by the shaft within the chambers. Oscillation of the shaft causes the vanes to sweep back and forth, forming in each arcuate chamber a pair of alternately enlarging and diminishing volumes adapted to receive and discharge a working fluid. Ports are provided in the chambers on opposite sides of each vane in communication with the enlarging and diminishing volumes. The oscillating shaft is operably connected to a crankshaft which rotates through  $360^\circ$ .

The embodiment illustrated in FIGS. 3 and 4 includes a hot expander cylinder 16, a refrigerator expander cylinder 17, and a compressor cylinder 18. The hot expander cylinder contains a pair of arcuate chambers 19 and 21, each extending through an angle on the order of  $110^\circ$ . An oscillating vane assembly 22, including diametrically disposed vane members 22a and 22b, is coaxially disposed in the cylinder 16. The vane members 22a and 22b divide the chambers 19 and 21 respectively, into pairs of alternately enlarging and diminishing volumes  $V_H$ . Ports 23 and 24 are provided at each end of the chambers 19 and 21, each of said ports being in fluid communication with one of the volumes  $V_H$ .

The refrigerator expander chamber 17 likewise contains a pair of arcuate chambers 26 and 27, each extending through an angle on the order of  $110^\circ$ . An oscillating vane assembly 28, including vane members 28a and 28b, is disposed coaxially in the cylinder. The vane members 28a and 28b divide the chambers 26 and 27, respectively, into pairs of alternately enlarging and diminishing volumes  $V_{\text{RF}}$ . Ports 29 and 30 are provided on opposite sides of the vane members in communication with the volumes  $V_{\text{RF}}$ .

The compressor cylinder 18 also contains a pair of arcuate chambers 31 and 32, each extending through an angle on the order of  $110^\circ$ . An oscillating vane assembly 33, including vane members 33a and 33b, is coaxially disposed in the cylinder. These vane members divide the chambers 31 and 32 into pairs of alternately enlarging and diminishing volumes  $V_C$ . Ports 34 and 35 are provided on opposite sides of the vane members in fluid communication with each of the volumes  $V_C$ .

The expander vane assemblies 22 and 28 are mounted on an upper oscillating shaft 36, and the compressor vane assembly 33 is mounted on a lower oscillating shaft 37. The upper oscillating shaft is operably connected to a crankshaft 38 by means of cranks 41 and 42 and a connecting rod 43. The crank 41 is fastened to the oscillating shaft 36, the crank 42 is secured to the crankshaft 38, and the connecting rod is pivotally mounted on crank pins 41a and 42a carried by the cranks. The lower oscillating shaft 37 is similarly connected to the crankshaft 38 by means of cranks 47 and 48 and a connecting rod 49. The oscillating shafts and crankshaft are mounted in conventional bearings and provided with suitable seals, such as those described in application Ser. No. 691,054, now U.S. Pat. No. 3,460,344.

As illustrated in FIG. 4, the expander vane assemblies 22 and 28 are constrained to move substantially in phase with each other. Hence, the variations of the hot expander volumes  $V_H$  and refrigerator expander volumes  $V_{\text{RF}}$  are substantially in phase with each other. The phase relationship between the volumetric variations of the expander chambers and the compressor chamber is determined by the positions of the cranks

41, 42, 47 and 48. In the preferred embodiment of the oscillating vane apparatus, these cranks are arranged in such manner that the expander volumes lead the compressor volumes by an angle on the order of  $110^\circ$  to  $140^\circ$ , although the machine will operate with phase angles ranging from about  $80^\circ$  to  $160^\circ$ .

First flow passage means 51 is provided for connecting one of the hot expander volumes  $V_H$  in free and open communication with one of the compressor volumes  $V_C$ . This means includes a first heat exchanger assembly 52 comprising a heater H, a regenerator R, and a cooler C. The heater is connected to the port 23 in communication with the volume  $V_H$ , the cooler is connected to the port 34 in communication with the volume  $V_C$ , and the regenerator is connected intermediate the heater and cooler.

Second flow passage means 53 connects a refrigerator expander volume  $V_{RF}$  with the compressor volume  $V_C$ . This means includes a second heat exchanger assembly 54 comprising a refrigerator heat exchanger  $R_F$  and regenerator R, and a cooler C. The refrigerator heat exchanger  $R_F$  is connected to the port 29 in communication with the volume  $V_{RF}$ , the cooler C is connected to the port 34 in communication with the volume  $V_C$ , and the regenerator is connected intermediate the refrigerator and cooler.

The oscillating vane machine is double acting, and additional flow passageways 56 and 57 are shown connected to the ports 24, 30 and 35. These passageways provide fluid communication between the expander and compressor volumes which are formed in the same chambers as the volumes which are interconnected through the passageways 51 and 53. The passageways 56 and 57 include heat exchanger assemblies which are similar in construction and connection to the heat exchanger assemblies 52 and 54.

Both sides of this double acting machine perform the thermal refrigeration process described hereinbefore in connection with FIGS. 1 through 2D. Thus, FIGS. 1 through 2D illustrate the operation of the individual sides of the double acting oscillating machine. Like reference characters have been used for the expander and compressor volumes, and the heat exchanger assemblies 52 and 54 correspond to the heat exchanger assemblies 11 and 12, respectively, in FIG. 2. Further, it should be noted that since the working volumes of the double acting machine are formed by single vane members, the two sides of the machine operate  $180^\circ$  degrees out of phase with each other.

The diametrically opposed working volumes in each of the expander and compressor cylinders operate in phase with each other and are interconnected by passageways, as indicated by dashed lines 59. This interconnection doubles the effective working volume of each of the expander and compressor chambers and further enhances the balance of the machine.

In FIGS. 3 and 4, the hot expander chamber and refrigerator expander chamber have been illustrated adjacent to each other in order to clarify the operation of the machine. However, in an actual machine, these chambers should preferably be isolated from each other in order to minimize heat loss. Thus, the hot expander and refrigerator expander units could be located on opposite ends of a crankcase, with the compressor unit on a separate axis. The thermodynamic cycle is not altered by the location of the cylinders.

If desired, work input can be added to the oscillating vane machine to supplement the thermal energy input. This is conveniently done by connecting a driving motor 60 to the crankshaft 38.

FIG. 5 illustrates one embodiment of a reciprocating piston machine incorporating the present invention. This machine includes means forming a hot expander cylinder 61, a refrigerator expander cylinder 62, and a compressor cylinder 63. Expander pistons 64 and 65 are slidably disposed in the expander cylinders 61 and 62, respectively, for varying the volume  $V_H$  and  $V_{RF}$  of these cylinders. A compressor piston 66 is slidably disposed in the cylinder 63 for varying the volume  $V_C$  of the compressor cylinder. The expander and compressor pistons are operably connected together by linking means comprising a crankshaft 68 and connecting rods 69. The phase relationships between the pistons are thus determined by the angles between the cranks on the crankshaft. In the preferred embodiment, as illustrated, the cranks are arranged so that the expander pistons are substantially in phase with each other and lead the compressor piston by an angle on the order of  $110^\circ$  to  $140^\circ$ . The machine will still operate, however, with phase angles ranging from about  $80^\circ$  to  $160^\circ$ .

Means forming a first flow passage 71 is provided for connecting the hot expander volume  $V_H$  in free and open communication with the compressor volume  $V_C$ . This means includes a first heat exchanger assembly 72 comprising a heater H, a regenerator R, and a cooler C. The heater is connected adjacent to the hot expander chamber, the cooler is connected to the compressor chamber, and the regenerator is intermediate the heater and cooler.

Second flow passage means 73 is provided for connecting the refrigerator expander volume  $V_{RF}$  in free and open communication with the compressor volume  $V_C$ . This means includes a second heat exchanger assembly 74 comprising a refrigerator heat exchanger  $R_F$ , a regenerator R, and a cooler C. The refrigerator heat exchanger is connected adjacent to the refrigerator expander volume, the cooler is connected to the compressor volume, and the regenerator is intermediate the refrigerator heat exchanger and cooler.

High temperature thermal energy  $Q_{IN}$  is supplied to the working fluid through the heater H in the first heat exchanger assembly 72, and low temperature thermal energy is added through the refrigerator heat exchanger  $R_F$ . This machine operates according to the process described hereinbefore in connection with FIGS. 1 through 2D. Accordingly, these figures illustrate the operation of the reciprocating piston machine, with the heat exchanger assemblies 72 and 74 corresponding to the heat exchanger assemblies 11 and 12, respectively, in FIG. 2.

If desired, the thermal energy input to the machine illustrated in FIG. 5 can be supplemented by work input by connecting a driving motor 76 to the crankshaft 68.

FIG. 6 illustrates one embodiment of a reciprocating piston machine with balancing chambers incorporating the present invention. This machine includes means forming cylinders 77, 78 and 79. Pistons are slidably disposed in each of these cylinders. Thus, a displacer piston 81 forms a hot expander chamber 82 and a balancing chamber 83 of variable volumes  $V_H$  and  $V_{B1}$ , respectively, in the cylinder 77. Similarly, a displacer piston 84 forms a refrigerator expander chamber 86 and

a balancing chamber 87 having variable volumes  $V_{RF}$  and  $V_{B2}$ , respectively, in the cylinder 78. The displacer pistons 77 and 78 are provided with elongated head portions because of the thermal differences existing on the opposing sides of these pistons. A power piston 88 forms a power chamber 89 of variable volume  $V_P$  in the cylinder 79.

The displacer and power pistons are operably interconnected by means of a crankshaft 90, connecting rods 91, and piston rods 92. The piston rods associated with the displacer pistons are provided with conventional seals where they pass through the walls of the balancing chambers. The phase relationship among the pistons is controlled by the phase angles between the cranks on the crankshaft. In the preferred embodiment, the cranks are arranged in such manner that the displacer pistons are substantially in phase with each other and lead the power piston 88 by an angle on the order of about 60° to 120°. Thus, the hot expander volume  $V_H$  and refrigerator expander volume  $V_{RF}$  are substantially in phase with each other and lead the power volume  $V_P$  by an angle on the order of 90°. The balancing chamber volumes  $V_{B1}$  and  $V_{B2}$  are 180° out of phase with the expander volumes and act as compressor volumes.

First flow passage means 93 is provided for connecting the volumes  $V_H$ ,  $V_{B1}$ , and  $V_P$  in free and open communication with each other. This means includes a heat exchanger assembly 94 comprising a heater H, a regenerator R, and a cooler C. The heater is connected adjacent to the hot expander volume  $V_H$ , and the cooler is connected to the balancing volume  $V_{B1}$  and to the power volume  $V_P$ . The regenerator is connected intermediate the heater and cooler.

Second flow passage means 96 is provided to connect the volumes  $V_{RF}$ ,  $V_{B2}$ , and  $V_P$  in free and open communication with each other. This means includes a heat exchanger assembly 97, comprising a refrigerator heat exchanger  $R_F$ , a regenerator R, and a cooler C. The refrigerator heat exchanger  $R_F$  is connected to the refrigerator expander volume  $V_{RF}$ , and the cooler is connected to the balancing volume  $V_{B2}$  and to the power volume  $V_P$ .

High temperature thermal energy is supplied to the working fluid in this machine through the heater H in the heat exchanger assembly 94, and low temperature thermal energy is supplied through the refrigerator heat exchanger  $R_F$ .

The relationships among the volumes in the machine during one complete cycle of operation are illustrated in FIG. 7. The total volume  $V_T$  is equal to the sum of the expander, power, balancing, and dead volumes. Because the sum of the expander and balancing volumes is constant, as is the dead volume, the total volume  $V_T$  varies in phase with the power volume  $V_P$ . This volume  $V_T$  passes through substantially constant volume phases between A and B and between C and D. Between B and C the total volume undergoes an expansion phase, and between D and A it undergoes a compression phase.

Since the volumetric variations in the expander chambers are offset by simultaneous variations in the balancing chambers, the constant volume shifts in the total working volume are not dependent upon the relative volumes of the expander and power chambers. Hence, the ratio of power volume to total expander volume is not critical in this machine.

Pressure changes in the working fluid during the constant volume shifts are cancelled by the interaction of

simultaneous heat addition and removal in the heat exchangers associated with the two displacer piston assemblies. Thus, the pressure of the working fluid remains substantially constant during the constant volume shifts. FIG. 7A describes the pressure changes in the machine through a complete cycle, or 360° of crankshaft rotation. For the sake of clarity, the pressure changes shown reflect perfect constant volume shifts.

FIGS. 8A and 8C are P-V diagrams showing the expansion cycles occurring in the hot expander chamber  $V_H$  and the refrigerator expander chamber  $V_{RF}$ , respectively. FIG. 8B shows the compression cycle occurring in the balancing volumes  $V_{B1}$  and  $V_{B2}$ .

FIG. 8D is a P-V diagram of the process occurring in the power volume  $V_P$ . As illustrated, the machine is producing only refrigeration, and FIG. 8D therefore reflects no net work. The machine can be made to produce work as well as refrigeration by increasing the ratio  $V_H/V_{RF}$ , as described hereinbefore, to make the process shown in FIG. 8D an expansion cycle. Likewise, the ratio  $V_H/V_{RF}$  can be decreased to make the process in FIG. 8D a compression cycle, in which case work input will be required as well as thermal energy to produce refrigeration.

If desired, the thermal energy input to the machine illustrated in FIG. 6 can be supplemented with work input by connecting a driving motor 98 to the crankshaft 90.

FIG. 9 shows a reciprocating piston machine having only one displacer piston and one balancing chamber. This machine includes means forming cylinders 101, 102, and 103. Pistons 104 and 106 form a hot expander chamber 107 of variable volume  $V_H$  and a compression chamber 108 of variable volume  $V_C$ , respectively, in the cylinders 101 and 103. A displacer piston 109 is slidably disposed in the cylinder 102 to form a refrigerator expander chamber 111 and a balancing chamber 112 of variable volumes  $V_{RF}$  and  $V_B$ , respectively.

The pistons are operably interconnected by means of a crankshaft 113 and connecting rods 114. As in the previous machines, the phase relationship among the pistons is controlled by the phase angles between the cranks on the crankshaft. In the preferred embodiment, the cranks are arranged in such manner that the pistons 104 and 109 are substantially in phase with each other and lead the piston 106 by an angle on the order of 120° to 140°. Thus, the hot expander volume  $V_H$  and the refrigerator expander volume  $V_{RF}$  are substantially in phase with each other and lead the compressor volume  $V_C$  by an angle on the order of 120° to 140°. Although the preferred phase angle is on the order of 120° to 140°, the machine will still operate with phase angles ranging from about 80° to 160°. The balancing chamber volume  $V_B$  is 180° out of phase with the expander volumes.

First flow passage means 116 is provided for connecting the hot expander volume  $V_H$  in free and open communication with the compressor volume  $V_C$ . This means includes a heat exchanger assembly 117 comprising a heater, a refrigerator, and a cooler. The heater is connected to the hot expander volume  $V_H$ , the cooler is connected to the compressor volume  $V_C$ , and the regenerator is intermediate the heater and the cooler.

Second flow passage means 118 is provided for interconnecting the volumes  $V_{RF}$ ,  $V_B$ , and  $V_C$  in free and open communication with each other. This means in-

cludes a heat exchanger assembly 119 comprising a refrigerator heat exchanger  $R_F$ , regenerator  $R$ , and a cooler  $C$ . The refrigerator heat exchanger  $R_F$  is connected to the refrigerator expander volume  $V_{RF}$ , the cooler is connected to the balancing volume  $V_B$  and to the compressor volume  $V_C$ , and the regenerator is intermediate the refrigerator and cooler.

The operation of this machine is illustrated in FIGS. 11 through 12E. FIG. 11 described the volumetric shifts which occur in the machine during one complete cycle, corresponding to  $360^\circ$  of crankshaft rotation. As with the other machines, it can be seen that the total volume  $V_T$  of this machine undergoes substantially constant volume phases between A and B and between C and D. Likewise, between B and C the total volume undergoes an expansion phase, and between D and A it undergoes a compression phase.

The phase of the working fluid remains substantially constant during the constant volume phases, and the pressure changes during one cycle of operation are shown in FIG. 11A. Again for the sake of clarity, the pressure changes shown reflect perfect constant volume shifts.

FIGS. 12A through 12D are P-V diagrams showing the thermodynamic processes occurring in the volumes  $V_H$ ,  $V_C$ ,  $V_B$ , and  $V_{RF}$ , respectively. The processes occurring in the volumes  $V_H$ ,  $V_C$ , and  $V_{RF}$ , are similar to the processes occurring in the corresponding volumes in the phase machines. The balancing volume  $V_B$  undergoes a compression cycle.

FIG. 12E is a P-V diagram illustrating the thermodynamic process for the entire working volume  $V_T$ . As illustrated, the machine is producing only refrigeration, and FIG. 12E therefore reflects no net work. This machine, like the others, can be made to produce net work as well as refrigeration by increasing the ratio  $V_H/V_{RF}$  to make the process shown in FIG. 12E an expansion cycle. Likewise, the ratio  $V_H/V_{RF}$  can be decreased, as described hereinbefore, to make the process a compression cycle, in which case work input will be required as well as thermal energy to produce refrigeration.

FIG. 10 illustrates another embodiment of a reciprocating piston machine with a balancing chamber incorporating the present invention. A well known arrangement of displacer piston and power piston mounted in a common cylinder is combined with a displacer piston mounted in a second cylinder. A typical linkage is shown to illustrate how the two assemblies may be arranged to operate together.

This machine includes means forming two cylinders 121 and 122. Both a displacer piston 123 and a power piston 124 are slidably mounted in the cylinder 121, forming a hot expander chamber 126 and a compressor chamber 127 of variable volumes  $V_H$  and  $V_C$ , respectively. As is described hereinafter, the pistons 123 and 124 move out of phase with each other, and the compressor chamber 127 is formed between them. A displacer piston 128 is slidably mounted in the cylinder 122 to form a refrigerator expander chamber 129 and a balancing chamber 131 of variable volumes  $V_{RF}$  and  $V_B$ , respectively.

This displacer and power pistons are operably interconnected by means of a crankshaft 132 and connecting rods 133 and 134. As in the other machines, the phase relationship among the pistons is controlled by the angles between the cranks on the crankshaft. In the

preferred embodiment, the cranks are arranged so that the displacer piston cranks are substantially in phase with each other and lead the power piston crank by an angle on the order of  $60^\circ$  to  $100^\circ$ . Thus, the expander chambers  $V_H$  and  $V_{RF}$  are substantially in phase with each other and lead the compressor chamber volume  $V_C$  by an angle on the order of  $120^\circ$  to  $150^\circ$ . The machine will still operate, however, if the expander chambers lead the compressor chamber by an angle ranging from about  $80^\circ$  to  $180^\circ$ . The balancing chamber volume  $V_B$  is  $180^\circ$  out of phase with the expander volumes.

First flow passage means 136 is provided for connecting the hot expander  $V_H$  in free and open communication with the compressor volume  $V_C$ . This means includes a heat exchanger assembly 137 comprising a heater  $H$ , a regenerator  $R$ , and a cooler  $C$ . The heater is connected to the hot expander chamber, the cooler is connected to the compressor chamber, and the regenerator is intermediate the heater and the cooler.

Second flow passage means 138 is provided for connecting the refrigerator expander volume  $V_{RF}$  in free and open communication with the balancing volume  $V_B$ . This means includes a heat exchanger assembly 139 comprising a refrigerator heat exchanger  $R_F$ , a regenerator, and a cooler. The refrigerator heat exchanger  $R_F$  is connected to the refrigerator expander volume, the cooler is connected to the balancing volume, and the regenerator is intermediate the refrigerator heat exchanger and cooler.

Additional flow passage means 141 is provided for connecting the compressor volume  $V_C$  in free and open communication with the balancing volume  $V_B$ .

The operation of this machine is substantially the same as that of the machine shown in FIG. 9 and is, therefore, illustrated in FIGS. 11 through 12E.

FIG. 13 shows another embodiment of a two cylinder reciprocating piston machine incorporating the present invention. This machine includes means forming cylinders 142 and 143. A displacer piston and a power piston are slidably mounted in each of these cylinders. Thus, displacer piston 144 and a power piston 146 forms a hot expander chamber 147 and a compressor chamber 148 of variable volumes  $V_H$  and  $V_C$ , respectively, in the cylinder 142. Likewise, a displacer piston 151 and a power piston 152 form a refrigerator expander chamber 153 and a compressor chamber 154 of variable volumes  $V_{RF}$  and  $V_C$ , respectively, in the cylinder 143.

The pistons are operably interconnected by means of a crankshaft 156 and connecting rods 157 and 158. The phase relationship among the pistons is determined by the angles between the cranks on the crankshaft. In the preferred embodiment, the cranks are arranged in such manner that the displacer pistons 144 and 151 are substantially in phase with each other and lead the power pistons 146 and 152 by a crank angle on the order of  $45^\circ$  to  $100^\circ$ . Thus, the hot expander volume  $V_H$  and refrigerator expander volume  $V_{RF}$  vary substantially in phase with each other and lead the compressor volumes  $V_C$  by an angle on the order of  $90^\circ$  to  $150^\circ$ . The machine will still operate, however, with phase angles between the expander and compressor volumes ranging from about  $80^\circ$  to  $180^\circ$ .

First flow passage means 161 is provided for connecting the hot expander volume  $V_H$  in free and open communication with the volume  $V_C$  of the compressor chamber 148. This means includes heat exchanger as-

sembly 162 comprising a heater H, a regenerator R, and a cooler C. The heater is connected to the hot expander volume  $V_H$ , the cooler is connected to the compressor volume  $V_C$ , and the regenerator is intermediate the heater and the cooler.

Second flow passage means 163 is provided for connecting the refrigerator expander volume  $V_{RF}$  in free and open communication with the volume  $V_C$  of the compressor chamber 154. This means includes a refrigerator heat exchanger  $R_F$ , a regenerator R, and a cooler C. The refrigerator heat exchanger  $R_F$  is connected to the refrigerator expander volume  $V_{RF}$ , the cooler is connected to the compressor volume  $V_C$ , and the regenerator is intermediate the refrigerator and cooler.

Additional flow passage means 164 is provided for connecting the volumes of the compressor chambers 148 and 154 in free and open communication with each other.

The operation of this machine is similar to the operation of the machines shown in FIGS. 3 through 5, and reference is made to FIGS. 1 through 2D and the associated disclosure for a description of this operation.

It is apparent from the foregoing that a new and improved thermal refrigeration process and apparatus has been provided which can produce either refrigeration or refrigeration and work directly from thermal energy. In addition, refrigeration can be produced from thermal energy supplemented by work input. The process can be applied over a broad temperature range, including air conditioning and cryogenic applications, with high performance and low cost over the entire range. While only the presently preferred embodiments of the process and apparatus have been described hereinbefore, as will be apparent to those familiar with the art, certain changes and modifications can be made without departing from the scope of the invention as defined by the following claims.

I claim:

1. In a process for producing refrigeration from thermal energy, the steps of confining a gas working fluid in free and open communication throughout a closed system comprising a high temperature hot expander chamber of variable volume, a low temperature refrigerator expander chamber of variable volume, a compressor chamber of variable volume, said compressor chamber operating at a temperature intermediate that of said hot expander and refrigerator expander chambers, a first heat exchanger assembly connected intermediate said hot expander chamber and said compressor chamber, and a second heat exchanger assembly connected intermediate said refrigerator expander chamber and said compressor chamber, said first and second said heat exchanger assemblies each including means for adding thermal energy to the working fluid, storing and releasing thermal energy, and removing thermal energy from the working fluid; varying the volumes of said hot and refrigerator expander chambers substantially harmonically and substantially in phase with each other from their minimum values to their maximum values and back to their minimum values in an expansion cycle of  $360^\circ$ ; varying the volume of said compressor chamber substantially harmonically from its minimum value to its maximum value and back to its minimum value in a compression cycle of  $360^\circ$ , with said compression cycle lagging said expansion cycle by an angle such that the net volume of said expander and compressor chambers undergoes a cycle comprising a

first substantially constant volume phase in which the working fluid is transferred from said compressor chamber through the heat exchangers to said expander chambers, an expansion phase, a second substantially constant volume phase in which the working fluid is transferred from said expander chambers through the heat exchangers to said compressor chamber, and a compression phase; adding high temperature thermal energy to the working fluid in said first heat exchanger assembly and removing thermal energy from the working fluid in said second heat exchanger assembly during said first substantially constant volume phase, thereby maintaining the pressure of said working fluid substantially constant during said first substantially constant volume phase; adding low temperature thermal energy to the working fluid in said second heat exchanger assembly and removing thermal energy from the working fluid in said first heat exchanger assembly during said second substantially constant volume phase, thereby maintaining the pressure of said working fluid substantially constant during said second substantially constant volume phase.

2. A thermal refrigeration process as in claim 1 wherein the relative working volumes of the expander chambers and the quantities of thermal energy added to and subtracted from the working fluid during the substantially constant volume phases are such that the work done in the expander chambers is substantially equal to the sum of the work done in the compressor chamber and the work required to overcome friction and leakage in the system; whereby said process is adapted for producing refrigeration alone.

3. A thermal refrigeration process as in claim 1 wherein the relative working volumes of the expander chambers and the quantities of thermal energy added to and subtracted from the working fluid during the substantially constant volume phases are such that the work done in the expander chambers is greater than the sum of the work done in the compressor chamber and the work required to overcome friction and leakage in the system, whereby said process is adapted for producing both refrigeration and work output, the work output cycle being out of phase with the refrigeration cycle.

4. A thermal refrigeration process as in claim 1 together with the additional step of adding work input to said system to supplement the thermal energy added in said heat exchangers in producing refrigeration.

5. A thermal refrigeration process as in claim 4 wherein said work input is provided by mechanically varying the volume of said expander and compressor chambers.

6. In a process for producing refrigeration from thermal energy, the steps of compressing a working fluid in at least one hot expander chamber, at least one refrigerator expander chamber and at least one compressor chamber, said compressor chamber operating at a temperature intermediate the temperatures of the expander chambers, transferring the working fluid from said compressor chamber to said expander chambers simultaneously in a first substantially constant volume and pressure stroke; expanding the working fluid in said expander and compressor chambers; transferring the working fluid simultaneously from said expander chambers to said compressor chamber in a second substantially constant volume and pressure stroke; adding high temperature thermal energy to the working fluid pass-

ing from said compressor chamber to said hot expander chamber and removing low temperature thermal energy from the working fluid passing from said compressor chamber to said refrigerator expander chamber during the first constant volume stroke; and adding low temperature thermal energy to the working fluid passing from said refrigerator expander chamber to said compressor chamber and removing high temperature thermal energy from the working fluid passing from said hot expander chamber to said compressor chamber during the second constant volume stroke.

7. In thermal refrigeration process as in claim 6 wherein the relative volumes of the expander chambers and the quantities of thermal energy added and subtracted are such that the work done in the expander chambers is substantially equal to the sum of the work done in the compressor chamber and the work required to overcome friction and leakage, whereby said process is adapted for producing refrigeration along with no work output.

8. A thermal refrigeration process as in claim 6 wherein the relative volumes of the expander chambers and the quantities of thermal energy added and subtracted are such that the work done in the expander chambers is greater than the sum of the work done in the compressor chamber and the work required to overcome friction and leakage whereby said process is adapted for producing refrigeration plus work output, the work output cycle being out of phase with the refrigeration cycle.

9. A thermal refrigeration process as in claim 6 together with the additional step of adding work input to supplement the thermal energy in producing refrigeration.

10. In apparatus for producing refrigeration from thermal energy, first means defining a hot expander chamber; second means defining a refrigerator expander chamber; third means defining a compressor chamber; the temperature of said compressor chamber being intermediate the temperatures of the expander chambers; first heat exchanger means connected to said first and third means in such manner that said hot expander chamber is in communication with said compressor chamber through said first heat exchanger means; second heat exchanger means connected to said second and third means in such manner that said refrigerator expander chamber is in communication with said compressor chamber through said second heat exchanger means; said chambers and heat exchanger means forming a closed system adapted for containing a gas working fluid; movable means carried by each of said first, second and third means for varying the volumes of said expander and compressor chambers substantially harmonically between maximum and minimum values; linking means connecting said movable means in such manner that said expander volumes vary substantially in phase with each other and lead said compressor volume by an angle such that the net volume of said chambers passes through a cycle comprising a first substantially constant volume phase in which the working fluid is transferred from said compressor chamber through said first and second heat exchanger means to said expander chambers, an expansion phase in which the working fluid is expanded in said expander and compressor chambers, a second substantially constant volume phase in which the working fluid is transferred from said expander chambers through said first and

second heat exchanger means to said compressor chamber, and a compression phase in which the working fluid is compressed in said expander and compressor chambers; and means forming a part of said heat exchanger means for adding thermal energy to and extracting thermal energy from the working fluid during said first and second constant volume phases.

11. Thermal refrigeration apparatus as in claim 10 wherein said first heat exchanger means includes means for adding high temperature thermal energy to the working fluid during said first constant volume phase and means for extracting thermal energy from the working fluid during said second constant volume phase; and wherein said second heat exchanger means includes means for extracting thermal energy from the working fluid during said first constant volume phase and means for adding low temperature energy to the working fluid during said second constant volume phase.

12. A thermal refrigeration apparatus as in claim 10 wherein the relative working volumes of the expander chambers and the quantities of thermal energy added and extracted during the substantially constant volume phases are such that the work done in the expander chambers is substantially equal to the sum of the work done in the compressor chamber and the work required to overcome friction and leakage in the apparatus, whereby said apparatus is adapted for producing refrigeration alone.

13. Thermal refrigeration apparatus as in claim 10 wherein the relative working volumes of the expander chambers and the quantities of thermal energy added and extracted during the substantially constant volume phases are such that the work done in the expander chambers is greater than the sum of the work done in the compressor chamber and the work required to overcome friction and leakage in the apparatus, whereby said apparatus is adapted for producing both refrigeration and work output, the work output cycle being out of phase with the refrigeration cycle.

14. Thermal refrigeration apparatus as in claim 10 together with driving means connected to said movable means for supplementing the thermal energy input to said apparatus with work energy input in producing refrigeration.

15. In thermal refrigeration apparatus, first means forming an arcuate hot expander chamber; second means forming an arcuate refrigerator expander chamber; third means forming an arcuate compressor chamber; oscillating vane means in each of said chambers movable between advanced and retracted positions to form in each chamber a pair of alternately enlarging and diminishing volumes each adapted to receive and discharge a working fluid; linking means connected to said oscillating vane means in such manner that the volumes of said expander chamber vary substantially in phase with each other and lead the volume of said compressor chamber, said linking means including an oscillating shaft connected to said oscillating vane means, a crankshaft, and connecting rod means connecting said oscillating shaft to said crankshaft; first flow passage means interconnecting one of the volumes in said hot expander chamber with one of the volumes in said compressor chamber; second flow passage means interconnecting one of the volumes in said refrigerator expander chamber with said one volume in said compressor chamber; and first and second heat exchanger

means disposed in said first and second flow passage means, respectively.

16. Thermal refrigeration apparatus as in claim 15 wherein said first heat exchanger means includes a heater adjoining said hot expander chamber, a cooler adjoining said compressor chamber, and a regenerator intermediate said heater and refrigerator; and said second heat exchanger means includes a refrigerator heat exchanger adjoining said refrigerator expander chamber, a cooler adjoining said compressor chamber, and a regenerator intermediate said refrigerator and cooler.

17. Thermal refrigeration apparatus as in claim 15 wherein the sum of the working volumes in said expander chambers is substantially equal to the working volume of said compressor chamber whereby during each 360° of crankshaft rotation the net volume of the interconnected expander and compressor volumes undergoes a first substantially constant volume phase, an expansion phase, a second substantially constant volume phase, and a compression phase.

18. Thermal refrigeration apparatus as in claim 17 wherein said first and second heat exchanger means are adapted for both adding thermal energy to and extracting thermal energy from the working fluid during each of the constant volume phases to maintain the pressure of the working fluid substantially constant during said phases.

19. Thermal refrigeration apparatus as in claim 18 wherein the relative volumes of the interconnected expander chambers and the quantities of thermal energy added and extracted in the heat exchanger means are such that the work done in the expander chambers is substantially equal to the sum of the work done in the compressor chamber and the work required to overcome friction and leakage in the apparatus, whereby said apparatus is adapted for producing refrigeration alone.

20. Thermal refrigeration apparatus as in claim 18 wherein the relative volumes of the interconnected expander chambers and the quantities of thermal energy added and extracted in the heat exchanger means are such that the work done in the expander chambers is greater than the sum of the work done in the compressor chamber and the work required to overcome friction and leakage in the apparatus, whereby said apparatus is adapted for producing refrigeration and work output, the work output cycle being out of phase with the refrigeration cycle.

21. Thermal refrigeration apparatus as in claim 18 together with driving means connected to said crankshaft for supplying work input to supplement the thermal energy in producing refrigeration.

22. Thermal refrigeration apparatus as in claim 15 together with means forming second arcuate expander and compressor chambers diametrically opposite the first named expander and compressor chambers in said first, second, and third means; said oscillating vane means being adapted for forming in each of said second chambers an additional pair of alternately enlarging and diminishing volumes adapted to receive and discharge the working fluid, and means connecting one of the volumes of each additional pair of fluid communication with the first named volume which is diametrically opposite and enlarging and diminishing in phase with the additional volume.

23. In apparatus for producing refrigeration from thermal energy, means forming a hot expander chamber, means forming a refrigerator expander chamber; means forming a compressor chamber; said compressor chamber operating at a temperature intermediate the temperature of the expander chambers; piston means in each of said chambers movable between advanced and retracted positions for enlarging and diminishing the volumes of said chambers, thereby adapting said chambers for receiving and discharging a working fluid; linking means including a crankshaft and connecting rods interconnecting said piston means in such manner that the volumes of said expander chambers vary substantially in phase with each other and lead the volume of said compressor chamber; first flow passage means interconnecting said hot expander chamber and said compressor chamber; second flow passage means interconnecting said refrigerator expander chamber and said compressor chamber; and first and second heat exchanger means disposed in said first and second flow passage means, respectively.

24. Thermal refrigeration apparatus as in claim 23 wherein said first heat exchanger means includes a heater adjoining said hot expander chamber, a cooler adjoining said compressor chamber, and a regenerator intermediate said heater and cooler and said second heat exchanger means includes a refrigerator heat exchanger adjoining said refrigerator expander chamber, a cooler adjoining said compressor chamber, and a regenerator intermediate said refrigerator and cooler.

25. Thermal refrigeration apparatus as in claim 23 wherein the sum of the working volumes in said expander chambers is substantially equal to the working volume of said compressor chamber, whereby during each 360° of crankshaft rotation the net volume of the interconnected expander and compressor chambers undergoes a first substantially constant volume phase, an expansion phase, a second substantially constant volume phase, and a compression phase.

26. Thermal refrigeration apparatus as in claim 25 wherein said first and second heat exchanger means are adapted for both adding thermal energy to and extracting thermal energy from the working fluid during each of the constant volume phase to maintain the pressure of the working fluid substantially constant during said phases.

27. Thermal refrigeration apparatus as in claim 26 wherein the relative volumes of the expander chambers and the quantities of thermal energy added and extracted in the heat exchanger means are such that the work done in the expander chambers is substantially equal to the sum of the work done in the compressor chamber and the work required to overcome friction and leakage in the apparatus, whereby said apparatus is adapted for producing refrigeration alone.

28. Thermal refrigeration apparatus as in claim 26 wherein the relative volume of the expander chambers and the quantities of thermal energy added and extracted in the heat exchanger means are such that the work done in the expander chambers is greater than the sum of the work done in the compressor chamber and the work required to overcome friction and leakage in the apparatus, whereby said apparatus is adapted for producing refrigeration and work output, the work output cycle being out of phase with the refrigeration cycle.

29. Thermal refrigeration apparatus as in claim 26 together with driving means connected to said crankshaft for supplying work input to supplement the thermal energy in producing refrigeration.

30. Thermal refrigeration apparatus as in claim 23 wherein said compressor chamber is formed in two sections, each of said first and second flow passageways being connected to a different one of said sections, each of said sections having piston means movable between advanced and retracted positions for enlarging and diminishing the volumes of said sections, said linking means connecting the pistons in said sections in such manner that the volumes of said sections vary substantially in phase with each other, together with flow passage means connecting the compressor chamber sections in free and open communication with each other.

31. Thermal refrigeration apparatus as in claim 30 wherein the total working volume of said compressor chamber is substantially equal to the total working volume of said expander chambers.

32. Thermal refrigeration apparatus as in claim 23 together with means forming a balancing chamber; piston means in said balancing chamber movable between advanced and retracted positions for enlarging and diminishing the volume of the chamber; the piston means in the balancing chamber and the refrigerator expander chamber being arranged in such manner that the volume of said balancing chamber varies substantially 180° out of phase with the volume of said refrigerator expander chamber; and flow passage means connecting said balancing chamber in free and open communication with said second flow passage means intermediate said compressor chamber and said second heat exchanger means.

33. Thermal refrigerator apparatus as in claim 32 together with means forming a second balancing chamber; piston means movably disposed in said chamber for varying the volume of said balancing chamber substantially 180° out of phase with the volume of said hot expander chamber; and flow passage means connecting said second balancing chamber in free and open communication with said first flow passage means intermediate said compressor chamber and said first heat exchanger means.

34. In apparatus for producing refrigeration from thermal energy, refrigeration and work from thermal energy, or refrigeration from thermal energy supplemented by input work energy, means forming first, second and third cylinders, piston means in said first and second cylinders movable between advanced and retracted positions to form, respectively, a hot expander chamber and a compressor chamber having substantially equal working volumes adapted for receiving and discharging a working fluid; displacer piston means in said third cylinder movable between advanced and retracted positions to form in said cylinder a refrigerator expander chamber and a balancing chamber of alternately expanding and diminishing volumes adapted for receiving and discharging a working fluid; said compressor chamber and balancing chamber operating at a temperature intermediate the temperatures of said expander chambers; linking means connecting said piston means and displacer piston means in such manner that the volumes of said expander chambers vary substantially in phase with each other and lead the

volume of said compressor chamber; first flow passage means interconnecting said hot expander chamber and said compressor chamber; second flow passage means interconnecting said refrigerator expander chamber, said balancing chamber and said compressor chamber, first heat exchanger means disposed in said first flow passage means; and second heat exchanger means disposed in said second flow passage means, one end of said heat exchanger means being connected to said refrigerator expander chamber and the other end being connected to said balancing chamber and to said compressor chamber.

35. In apparatus for producing refrigeration from thermal energy, refrigeration and work thermal energy, or refrigeration from thermal energy supplemented by work energy, means forming first, second, and third cylinders; displacer piston means in said first cylinder movable between said advanced and retracted positions to form in said cylinder a hot expander chamber and a first balancing compressor chamber of alternately expanding and diminishing volumes adapted for receiving and discharging a working fluid at high and intermediate temperatures, respectively; piston means in said second cylinder movable between advanced and retracted positions to form a power chamber of expanding and diminishing volume adapted for receiving and discharging a working fluid; displacer piston means in said third cylinder movable between advanced and retracted positions to form in said cylinder a refrigerator expander chamber and a second balancing compressor chamber of alternately expanding and diminishing volumes adapted for receiving and discharging a working fluid at a low temperature and at said intermediate temperature, respectively; linking means connecting said piston means and displacer piston means in such manner that the volumes of said expander chambers vary substantially in phase with each other and lead the volume of said power chamber by an angle on the order of 90 degrees; first flow passage means interconnecting said hot expander chamber, said first balancing compressor chamber, and said power chamber; second flow passage means interconnecting said refrigerator expander chamber, said second balancing compressor chamber, and said power chamber; heat exchanger means disposed in each of said first and second flow passage means, one end of each heat exchanger means being connected to the expander chamber and the other end being connected to the balancing compressor chamber.

36. In apparatus for producing refrigeration from thermal energy, refrigeration and work from thermal energy or refrigeration from thermal energy supplemented by work energy, means forming first and second cylinders; displacer and power piston means in said first cylinder movable between advanced and retracted positions to form a hot expander chamber and a compressor chamber of expanding and diminishing volumes adapted to receive and discharge a working fluid at high and intermediate temperatures; displacer piston means in said second cylinder movable between advanced and retracted positions to form a refrigerator expander chamber and a balancing chamber of alternately expanding and contracting volumes adapted to receive and discharge a working fluid at a low temperature and at said intermediate temperature, respectively; linking means connecting said displacer and power piston means in such manner that the volumes of said ex-

pander chambers vary substantially in phase with each other and lead the volume of said compressor chamber first flow passage means interconnecting said hot expander chamber and said compressor chamber; first heat exchanger means disposed in said first heat flow passage means; second flow passage means interconnecting said refrigerator expander chamber, said balancing volume, and said compressor chamber; and second heat exchanger means disposed in said second flow passage means, one end of said heat exchanger means being connected to said refrigerator expander chamber and the other end being connected to said balancing chamber and to said compressor chamber.

37. In apparatus for producing refrigeration from thermal energy, refrigeration and work from thermal energy, or refrigeration from thermal energy supplemented by work energy, means forming first and second cylinders; displacer and power piston means in each of said cylinders movable between advanced and retracted positions to form in said first cylinder a hot expander chamber and a first compressor chamber and in said second cylinder a refrigerator expander chamber and a second compressor chamber of expanding and diminishing volumes adapted to receive and discharge a working fluid at three temperature levels; linking means connecting said displacer and power piston means in such manner that the volumes of said expander chambers vary substantially in phase with each other and lead the volumes of said compressor chambers by an angle on the order of  $90^\circ$  to  $120^\circ$ , the volumes of said compressor chambers varying substantially in phase with each other; first flow passage means interconnecting said hot expander chamber and said

first compressor chamber; second flow passage means interconnecting said refrigerator expander chamber and said second compressor chamber; heat exchanger means disposed in each of said first and second flow passage means; and additional flow passage means interconnecting said first and second compressor chambers.

38. In apparatus for producing refrigeration from thermal energy, refrigeration and work from thermal energy, means forming first and second cylinders; displacer and power piston means in each of said cylinders movable between advanced and retracted positions to form in said first cylinder a hot expander chamber and a first compressor chamber and in said second cylinder a refrigerator expander chamber and a second compressor chamber of expanding and diminishing volumes adapted to receive and discharge a working fluid at three temperature levels; said displacer and power piston means being arranged in such manner that the volumes of said expander chambers vary substantially in phase with each other and lead the volumes of said compressor chambers, the volumes of said compressor chambers varying substantially in phase with each other; first flow passage means interconnecting said hot expander chamber and said first compressor chamber; second flow passage means interconnecting said refrigerator expander chamber and said second compressor chamber; heat exchanger means disposed in each of said first and second flow passage means; and additional flow passage means interconnecting said first and second compressor chambers.

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