United States Patent [19]

Schuman

[11] **3,782,859**

[45] **Jan. 1, 1974**

[54] **FREE PISTON APPARATUS**

[76] Inventor: **Mark Schuman,** 101 G St., S.W., Washington, D.C.

[22] Filed: **Dec. 7, 1971**

[21] Appl. No.: **205,651**

Related U.S. Application Data

[63] Continuation-in-part of Ser. No. 169,003, Aug. 4, 1971, abandoned.

[52] **U.S. Cl.** **417/207,** 60/24, 417/375
[51] **Int. Cl.** **F04b 19/24**
[58] **Field of Search** 60/24; 417/207, 375

[56] **References Cited**

UNITED STATES PATENTS

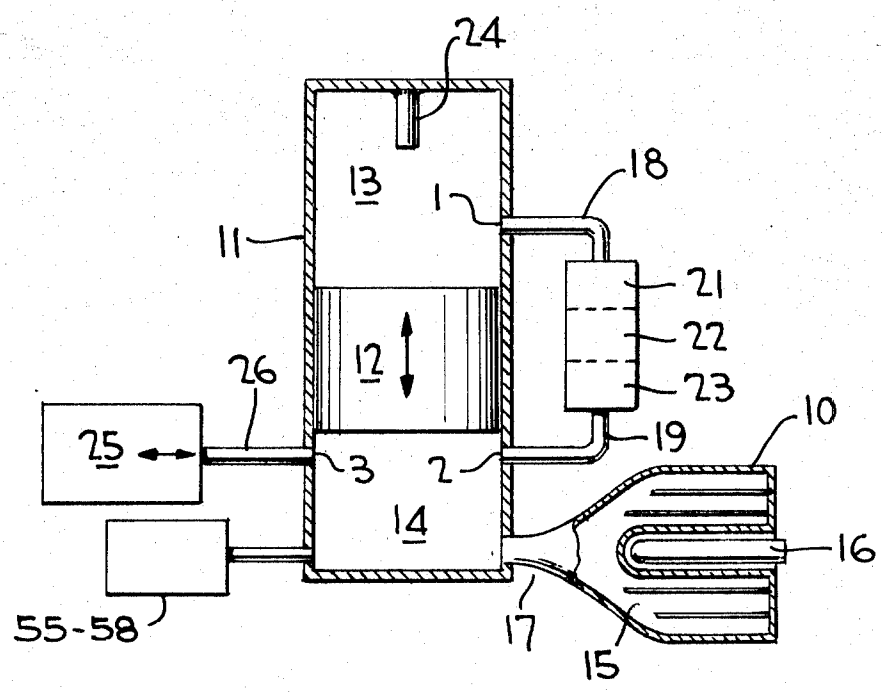
3,563,028	2/1971	Goranson et al.	60/24
3,552,120	1/1971	Beale	60/24
3,525,215	8/1970	Conrad	60/24
3,484,616	12/1969	Baumgardner et al.	60/24
3,559,398	2/1971	Meijer et al.	60/24
3,583,155	6/1971	Schuman	60/24
3,604,821	9/1971	Martini	60/24
3,597,766	8/1971	Buck	60/24
3,608,311	9/1971	Roesel	60/24

Primary Examiner—William L. Freeh
Assistant Examiner—Gregory LaPointe
Attorney—Lowe and King

[57] **ABSTRACT**

A by-pass including a heat regenerator is provided for a cylinder in which an oscillating free piston of a gas pump is driven. Shunting of gas back and forth through the by-pass between disparate hot and cold regions of the cylinder while the piston coasts in the by-pass region produces a periodic temperature and pressure variation which may be applied to a load. In one embodiment the by-pass includes in seriatim cold, regenerator, and hot chambers. In another embodiment the heating and/or cooling means are located within the cylinder. In still another embodiment preheated gas is drawn through an inlet into the hot end portion of the cylinder, and cool gas is drawn into the cool end portion of the cylinder. In one embodiment piston oscillation is sustained by means of a thermal leg chamber included in the piston. In another embodiment pressure pulses are fed back to the cylinder to sustain oscillation. In one embodiment the amplitude and frequency of pump pressure variations are controlled by varying the coasting portion of the cycle. In another embodiment the pump output is controlled by shunting a variable amount of gas around the heater and/or around the cooler, or to a second by-pass containing only a regenerator. In a further embodiment, the pressure variations are controlled by varying the feedback of pressure pulses. In still another embodiment pumping is controlled by varying the geometry of a thermal leg chamber or the portion of the cycle during which it is operative. Variations in temperature or geometry of pump components are also used as pump control means in certain embodiments. In one embodiment the pump produces electrical power by driving a free piston linear alternator wherein pumping controls such as the above can be used to maintain relatively constant voltage and frequency under conditions of variable load. In another embodiment the pump may be used to pump gases from one region to another, or from one pressure to another. In a further embodiment the pump may be used as a cooling device. In certain embodiments the pump may be used to produce pressure variations in a medium. In embodiments utilizing a thermal lag chamber for sustaining oscillation of the free piston, the relatively little coupling between the load and the free piston results in a pump which is essentially stall-free.

119 Claims, 17 Drawing Figures



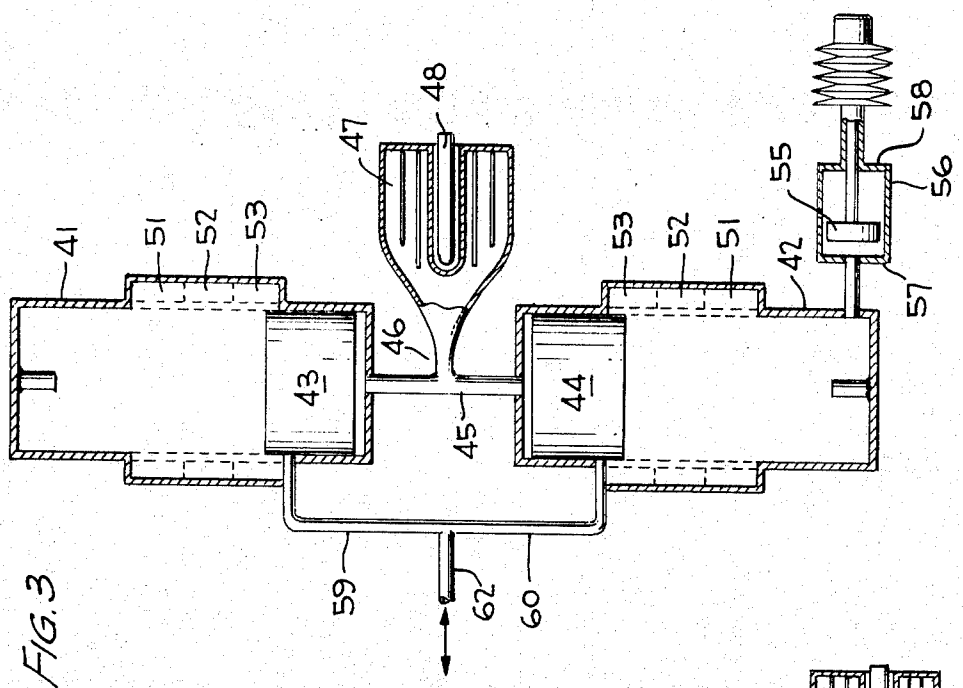


FIG. 3

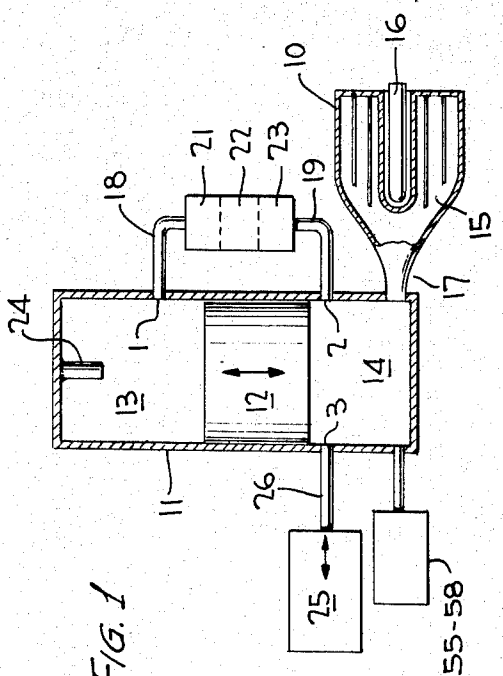


FIG. 1

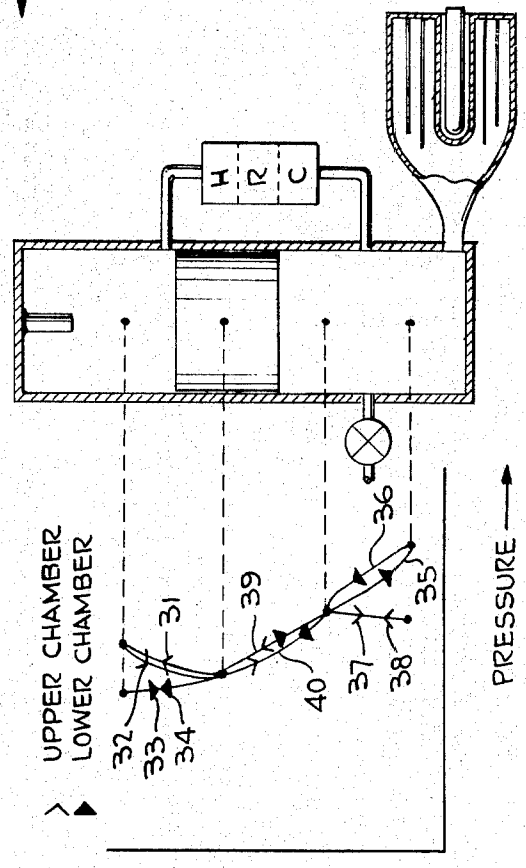


FIG. 2

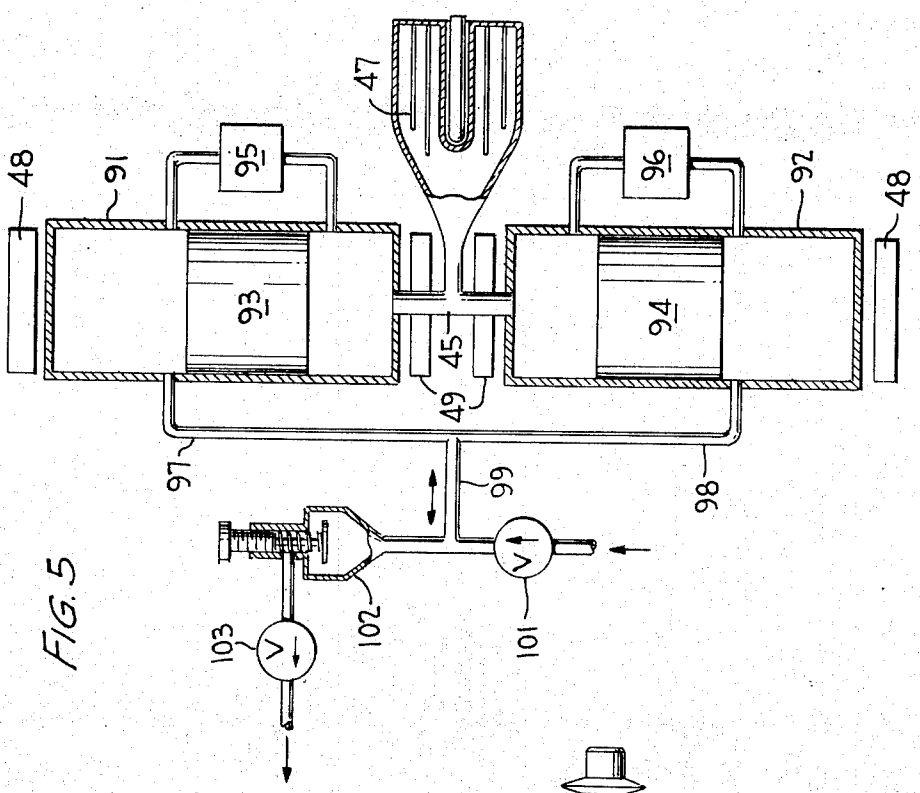

47
48
91
95
96
92
48
93
94
45
49
97
99
98
V
103
102
101

FIG. 5

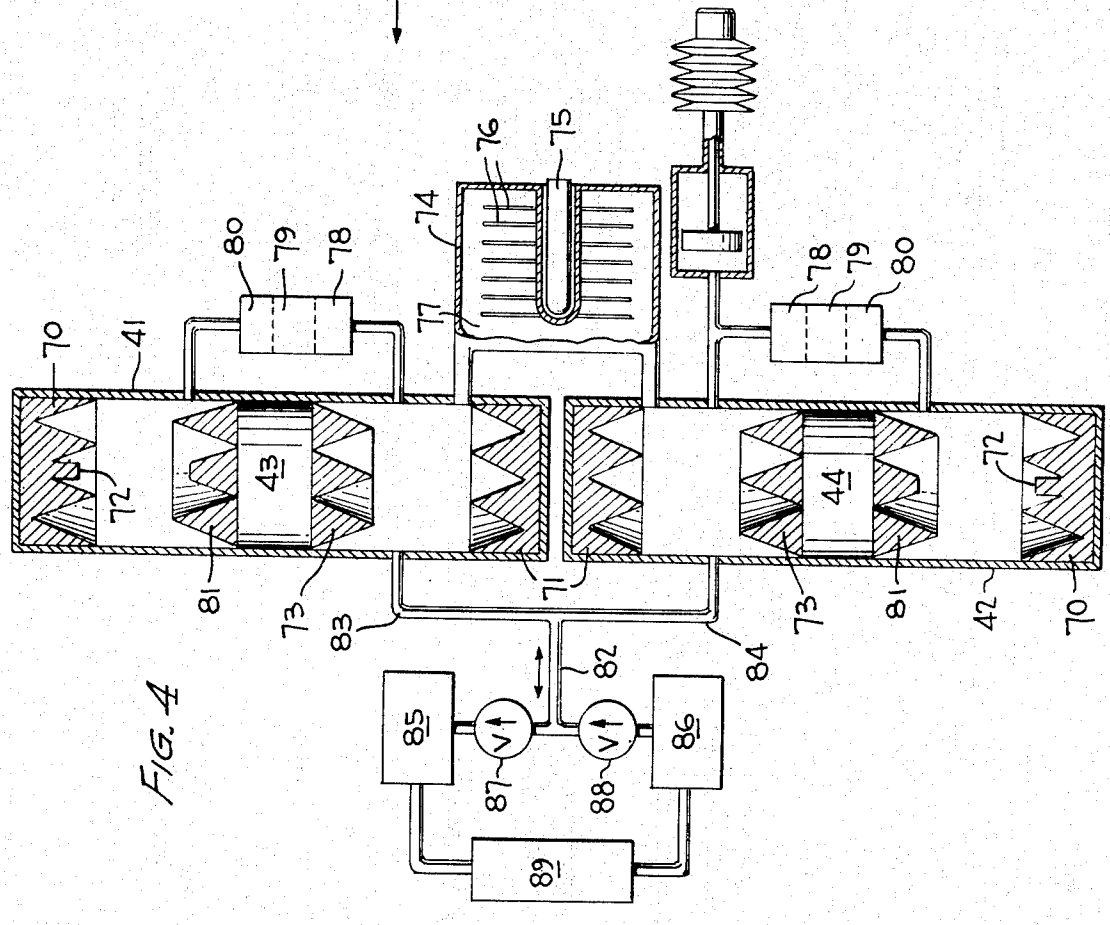

76
75
74
80
79
78
77
78
79
80
70
41
72
43
44
72
81
73
83
71
73
81
42
70
82
84
85
86
87
88
89

FIG. 4

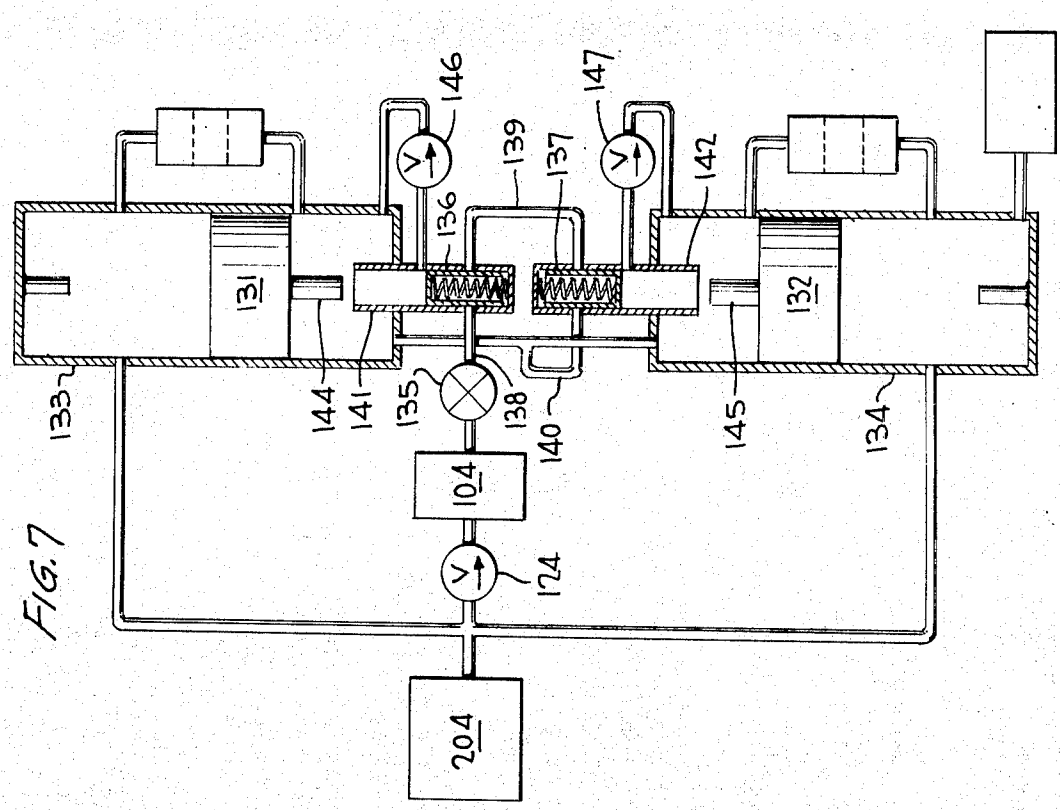

146
147
139
137
142
136
131
132
133
144
141
135
138
140
145
134
104
124
204

FIG. 7

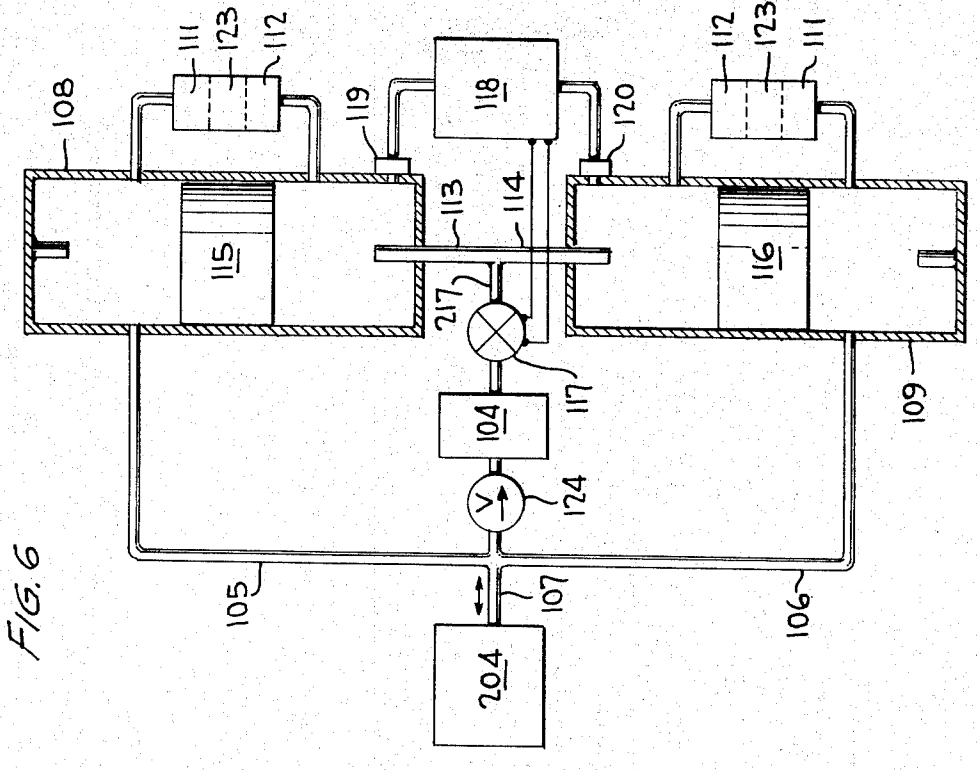
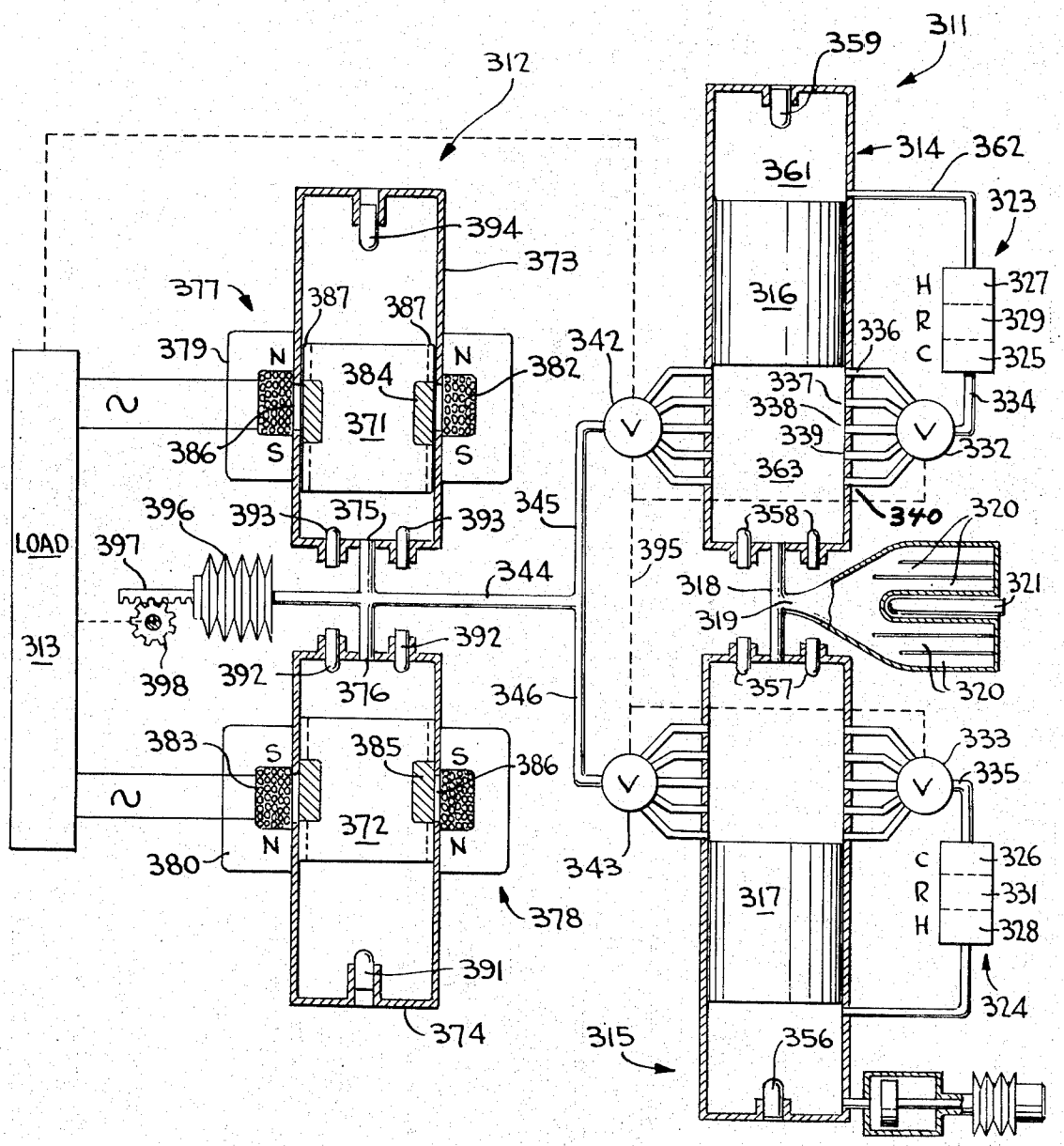

111
123
112
108
119
118
120
112
123
111
113
114
115
116
217
117
109
104
124
105
107
106
204

FIG. 6

FIG. 8


312
359
311
314
362
361
323
316
H
R
C
327
329
325
336
342
337
338
339
334
332
363
340
320
358
345
395
318
321
319
357
320
344
346
333
335
343
326
331
328
324
317
315
356
LOAD
313
397
396
393
375
393
398
392
392
376
383
385
386
372
380
378
391
374
377
387
387
379
384
382
371
386

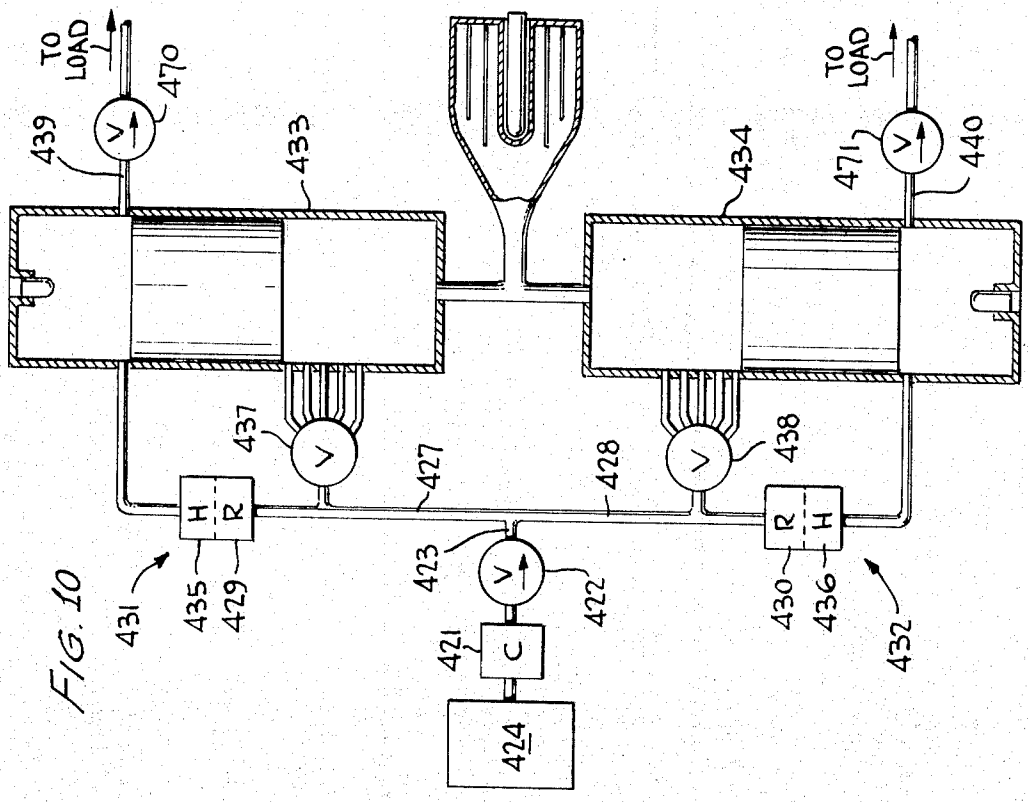

TO LOAD
470
439
433
434
471
440
437
438
427
428
H
R
423
422
421
C
424
431
435
429
430
436
432

FIG. 10

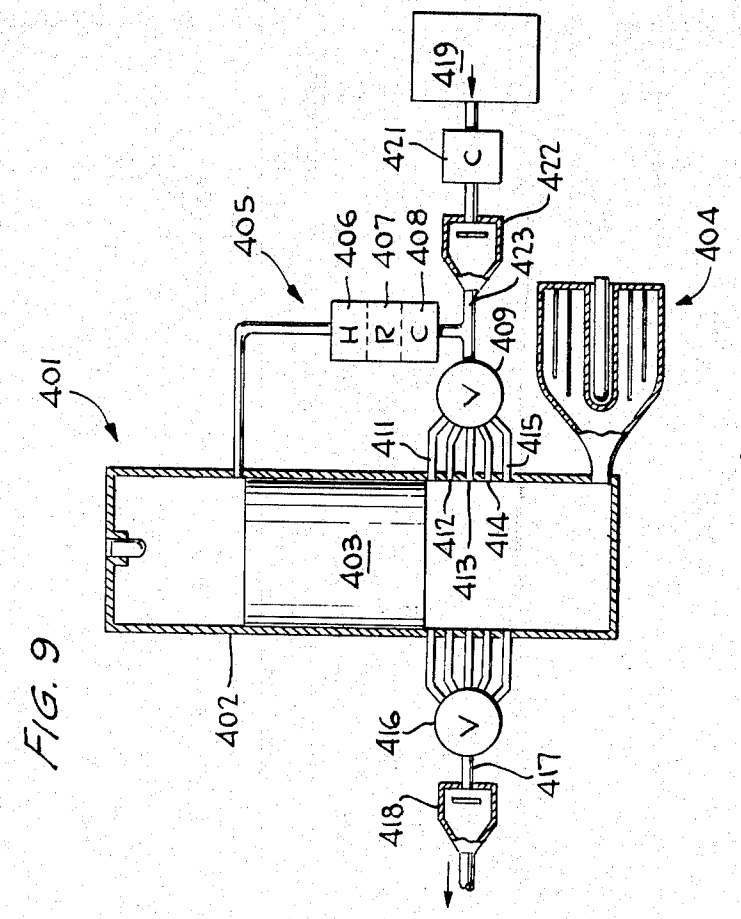

419
421
C
422
405
406
407
408
423
404
H
R
C
409
401
411
415
412
413
414
403
402
416
417
418

FIG. 9

FIG. 11

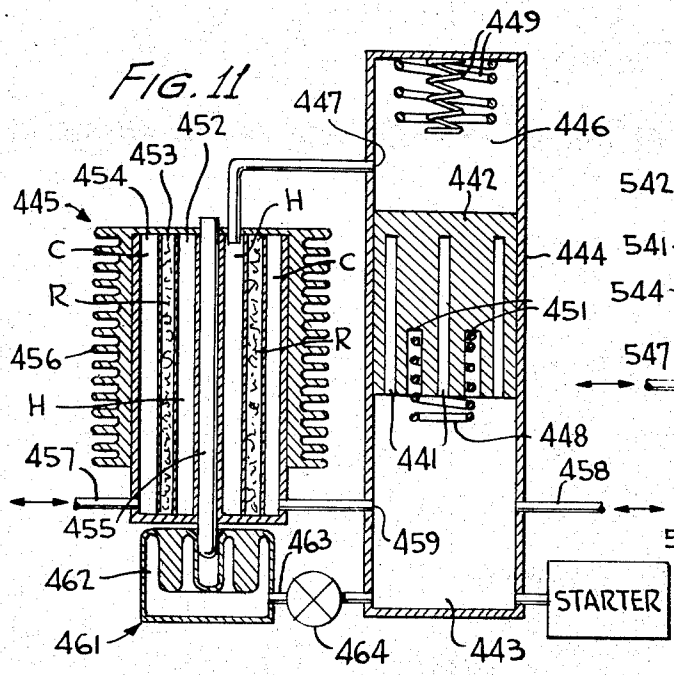


FIG. 14

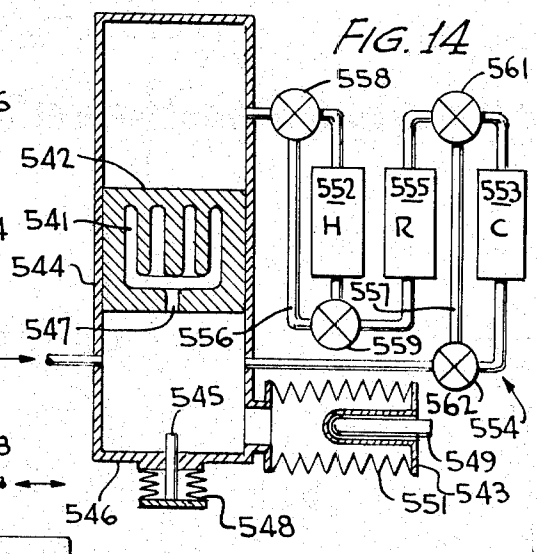


FIG. 12

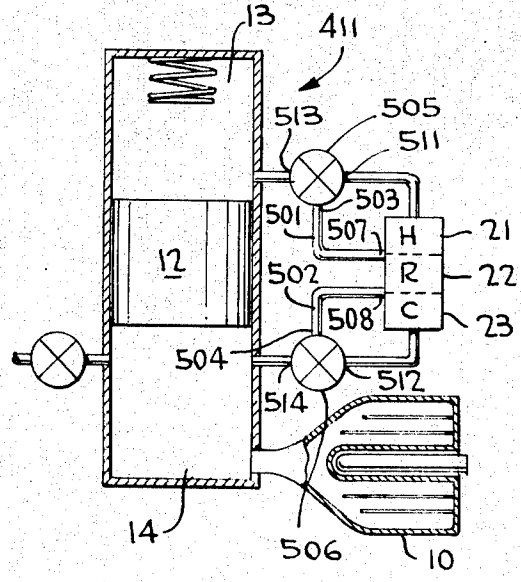


FIG. 13

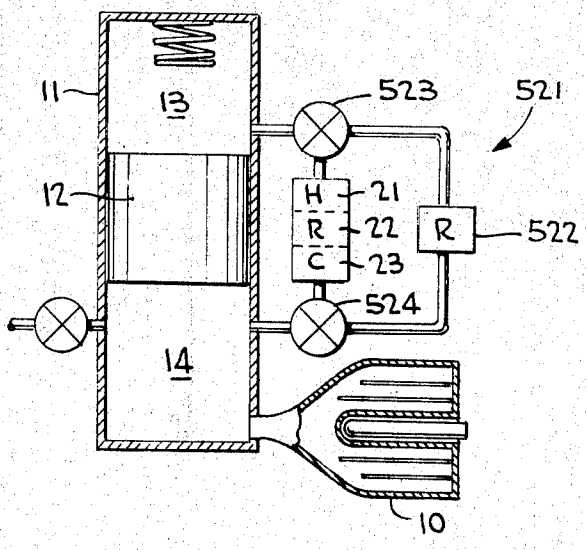


FIG. 15

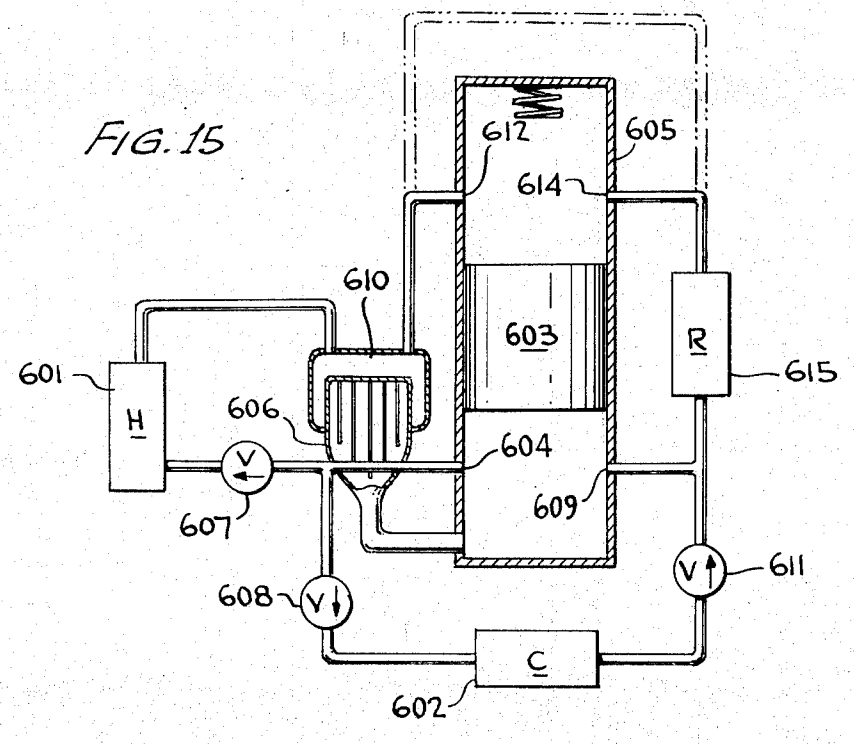


FIG. 16

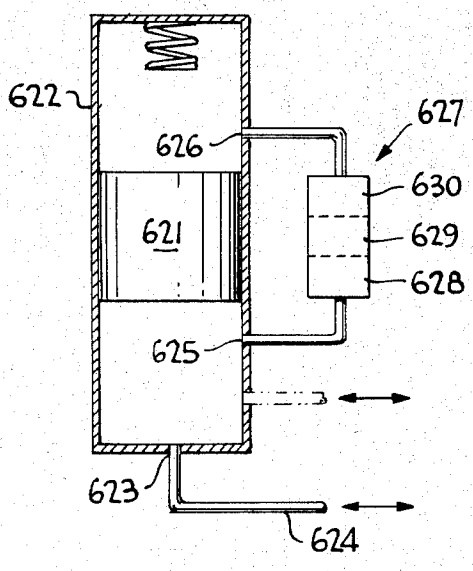
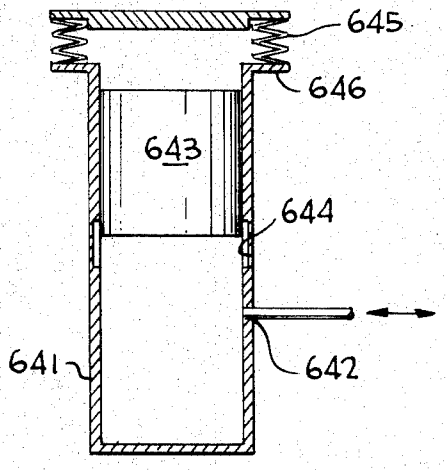


FIG. 17



FREE PISTON APPARATUS

RELATIONSHIP TO COPENDING APPLICATIONS

The present application is a continuation-in-part of my copending application entitled "Free Oscillating Piston Apparatus", Ser. No. 169,003 filed Aug. 4, 1971, now abandoned and is an improvement on my copending application entitled "Oscillating Piston Apparatus," Ser. No. 121,371, filed Mar. 5, 1971 now abandoned.

FIELD OF INVENTION

The present invention relates generally to free oscillating piston devices for converting externally applied thermal energy into work or another form of energy and, more particularly, to a free oscillating piston apparatus wherein the piston coasts in a by-pass region of a cylinder, which by-pass includes a regenerator.

BACKGROUND OF THE INVENTION

Efficient free piston devices including the principles of a Stirling engine have been reported. A free piston is one having no mechanical coupling to drive sources outside of the cylinder in which it travels. One of the reported devices is a regenerative device wherein a working gas is forced through the series combination of a heater, regenerator and cooler by a displacer piston that moves relative to a working piston in a cylinder. In this device the free piston is a combination of the displacer and working pistons. The use of a displacer piston that moves relative to a working piston obviates some of the inherent advantages of a free piston, namely, the simplicity, low friction, and low wear typically resulting from the use of only a single moving member in each cylinder. The coupling and contact between the displacer and working pistons during each cycle of the reported device absorb energy to reduce efficiency. In addition, contact between the displacer and working piston may have a tendency to produce unwanted vibrations. Further, the coupling between the displacer and working pistons makes the device more susceptible to being stalled by a severe load.

Other modern free piston regenerative devices include a free piston that is divided into a displacing and reversing pistons which are rigidly interconnected to each other. In response to varying differential pressure against its faces, the reversing piston changes its direction of movement and that of the displacer piston also. The displacing and reversing pistons do not ride in exactly the same cylinders, and do not always see the same pressures. During at least a portion of the cycle the reversing is done concurrently with the displacing, and the energy for reversing is derived from the pump output. The free piston of these devices is of a relatively complex shape and therefore difficult to fabricate. It is also susceptible to stalling.

BRIEF DESCRIPTION OF THE INVENTION

In accordance with the present invention, a regenerator is incorporated in an efficient piston device wherein a single, free piston is oscillated in each pump cylinder. Each cylinder is divided into first and second variable volume chambers by the opposite faces or edges of the single piston. The single piston, which is preferably of simple cylindrical shape, functions as both a displacer piston and as a reversing piston during different portions of an oscillating cycle. A by-pass, including the regenerator, is established around the piston faces between the first and second chambers while the piston is moving through a center region of the cylinder between the ends thereof. Thereby, gas in the cylinder flows through the by-pass from the first chamber to the second chamber while the piston is moving through the central region in a direction tending to increase the volume of the second chamber and decrease the volume of the first chamber. When the piston is moving through the central region in the opposite direction, so that the volume of the first chamber is increasing and the volume of the second chamber is decreasing, gas flows through the by-pass from the second chamber to the first chamber. The by-pass is blocked twice during each cycle of piston oscillation after the piston has coasted through the central region of the cylinder. Gas flowing into the second chamber through the by-pass is cooled while the piston is moving through the central region in a direction to increase the volume of the second chamber. Gas flowing into the first chamber through the by-pass while the piston is moving through the central region in the opposite direction is heated.

The regenerator is a heat storage and release mechanism, whereby during each cycle heat is applied to the cool gases and withdrawn from the hot gases flowing through the by-pass. The regenerator enables the temperature and pressure of the cooled gas to increase before the gas is heated by externally applied heat while flowing from the second chamber through the by-pass to the first chamber during a portion of one half cycle of the piston oscillation through the center region of the cylinder. Conversely the regenerator enables a decrease in the gas pressure to occur before the gas is cooled in an externally cooled cooling chamber during the opposite half cycle of the piston travel through the central region of the cylinder. The by-pass is also important to enable the center position of the oscillating free piston to be established in a manner similar to that disclosed in my co-pending application entitled "Oscillating Piston Apparatus," filed Mar. 5, 1971.

Oscillation of the free piston can be sustained by utilizing previously developed passive thermal lag techniques, such as disclosed in my U. S. Pat. No. 3,489,335, issued Jan. 13, 1970, or as disclosed in my aforementioned co-pending applications. In addition, the free piston can be driven from other energy sources, e.g., a pressurized source of gas. In accordance with one embodiment of the present invention, the gas source is pressurized by a feedback arrangement from the pump outlet.

To minimize vibration, in accordance with several embodiments of the invention, a pair of free oscillating pistons and a pair of cylinders are provided. The free oscillating pistons are driven in synchronism with each other so that the net vibration of the entire structure is virtually zero.

In accordance with another aspect of the invention, in certain embodiments thereof, the free piston or pistons are the only moving parts. There is no need for valves to control the flow through the by-pass because the free, oscillating piston and the by-pass are dimensioned in such a manner as to enable the piston itself to selectively block and unblock the by-pass.

Another feature of the invention is that most load variations are incapable of stalling the free, oscillating piston. The load is connected to the cylinder through a port which is in general positioned in the second

chamber in such a manner as to be responsive to cooled gas while the piston is coasting through the central portion of the cylinder. The load is responsive to the cooled gas in the second chamber primarily while the piston is coasting in the central region while the by-pass is unblocked. Since the by-pass is unblocked, load variations have substantially the same effect on both sides of the coasting piston and usually cannot stall the coasting piston.

In accordance with another aspect of the present invention, power modulation can be performed quickly, easily, over a wide range of values, and with a relatively high degree of accuracy when the free piston device is connection as an oscillatory pressure source driving a load. Thereby, the amount of power supplied to the load and the frequency of pressure variation can be controlled.

Power and frequency control can be achieved by varying the length and position of the by-pass region through which the free oscillating piston coasts in a cylinder. Increasing the length of the by-pass region, which increases the coasting time of the piston through the by-pass region, results in a greater output power being supplied by the oscillating piston pump to the load. Conversely, decreasing the length of the coasting region, which decreases the amount of time during which gas is supplied to the load by the pump, results in the application of a lesser amount of power to the load. Changing the by-pass length changes the mass of gas forced through the by-pass and heated or cooled thereby. Therefore, the amount and time duration of pressure variation applied to a load are changed. If heating and cooling chambers in the by-pass are heated and cooled at a sufficient rate while the by-pass is relatively short, appreciable heat energy is stored in the chamber walls because there is a relatively low gas flow through the by-pass. The stored energy in the chamber walls is available for rapid conversion into pneumatic power in response to the length of the by-pass region increasing. Efficiency is considerably greater at low power levels than would be attained with conventional flow modulation techniques, e.g., restricting flow by a valve.

There is a tendency for changes in the by-pass length or position to effect the frequency of piston oscillation. In my devices employing pneumatic springs at the ends of the cylinder to reverse the direction of piston movement, as disclosed in my co-pending applications, the effects on piston oscillation frequency due to changes in the by-pass length have a tendency to be cancelled. This is because changing the length of the by-pass tends to change the pressures at the ends of the cylinders which alters the pneumatic spring constants of gases at the ends of the cylinders. The alteration in spring constants is offset to a certain extent by the change in coasting time of the piston through the cylinder by-pass region. If the load to be driven requires precise frequency regulation, a frequency controlled system can be provided by responding to the load frequency and varying the location of the by-pass orifices or other changes in pump geometry in order to vary the pressure, spring constants or coasting time.

An additional aspect is a free piston electrical alternator responsive to oscillatory or periodic pressure variations such as may be derived from the free piston pump. Such a combination results in a thermally driven alternator which is sealed and which can automatically maintain substantially constant voltage and frequency under variable load conditions. The alternator portion comprises a magnetic circuit including magnetic poles displaced along the walls of the alternator cylinder. One component of the magnetic circuit is the free, oscillating piston driven by the periodic pressure variations. The magnetic circuit also includes coil means magnetically coupled through the alternator cylinder walls to the free piston, whereby there is induced an a.c. voltage in the coil means in response to the free piston being driven. The free piston may include poles of a permanent magnet, or the free piston can be of a material having high magnetic permeability coupled to pole faces in or outside of the cylinder walls. The alternator piston could have a by-pass for maintaining its desired center of oscillation in spite of variable pressure and load, but not for heating or cooling gases in the by-pass as in the case of the pump.

In accordance with another aspect of the present invention, a free oscillating piston in a cylinder including a by-pass having a heater is responsive to an external source of cool gas or gas from an external source as fed to the cool end cylinder through a cooling chamber. By using cool gas or by cooling gas from an external source prior to supplying it to the cylinder, a cooling chamber in the by-pass becomes less important and can even be eliminated. Heating, as well as cooling, may similarly be derived from gases being pumped.

In accordance with still a further embodiment of the invention, a thermal lag heater to sustain piston oscillation comprises hot passageways in the face of the piston communicating with the cold end of the cylinder. The piston and, therefore, the passageway walls are heated by hot gases from the by-pass and by radiant and conductive heat transfer from the hot cylinder walls.

In accordance with an additional embodiment, at least one controlled flow shunt path is provided around the heating and cooling chambers in the by-pass to control the amount of heating and cooling in the by-pass, and thus the average and differential pump pressure. In one arrangement, separate variable shunt flow paths are provided around the hot and cold chambers of the by-pass. In a second arrangement, another regenerator is provided in a variable shunt flow path around the hot, regenerator and cold chambers in seriatim. The first arrangement is particularly advantageous since oscillating frequency and output power can be controlled independently by varying the amount of heating and cooling individually.

In accordance with another aspect of the invention, modulation is achieved by varying one or more properties of the thermal lag device. To this end, the geometry, e.g., volume or passageway shape, of the thermal lag device can be controlled or the portion of the cycle over which the thermal lag device is operative can be controlled.

It is, accordingly, an object of the present invention to provide a new and improved free oscillating piston apparatus.

It is another object of the present invention to provide a new and improved simple and efficient free oscillating piston pump utilizing a heating chamber, regenerator, and cooling chamber.

Another object of the invention is to provide a new and improved free oscillating piston device having a minimum amount of vibration.

Another object of the invention is to provide a new and improved, efficient, free oscillating piston pump having as its only moving part a piston which functions alternately as a displacer and a reversing piston.

A further object of the invention is to provide a free oscillating piston energy conversion device employing a regenerator connected in a by-pass around the piston, which by-pass also functions to determine the approximate center of piston oscillation. **105.** An additional object of the invention is to provide an improved free, oscillating piston energy conversion device which is relatively insensitive to load variations and is path beginning stall-free.

Another object of the present invention is to provide a new and improved free oscillating piston device wherein the amount of power derived from the device can be varied rapidly in a simple manner.

Another object of the invention is to provide new and improved devices for rapidly controlling the oscillation frequency of a free piston device.

A further object of the invention is to provide a free oscillating piston device wherein power and frequency are controlled by varying the coasting of a piston in a cylinder containing a by-pass, which by-pass includes a regenerator and possibly a cooler and/or heater.

An additional object of the invention is to provide a new and improved, pressure driven, linear alternator employing a free piston which is centered in its cylinder in a simple, reliable manner.

Another object of the invention is to provide a new and improved system wherein an oscillating free piston pump drives and accurately controls an oscillating free piston linear alternator under conditions of variable load.

Yet another object of the invention is to provide a new and improved system for driving an electrical load in response to current derived from an alternator including free, oscillating pistons driven by oscillatory pressure variations derived from an oscillatory pressure source.

A further object of the invention is to provide a free oscillating piston pump that is controlled in power and frequency by varying the amount of heating and/or cooling of gas flowing in a by-pass.

Another object of the invention is to provide a new and improved, highly efficient and simply constructed free piston energy converter wherein oscillation of the free piston is sustained and controlled by a thermal lag means.

An additional object of the invention is to provide a free oscillating piston pump that is controlled by varying, during its cycle, the mass flow rate of gas through a by-pass.

A further object of the invention is to provide a new and improved thermally driven pump wherein the thermal energy to operate the pump is derived in part or in full from the fluid or fluids being pumped.

An additional object of the invention is to provide a new and improved cooling apparatus employing a free piston regenerative cycle device.

The above and still further objects, features and advantages of the present invention will become apparent upon consideration of the following detailed description of several specific embodiments thereof, especially when taken in conjunction with the accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWING

FIG. **1** is a schematic diagram of one embodiment of the present invention employing a single piston and a thermal lag heater to sustain oscillation;

FIG. **2** illustrates pressure vs. displacement curves for the device of FIG. **1** when the outlet is closed;

FIG. **3** is a schematic diagram of a modification of FIG. **1**, wherein a pair of synchronized pistons is provided to cancel vibration and a loose piston is employed as a starter;

FIG. **4** is a modification of the pump of FIG. **3**, wherein cylinders carrying the synchronized pistons communicate with each other via a thermal lag heater, heating and cooling chambers are provided in the cylinders, and the connections to a load are illustrated;

FIG. **5** is a modification of the system of FIG. **4**, wherein heating and cooling means are provided only in the cylinders and alternative load connections are illustrated;

FIGS. **6** and **7** are other synchronized piston embodiments wherein oscillation is sustained by pressure feedback from a load;

FIG. **8** is a diagram of an embodiment wherein pressure variations supplied to a load are affected by controlling the coasting region of the free piston and wherein a free piston alternator is responsive to oscillatory pressure variations;

FIG. **9** is a diagram of a modification of the system of FIG. **8**, wherein a separate pump inlet and outlet are provided and gas entering the pump through the inlet is precooled;

FIG. **10** is a diagram illustrating how the device of FIG. **9** can be modified so that there is no cooling chamber in the by-pass and also illustrates how the oscillating piston pump may utilize two synchronized free pistons;

FIG. **11** is a diagram of an embodiment employing a thermal lag heater carried by the free piston;

FIGS. **12, 13** and **14** are diagrams of further embodiments wherein pressure amplitude and frequency are controlled by varying the amount of heating and cooling of gas flowing in the by-pass;

FIG. **14** also illustrates two thermal lag chambers having variable properties for varying piston amplitude and frequency.

FIG. **15** illustrates how the present invention may be employed for cooling or warming a remote body.

FIG. **16** illustrates how the present invention may be used as a heat pump or refrigerator.

FIG. **17** is a diagram illustrating another embodiment of the invention for transforming pressure variations.

DETAILED DESCRIPTION OF THE DRAWING

Reference is now made to FIG. **1** of the drawing wherein there is illustrated a gas pump including a sealed cylinder **11** through which free piston **12** is oscillated. The opposite faces of piston **12** divide cylinder **11** into a first and second chamber **13** and **14.** At the end of chamber **14** there is provided a series of elongated passageways **15** located in a double walled chamber **10.** A heating source **16** extends between the interior walls of chamber **10** so that passageways **15** are heated to establish a thermal lag heating chamber to sustain oscillation of piston **12.** As described in my application filed Mar. 5, 1971, passageways **15** may be heated along their lengths, and the cylinder walls may

be cooled in proximity to conduit 17, and cooling fins may be added. Cooling chamber 23 also contributes to cooling of the gases in chamber 14. The cooling of gases in chamber 14 augments the subsequent heating of these gases upon the gases being forced by piston 12 into the heated passageways 15. Thus the cooling of gas in chamber 14, as well as the heating of passageways 15, increases the kinetic energy supplied to piston 12 by the thermal lag heating chamber 10, and this increase in kinetic energy tends to increase the piston amplitude and frequency.

The walls of cylinder 11 and piston 12 are so close to each other that the piston provides a reasonably good seal between the opposite faces thereof. Therefore, while piston 12 is near the top and bottom respectively of chambers 13 and 14, there is a substantial difference between gas pressures at opposite faces of the piston. In a central region of cylinder 11, the pressure difference across piston 12 is virtually zero due to a by-pass formed by lines 18 and 19, between which are connected in seriatim heater 21, regenerator 22 and cooler 23. Heater 21 can be a chamber containing one or more heated passageways, regenerator 22 can be a chamber containing a mesh of wire through which gas flows between heater 21 and cooler 23, and cooler 23 can be a chamber containing one or more passageways and having cooling fins or other means for external cooling.

Orifices 1 and 2 are positioned in the side wall of cylinder 11 leading to conduits 18 and 19 and the length of piston 12 is determined in such a manner that both conduits simultaneously communicate with chambers 13 and 14 while piston 12 is coasting in the cylinder central region between the orifices. The positions of orifices 1 and 2 and the length of piston 12 are such that the piston covers and blocks conduit 18 when the volume in chamber 13 is a minimum and for a certain time interval on either side of minimum volume, and the piston blocks conduit 19 while the volume in chamber 14 is a minimum and for a certain time interval on either side of minimum volume. Chambers 13 and 14 (the volume of chamber 14 includes the volume of passageways 15) form gaseous springs at each end of cylinder 11. The upper gaseous spring is not always effective during piston start up, however, whereby a resilient stop 24 may be provided at the end of chamber 13 to guarantee a minimum amount of gas for this gaseous spring.

A load 25 is driven by pressure variations in the cylinder 11 through a conduit 26 having an orifice 3 disposed in the side wall of cylinder 11 approximately the same distance from the lower end of cylinder 11 as orifice 2 so that orifices 2 and 3 are open and closed at about the same time. Thereby, as piston 12 coasts between orifices 1 and 2 in a downward direction, compressed gas is fed through conduit 26 to load 25 at relatively high pressure and substantially no gas is fed to conduit 26 while the lower face of piston 12 is below orifices 2 and 3. Gas is drawn from the load 25 at relatively low pressure while the piston 12 is moving upward above the orifices 2 and 3. If the gas drawn from the load is cool, it assists in cooling the gas in chamber 14.

While the orifice feeding pressure variations from the pump to load 25 is preferably in the position illustrated to substantially prevent stalling, it is to be understood that the orifice in this and other embodiments can be located at other positions.

A suitable starting means 55–58 for piston 12, as described in my previously mentioned copending applications as is described herein infra, can be employed. To describe the operation of the embodiment of FIG. 1, it is initially assumed that piston 12 is at its greatest upward travel to minimize the volume of chamber 13 and close off orifice 1. To simplify the operating description, it is assumed that orifice 26 is blocked so that load 25 has no effect on operation. With piston 12 so located, the inlet 1 from chamber 13 to the by-pass is substantially closed due to the sealing action of piston 12 against the walls of cylinder 11. After piston 12 has assumed the described initial position, the gases above the piston in chamber 13 expand and force the piston downwardly. The gases in chamber 13 continue to expand and exert a force on the upper face of piston 12 until the upper face of the piston passes orifice 1. At this time, the gases in chamber 13 communicate with conduit 18 and the by-pass becomes operative around piston 12. The piston continues to travel, essentially by coasting, through the by-pass region between orifices 1 and 2 because of the momentum of the piston resulting from the force applied thereto as the piston began its downward stroke in chamber 13.

Because of the by-pass around piston 12, there is only a slight difference in pressure upon the upper and lower piston faces; the pressure at the upper face is slightly less than that of the lower face as piston 12 coasts downwardly through the region between orifices 1 and 2. In response to the pressure difference between the faces of piston 12 as the piston coasts downwardly through the central by-pass region, there is a flow of gas from chamber 14 through line 19, cold chamber 23, regenerator chamber 22, hot chamber 21 and conduit 18 into chamber 13. This heating of gas flowing upward through the by-pass increases the pneumatic pressure throughout the pump. Because of regenerator 22, which establishes a thermal gradient between hot chamber 21 and cold chamber 23, heat is applied to the gases flowing upwardly through the regenerator 22. Regenerator 22 enables the temperature of the gas flowing through the by-pass to increase before it is fed to heater chamber 21 and upper cylinder chamber 13. Since external heat is applied to gas already preheated by the regenerator, a higher peak gas temperature and pressure may be reached and efficiency of the pump is enhanced.

After the bottom face of piston 12 has moved downwardly past orifice 2, the by-pass is cut off and the bottom face of piston 12 compresses the gases in chamber 14 (including passageways 15) to reduce the piston velocity. The piston velocity is zero when minimum volume of chamber 14 is reached. The gases in chamber 14, now being almost at maximum compression, begin to expand and drive piston 12 upwardly as a result of gaseous or pneumatic spring effect and a thermal lag effect caused by passageways 15. As piston 12 translates upwardly in response to the spring and thermal lag effect of the gases at the bottom of chamber 14 and before the bottom face of piston 12 passes orifice 2, gas being heated in the thermal lag passageways 15 and escaping from the thermal lag device through conduit 17 continues to expand in cylinder 11 to provide additional force to maintain piston 12 in an oscillating state. Passageways 15 provide a thermal lag, as described in

my copending applications, and this increases the upward force on piston **12** to maintain the piston oscillation. The pressure in chamber **14** is augmented by the heated, expanding gases from passageways **15.** By the time the bottom face of piston **12** is at the level of orifice **2** the pressure in chamber **14** has decreased almost to the same value as the pressure in chamber **14** during the downward stroke of piston **12** when it reached orifice **2.**

As piston **12** moves upwardly so that its bottom face is above orifice **2,** the by-pass is again unblocked and the pressures in chambers **13** and **14** both decrease in response to cooling of the gas therein. As piston **12** moves upwardly through the by-pass region, there is a slightly greater pressure on the upper piston face than on the lower face. Despite this pressure difference, piston **12** moves upwardly through the by-pass region in response to the momentum imparted thereto by the spring effect at the end of chamber **14** and the effects of the thermal lag created by passageways **15.** The slightly higher pressure of gases in chamber **13** above the upper face of piston **12** relative to the pressure in chamber **14** at the time piston **12** coasts through the by-pass region results in gas flow from chamber **13** to chamber **14** through conduit **18,** heating chamber **21,** regenerator chamber **22,** cold chamber **23** and conduit **19.** Thereby, heat in the gases flowing from chamber **13** through conduit **19** into chamber **14** is withdrawn from the gas by the regenerator and the cooling chamber **23** and pump pressure decreases. By using the regenerator a greater temperature differential is established between the hot and cold gases in chambers **13** and **14,** which results in a greater pressure change during coasting of piston **12,** to provide increased efficiency.

Gas continues to flow from chamber **13** to chamber **14** through the by-pass until the upper face of piston **12** passes orifice **1,** at which time the by-pass is blocked and the pressure in chamber **13** increases with further upward movement of piston **12.** The increased pressure in chamber **13** decreases the velocity of piston **12** until substantially maximum compression exists in chamber **13** at the top of the stroke of piston **12** at which time the piston velocity is zero. After maximum compression of the gases in chamber **13** is achieved, the gases in chamber **13,** due to the pneumatic spring effect, begin to expand and drive piston **12** downwardly to repeat the cycle described.

Under the assumed conditions of conduit **26** being blocked, the pressures in chambers **13** and **14** versus position of the center of the piston are illustrated in FIG. **2.** The arrows in FIG. **2** indicate the direction of piston movement, i.e., the downwardly directed arrows indicate the pressure in chambers **13** and **14** as piston **12** is moving downwardly, while the upwardly directed arrows indicate the pressures in the two chambers as the piston stroke is upward. The dashed lines and dots show the piston position in the cylinder and on a plot of pressure versus position. The pressure in chamber **13** as piston **12** moves in the region above the orifice of conduit **18** is defined by the line segments **31** and **32,** while the pressure in the lower chamber **14** during this interval is defined by the line segments **33** and **34.** While the lower face of piston **12** is below orifice **2,** the lower chamber **14** pressure is indicated by line segments **35** and **36,** while the upper chamber **13** pressure at this time is indicated by line segments **37** and **38.** While piston **12** is coasting between the by-pass in an upward direction, the pressures of upper chamber **13** and lower chamber **14** are given by segments **39** and **40.** As piston **12** coasts in the by-pass in a downward direction the pressures in the lower chamber **14** and upper chamber **13** are indicated by line segments **39** and **40,** respectively.

It is noted from an inspection of FIG. **2** that there is a substantial change in pressure in chambers **13** and **14** while piston **12** is coasting through the by-pass region. Under the assumed conditions of conduit **26** being blocked, this pressure change is maximized. If conduit **26** were unblocked and connected directly to the atmosphere, there would be almost no pressure change in chambers **13** and **14** as piston **12** coasts through the by-pass region.

In both previously described conditions, substantially no work would be produced by the device. When an actual load is connected to conduit **26** the range of pressure in chamber **14** as piston **12** coasts in the by-pass region is less than when conduit **26** is sealed and greater than when conduit **26** is open to the atmosphere. In addition, the drop in pressure in chamber **14** occurs primarily in the early part of the upward coasting motion of piston **12,** and the increase in pressure in chamber **14** occurs primarily in the early part of its downward coasting motion, according to the nature of the load. The resulting curve forms a clockwise type cycle. The pump can be designed to have a given thermopneumatic impedance. Matching the load to the pump is a factor in determining the power applied to the load by the pump.

Carnot efficiency of the device illustrated in FIG. **1** is approached because regenerator **22** is included in the by-pass, which allows a greater variation in temperature of the gas during the cyclic motion of piston **12** for a given rate of external heating and cooling. This results from applying heat to hot gases and removing heat from cooler gases to cause a greater temperature and pressure in the gases before the gases are fed to chamber **13** and a reduction of the temperature and pressure of the gas fed to chamber **14.** The by-pass also enables the center of oscillation of piston **12** to be determined approximately. In particular, if the two spring constants are equal, and there is no axial gravitational force, the piston **12** oscillates about a point approximately midway between the orifices of conduits **18** and **19** into cylinder **11.**

Those familiar with prior art, gaseous regenerative thermodynamic cyclic devices will appreciate that piston **12** performs the dual functions of a displacer piston and a reversing piston, two distinct elements usually included in devices having regenerative cycles wherein the displacer piston is reversed in part pneumatically. In such prior art devices, the displacer piston functions during most or all of the cycle. Concurrently during at least a portion of the cycle, pressure changes across the reversing piston, which usually has at least one face in a chamber different from the chamber for the displacer piston, cause reversals in the direction of the displacer piston. In the oscillating piston pump of FIG. **1,** however, piston **12** functions as a displacer piston while the by-pass is open and the same piston functions as a reversing piston when it moves beyond the coasting region into the region of cylinder **11** where pneumatic springs are formed by gases that cannot escape from the cylinder ends. Since the reversing chambers containing the pneumatic springs are merely extensions of

the displacer cylinder, simplicity, reliability, and possibly efficiency, are greater than in prior art devices requiring structurally distinct reversing piston and a more complex reversing chamber and associated parts, accurate sealing surfaces, and thermal or frictional losses. In addition, the reversing process does not require an electric motor or depend on load pressure variations, as with such prior art devices, and the pump is essentially stall-free and operates at a frequency which is relatively independent of load.

Reference is now made to FIG. 3 of the drawing wherein there is illustrated a modification of the device illustrated in FIG. 1. In FIG. 3, a pair of cylinders 41 and 42, each respectively including free pistons 43 and 44, are interconnected with each other via conduit 45. Conduit 45 is provided with a T to which is connected conduit 46 that leads to passageways 47 which form a thermal lag heater responsive to heat generated by source 48. The thermal lag device comprising passageways 47 and heat source 48 simultaneously applies increased pressure to pistons 43 and 44 as the pistons are moving away from conduit 45 and thereby maintains the pistons in synchronized oscillation, whereby both pistons travel toward and away from conduit 45 simultaneously. Because of the synchronized, in-line motion of pistons 43 and 44 there is virtually no vibration or torque applied to the pump as a result of piston oscillation.

Cylinders 41 and 42 are provided with by-passes comprising heaters 51, regenerators 52 and coolers 53. The heaters, regenerators and coolers are formed by enlarging the walls of cylinders 41 and 42 in central regions thereof to form an annular space or passageways. Guide walls are provided in cylinders 41 and 42 to control the motion of pistons 43 and 44 to prevent possible entry of the piston into the chambers 51–53. Outside of the central region, pistons 43 and 44 are guided by the cylinder wall. Chambers 51–53 function effectively in the same manner as described supra with regard to chambers 21–23. Cold chambers 53 are in closest proximity to conduit 45 relative to regenerator and hot chambers 52 and 51. The hot chambers 51 are farthest from conduit 45, while regenerator chambers 52 are located intermediate of the hot and cold chambers.

A starter is provided in the system of FIG. 3 at the bottom of cylinder 42. The starter comprises a loose piston 55 that is translatable in cylinder 56 between stops 57 and 58 at the ends of the cylinder. Stop 57 is provided with an aperture to admit gas in cylinder 56 into cylinder 42. During normal operation, piston 55 is locked by a means (not shown) against stop 57 so that gas from cylinder 42 cannot escape into the starter housing and gas is not fed from the starter into the cylinder 42. In starting, loose piston 55 is oscillated back and forth to allow leakage of air from cylinder 42 through the passageway defined by the walls of piston 55 and cylinder 56 and provide a pressure impulse to piston 44. Oscillation of pistons 43 and 44 in a synchronized manner is quickly achieved by virtue of the thermal lag effects of passageways 47 and the cooling of gases by cold chambers 53. Other starter locations are also possible.

To enable useful work to be derived with the system of FIG. 3, conduits 59 and 60 are connected to orifices in the walls of cylinders 41 and 42 between conduit 45 and cooling chambers 53, preferably a point immediately adjacent cooling chambers 53. Conduits 59 and 60 are connected to a common conduit 62, which in turn is connected to a load (not shown).

Reference is now made to FIG. 4 of the drawing wherein the system of FIG. 3 is modified whereby the near ends of cylinders 41 and 42 are provided with internal cooling fins, in the form of teeth 71 and the remote ends of the cylinders are provided with internal heating fins in the form of teeth 70, both of which sets of teeth have a relatively large surface area. Teeth 70 are provided with resilient stops 72 at the bases thereof for easier starting. Teeth 71 mate with cooling teeth 73 carried by pistons 43 and 44. Teeth 70 mate with heating teeth 81 carried by the pistons. Teeth 81 opposite stops 72 may be truncated to reduce stress upon contact with the stop. The teeth at the top of cylinder 41 and the bottom of cylinder 42 can be heated by an external source while the teeth in the center of the pump assembly, at the bottom and top of cylinders 41 and 42, can be cooled by external sources.

In the pump of FIG. 4, cylinders 41 and 42 are connected together via a thermal lag device 74. Thermal lag device 74 is responsive to heat derived from source 75 and includes a number of heated passageways 76 which communicate with a common conduit that extends between orifices provided in cylinders 41 and 42 just above the crowns of teeth 71.

A by-pass structure, of the type described in conjunction with FIG. 1 or FIG. 3 is provided in each of cylinders 41 and 42 and is illustrated as comprising cold chamber 78, regenerator chamber 79 and hot chamber 80. Gas pumped by pistons 43 and 44 of FIG. 4 is fed to a common line 82 via conduits 83 and 84. Lines 83 and 84 are connected to cylinders 41 and 42 through orifices positioned on the cylinder wall at approximately the same vertical location as the orifices to conduits leading between cold chambers 78 and cylinders 41 and 42. Thereby, gas is simultaneously fed by lines 83 and 84 in synchronism to common conduit 82 as pistons 43 and 44 approach conduits 83 and 84 in their travel towards the center of the piston assembly.

An exemplary load driven by the gases flowing to conduit 82 will now be described as comprising high pressure gas storage chamber 85 and low pressure gas storage chamber 86. Conduit 82 is connected to high pressure chamber 85 via check valve 87, while the low pressure chamber 86 is connected to conduit 82 via check valve 88. Check valve 87 prevents the flow of gas from high pressure chamber 85 to conduit 82. While pistons 43 and 44 have unblocked conduits 83 and 84 in their travel away from the center of the piston assembly, gas from low pressure source 86 is sucked through check valve 88 and conduit 82 into cylinders 41 and 42. During the portion of the cycle while conduits 83 and 84 are not blocked by pistons 43 and 44 and the pistons are moving towards the center of the pump assembly, compressed gas is fed to conduits 83 and 84 and fed through check valve 87 to high pressure chamber 85. Compressed gas flowing in conduit 82 at this time is not fed to low pressure chamber 86 because of check valve 88. High pressure and low pressure chambers 85 and 86 are connected to the inlet and outlet of an appropriate load 89, which may comprise a heart pump or other pneumatic device driven by a relatively constant pressure differential.

Reference is now made to FIG. 5 of the drawing wherein there is illustrated a modification of the embodiment illustrated by FIG. 4. In FIG. 5, upper and

lower cylinders **91** and **92** in which free pistons **93** and **94** oscillate in synchronism are provided with internal heating and cooling fins (not shown) and external heating sources **48** and cooling sources **49.** The external heating sources are located at the upper and lower ends of cylinders **91** and **92,** respectively, while the external cooling means are provided at the lower and upper ends of cylinders **91** and **92,** respectively. Some thermal lag effect may be obtained from the internal fins, thereby helping to drive the pistons. The cooler ends of cylinders **91** and **92** are connected together via conduit **45** which is connected to thermal lag device **47** in the same manner indicated supra, in FIG. **3.** A by-pass is provided in cylinders **91** and **92** at the same places as indicated supra regarding FIG. **1.** The by-passes **95** and **96** for cylinders **91** and **92** include only regenerating chambers **95** and **96** and appropriately located conduits. The heating and cooling chambers of the previously discussed embodiments are incorporated directly in cylinders **91** and **92** and comprise the heating and cooling sources referred to above.

In the embodiment of FIG. **5,** hot, rather than cool, gases are the pumped gases. To this end, conduits **97** and **98** are respectively provided with outlets connected to orifices in cylinders **91** and **92.** The outlets of conduits **97** and **98** are respectively connected to the heated upper and lower segments of cylinders **91** and **92,** at a position approximately aligned with the outlets of regenerators **95** and **96** into the heated portions of the cylinders. Hot gas may be pumped in other embodiments also.

In FIG. **5** there is illustrated another type of device for feeding a load in response to oscillatory gases fed by conduits **97** and **98** to a common outlet conduit **99.** The load driving device comprises check valve **101** poled in a direction to feed low pressure gas into conduit **99** and prevent the flow of high pressure gas from conduit **99** through it. A flow limiter **102** is connected in series with a further check valve **103,** poled in such a manner as to pass high pressure gas from the pump to a load. Thereby, regulation is provided as to the flow of gas from the pump into a chamber or other region of space.

Reference is now made to FIG. **6** of the drawing wherein is illustrated a modified two piston system.

In the embodiment of FIG. **6,** pressure pulses are fed to a piston from a pressurized fluid source to maintain the piston in oscillation. The source may be pressurized by a feedback arrangement from the load. The pressure pulse arrangement replaces the passageways in the previously discussed embodiments in which the pressure effect of thermal lag was used to sustain oscillation.

In FIG. **6,** load **204** is driven by outlet conduits **105** and **106** via conduit **107.** Conduits **105** and **106** are respectively connected to orifices at the top and bottom of cylinders **108** and **109.** The orifices leading to conduits **105** and **106** are approximately in alignment with orifices leading from cold chambers **111** of by-pass devices of the type described with regard to FIG. **1.** The bottom and top of cylinders **108** and **109** are connected via conduits **113** and **114,** which extend into the cylinders by an amount necessary to establish stops for free oscillating pistons **115** and **116.** Conduits **113** and **114** are connected via a T to conduit **217** that is in turn connected to high pressure gas source **104** via solenoid controlled valve **117.** Solenoid controlled valve **117** is normally closed, and is actuated to an open condition for a relatively short predetermined length of time in response to a pulse fed thereto by electrical network **118.** Electrical network **118** is responsive to signals derived by proximity sensors **119** and **120** at the bottom and top of cylinders **108** and **109,** respectively. Sensors **119** and **120** may be of any conventional type, e.g., capacitive, that derive output signals indicative of displacement between the pistons and sensors.

The electrical circuit includes a pair of differentiators, each responsive to one of sensors **119** and **120.** The differentiators derive output pulses when pistons **115** and **116** come closest to the proximate ends of cylinders **108** and **109** and begin moving away from the ends of the cylinders.

If pistons **115** and **116** are perfectly synchronized, the output signals of the two differentiators occur simultaneously and are combined to derive an electrical pulse. The pulse is fed to valve **117** to open the valve momentarily to enable a pressure pulse to be fed from chamber **104** through conduits **113** and **114** to maintain pistons **115** and **116** in oscillation. The pressure pulse occurs at a time while pistons **115** and **116** are both moving away from the bottom and top of cylinders **108** and **109** by virtue of the gas springs at the cylinder ends. The pressure pulse is timed to add drive force to the pistons in such a manner that synchronized oscillation is sustained in response thereto.

If pistons **115** and **116** are not perfectly synchronized, the instants of maximum proximity sensed by sensors **119** and **120** do not occur simultaneously. Circuit **118** is provided with a means e.g., an AND gate responsive to signals from the differentiators indicating both pistons are moving away from sensors **119** and **120,** for enabling only the last occurring pulse derived by the differentiators to be utilized to open valve **117.** The pressure pulse fed by valve **117** to line **217** has a greater effect on the lagging piston. This is because the lagging piston is in a smaller volume than the faster piston and velocity of the slower piston is less than the faster piston, to provide higher fluid pressure for the pulse. The pressure pulse fed by line **217** to conduits **113** and **114** can be heated by feeding the conduit **217** through apertures provided in a regenerator **123** and heater **112** so that it supplies added energy to the piston and higher efficiency is attained.

Chamber **104** is pressurized by gas fed back from or diverted from load **204** while piston **116** is coasting downwardly in the cylinder **109,** i.e., during the power stroke of piston **116.** The gas fed back from the load is fed through check valve **124,** poled in such a manner as to permit gas to flow from conduits **105** and **106** into chamber **104,** but not vice versa. Since the maximum pressure of the hot ends of cylinders **108** and **109** is typically less, during pneumatic spring action, than the pressure in storage chamber **104,** while the maximum spring pressure in the cold ends would be greater than pressure in chamber **104,** the hot cylinder ends are made common and used for synchronizing rather than utilizing the cold cylinder ends for this purpose as was illustrated previously in the embodiments of FIGS. **3-5.**

In the embodiment of FIG. **6,** there may be greater efficiency than in the previously described embodiments because of the more efficient structure utilized for feeding added energy to the pistons to maintain them in an oscillatory state. Greater efficiency may result since the pressure in chamber **104** is developed from the regenerative gas cycle, i.e., feeding gas

through regenerators 123 instead of from a thermal lag type heater. The system of FIG. 6 has particular advantage with devices having greater power output than the previously described systems, where greater thermodynamic efficiency is required and the additional moving parts required to attain the added efficiency do not detract from overall performance of the device.

Reference is now made to FIG. 7 wherein the pump of FIG. 6 is modified to eliminate the requirement for an electrical network and a pneumatic device is employed to control the pulses of pressurized gas which maintain pistons 131 and 132 in synchronous oscillation in cylinders 133 and 134. Pressure pulses from storage chamber 104 are periodically applied, once during each cycle of oscillation of the slowest one of pistons 131 or 132, to the bottom and top of cylinders 133 and 134 via flow limiter 135, series connected spring loaded cup shaped valves 136 and 137, and conduits 138, 139 and 140. Valves 136 and 137 are located at sealed ends of tubes 141 and 142, the other ends of which extend into the bottom and top of cylinders 133 and 134 to form stops for pistons 131 and 132 for easier starting of the device. Pistons 131 and 132 are provided with cylindrical appendages 144 and 145 which, during a small portion of the cycle, extend into tubes 141 and 142. Valves 136 and 137 are pulled against the closed ends of tubes 141 and 142 by virure of the spring tension and normally closed orifices in tubes 141 and 142 leading to conduits 138, 139 and 140. To open valves 136 and 137 in response to movement of pistons 131 and 132, tubes 141 and 142 are connected to the ends of cylinders 133 and 134 via check valves 146 and 147. Orifices in tubes 141 and 142 that communicate with check valves 146 and 147 are positioned so that they are near valves 136 and 137 but are not blocked while valves 136 and 137 are closed.

In operation, valves 136 and 137 are normally closed. As appendages 144 and 145 enter tubes 141 and 142, they compress gases in the tubes to a greater extent than the gases at the bottom and top of cylinders 133 and 134. Thereby, valves 146 and 147 open, tending to equalize the pressures in tube 141 and cylinder 133, as well as the pressures in tube 142 and cylinder 134. Due to the pressure in tubes 141 and 142 as well as cylinders 133 and 134, the velocity of pistons 131 and 132 drops to zero and, due to the gaseous spring effects, the pistons begin thereafter to move away from the central portion of the assembly. In response to movement of pistons 131 and 132 away from the central portion of the assembly and while appendages 144 and 145 are in tubes 141 and 142, the pressure in tubes 141 and 142 is reduced and a suction force is exerted on valves 136 and 137 to open the valves and allow gas to flow through them. Valves 136 and 137 stay open until appendages 144 and 145 are withdrawn from tubes 141 and 142, at which time gases from cylinders 133 and 134 flow into tubes 141 and 142 and valves 136 and 137 are closed, because of their spring tension.

If pistons 131 and 132 are synchronized, valves 136 and 137 open and close simultaneously. In response to valves 136 and 137 both being open, gas flows from pressurized chamber 104 through flow limitor 135, conduits 138–140, and valves 136 and 137 to cylinders 133 and 134. The response of valves 136 and 137 only to suction and the resulting gas flow are such that gas is not delivered into cylinders 133 and 134 by conduit 140 until both pistons 101 and 132 have begun to move away from the center of the assembly. Thereby, the gas flow assists the gaseous spring effect to sustain oscillation. Gas continues to flow through conduit 140 into cylinders 133 and 134 until one of appendages 144 and 145 is out of tube 141 or 142. Pistons 131 and 132 cannot be overdriven, despite the relatively long time interval during which drive pressure is applied to them by conduit 140 by proper overall design.

If pistons 131 and 132 are out of synchronism, whereby one piston begins to move away from the center of the assembly before the other, gas does not flow into cylinders 133 and 134 via conduit 140 until the lagging piston has begun to move away from the center. Gas flow is prevented until this time because of the series connection of valves 136 and 137. The gas flow into cylinders 133 and 134 has the greatest effect on the lagging piston because of the lower velocity of the lagging piston, and the lesser volume between the end of the cylinder nearest conduit 140 and the closer face of the lagging piston contributes to a slower flow rate of gas into this lagging chamber than the leading chamber. Therefore, a pressure differential along the conduit 140 results in a greater pressure in the lagging chamber than the leading chamber, so that the lagging piston receives a pneumatic impulse, which is relatively greater as well as earlier in its cycle, thereby tending to maintain synchronism.

Reference is now made to FIG. 8 of the drawing wherein there is illustrated an oscillatory gas pump 311 which feeds oscillatory pressure variations to power converting alternator 312, which in turn supplies alternating electric current to an electric load 313. Separate consideration will be given to pump 311 and alternator 312, both of which are free oscillating piston devices.

Pump 311 includes a pair of sealed cylinders 314 and 315, each respectively including free pistons 316 and 317. The ends of cylinders 314 and 315 in closest proximity to each other are interconnected with each other via conduit 318. Conduit 318 includes a T connection, to which is connected conduit 319 that leads to a thermal lag device comprising passageways 320 and heat source 321. The thermal lag device applies increased pressure to pistons 316 and 317 as the pistons are moving simultaneously away from conduit 318. Thereby, pistons 316 and 317 are operated synchronously so there is virtually no vibration or torque applied to pump 311 as a result of piston oscillation.

Cylinders 314 and 315 are respectively provided with variable length by-passes 323 and 324. In each of by-passes 323 and 324 there is included in seriatim a heating chamber, a regenerator chamber, and a cooling chamber. Cooling chambers 325 and 326 are located in by-passes 323 and 324, respectively, at the ends of the by-passes closest to conduit 318, while heating chambers 327 and 328 are located in the by-passes at the ends of the by-passes closest to the ends of cylinders 314 and 315 most remote from conduit 318. In by-pass 323, between cooling chamber 325 and heating chamber 327, is regenerator chamber 329, while regenerator chamber 331 is located in by-pass 324 between cooling chamber 326 and heating chamber 328.

To control the lengths of by-passes 323 and 324, the ends of the by-passes closest to conduit 318 are connected through multiport valves 332 and 333 to the cylinder walls near the ends of cylinders 314 and 315 closest to conduit 318. Valves 332 and 333 include ports 334 and 335 respectively connected to cold chambers

325 and 326. Valve 332 includes five additional ports 336–340 having connections to cylinder 314 at different locations along the length of the cylinder. The distance between port 336 and the other end of by-pass 323, the end connecting hot chamber 327 to cylinder 314 is great enough to position the center of piston oscillation and to provide the minimum desired pumping power. During normal operation this distance is equal to or greater than the length of piston 316 to enable piston 316 to coast through a portion of the cylinder by-pass region. In certain instances, however, the distance between port 336 and the other end of by-pass 323 may be less than the length of piston 316, if it is desired to minimize coupling of pressure variations from pump 311 to alternator 312. Valve 332 is constructed so that a flow path is established from inlet port 334 to only one or two of ports 336–340 at a time, whereby adjustment of valve 332 effectively changes the length and position of by-pass 323. Changing the length of by-pass 323 alters the magnitude and time duration of the pressure variations supplied by pump 311 to alternator 312, thereby to modulate the amount of voltage or power supplied by the alternator to load 313.

Valve 333 controls the length of by-pass 324 for cylinder 315 in exactly the same manner that valve 332 controls the length of by-pass 323 for cylinder 314. To enable pistons 316 and 317 to be maintained in synchronism and approximately the same amount of gas pressure to be supplied by each of cylinders 314 and 315 to alternator 312, valves 332 and 333 are generally adjusted in a like manner so that the lengths of by-passes 323 and 324 are always substantially the same.

Oscillatory pressure variations from cylinders 314 and 315 are supplied to alternator 312 via multiport valves 342 and 343, having ports connected to alternator 312 via conduits 344, 345 and 346. Multiport valves 342 and 343 are identical with valves 332 and 333 and include five ports having connections to different positions along the length of cylinders 314 and 315 opposite from the inlet ports to valves 332 and 333 from the cylinder walls. The positions of valves 342 and 343 are generally controlled in synchronism with those of valves 332 and 333 so that gas communicates through corresponding ports to the cylinders; thus, in general, all of the valves simultaneously have connections to cylinders 314 and 315 through the ports equidistant from conduit 318. By synchronizing the positions of valves 332, 333, 342 and 343 gas is efficiently transferred from the ends of cylinders 314 and 315 proximate conduit 318 to alternator 312 since the conduits leading to the alternator are opened and closed at the same time as the by-pass. To enable the oscillatory pressure variations derived from cylinders 314 and 315 to be applied in synchronism to alternator 312, the lengths of the flow paths through valves 342 and 343 and conduits 345 and 346 are maintained substantially the same. In theory, however, an alternator or other load may be connected to any portion of the pump.

The alternator 312 and conduits 344, 345 and 346 may be cooled to improve the efficiency of the alternator and pump. In an alternative embodiment, not shown, the valves 342 and 343 may be eliminated, and the conduits 345 and 346 connected directly to the conduits which connect cold chambers 325 and 326 to ports 334 and 335 in valves 332 and 333. As an additional alternative, the axial location of the hot end of the by-pass region may be varied by means of a valve and porting arrangement. The axial location of the hot end may be varied simultaneously with, or instead of, the valve and porting arrangement at the cool end of the by-pass. Further, the pump inlet/outlet may be connected to the hot, rather than cool, end of the by-pass region, for pumping hot gas. Also, by combining certain functions, some of the duplication inherent in the two by-passes, and the two load connections, may be eliminated, e.g., the functions of valves 332 and 333 may be performed by a single valve, or the heating, cooling, or regenerator chambers and/or by-passes may be combined.

To prevent pistons 316 and 317 from contacting end faces of cylinders 314 and 315, and for easier starting, the cylinder end faces are provided with resilient stops 356–359. Stops 356–359 are normally inoperative during a typical oscillatory cycle of pistons 316 and 317 since the pistons are prevented from reaching the ends of cylinders 314 and 315 by pneumatic springs caused by gases in the cylinders.

After starting, pistons 316 and 317 are driven in synchronism, so that both approach conduit 318 at approximately the same time and both recede from the conduit simultaneously. Thereby, gas is supplied by cylinders 314 and 315 to conduits 345 and 346 at substantially equal rates. These in-phase gas flows are combined in conduit 344.

The thermally induced change in the pressure of gases in chambers 361 and 363 while piston 316 is coasting in the by-pass region essentially provides the pumping power. The pressure change during each cycle is dependent upon the amount of gas heated and cooled in the by-pass and therefore upon the length of the by-passed region of the cylinder. In particular, if the length of by-pass 323 is a minimum, whereby port 336 is connected through valve 332 to orifice 334, the pump output power is a minimum, and may be substantially zero if there is no coasting of piston 316. The pressure variations in chambers 361 and 363, and therefore the pump output power available, are maximized when the length of by-pass 323 is greatest, as occurs by connecting port 340 through valve 332 to orifice 334. The pressure changes in chambers 361 and 363 are dependent upon the amount of heating and cooling of the gas in the by-pass, and therefore the length of the coasting region. The fraction of the cycle during which pneumatic power is applied to a load also increases with the length of the by-pass, and this contributes to the increase in output power when the coasting region is lengthened by means of the valves 332 and 342. Power and frequency are also affected by a change in mean position of the by-pass, partly due to its effect on average gas temperature and therefore pressure. Independent control of by-pass length and position can be used to obtain independent control of power and frequency. In particular, when the length of by-pass 323 and therefore the piston coasting region is minimized, whereby piston 316 blocks the flow of gas from chamber 363 to valve 342 for a relatively long time interval during each oscillatory piston cycle, pressure variations are coupled from chamber 363 through valve 342 to conduit 345 for only a relatively short time interval. In contrast, when valves 332 and 342 are arranged so that the by-pass length is maximized, pressure is supplied from chamber 363 to conduit 345 for a relatively long time interval.

In general, in order to reduce susceptibility to stalling, the dimensions of the various pump embodiments are such the piston never completely traverses a load port during rebounding. Thus, in FIG. **8**, the upper face of piston **316** never goes below the load port selected by valve **342**, except for special applications. This decoupling of the load from the device during this piston rebound reduces susceptibility of the free piston pump to stalling (in addition to balancing of the load pressures on the two piston faces during coasting).

The oscillatory pressure variations derived by free piston pump **311** are coupled to alternator **312** through conduit **344.** Alternator **312** includes a pair of high magnetic permeability (e.g., ferromagnetic) free pistons **371** and **372,** respectively located in cylinders **373** and **374.** Cylinders **373** and **374** are connected by conduits **375** and **376** to conduit **344** so that the periodic pressure variations in conduit **344** tend to produce synchronous oscillation of pistons **371** and **372.**

Pistons **371** and **372** are part of magnetic circuits **377** and **378,** respectively. Magnetic circuits **377** and **378** include ring shaped permanent magnets **379** and **380,** having pole faces fixedly positioned along the length of cylinders **373** and **374.** Positioned in a hollow central portion of magnets **379** and **380** are circularly wound coils **382** and **383** in which are developed a.c. voltages in response to the oscillatory motion of pistons **371** and **372** varying the flux in the magnetic circuits in an oscillatory manner. To prevent magnetic short circuits between the poles of magnets **379** and **380,** pistons **371** and **372** are provided with ring-like cutout segments in which are mounted rings **384** and **385** of substantially nonmagnetic material. To provide good efficiency in generating a.c. voltages in coils **382** and **383,** the lengths of rings **384** and **385** along the axis of cylinders **373** and **374** are approximately equal to the minimum distance between the north and south poles of the magnets.

To enable the center of oscillation of pistons **371** and **372** to be fixed, by-pass regions are provided in cylinders **373** and **374.** The by-passes may have a length equal to, greater than or less than the length of pistons **371** and **372.** The by-passes may take the form of a passage external to cylinders **373** and **374,** as disclosed in conjunction with by-passes **323** and **324,** or they may be wholly within the cylinders, as described in my copending application, Ser. No. 121,371.

In the specific embodiment illustrated in FIG. **8,** the by-pass of cylinder **373** is provided by axial grooves **386** in the cylinder wall portion surrounded by coil **382.** Axial grooves **387** in the sides of piston **371** near its ends shorten the effective, or sealing length of the piston. Grooves **387** are long enough to allow gas flow in the by-pass sufficient to maintain piston position under conditions of variable system pressure or other forces on the piston, but do not extend over the entire length of the piston in order to maintain a reasonably good seal between the circumference of piston **371** and the walls of cylinder **373** beyond the by-pass.

In response to synchronized, oscillatory motion of free pistons **371** and **372** resulting from the periodic pressure variations in conduit **344** and the pneumatic springs at the remote ends of cylinders **373** and **374,** currents of like phase are induced in coils **382** and **383.** The voltages generated in coils **382** and **383** can be supplied to separate loads, or can be supplied to the same load **313** in an additive manner.

Linear alternator magnetic circuit designs other than that of magnetic circuits **377** and **378** can be employed in alternator **312.** If desired for certain designs, oscillation of the alternator free pistons can extend outside of the by-pass region. The extent of travel of the free pistons in general depends on the load, the design of the pistons, cylinders and by-passes, as well as the amplitude of pressure variations supplied to the alternator by the external pressure source, pump **312.**

To prevent damage to cylinders **373** and **374** and pistons **371** and **372** and for easier starting, the cylinders are provided with resilient stops **391–394** on their top and bottom faces. Normally, pistons **371** and **372** do not engage the stops, which are provided primarily as a safety means and for easier starting.

While free piston pump **311** is shown as a source for feeding oscillatory pressure pulses into conduit **344,** it is to be understood that other oscillatory pressure sources can be employed.

In many instances, it is necessary to modulate the amount of power derived from alternator **312,** which functions as a pneumatic to electric power converter. Changes in the amplitude and frequency of pressure variations fed to alternator **312** via conduit **344,** cause the length of the stroke and frequency of pistons **371** and **372** to be varied. In accordance with another aspect of the present invention, the pressure variation amplitudes in conduit **344** are controlled in response to changes in load **313.** To this end, the power supplied to load **313** by coils **382** and **383** is monitored by any suitable means (not shown), such as voltage and frequency comparators. If the voltage across load **313** is above or below a predetermined value necessary to power the load, the amount of power derived from the pump **311** is varied by controlling the length and/or position of by-passes **323** and **324** for cylinders **314** and **315.** The lengths of the by-passes are varied by controlling the positions of valves **332** and **333, 342** and **343,** which are preferably ganged together by mechanical connection **395.** Mechanical connection **395** can be responsive to an operator manually monitoring the voltage or a servo mechanism can be provided that is responsive to the deviation between the load voltage and a target value therefor and automatically operates the valves. Similarly, frequency may be simultaneously controlled.

In response to changes in the load, the lengths of by-passes **323** and **324,** or in gas average temperature, there may be a tendency for the frequency of oscillation of pistons **316, 317, 371** and **372** to change. The effect on frequency of a change in by-pass length is compensated to some degree by the fact that a change in pump pressure differential increases the pneumatic spring constant at one end of the cylinder and decreases the pneumatic spring constant at the other end of the cylinder. In addition, the change in coasting time is compensated to a degree by the change in cold pneumatic spring constant in chamber **363** resulting from the change in volume of the cold reversing chamber when the by-pass length is altered.

In certain instances, however, changes in the average temperature and therefore pressure of gas in the system resulting from changes in the mean position of by-passes **323** and **324** when the lengths of these by-passes are varied may materially change the frequency of oscillation of the pistons. To enable the frequency of oscillation to be maintained constant despite changes in load, thermal input, by-pass length, or other factors,

the by-pass mean position may be varied independently of by-pass length to control the average pressure of working gas and the values of pneumatic spring constants. By-pass position and length may be controlled independently by using a multiport valve at the hot end of each by-pass in addition to valves **332** and **333.**

Freqeuncy may also be varied in the embodiment of FIG. **8** by changing the gas volume of conduit **344** and therefore the density of working gas and values of the spring constants. To this end, there is provided a bellows **396** which is connected to an intersection between conduits **344, 375** and **376.** Bellows **396** is mechanically driven by rack and pinion **397** and **398,** the former being fixedly connected to one face of the bellows. Pinion **398** is driven by comparing a characteristic(s) of load **313,** e.g., the frequency and/or voltage across load **313,** as derived by a suitable meter(s) (not shown) with a predetermined value therefor. In response to the load characteristic(s) differing from the predetermined value, pinion **398** is driven by a manual control or in response to a signal automatically derived from a servo mechanism responsive to a comparison of the load frequency and desired frequency. Changes in average pressure within pump **311** and alternator **312** also affect the amount of power derived from the pump and alternator. Conversely, changes in by-pass length affect the frequency of the pump and alternator. By using a dual feedback arrangement the frequency and voltage supplied to the load by the pump alternator combination can be maintained relatively constant under variable load conditions. It is also possible, in order to lessen the temporary displacements of the centers of oscillation of the alternator pistons as a result of a rapid change in system pressure, to operate simultaneously three bellows, rather than one, for correcting voltage and frequency. The second and third bellows would be relatively small and be connected to the top and bottom of cylinders **373** and **374.** Thus, average system pressure can be changed approximately equally on both sides of each alternator piston, to help maintain their centers of oscillation during conditions of variable load, and decrease the required lengths of alternator cylinder by-passes. Further, under certain conditions, control of the position of one or both ends of the by-pass in the pump, as well as heat exchanger temperatures, can be employed to maintain proper voltage and frequency under conditions of variable load, without use of a bellows. Alternatively, the use of bellows or other means of varying system pressure, coupled with control of the heat exchanger temperatures, may be sufficient for controlling frequency and voltage without varying the length and position of the pump by-passes. In addition, performance may be controlled by varying the rebound chamber volumes by means of adjustable pistons or bellows at the ends of the cylinders, thereby affecting penumatic spring constants as well as overall pressure in the system. Also, the impedance of the pneumatic connection between the pump and alternator can be controlled, as by means of a valve, to vary the power and frequency. Or, means of varying the heater, regenerator, and/or cooler capacities in the by-pass can be used to vary power. Still other means for controlling pump output power and frequency are described below. The variable by-pass length has good heat economy at low, as well as high, power levels, and its stored thermal energy is avialable for rapid power increase, but this control technique is more complex than those described below.

An alternative method of starting the pump driven alternator **312** is to apply electrical power, of proper waveshape, to the alternator so that its pistons will start those of the pump.

In certain instances, an oscillating free piston pump has an inlet and outlet which are partially or completely separate from each other, and is used to pump gases from one point to another, as illustrated in FIG. **9.** In FIG. **9,** a free piston pump **401** includes a single cylinder **402** through which free piston **403** is translated in an oscillatory manner in response to pressure applied to the piston by thermal lag device **404,** located at one end of cylinder **402.** Variable length by-pass **405,** including hot chamber **406,** regenerator chamber **407,** and cold chamber **408,** is provided around piston **403.** An orifice at the bottom of chamber **408** is connected through valve **409** to one of orifices **411–415** spaced at different points along the length of cylinder **402.** Disposed in approximate alignment with orifices **411–415** are five orifices in the cylinder wall connected to valve **416** which includes an output orifice **417** that is connected to a suitable load. Preferably, orifice **417** is connected to the load through check valve **418.** Alternatively, valve **416** may be eliminated and orifice **417** connected directly to conduit **423,** or to any portion of the cylinder wall.

In the embodiment of FIG. **9,** gas is drawn into cylinder **402** from source **419.** Gas from source **419** is fed through cold chamber **421** and check valve **422** to conduit **423,** which is also connected to cold chamber **408** and valve **409.** Gas is supplied to conduit **423** from source **419,** after having been cooled in chamber **421,** during the portion of the oscillatory cycle of piston **403** when the pressure in conduit **423** is less than the pressure of source **419,** i.e., primarily when piston **403** is travelling upwardly in the by-pass region. Thereby, cold gas from chamber **421** mixes with cold gas flowing from chamber **408** into conduit **423** and the mixture is fed into the portion of cylinder **402** below the lower face of piston **403.** When piston **403** is coasting in a downward direction in the by-pass region of cylinder **402** the pressure in conduit **423** becomes greater than the pressure of source **419,** whereby check valve **422** is closed. By feeding cold, and therefore dense, gas from source **419** and cold chamber **421** into cylinder **402** through conduit **423** while piston **403** is coasting upwardly in the by-pass region, the amount of gas drawn into the cylinder for pumping, and the differential temperature of gases in the cylinder over a cycle are increased, to increase the efficiency of the pump.

Check valve **418** supplies gas from cylinder **402** to a load primarily while piston **403** is coasting in a downward direction through the by-pass region of cylinder **402.** With piston **403** coasting in cylinder **402,** the pressure of the gas supplied through valve **416** to the check valve **418** generally becomes greater than the load pressure, and only then is the load responsive to the pressurized gas in the portion of cylinder **402** below the lower face of piston **403** because check valve **418** oth-

erwise is closed by the higher pressure of the load. Alternatively, the pump outlet may be located near the top end of cylinder **402** for pumping hot gases.

The arrangement of FIG. **9** can be adopted to a synchronized, two free piston arrangement, as illustrated in FIG. **10.** The pump of FIG. **10** is essentially the same as pump **311** of FIG. **8,** except that gas from an external source **424** is fed through cold chamber **421** and check valve **422** to conduits **427** and **428,** which preferably are of equal length to connect the output orifice of the check valve **422** to orifices of regenerators **429** and **430.** Regenerators **429** and **430** are connected in bypasses **431** and **432** for cylinders **433** and **434.** Bypasses **431** and **432** respectively include heating chambers **435** and **436** which are connected to the top and bottom portions of cylinders **433** and **434.** The bypasses **431** and **432** are completed through variable position valves **437** and **438,** the functions of which may alternatively be performed by a single valve.

It is to be noted that check valve **422** is connected directly to regenerators **429** and **430,** without being connected to a cooling chamber in the by-pass. In many instances, the increased efficiency attained through the use of cooling chamber **421** obviates the requirement of cooling chambers in the by-passes **431** and **432.** In certain systems, regenerators **429** and **430** can be eliminated because of the inherent heat storage properties of conduits **427** and **428.** Similarly, in FIG. **9,** it is possible in certain instances to connect an orifice of hot chamber **406** directly to conduit **423,** thereby eliminating regenerator **407** and cold chamber **408.** In the alternative, it is possible in the system of FIG. **9** to eliminate cold chamber **408.** If a source of hot gas is available for pumping, the need for heating chambers in the bypasses of the pumps in FIGS. **9** and **10** may similarly be obviated.

Pressure variations in cylinders **433** and **434** are supplied to separate loads via output ports **439** and **440** and check valves **470** and **471,** or the ports may be connected to a common load via a T connection (not shown) so that in phase flows from cylinders **433** and **434** are combined. Ports **439** and **440** are positioned in the hot ends of cylinders **433** and **434** in axial alignment with ports from the hot ends of the cylinders to by-passes **431** and **432.** In the alternative, multiple output ports can be provided in the cold ends of cylinders **433** and **434,** as illustrated in FIGS. **8** and **9.**

In accordance with a further embodiment of the invention, illustrated in FIG. **11,** the thermal lag heater comprises blind passageways **441** located in free piston **422.** Passageways **441** extend longitudinally in piston **442** from the face of the piston positioned to be responsive to cold gas flowing into lower chamber **443** of cylinder **444** from by-pass **445** and from a load connected to cylinder port **458.** Passageways **441** extend through the length of piston **442** for a distance sufficient to provide adequate temperature and volume of the thermal lag chamber.

Preferably, in normal operation piston **442** is indirectly heated so that no additional heat source is required for the thermal lag device. To this end, the upper face of piston **442** forming an end wall of upper chamber **446** of cylinder **444** is heated by hot gas flowing out of by-pass **445** into upper cylinder chamber **446** via port **447.** The piston is also heated by the hot cylinder wall surfaces. Heat from the upper piston face is conducted through the piston to the walls of passageways **441** to heat cold gas fed into the bottom chamber **443** and thence into the passageways from by-pass **445** as the piston travels downwardly past bottom port **459** that connects the by-pass to cylinder **444.** The cold gas flows into passageways **441,** is heated therein and expands to flow out of the passageways as piston **442** begins its upward travel, thereby sustaining oscillation of the free piston.

Stops in chambers **443** and **446** are formed by compression spring means **448** and **449** fixedly mounted to extend longitudinally from the lower end face of piston **442** and the upper end face of cylinder **444.** Spring means **448** is a single compression spring fixedly mounted in short passageways **451** of piston **444.** Spring means **448** is preferably mounted on the piston, rather than the bottom end face of cylinder **444,** to maintain it at a higher temperature and to use the thermal lag effect from its hot surfaces and those of its mounting passageways to help sustain piston oscillation.

Spring means **449** comprises several metal compression springs positioned on the upper face of cylinder **444** to form thermal lag passageways for helping to sustain piston oscillation. To this end spring means **449** may comprise plural coaxial springs having different diameters or plural springs having different axes relative to the upper end face of cylinder **444.** Preferably springs **449** are indirectly heated by hot gas flowing from by-pass **445** and by radiant heating. The thermal lag device formed by the passageways of spring means **449** may also cool some of the hot gas in the cylinder as piston **442** is rebounding from the lower end face of cylinder **444.** Cooling of the hot gas during this period tends to decrease the pressure of the gas upon the upper piston face to assist in sustaining oscillation of piston **442.** While spring means **449** is preferably indirectly heated, it is to be understood, however, that spring means **449,** or other thermal lag means opening into the top of cylinder **444,** can be directly heated if necessary.

The structure of FIG. **11** can be incorporated in a device having two or more synchronized pistons by providing a cylinder for each piston and by connecting one chamber of each cylinder together. Preferably, chambers having the same configurations are interconnected and greater synchronization can be attained by interconnecting the chambers providing the greatest thermal lag forces. In some instances, however, to achieve synchronization, it may be necessary to utilize a common thermal lag chamber that is connected to one chamber of each cylinder. The single directly heated chamber could be employed in combination with separate thermal lag devices for each cylinder. For example, the directly heated thermal lag device of FIG. **8** could be provided between a pair of cylinders including free pistons and thermal lag springs illustrated in FIG. **11.**

The by-pass **445** illustrated in FIG. **11** is of a form which, although functionally similar, is structurally different from that suggested in the other embodiments. In particular, by-pass **445** includes concentric chambers **452, 453** and **454** forming the hot, regenerator and cold chambers. Inner, hot chamber **452** surrounds centrally located heater **455,** while outer, cold chamber **454** includes cooling fins **456** on its exterior surface. Gas flows axially and/or radially within chambers **452–454,** through passageways in the chambers. Gas

also flows between hot chamber **452** and cylinder chamber **446** via port **447**, as well as between cold chamber **454** and cylinder chamber **443** via port **459**.

Oscillatory gas pressure variations derived by the pump of FIG. **11** can be supplied to a load via conduit **457** connected to cold chamber **454**. If, however, there is a significant load, this connection might stall piston **442** since the piston is connected to the load through port **447** and by-pass **445** while the piston is rebounding near the bottom face of cylinder **444**. The possibility of stall can be substantially eliminated by connecting the load to port **458**, in the side wall of cylinder **444** opposite port **459**. Port **458** is not in by-pass **445** so that the load is decoupled from the piston while the piston rebounds from the bottom face of cylinder **444**.

The thermal lag device formed by spring means **449** operates in conjunction with load variations to help sustain oscillation of piston **442**. If there is a heavy load, e.g., the atmosphere in a very extreme case, there is little gas flow through by-pass **445** so that the by-pass pressure drop is minimized and there is minimum load on piston **442**. In response to the atmospheric load, pressure variations and resulting adiabatic temperature variations in chamber **446** over a cycle of piston oscillation are minimized and minimum or possibly negative rebound energy is applied to the piston from the thermal lag passageways formed by spring means **449**. Minimum or negative force from these thermal lag passageways compensates for the light load on the piston to prevent excessive oscillation. If there is a light load, e.g., the outlet port to the load is closed in the extreme case, there is appreciable flow through by-pass **445** so that the by-pass pressure drop is maximized and there is maximum load on piston **442**. In consequence of the greater pressure variations and adiabatically induced temperature variations in chamber **446** over a cycle of piston oscillation in this case, a greater rebound energy is derived from the thermal lag device formed by spring means **449**. Greater force from this thermal lag device compensates for the heavy load imposed on the piston by by-pass **445** to sustain piston oscillation.

To enable oscillation of piston **442** to be sustained upon starting and prior to the piston being heated sufficiently by hot gas from by-pass **454** to establish a sufficient temperature in the thermal lag passageways **441**, a directly heated thermal lag device **461** is provided. Thermal lag device **461** includes passageways **462** that are responsive to heat from source **455** and possibly hot chamber **452** and regenerator chamber **453**. Added heat is supplied to thermal lag device **461** by providing the device with a central indention that receives the lower end of heat source **455** during starting. Passageways **462** are connected to the lower end of cylinder **444** via conduit **463** and valve **464**. Valve **464** is open at the beginning of the operation of piston **442** until the piston is heated sufficiently by hot gases from by-pass **445** to establish the thermal lag device in the piston. During this period thermal lag heater **461** functions as the thermal lag heaters described in conjunction with FIGS. **8-10**. After piston **442** has been heated sufficiently valve **464** may be closed and heater **455** is moved upwardly out of the indentation of thermal lag device **461** so that less heat is withdrawn from by-pass **445** by passageways **462** and thermal lag device **461** becomes inoperative. This movement of heater **455** may increase efficiency of the device.

Reference is now made to FIG. **12** of the drawing wherein the by-pass of FIG. **1** is modified to include a shunt path **501** around the hot chamber **21** and a variable ratio three-way valve **505** for shunting a variable fraction of gas around the hot chamber, and further includes a similar valve **506** and shunt path **502** for diverging a selected fraction of gas around the cold chamber **23**. In this way the amount of heating and the amount of cooling of gases flowing in the by-pass are each controlled to modulate differential and average temperature and pressure during the cycle and therefore the amplitude and frequency of pressure variations derived from the pump. The variable flow shunt paths comprise conduits **501** and **502**, respectively connected between ports **503** and **504** of three-way valves **505** and **506** and ports **507** and **508**. Ports **507** and **508** are respectively provided at the intersections between regenerator **22** and each of hot chamber **21** and cold chamber **23**. Ports **511** and **512** of valves **505** and **506** are connected to hot and cold chambers **21** and **23**, while ports **513** and **514** are connected to chambers **13** and **14**. Alternatively, ports **507** and **508** could each be connected directly to the cylinder wall by means of a shunt path and separate valve for each. The axial location of the entry points into the cylinder of the by-pass and shunt path need not then be identical.

By controlling the positions of valves **505** and **506** independently, the average gas temperature and pressure can be controlled independently of the differential temperature and pressure occurring during the cycle, so that the frequency and amplitude are independently controlled. For example, the pressures in chambers **13** and **14** during piston rebound affect the spring constants of these rebound chambers, which affect piston frequency. The temperature of the gas in chamber **14** relative to the temperature of the heated passageways of thermal lag chamber **10** affects the kinetic energy supplied to the piston each cycle by the heating of cool gas in chamber **10**, and this also affects the piston frequency. Valves **505** and **506**, by controlling the amount of heating and cooling, can be used to control these pressures and the relative temperature, as well as the change in pressure during coasting, and thus the amplitude of the pump output. Thus, if this pump were driving the free piston linear alternator described above, the two valves could each be servo operated to maintain a relatively constant voltage and frequency under conditions of rapidly varying electrical load.

In the embodiment of FIG. **13**, the amount of heating and cooling of gas in the by-pass to cylinder **11** is controlled by including a variable gas flow path **521**, including regenerator **522**, which shunts the hot, regenerator and cold chambers **21**, **22** and **23**. Synchronously driven three-way variable ratio valves **523** and **524**, constructed identically to valves **505** and **506**, can thus vary the power and frequency of the device. To maximize the temperature difference over a cycle of the hot and cold gases, whereby maximum output power is attained, valves **523** and **524** are positioned so that there is no flow through the valves to regenerator **522** and all of the flow is through chambers **21**, **22** and **23**. In the other extreme, substantially zero temperature difference between hot and cold gases (essentially zero output power) occurs if valves **523** and **524** are positioned so no gas flows through chambers **21–23**. One of valves **523** or **524** could be replaced with a T connection with the deleterious effect, under maximum output power,

of wasting energy to fill a gas dead space comprising most of shunt path **521.** The modulation systems of FIGS. **12** and **13** may not be quite as efficient as the variable length and position by-pass system of, e.g., FIG. **8,** but should be easier to construct and operate.

Reference is now made to FIG. **14** of the drawing wherein modulation can be achieved by varying one or more properties of the thermal lag heaters which include hot passageways **541** in free piston **542** and hot chamber **543** near the end of cylinder **544.**

To control the portion of the oscillatory cycle over which thermal lag passageways **541** are operative plug **545** is mounted on fixed end wall **546** of cylinder **544** so that it is axially aligned with and can extend into central passageway **547** of piston **542.** The axial position of plug **545** is controlled by any suitable means, such as sealing bellows **548** that is secured to the end wall and is controllable during operation, possibly by automatic servo means. Plug **545** has a snug fit with passageway **547** while piston **542** is at the lower end of its travel to prevent gas flow from passageways **541** to cylinder **544.** By raising and lowering plug **545** control is obtained over the portion of each oscillatory cycle of piston **542** over which the thermal lag device comprising passageways **541** is operative. This is one means of varying piston amplitude and frequency, and therefore pump pressure amplitude and frequency.

An alternative or supplementary means for controlling the thermal lag device involves varying the geometry of thermal lag chamber **543** that is heated by heater **549.** The geometry of chamber **543** is varied by forming a portion of its walls as bellows **551** that may have sufficient stiffness and a position keeper (not shown) to prevent expansion and contraction thereof in response to pressure variations, to enable the geometry of chamber **543** to be fixed by an operator or servocontroller. By expanding and contracting bellows **551** the volume of chamber **543** is altered, as are the thermal lag characteristics of passageways formed by the folds of the bellows. By changing the volume of chamber **543** and the relative length to width relation of passageways formed by bellows **551** the time and perhaps amplitude of the maximum pressure variations supplied to piston **542** by gases escaping from chamber **543** are altered. The thermal lag effect of chamber **543** can also be changed by providing a multiport valve, similar to valve **332,** FIG. **8,** between a port of the chamber and different points along the length of cylinder **544.** Varying the multiport valve changes the portion of the cycle over which the thermal lag heater comprising chamber **543** is operative.

In FIG. **14,** there is also shown a modified arrangement for shunting gases around hot and cold chambers **552** and **553** that are included in by-pass **554,** which also contains regenerator **555.** Hot and cold chambers **552** and **553** are variably shunted by gas flow paths comprising pipes **556** and **557,** respectively. Three way valves **558** and **559,** preferably having synchronized movement, connect opposite ends of hot chamber **552** and pipe **556** to a by-pass port located in the side wall of the upper chamber of cylinder **544** and to regenerator **555.** Three way valves **561** and **562,** preferably having synchronized movement independent of the movement of valves **558** and **559,** respectively interconnect regenerator **555** and the other by-pass port of cylinder **544** to opposite ends of pipe **557** and cold chamber **553.** Two three way valves are provided for each of shunt gas flow paths **556** and **557** to prevent dead spaces in chambers **552** and **553;** if desired, however, one of the three way valves in one or both shunt paths can be eliminated. The amount of heating and the amount of cooling of gas flowing in the by-pass are thereby independently controlled in the by-pass arrangement of FIG. **14.** The three way variable ratio valves **558, 559, 561,** and **562** facilitate a modulation effect similar to that attained with the embodiment of FIG. **12.**

Reference is now made to FIG. **15** of the drawing wherein the free piston pump of the present invention is thermally operated by two substances or bodies of differing temperatures which may be located remotely from the pump. The configuration also shows how the technique may be used as a means for cooling a hot body or heating a cold body. In particular, gas is pumped through or around a hot body **601** and also through or around a cold body **602.** Hot body **601** and cold body **602** transfer heat to and from gases pumped by free oscillating piston **603** via port **604** near the end of cylinder **605** coupled to thermal lag heater **606.** Port **604** is connected to bodies **601** and **602** via check valves **607** and **608.** Check valves **607** and **608** are positioned so that gas flows through them in response to the pressure at port **604** being greater than the pressures in the passageways through or around the hot and cold bodies.

Gas flowing through or around body **602,** after having been cooled by the body, is fed back via check valve **611** to cylinder port **609** that is approximately aligned with port **604.** Check valve **611** is positioned so that gas from cold body **602** is fed to port **609** in response to the pressure in the cold body passageways being greater than the pressure at port **609.** Gas from hot body **601,** after having been heated by the hot body, flows through heat exchange jacket **610** for thermal lag heater **606** and thence back into the cylinder via port **612** at the opposite end of cylinder **605** from ports **604** and **609.** Thermal lag heater **606** is thereby heated to a temperature sufficient to maintain oscillation of piston **603.** In approximate alignment with port **612** is cylinder port **614** that is connected to one end of regenerator **615,** the other end of which is connected to port **609.** In the alternative, the outlet of thermal lag jacket **610** can be connected to port **614** and port **612** can be eliminated, as indicated by the dotted line representation. By connecting the outlet of heat exchanger **610** to port **614** flow occurs in the heating path through hot body **610** while the piston is rebounding at the top of cylinder **605.**

The distance in cylinder **605** between aligned ports **604** and **609** and aligned ports **612** and **614** forms a by-pass region of the cylinder. The by-pass includes two parallel paths, namely a first path including regenerator **615,** and a heating path including check valve **607,** passageways in or around hot body **601,** and heat exchanger jacket **610.** A third, or cooling, path includes check valve **608,** passageways in or around cold body **602,** and check valve **611.** In operation, gas flows through regenerator **615** while piston **603** is coasting in either direction through the by-pass region. The heating and cooling paths are charged via check valves **607** and **608** while piston **603** coasts downwardly through the by-pass region in response to the increasing pressure in the cylinder and the relatively greater pressure in the bottom of the cylinder at this time resulting from

the downward motion of the piston. As piston **603** coasts downwardly through the by-pass region, the pressure at the outlet of check valve **611** is equal to, or possibly greater, than the pressure at its inlet, whereby there is no substantial gas flow from the cooling path. Due to the pressure differential across the piston **603,** gas flows through the heating path during this time. In response to piston **603** coasting upwardly through the by-pass region and the resulting drop in cylinder pressure, gas flows into the top and bottom of the cylinder via the heating and cooling paths.

It is thus seen that piston **603** is driven in response to hot and cold gases coupled into the upper and lower portions of cylinder **605** and in response to gases flowing through regenerator **615** in substantially the same manner as in the previously disclosed embodiments. It is to be understood that the principles of FIG. **15** are applicable to two or more cylinders having two or more synchronized free pistons. Also, the by-pass containing the regenerator may be eliminated. Further, to increase heat exchange rate, a surge tank may be added to the heating path just after check valve **607** and/or to the cooling path just after check valve **608.** In addition, the location of the ports connecting the hot and cold paths to the cylinder should be considered variable. Further, the cooling path and/or the heating path could originate at a port in the hot, rather than cold, end of the cylinder. Finally, flow limiter valves could be added anywhere in the lines to restrict or limit flow for adjustable or better performance.

The principles of the present invention are also applicable to a heat pumping or refrigeration cycle, as disclosed in the embodiment of FIG. **16.** In FIG. **16,** free piston **621** is driven in cylinder **622** in response to oscillatory pressure variations coupled via conduit **624** to the bottom end of the cylinder through port **623.** The source of oscillatory pressure variations could be a thermally driven pump, as disclosed in other embodiments herein or in my previously mentioned copending patent applications.

Cylinder **622** is provided with a by-pass **627** formed between ports **625** and **626.** By-pass **627** includes, in seriatim, cooled chamber **628,** regenerator chamber **629,** and heated chamber **630.** By-pass **627** is blocked in response to piston **621** blocking port **625** or **626,** while the volume of cylinder **622** below or above piston **621** is in a minimum range.

Chambers **628** and **630** are respectively cooled and heated in response to piston **621** being driven by the oscillatory pressure variations coupled through port **623** by a process substantially reversed during coasting from that described with regard to FIGS. **1** and **2.** The oscillatory cycle during coasting is substantially reversed in order to convert pneumatic energy from an external source into thermal energy rather than vice versa. By driving the system of FIG. **16** with a thermally driven pump, as described herein, an efficient refrigerator or heat pump should result because of the regenerative nature of the pump and refrigeration actuation cycles.

Port **623** could alternatively be located in the cylinder side wall between the bottom end of cylinder **622** and by-pass port **625,** for varying the rebound pressure and/or piston phase relative to that of the external oscillatory pressure source, to obtain modified performance characteristics.

Reference is now made to FIG. **17** of the drawing wherein there is illustrated a further embodiment of the present invention wherein oscillatory pressure variations from an external source are applied to the input of a "pressure transformer" in accordance with the present invention. The oscillatory pressure variations from the external, independent source are fed into closed cylinder **641** via port **642** located in the cylinder side wall between the bottom of the cylinder and the center of oscillation of free piston **643,** having side walls in sealing arrangement with the side walls of cylinder **641,** except possibly for a short distance along the cylinder side wall. The center of oscillation of free piston **643** is controlled by any suitable means, such as check valves or a by-pass **644** in the walls of cylinder **641.** By-pass **644** may be any convenient length, but is preferably shorter than the length of piston **643** to prevent substantial gas leakage around piston **643.**

A load can be provided at either or both ends of closed cylinder **641.** In FIG. **17,** the load is in the form of a bellows **645** sealingly connected to flange **646** at the upper edge of cylinder **641.** In the alternative, a check valve and flow limiter arrangement, as illustrated in FIG. **5,** could be located at either end of cylinder **641.** Or, the oscillating bellows or cylinder could be used as a vibrator or hammer.

In operation, relatively low pressure, high flow rate, periodic variations applied at port **642** are transformed and applied to a load as high differential pressure, low flow rate pressure variations. In response to the gaseous spring at the bottom of cylinder **641** and pressure of the external source being applied at port **642,** piston **643** is driven upwardly to its greatest extent, whereby the volume in cylinder **641** above the upper face of piston **643** is minimized and the gas pressure in this volume is maximized. In response to a reduction in the pressure of the external source and the gaseous spring at the top of cylinder **641,** piston **643** rebounds downward to again compress the gaseous spring at the bottom of the cylinder. Depending on design and operating conditions, the upper face of piston **643** may go below port **642** for a small fraction of the cycle, whereupon the varying pressure from the external source contributes to the oscillation during this short interval of time. Because the piston **643** oscillates between gaseous springs, the external source need only supply enough energy to make up for internal losses and energy applied to a load in order to maintain the oscillation. Under this oscillating condition wherein the pressure differential of gaseous springs adds to the pressure effects from the external source, the peak-to-peak pressure in either end of the cylinder may be greater than that of the external source, and the device acts as a pressure transformer.

While there have been described and illustrated several specific embodiments of the invention, it will be clear that variations in the details of the embodiments specifically illustrated and described may be made without departing from the true spirit and scope of the invention as defined in the appended claims. For example, certain liquids, e.g., water, can be mixed with the gases or liquid and its vapor can be employed to increase efficiency by cyclic evaporation and condensation of the liquid. In certain instances when liquids are employed, it would be desirable to maintain the average temperature of the device substantially constant so that the liquid vapor ratio averaged over the cycle re-

mains the same. Alternatively, the power output may be varied by varying the temperatures and thus the average or the instantaneous liquid vapor ratio. When using a liquid-vapor mixture in the apparatus of this invention, the change of phase which may occur in the by-pass and in the thermal lag heater produces a greater pressure variation, for a given mass of fluid, than if change of phase did not occur. Also, the by-pass could be formed within the oscillating piston itself. In such an instance, the opposite ends of the cylinder are supplied with external heating and cooling means as in FIGS. **4** and **5** and the oscillating piston contains the regenerator. A valving structure is connected to the oscillating piston to be responsive to the movement of the piston in the cylinder to enable hot and cold gases from opposite ends of the cylinder to be fed through the regenerator in the piston primarily during a central coasting region of the cylinder. The valving structure would function in the same time relationship as the movement of the piston relative to the conduits connecting the cylinder to the cold and hot chambers in the by-pass of the specifically illustrated embodiments, in order to reverse the piston outside of the coasting region. Also, if reduced efficiency can be tolerated, the regenerator chamber can be eliminated. Certain principles of the invention are applicable to systems wherein the free piston is reversed at one end of the cylinder by gravity, rather than with a gaseous spring. Gravity can be used as the reversing mechanism if the length of the by-pass is such that the kinetic energy imparted to the free piston at the bottom of the cylinder is reduced to zero prior to the piston coasting to the by-pass port at the top of the cylinder. Another possible variation to control piston amplitude or frequency is to alter the position of the connection of the thermal lag heater along the length of the cylinder by a valving device similar to that illustrated for the variable length by-pass. The modulation systems of FIGS. **12, 13** and **14** could be employed in a multi-piston, multi-cylinder pump of the type shown in FIG. **8** so that the shunt paths of the multiple cylinders are controlled in the same manner. Each of the embodiments illustrated herein may be modified using the principles disclosed to incorporate one or more pistons and cylinders.

I claim:

1. An oscillating piston apparatus comprising a cylinder, a free piston in said cylinder dividing the cylinder into first and second variable volumes, means for sustaining oscillation of the piston in the cylinder, a by-pass between the first and second volumes such that the piston coasts through a region of the cylinder between ends of the cylinder, a regenerator means in the by-pass, means for restricting the by-pass during the piston oscillation while at least one volume has a value in a minimum range, means for feeding cool fluid into the second volume, and means for feeding hot fluid into the first volume.

2. The apparatus of claim **1** wherein the by-pass includes a port in a side wall of the cylinder, said port being blocked by the piston while one of the volumes is in the minimum range to form the restricting means and said port being unblocked while the piston is coasting through the region.

3. The apparatus of claim **1** further including means for feeding fluid from one of said volumes to a load as the piston coasts through the region.

4. The apparatus of claim **1** wherein the means for sustaining oscillation comprises thermal lag means having an inlet communicating with one of the volumes while the one volume has values in a minimum range.

5. The apparatus of claim **1** wherein the means for sustaining oscillation includes a pressurized gas source means responsive to the piston substantially minimizing one volume for applying a pressure pulse from the source to the cylinder to drive the piston in a direction tending to increase said one volume as the piston is moving in a direction to increase said one volume.

6. The apparatus of claim **5** further including means for feeding pressurizing gas from one of the volumes to the pressurized gas source.

7. The apparatus of claim **1** wherein the means for feeding hot fluid includes a hot chamber in the by-pass.

8. The apparatus of claim **1** wherein the means for feeding cool fluid includes a cold chamber in the by-pass.

9. The apparatus of claim **1** wherein the means for feeding hot fluid comprises a hot chamber in the by-pass and the means for feeding cold fluid comprises a cold chamber in the by-pass.

10. The apparatus of claim **1** wherein the means for feeding hot fluid includes means for heating one of the volumes.

11. The apparatus of claim **1** wherein the means for feeding cold fluid includes means for cooling the other of said volumes.

12. The apparatus of claim **1** further including means for connecting a load to be responsive to fluid being decompressed as the piston coasts through the by-pass in a direction to decrease the first volume.

13. The apparatus of claim **1** further including means for connecting a load to be responsive to gas being compressed as the piston coasts through the by-pass in a direction to decrease the second volume.

14. The apparatus of claim **1** further including means for feeding gas pressure variations from one of said volumes to a load as the piston coasts through the region, and means for modulating a characteristic of the pressure variations.

15. The apparatus of claim **14** wherein the means for modulating includes means for controlling the average pressure of gas in the cylinder.

16. The apparatus of claim **14** wherein the means for modulating includes means responsive to a characteristic of the load for controlling the average pressure of gas in the cylinder.

17. The apparatus of claim **14** wherein the means for modulating includes means for controlling the amplitude of the pressure variations.

18. The apparatus of claim **14** wherein the means for modulating includes means responsive to a characteristic of the load for controlling the amplitude of the pressure variations.

19. The apparatus of claim **14** wherein the means for modulating includes means for controlling the frequency of the pressure variations.

20. The apparatus of claim **14** wherein the means for modulating includes means responsive to a characteristic of the load for controlling the frequency of the pressure variations.

21. The apparatus of claim **14** wherein the means for modulating includes means for controlling the amplitude and frequency of the pressure variations.

22. The apparatus of claim 14 wherein the means for modulating includes means responsive to a pair of characteristics of the load for controlling the amplitude and frequency of the pressure variations.

23. The apparatus of claim 14 wherein the means for modulating includes means for varying a property of the by-pass.

24. The apparatus of claim 14 wherein the means for modulating includes means for varying the length of the by-pass.

25. The apparatus of claim 14 wherein the means for modulating includes means for varying the position of the by-pass along the length of the cylinder.

26. The apparatus of claim 14 wherein the means for modulating includes means for varying the temperature of the fluid fed into one of the volumes.

27. The apparatus of claim 26 wherein the by-pass includes means for changing the temperature of fluid in the by-pass, said temperature changing means being connected with said regenerator means and means for controlling the relative fluid flow between the temperature changing means and the regenerator means.

28. The apparatus of claim 27 wherein the temperature changing means includes a heating chamber.

29. The apparatus of claim 27 wherein the temperature changing means includes a cooling chamber.

30. The apparatus of claim 27 wherein the temperature changing means includes means for heating the fluid in the by-pass, and means for cooling the fluid in the by-pass, said regenerator means including a regenerator chamber between the heating means and the cooling means.

31. The apparatus of claim 30 wherein the flow control means includes means for separately controlling the relative fluid flows through the heating means and the cooling means.

32. The apparatus of claim 27 wherein the regenerator means includes a first regenerator chamber connected in seriatim with the temperature changing means to form a first fluid flow path, and a second regenerator chamber connected in shunt with at least a portion of the first fluid flow path, said means for controlling including means for controlling the relative fluid flow between the second regenerator chamber and at least a portion of the first fluid flow path.

33. The apparatus of claim 27 wherein the regenerator means includes a regenerator chamber connected in seriatim with the temperature changing means, said means for controlling including a variable flow fluid path around the temperature changing means.

34. The apparatus of claim 14 wherein the means for sustaining oscillation comprises thermal lag means having an inlet communicating with one of the volumes while the one volume has values in a minimum range, wherein the means for modulating includes means for controlling a property of the thermal leg device.

35. The apparatus of claim 14 wherein the means for sustaining oscillation comprises thermal lag means having an inlet communicating with one of the volumes while the one volume has values in a minimum range, wherein the means for modulating includes means for controlling the volume of the thermal lag device.

36. The apparatus of claim 14 wherein the means for sustaining oscillation comprises thermal lag means having an inlet communicating with one of the volumes while the one volume has values in a minimum range, wherein the means for modulating includes means for controlling the geometry of the thermal lag device.

37. The apparatus of claim 14 wherein the means for sustaining oscillation comprises thermal lag means having an inlet communicating with one of the volumes while the one volume has values in a minimum range, wherein the means for modulating includes means for controlling the portion of the cycle that the thermal lag device is operative.

38. The apparatus of claim 14 wherein the means for modulating includes means for varying the stiffness of gaseous spring means included in at least one of the volumes.

39. The apparatus of claim 1 further including a port in a side wall of the cylinder for feeding gas from the cylinder to a load, said port being blocked by the piston while one of the volumes is in the minimum range and being unblocked while the piston is coasting through the region.

40. The apparatus of claim 39 including means for controlling the position of the port along the length of the cylinder.

41. The apparatus of claim 39 including means for controlling the position of the port along the length of the cylinder so that the port is blocked by the piston substantially while the by-pass is restricted.

42. The apparatus of claim 39 including means for controlling the position of the port along the length of the cylinder so that the port is blocked by the piston substantially while the by-pass is restricted and the port is not blocked by the piston while the piston coasts through the region.

43. The apparatus of claim 1 further including means for changing the temperature of fluid from an external source of fluid, and wherein one of said means for feeding is responsive to fluid in said means for changing and said fluid is changed in temperature by the temperature changing means prior to being fed into the cylinder.

44. The apparatus of claim 1 wherein at least one of the means for feeding includes means for feeding fluid from an external source of fluid.

45. The apparatus of claim 44 further including check valve means for feeding fluid from the external source into the cylinder only in response to the pressure of the fluid in the external source exceeding the pressure of the fluid in the cylinder.

46. The apparatus of claim 1 including means for feeding fluid from one of said volumes to a load as the piston coasts through the region, and wherein said means for feeding fluid into at least one of said volumes includes means for feeding fluid from the load.

47. The apparatus of claim 46 further including check valve means for feeding fluid from the load into the cylinder only in response to the pressure of the fluid returning from the load exceeding the pressure of the fluid in the cylinder.

48. The apparatus of claim 1 wherein at least one of the means for feeding includes means for varying the temperature of fluid flowing into the volume.

49. The apparatus of claim 48 further including check valve means for feeding gas from the temperature changing means into the cylinder only in response to the pressure of the fluid in the temperature changing means exceeding the pressure of the fluid in the cylinder.

50. The apparatus of claim 1 further including check valve means for feeding fluid only in one direction be-

tween one of said volumes and a load as the piston coasts through the region.

51. The apparatus of claim **1** further including a check valve means wherein fluid is pumped substantially in only one direction in a flow path through a load.

52. The apparatus of claim **1** wherein the means for sustaining comprises a thermal lag heater including passageways open at one end to be responsive to fluid in one of said volumes and for supplying heated fluid to said one volume as the piston is moving in a direction to increase said one volume.

53. The apparatus of claim **52** wherein the passageways are located in the piston.

54. The apparatus of claim **52** wherein the passageways are located in the piston and are heated in response to the hot fluid fed to the cylinder.

55. The apparatus of claim **1** further including means for controlling the portion of the piston oscillation cycle during which the piston coasts in the by-pass region.

56. The apparatus of claim **1** further including means for feeding fluid from one of said volumes to a load as the piston coasts through the region, a second cylinder, a second free piston in the second cylinder, said second cylinder and second piston being responsive to pressure variations derived from said means for feeding fluid, said second piston dividing the second cylinder into third and fourth variable volumes, and means tending to center oscillation of the second piston.

57. The apparatus of claim **56** wherein the means tending to center oscillation includes a by-pass between the third and fourth volumes.

58. The apparatus of claim **56** further including a magnetic circuit comprising: a magnetic component included in the second free piston, whereby magnetic flux variations are induced in the magnetic circuit in response to oscillation of the second free piston.

59. The apparatus of claim **58** further including coil means responsive to the magnetic flux variations for driving an electric load in response to the magnetic flux variations.

60. The apparatus of claim **58** further including means for controlling the average pressure in both said cylinders.

61. The apparatus of claim **1** wherein both of said volumes are substantially enclosed chambers.

62. The apparatus of claim **1** wherein the means for sustaining oscillation comprises thermal lag means having an inlet communicating with one of the volumes while the one volume has values in a minimum range, and means for varying the time during each oscillation cycle while the thermal lag means is operative.

63. An oscillating piston apparatus comprising a pair of cylinders, a different free piston in each of said cylinders dividing each of the cylinders into first and second variable volume chambers, means for providing a fluid flow path between the second chambers of each cylinder while volumes of both of the second chambers have values in a minimum range, means for sustaining synchronized oscillation of the pistons in the cylinders, by-pass means between the first and second chambers of each cylinder while the piston of the cylinder is moving through a predetermined region of the cylinder between the ends of the cylinder, a regenerator means in the by-pass means, means for each cylinder for restricting the by-pass while the volumes of the first and second chambers have values in a minimum range, means for each cylinder for feeding hot fluid into the first chamber, and means for each cylinder for feeding cold fluid into the second chamber.

64. The oscillating piston apparatus of claim **63** further including means for controlling like properties of the by-pass in a like manner.

65. The oscillating piston apparatus of claim **63** wherein the means for controlling includes means for adjusting the lengths of the two by-passes so that they are substantially the same.

66. The oscillating piston apparatus of claim **63** wherein the means for controlling includes means for adjusting the positions of the two by-passes so that they are substantially the same.

67. The oscillating piston apparatus of claim **66** wherein the means for controlling further includes means for adjusting the lengths of the two by-passes so that they are substantially the same.

68. The oscillating piston apparatus of claim **63** further including means for coupling pressure variations from both cylinders to a single load.

69. The oscillating piston apparatus of claim **63** wherein each of said cylinders includes a port in its side wall, each of said ports being responsive to pressure variations in the respective cylinders while the pistons coast through the regions, and means for combining pressure variations at said ports in a conduit and for driving a load in response to the combined variations.

70. The oscillating piston apparatus of claim **69** wherein the means for sustaining includes a common thermal lag heater connected to said second chambers.

71. The oscillating piston apparatus of claim **69** further including means for together and in the same manner controlling the positions of said ports along the lengths of the cylinders.

72. The oscillating piston apparatus of claim **71** further including means for controlling the lengths of the by-passes in synchronism with the positions of the ports.

73. The oscillating piston apparatus of claim **71** further including means for controlling the positions of the by-passes in synchronism with the positions of the ports.

74. The oscillating piston apparatus of claim **69** wherein the load comprises a second cylinder, a second free piston in the second cylinder responsive to pressure variations derived from the combining means, means for centering the second free piston, a magnetic circuit including: magnetic poles displaced along the walls of the second cylinder, and a magnetic component included in the second free piston whereby magnetic flux variations are induced in the magnetic circuit in response to oscillation of the second free piston.

75. The oscillating piston apparatus of claim **69** wherein the load comprises an alternator, said alternator including a pair of second cylinders having ends connected together and to the combining means so as to be responsive to pressure variations in the combining means, each of said second cylinders including a second free piston responsive to pressure variations in the combining means so that the second free pistons of the pair of cylinders are synchronously driven in response to the pressure variations, means for centering each of the second free pistons, each of said second cylinders being in a magnetic circuit including: magnetic poles on the second cylinders, and a magnetic component in-

cluded in the second free pistons whereby magnetic flux variations are induced in the magnetic circuit in response to oscillation of the second free pistons, and coil means responsive to the magnetic flux variations, whereby an a.c. voltage is induced in each coil means in response to oscillation of the second free pistons.

76. The oscillating piston apparatus of claim **75** wherein the means for centering includes a by-pass in each second cylinder, said by-pass providing for each second free piston, a coasting region near the center of oscillation of the second pistsons.

77. The apparatus of claim **75** wherein the means for sustaining includes a thermal lag heater communicating with said second chambers.

78. The apparatus of claim **75** including means responsive to voltage supplied to a load to automatically vary a property of the apparatus so as to maintain a more constant voltage.

79. The apparatus of claim **75** including means responsive to the frequency of electrical power supplied to a load to vary a property of the apparatus so as to maintain a substantially constant frequency of piston oscillation.

80. The apparatus of claim **75** including means responsive to load voltage and frequency to vary at least one property of the apparatus to maintain a more constant voltage and frequency

81. An oscillating piston device comprising a cylinder, a free piston in said cylinder dividing the cylinder into first and second variable volume chambers, means for sustaining oscillation of the piston in the cylinder, said means for sustaining including a pressurized gas source, means responsive to the piston substantially minimizing the volume of the first chamber for applying a pressure pulse from the source to the cylinder to drive the piston in a direction tending to increase the volume of said first chamber as the piston is moving in a direction to increase the volume of said first chamber, and means for establishing a by-pass between the first and second chambers while the piston is moving through a predetermined region of the cylinder between ends of the cylinder.

82. A system for supplying energy to a load comprising a source for deriving oscillatory gas pressure, said source including: a regenerative gas cycle piston device comprising a cylinder, a free piston in the cylinder dividing the cylinder into first and second variable volumes, a by-pass including regenerator means, said by-pass connecting disparate parts of the cylinder together, the length of the by-passed portion of the cylinder being at least equal to the piston length, means for sustaining oscillation of the piston in the cylinder, means for restricting the by-pass while at least one of the volumes has values in a minimum range, wherein the piston functions as a displacer piston while completely within the length and the same piston functions as a reversing piston while any portion of the piston is outside of said length.

83. The system of claim **82** further including means for feeding hot fluid into the first volume, and means for feeding cold fluid into the second volume.

84. The system of claim **83** wherein the means for sustaining includes a thermal lag heater connected to one of the volumes.

85. The system of claim **82** wherein ports to the by-pass are provided in the cylinder side wall at both ends of said length, said ports being blocked and unblocked by said piston to form said restricting means.

86. The system of claim **85** wherein the means for feeding cold fluid is connected to one of the ports and the means for feeding hot fluid is connected to the other of the ports.

87. The system of claim **86** including an outlet port for driving a load, said outlet port being positioned on the cylinder side wall at substantially the same axial location as one of the by-pass ports and being blocked and unblocked by the piston.

88. The system of claim **82** including an outlet port for driving a load, said outlet port being positioned on the cylinder side wall so that it is responsive to fluid in the cylinder while the piston is in said length and is blocked by the piston while one of the volumes is in the minimum range.

89. In combination, a cylinder, a free piston in the cylinder, an independent source of oscillatory pressure variations, and means for feeding pressure variations of the source into the cylinder to drive the piston in the cylinder in an oscillatory manner.

90. The combination of claim **89** wherein the cylinder is divided into a pair of variable volumes by the piston, and the means for feeding includes a port in a side wall of the piston, said port being positioned so that the pressure variations of the source as coupled to one of said volumes are increased in amplitude.

91. The combination of claim **90** further including means for controlling the center of oscillation of the piston.

92. The combination of claim **89** further including means for controlling the center of oscillation of the piston.

93. The combination of claim **92** wherein the means for controlling includes a by-pass around a portion of the cylinder side wall.

94. An oscillating piston apparatus comprising a closed cylinder, a free piston in said cylinder dividing the cylinder into first and second variable volumes, means for sustaining oscillation of the piston in the cylinder, a by-pass between the first and second volumes such that the piston coasts through a region of the cylinder between ends of the cylinder, means for restricting the by-pass during the piston oscillation while at least one volume has a value in a minimum range, means for feeding cool fluid into the second volume, and means for feeding hot fluid into the first volume.

95. The apparatus of claim **94** wherein the means for feeding hot fluid is in the by-pass.

96. The apparatus of claim **94** further including a regenerator in the by-pass.

97. The apparatus of claim **96** wherein the means for feeding hot fluid includes a hot body in heat exchange relationship with gas in the by-pass, whereby the hot body is cooled.

98. The apparatus of claim **97** further including check valve means for feeding cold fluid from the second volume to the first volume via a path in heat exchange relationship with the hot body.

99. The apparatus of claim **98** wherein the means for feeding cold fluid includes a cold body in heat exchange relationship with gas flowing in a circuit beginning and ending at the second volume, whereby the cold body is heated, and check valve means for controlling the flow of the fluid that is in heat exchange relationship with the cold body.

100. An oscillating piston apparatus comprising a closed cylinder, a free piston in the cylinder dividing the cylinder into first and second variable volumes, a by-pass between the first and second volumes such that the piston coasts through a region of the cylinder between ends of the cylinder, a regenerator in the by-pass, a hot chamber located in the by-pass between the regenerator and the first volume, a cold chamber located in the by-pass between the regenerator and the second chamber, means for restricting the by-pass during the piston oscillation while at least one volume has values in a minimum range, an external source of oscillatory pressure variations connected to the cylinder such that oscillation of the piston is sustained and fluid flows in the by-pass in a direction from the hot chamber to the cold chamber at a higher average pressure than the average pressure of fluid flowing in the by-pass in the opposite direction, whereby the hot chamber is heated and the cold chamber is cooled by fluid flowing in the by-pass.

101. The apparatus of claim 100 wherein the external source is connected to a port in an end wall of the cylinder.

102. The apparatus of claim 100 wherein the external source is connected to a port in the side wall of the cylinder.

103. The apparatus of claim 96 wherein the by-pass includes two parallel paths, one of said paths including said regenerator, the other of said paths including said means for feeding hot fluid.

104. The apparatus of claim 96 wherein the means for feeding cold fluid includes a cold body in heat exchange relationship with gas flowing in a flow path beginning and ending at the second volume, whereby the cold body is heated, and check valve means for controlling the flow of the fluid that is in heat exchange relationship with the cold body.

104. The apparatus of claim 96 wherein the means for feeding cold fluid includes a cold body in heat exchange relationship with gas flowing in a flow pathbeginning in one of said volumes and ending at the second volume, whereby the cold body is heated, and check valve means for controlling the flow of the fluid that is in heat exchange relationship with the cold body.

106. The apparatus of claim 96 wherein the means for feeding hot fluid includes a hot body in heat exchange relationship with gas flowing in a flow path beginning in one of said volumes and ending at the first volume, whereby the hot body is cooled, and check valve means for controlling the flow of the fluid that is in heat exchange relationship with the hot body.

107. The apparatus of claim 94 including regenerator means in the by-pass, wherein at least one of the means for feeding includes a gas flow path beginning at one of said volumes and ending at one of said volumes, check valve means for attaining a net flow in said path in one direction, and heat exchange means between an external body and gas flowing in the path.

108. The apparatus of claim 1 wherein the piston is of integral construction.

109. The apparatus of claim 1 wherein the piston has a substantially uniform cross section throughout substantially all of its length.

110. The apparatus of claim 1 wherein the piston is of substantially integral construction and has a substantially uniform cross section throughout substantially all of its length.

111. The apparatus of claim 1 wherein at least one of said volumes is a substantially closed chamber during a portion of the piston oscillation cycle while said one volume is in a minimum range to form a gaseous spring for reversing the direction of motion of the piston.

112. The apparatus of claim 1 wherein the by-pass includes ports in the side wall of the cylinder for establishing fluid flow paths from the cylinder to the regenerator, and a further port in the side wall for establishing a fluid flow path from the cylinder to a device external to the cylinder.

113. The apparatus of claim 112 wherein the further port is subsatntially aligned with one of the by-pass ports.

114. The apparatus of claim 63 wherein each of the pistons is of substantially integral construction.

115. The apparatus of claim 63 wherein each of the pistons has a substantially uniform cross section throughout substantially all of its length.

116. The apparatus of claim 63 wherein the means for feeding fluid includes by-pass ports in the side wall of each cylinder for connecting the cylinder by-pass in fluid flow relation with the cylinder.

117. The apparatus of claim 116 including at least one additional port in the wall of each cylinder for coupling pressure variations within the cylinder to a load.

118. The apparatus of claim 117 further including means for connecting at least one of said additional ports of each cylinder together to couple pressure variations within each cylinder to a common load.

119. The apparatus of claim 63 wherein the by-pass means includes a separate by-pass for each of the cylinders.

* * * * *