

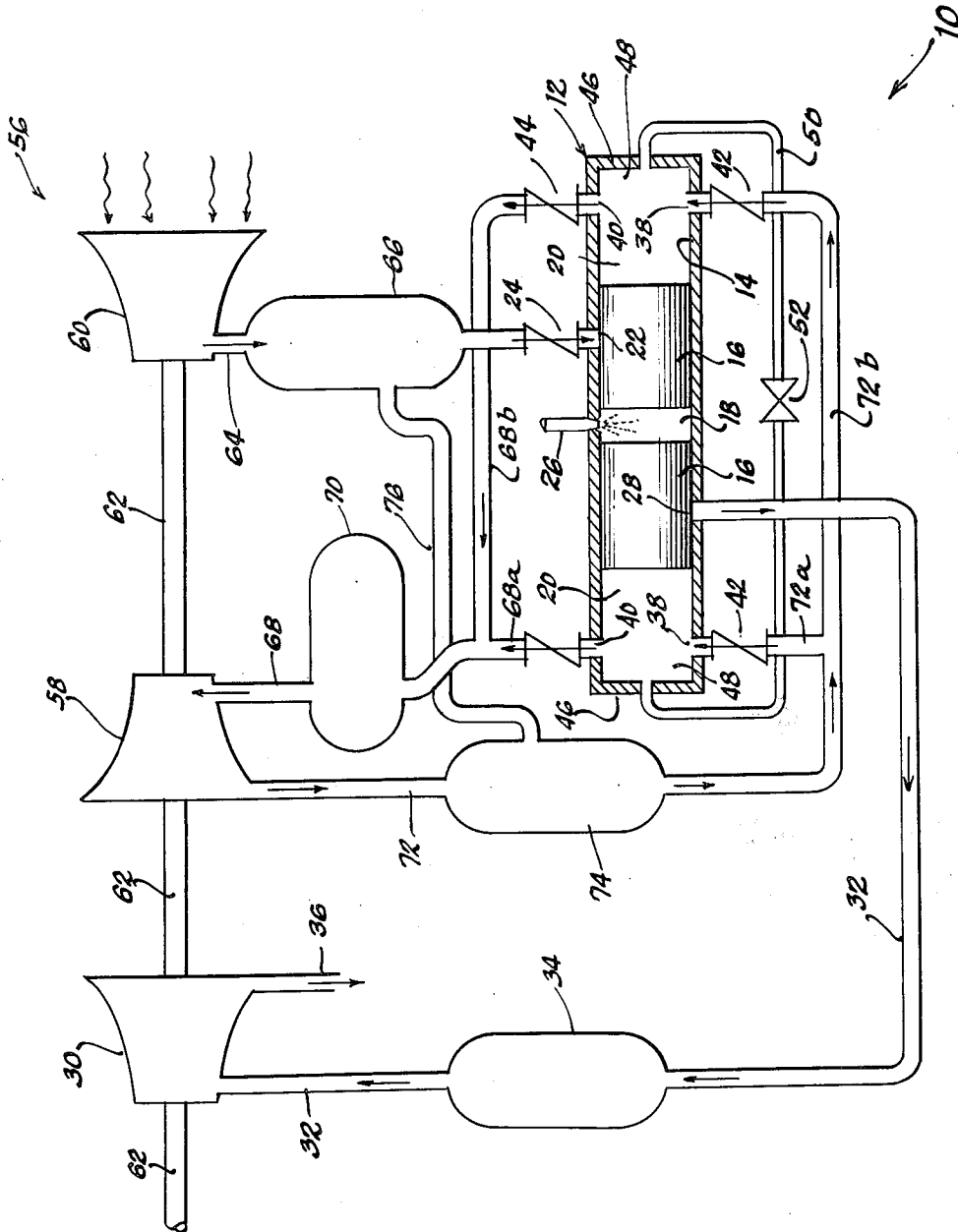
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FREE PISTON ENGINE POWER UNIT

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FREE PISTON ENGINE POWER UNIT
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This invention relates to free piston engines of the type in which hot gases under pressure are generated by combustion in a cylinder containing a pair of free pistons adapted to oscillate toward and away from each other in a symmetrical manner, the hot gases being employed to operate a turbine or the like.

One object of the present invention is to provide a new and improved free piston engine which achieves numerous advantages, including flexibility in design, stability of operation, adaptability to load changes, adjustability for maximum output, and highly efficient operation, even under low load conditions.

A further object is to provide a new and improved free piston engine, in which these and other advantages are achieved in a construction which utilizes a power turbine operable by the hot gases, an air turbine operable by air compressed in the cylinder, and a turbo-compressor, adapted to compress atmospheric air for use in charging the cylinder, the two turbines and the turbo-compressor being all on the same shaft, or otherwise being mechanically coupled.

Further objects and advantages of the present invention will appear from the following description, taken with the accompanying drawing, in which the single figure is a diagrammatic view illustrating a free piston engine to be described as an illustrative embodiment of the present invention.

The present invention may be regarded as an improvement upon the invention disclosed in my copending U.S. application, Serial Number 483,987, filed January 25, 1955, and entitled Free Piston Engine. Thus, the present drawing illustrates a free piston engine 10 comprising a cylinder 12 having a bore 14 therein which is substantially the same diameter throughout its entire length. This is in contrast with many of the prior, conventional free piston engines, which usually have a stepped bore.

In the present engine, two identical pistons 16 are symmetrically disposed in the bore 14 for reciprocation therein. It will be seen that the pistons 16 are simply in the form of substantially cylindrical slugs, having the same outside diameter along their entire length. This is in contrast with the stepped pistons of many prior, conventional free piston engines. In the present engine, the pistons 16 are adapted to oscillate in synchronism, toward and away from each other. Thus, the oscillation of the pistons is symmetrical about the center of the cylinder. The symmetrical oscillation of the pistons balances out the vibrations that would be imparted to the cylinder if the cylinder contained only a single piston. Thus, the engine operates with very little vibration. It will be understood, however, that the invention is applicable to a free piston engine utilizing only one piston adapted to oscillate in a cylinder.

The present pistons 16 are entirely free, in the sense that no mechanical or hydraulic connection is made to the pistons. This arrangement enables the pistons to oscillate with a minimum of friction between the pistons and the cylinder.

The cylinder 12 may be regarded as containing a combustion chamber 18 and a compression chamber 20 on opposite sides of each piston 16. In this case, the combustion chamber 18 is located in the center of the cylinder 12, between the two pistons 16. Thus, a single combustion chamber serves both pistons. However, there are two compression chambers 20, in the opposite end portions of the cylinder 12. For some purposes, the combustion and compression chambers could be interchanged, but the illustrated arrangement is preferable.

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Air is introduced into the combustion chamber 18 through an inlet port 22 adapted to be covered and uncovered by the right-hand piston 16. A check valve 24 may be provided at the inlet port 22 to prevent backflow of air into the inlet port. It will be understood that the inlet port 22 is uncovered when the pistons 16 move apart to the outward limits of their travel. As the pistons 16 move together, the inlet port 22 is covered by the right-hand piston so that the charge of air is compressed between the pistons. The increasing pressure of the charge of air between the pistons slows them down and eventually arrests their inward movement. At this point, fuel is introduced into the combustion chamber 18 by a fuel injector 26, located at the midpoint of the cylinder 12. The fuel injector may be timed and controlled in the manner disclosed in my above-mentioned copending application, Serial Number 483,987, filed January 25, 1955. The fuel is ignited as it is injected, due to the high temperature of the compressed air between the piston 16. Thus, the illustrated engine utilizes a diesel combustion cycle. The pistons 16 are driven apart by the pressure of the hot products of combustion.

After the products of combustion have expanded to a considerable extent, the left-hand piston 16 uncovers an outlet port 28 through which the products of combustion are discharged. A new charge of air is then introduced through the inlet port 22. It will be understood that the incoming compressed air scavenges the combustion chamber 18 by forcing the products of combustion out the discharge port 28.

The hot gaseous products of combustion may be employed to operate a power turbine 30, or some other machine capable of converting the energy of the hot gases into mechanical work. A pipe 32 is connected between the outlet port 28 and the turbine 30. As shown, a reservoir 34 may be connected into the pipe 32 to accumulate the products of combustion and smooth out the pressure pulsations at the inlet of the turbine 30. After being expanded in the turbine 30, the products of combustion may be discharged to the atmosphere through the outlet 36.

It will be evident that compressed air is needed to charge the combustion chamber 18. This compressed air is derived indirectly from the compression chambers 20 of the cylinder 12. It will be seen that each compression chamber 20 is formed with inlet and outlet ports 38 and 40. Air enters each compression chamber 20 through the inlet port 38. A check valve 42 may be provided at the inlet port 38 to prevent backflow of air. The air passes into the compression chambers 20 through the ports 38 as the pistons move inwardly toward the midpoint of the cylinder. When the pistons move outwardly, the air is compressed in the compression chambers 20. The air leaves the compression chambers 20 through the outlet ports 40. Check valves 44 may be provided at the outlet ports 40 to prevent backflow of the compressed air into the cylinder. Thus, each compression chamber 20 operates on an air compression cycle.

It will be seen that the inlet and outlet ports 38 and 40 are spaced inwardly from the ends of the cylinder. End walls 46 are provided to close the ends of the cylinder. The space between each end wall 46 and the corresponding ports 38 and 40 constitutes a bounce chamber 48. After the pistons 16 cover the ports 38 and 40, the air trapped in the bounce chambers 48 is highly compressed. The pressure of the compressed air in the bounce chambers slows down the pistons and eventually drives them inwardly toward the midpoint of the cylinder.

To assist in synchronizing the oscillation of the pistons about the midpoint of the cylinder, a small pipe 50

is connected between the bounce chambers 48 at the opposite ends of the cylinder. An adjustable restriction 52 may be introduced into the pipe 50. When the pistons 16 are perfectly synchronized, there is no flow of air through the pipe 50, although the air in the opposite end portions of the pipe is compressed along with the air in the bounce chambers 48. Thus, the restriction 52 does not introduce any loss of energy. However, any deviation of the pistons from synchronism causes flow of air through the pipe 50, with an accompanying loss of energy. Thus, the synchronized condition of the pistons involves a minimum loss of energy. This factor assists in synchronizing the pistons, as described in greater detail in my above-mentioned copending application Serial Number 483,987, filed January 25, 1955.

In general, the charging of the combustion chamber 18 requires a relatively large volume of air at a relatively low pressure. The compression chambers 20 are capable of delivering compressed air at a power level sufficient to charge the combustion chamber, but the compressed air derived from the compression chambers is at a relatively high pressure and a relatively low volume. This factor accounts for the prevailing use of stepped pistons in conventional free piston engines. Conventionally, the diameter of the compression chambers is much greater than the diameter of the combustion chamber, so that the compression chambers will supply air at a low pressure and a high volume, suitable for charging the combustion chamber. However, the use of stepped pistons increases the frictional losses in the free piston engine, tends to decrease the oscillatory frequency of the pistons, and introduces serious alignment problems, particularly when the engine is heated up to its relatively high operating temperature.

In the present engine, the high-pressure, low-volume compressed air from the compression chambers 20 is converted into low-pressure, high-volume compressed air by a mechanical pressure converter 56. The pressure converter may take various forms, but is illustrated as a pair of rotary units 58 and 60, which are connected to a common shaft 62, or are mechanically coupled together in some other suitable manner. The rotary unit 58 operates as an air turbine driven by the compressed air from the compression chambers 20, and is effective to deliver work to the shaft 62. The rotary unit 60 is driven by the shaft 62 and operates as a compressor unit adapted to compress atmospheric air for use in supercharging the combustion chamber 18. Thus, the outlet of the compressor 60 is connected to the inlet port 22 through a pipe 64. A reservoir 66 may be connected into the pipe 64, to accumulate the compressed air and smooth out pulsations in the flow of air from the compressor 60.

The compression chambers 20 are connected to the inlet of the air turbine 58 by a pipe 68 having branches 68a and 68b connected to the outlet ports 40 at the opposite ends of the cylinder 12. A reservoir 70 may be connected into the pipe 68 to smooth out the flow of air to the turbine 58.

A pipe 72 runs from the outlet of the turbine 58 and is formed with branch pipes 72a and 72b connected to the inlet ports 38 at the opposite ends of the cylinder. Here again, a reservoir 74 is connected into the pipe 72.

It is often advantageous to provide for an interchange of air between the outlet of the compressor 60 and the outlet of the air turbine 58. Such interchange will occur primarily during changes of load and the like. Thus, the combustion chamber 18 may be charged in part by air discharged from the turbine 58. Likewise, the compressor chambers 20 may be charged by air derived from the outlet of the compressor 60. In this case, a pipe 76 is connected between the reservoirs 66 and 74 to provide for such interchange of air. If interchange is not desired, the pipe 76 may be omitted. Of course, the inclusion of the pipe 76 has the effect of combining the reservoirs 66 and 74, so that a single, common reservoir could be employed. With the present arrangement, the

discharge pressure of the turbine 58 is kept the same as the discharge pressure of the compressor 60. Thus, the combustion chamber 18 and the compression chambers 20 are charged at substantially the same pressure.

The general arrangement described thus far is also disclosed in my above-identified copending application, Serial Number 483,987, filed January 25, 1955. However, in the present engine, the work turbine 30, operated by the hot gaseous products of combustion, is mechanically coupled to the air turbine 58 and the compressor 60. This may advantageously be done by utilizing the shaft 62 as the common shaft of the three units 30, 58 and 60. Any suitable device, such as an electric generator, may be driven by the shaft 62, to utilize the mechanical power developed by the engine.

With the three units coupled together, the power for driving the compressor 60 need not be derived entirely from the air turbine 58, but may be derived in part from the output turbine 30. This makes possible various improvements in the design and the operation of the engine. For example, the compressor 60 may be arranged to supply supercharging air at a higher pressure than could be developed if the compressor 60 were driven solely by the air turbine 58. The extra power to drive the compressor is derived from the output turbine 30. In many cases, the greater compression ratio resulting from increasing the supercharging pressure increases the efficiency of the engine to such an extent that the turbine is able to deliver the extra power to the compressor, while still delivering greater output in the form of available mechanical work.

The mechanical coupling of the three units 30, 58 and 60 also makes it possible to arrange the engine so that the air turbine 58 will produce more power than is necessary to operate the compressor 60, in which case the air turbine 58 will deliver available mechanical work to the shaft 62, over and above the work delivered by the turbine 30. This mode of operation is useful, for example, when the engine is lightly loaded.

The illustrated arrangement also tends to stabilize the operation of the engine at partial loads, when the fuel supply is cut back. With the present engine, there is no need to dispose of excess compressed air by dumping the air or throttling the intake to the compression chambers. Any excess energy in the compressed air from the compression chambers 20 can be converted into available work output by the air turbine 58.

The present arrangement also tends to stabilize the oscillator frequency of the pistons 16 despite variations in loading on the engine. There is also a tendency to stabilize the amplitude of the oscillation of the pistons.

The advantages of the present invention may best be made clear by comparing the arrangement of the present invention with prior free piston engines.

Essentially every free piston engine consists of two parts, an oscillator or gasifier which supplies compressed gas at relatively high temperature, and a turbine or equivalent which uses this gas to produce useful power output.

The conventional oscillator consists of two pistons in a cylinder oscillating symmetrically with respect to the center line of the cylinder. The central volume between the pistons comprises a supercharged diesel cycle. Compressed air is admitted to the central volume, further compressed as the pistons approach, heated by fuel input as expansions occurs, and finally exhausted at approximately the inlet pressure but at an elevated temperature. Such a cycle generates power. This power is used to overcome friction and maintain the pistons in oscillation, and also to compress air in the outer volumes of the cylinder, from atmospheric pressure to the inlet supercharge pressure. For this purpose, in the conventional engine, each piston is of two diameters, and the portion acting in the outer volumes is of larger diameter than the portion participating in the diesel cycle, in order to furnish enough gas, at the elevated pressure, to scavenge the central volume at the end of each stroke.

There is a point here which is not always fully appreciated. The central volume must have fuel injected, and hence must develop power, and this power must be absorbed in the oscillator if the device is to work at all. Consider such a device in which there is no power used for compression, the valves for example being left continuously open on the compressor ends, and in which the supercharged air is supplied from a separate source. Evidently, if no fuel is supplied, the energy in the gas exhausted from the central volume will be equal, except for losses, to the energy in the compressed gas used for supercharging. The output turbine can develop only the power necessary for compression, in fact less power, due to losses. The engine has not power output. But suppose fuel is supplied. Now the central volume develops power, and it is not absorbed in compression. It must go somewhere, and it goes to increase the energy, potential and kinetic, of the gas volumes and pistons. The speed of the pistons, and their amplitudes, will increase without limit, until the power is absorbed by losses or the engine destroys itself. In order to have a workable oscillator, there must be power absorption in the oscillator, and this occurs in practice because of the power used for compression.

As a corollary, this involves a nice balance. With a given supercharge pressure, and a given amplitude of oscillation, and full scavenge, which occurs at full load, the compression power is fixed. This means that the allowable fuel input is fixed, and, through this the temperature of the exhaust. Hence the initial compression ratio, that is the pressure of supercharge, is practically limited, for otherwise the exhaust temperature would be too high for economical turbine design. The initial compression ratio affects directly the power that can be developed in a given size of cylinder, with a given stroke and frequency of oscillation. The temperature allowable at full load thus determines design directly. The intimate connection between this and the initial compression ratio thus imposes severe limits on design.

Another point is equally important. Consider such an engine operating on low loads. If the supercharge pressure is held the same as at full load, the output turbine will require a much lower rate of flow of gas to develop the low output power. Partial scavenging is allowable, since the fuel input will be low. But, if frequency and amplitude of oscillation are maintained, the compressor ends still develop full volume of gas flow to the input. What is to be done with the excess? The frequency may drop somewhat, but nowhere enough to take care of the matter, and there are also severe limits to what can be done to change amplitude, so there is no cure here. One might consider dropping the supercharge pressure, so that the turbine, on lower input pressure, would require a greater flow of gas. But, if supercharge pressure is dropped, the volume of air delivered by the compressor is correspondingly increased, so this change does no great good. Actually, in large European installations, the excess is dumped. This is obviously a crude procedure, and it greatly reduces efficiency at low loads.

In order to maintain balance in the oscillator, at low loads, the power absorbed in the compressor ends must be decreased. This can be done by throttling the input to these volumes. When this is done the output is correspondingly decreased in pressure and volume. Thus, there is a compromise situation and the initial compression ratio drops. But, as a practical matter, it still becomes necessary to dump some of the compressed gas at low loads. The necessity for maintaining power balance in the oscillator simply cannot be made to fit, without compromise in design and loss of efficiency, over a wide range of loads.

These two limitations impose a serious burden on design and operation of the conventional engine.

Now the engine considered in this application differs radically from the conventional form. The cylinder is

of uniform diameter, and the pistons are simple cylindrical slugs. The volumes of the compression chambers, taken together, are equal, except for matters such as port overlap, to the central volume in which the diesel cycle occurs. The compression chambers develop the power, but not the volume, for supercharge. In one form of engine this power from the compression chambers is used to drive a power turbine of an auxiliary unit, and the compressor of this unit, on the same shaft, delivers this power, less losses, to the combustion chamber in the form of a greater volume at lower pressure, at a pressure in fact suitable for supercharge. The mean pressure existing in the compression chambers is now chosen so as to cause these chambers, in the light of their diameter and stroke, to absorb the power necessary for supercharging and losses. The condition of balance in the oscillator thus remains as before.

However, there is a great difference in operation at once apparent. The compressor of the auxiliary unit is essentially a constant pressure device, rather than a constant volume device, such as the compression chambers of the conventional engine. At a given speed it tends to hold a constant pressure of output, dropping somewhat of course with increased flow. Hence, if the power turbine on low loads calls for less flow of gas, a lesser flow will occur automatically. There will be no need to dump an excess.

These important advantages are achieved by the engines of my above-mentioned copending application, Serial No. 483,987, as well as by the engine of the present invention. As to the engines of my above-mentioned copending application, however, it still holds that, with low fuel input to the diesel volume, the power absorbed in the compressor ends must be correspondingly decreased. In fact this is also necessary because the power turbine of the auxiliary unit now requires less input in order to supply the lower volume of supercharging air. But this can readily be accomplished by reducing the mean pressure in the compression chambers, and this does not involve a loss, as does throttling or dumping. With an auxiliary unit, therefore, the increased flexibility allows the design and control to be such that no dumping is necessary at low loads. When low loads occur, all pressures can be made to drop together, balance is maintained, and flow in the output turbine adjusts automatically to the load. Thus, while the auxiliary unit involves extra losses, since there is an extra power conversion in connection with supercharging, losses which incidentally are partially or wholly offset by the higher compression ratio, and other features, inherent in the use of single diameter pistons, the use of this auxiliary unit introduces a new independent variable in design, the use of which avoids one of the limitations of the conventional engine. As to the engines of my above-mentioned prior copending application, Serial Number 483,987, the other limitation, in the relation between initial compression ratio and power turbine input temperature, still remains.

Suppose now, however, that the auxiliary unit and the power turbine are mounted on the same shaft or otherwise coupled together, forming a three rotary unit assembly, in accordance with the present invention. One more independent variable is now introduced. The power to drive the compressor is no longer necessarily derived entirely from the compression chambers of the oscillator. Such power can be derived partially from that source, and partially from the power turbine. Alternately, the turbine driven by gas from the compression chambers may supply power both to the compressor and to the output shaft. This allows the power of the assembly to be adjusted at will. It will be adjusted, in fact, so as to absorb the correct amount of power from the compressor and the compression chambers in order to maintain balance at all loads and hence at all rates of fuel input, or to maintain constant amplitude of oscillation

for example. This power is adjusted by adjusting the mean pressure in the compression chambers. The compressor takes the power it needs from the shaft, in light of the supercharge pressure and the flow to the power turbine. The difference, plus or minus, of the power in these two units, appears on the shaft to add or subtract to the power of the output turbine.

Now one is freed from the limitation on design noted earlier. Suppose one chooses an initial compression ratio for full load. This no longer fixes the compressor power absorption of the oscillator, and hence the fuel input, the power output, and the exhaust temperature. Instead one may independently choose the compression chamber power absorption. This may be done, in fact, in such manner as to produce an optimum turbine input temperature. In design, one must hold this temperature within limits, and then design for maximum output of an oscillator of given size, or minimum size of an oscillator of given output. In general, increasing the supercharge pressure increases the output of a diesel cycle, even when overall compression ratio is maintained constant, and this latter is presumably fixed by allowable maximum temperatures and pressures. However, suppose one has an engine in which the power turbine temperature is at its allowable top limit, and that he then increases the initial compression ratio, and adjusts the fuel input to the same value per pound of gas as before. The power input will be increased, but the turbine temperature will now exceed its allowable limit. One would wish to decrease the fuel input somewhat, to retain the increased output per pound of weight of both oscillator and turbine due to higher initial compression ratio. But when the full compressor power has to be derived from the oscillator, as in the engines of my above-mentioned prior application, this cannot be done, or balance would be departed from. However, with three units on the same shaft, in accordance with the present invention, it can be done, for the excess power needed for supercharging may now be drawn from the output turbine. In this way the turbine temperature may be adjusted back to its allowable value.

This discussion is necessarily qualitative and approximate. The actual situation is complicated by such factors as changes of turbine speed and oscillator frequency with load. However, the main point is that, with the three turbine unit, an additional factor is introduced, which may be chosen in making a design, and this added flexibility avoids the constriction imposed on design, if the oscillator power situation must be exactly balanced at all loads.

Various other modifications, alternative constructions and equivalents may be employed without departing from the true spirit and scope of the invention as exemplified in the foregoing description and defined in the following claims.

I claim:

1. In a free piston engine comprising a cylinder, a pair of free pistons reciprocable therein, said cylinder having a compression chamber and a combustion chamber therein on the opposite sides of each of said pistons, a compressor for supplying compressed air to said combustion chamber, means for introducing the compressed air and fuel into said combustion chamber for combustion therein, and an output motor for receiving and expanding the gaseous combustion products from said combustion chamber to convert the energy of said products into mechanical work, the combination of: an air motor for receiving compressed air from said compression chamber and expanding such compressed air, said expanded air being supplied to said chambers, and means mechanically interconnecting said output motor, said air motor and said compressor for common mechanical operation, so that both the output motor and the air motor can drive the compressor.

2. In a free piston engine, the combination comprising

a cylinder having substantially the same inside diameter along its entire length, a pair of substantially cylindrical pistons reciprocable in said cylinder, each of said pistons having substantially the same diameter along its entire length, said cylinder having compression chambers in the ends thereof and a combustion chamber between said pistons, means for admitting air to said compression chambers for compression by said pistons, means for withdrawing compressed air from said compression chambers, means for introducing air and fuel into said combustion chamber for combustion therein, means for withdrawing gaseous combustion products from said combustion chamber, a power turbine for receiving and expanding said combustion products to convert the energy of said products into mechanical work, an air turbine for receiving and expanding the compressed air from said compression chambers, a compressor for compressing atmospheric air, means for carrying air from the exhaust of said air turbine and the output of said compressor to said cylinder for supplying said chambers, and a common shaft mechanically interconnecting said power turbine, said air turbine and said compressor for common mechanical operation.

3. In a free piston engine comprising a cylinder having substantially the same inside diameter along its entire length, a pair of substantially cylindrical pistons reciprocable in said cylinder, each of said pistons having substantially the same diameter along its entire length, said cylinder having compression chambers in the ends thereof and a combustion chamber between said pistons, means for admitting air to said compression chambers for compression by said pistons, means for withdrawing compressed air from said compression chambers, means for introducing air and fuel into said combustion chamber for combustion therein, means for withdrawing gaseous combustion products from said combustion chamber, a power motor for receiving and expanding said combustion products to convert the energy of said products into mechanical work, a compressor for compressing atmospheric air, means for carrying air from the output of said compressor to said cylinder for charging said chambers, the combination of an air motor for receiving and expanding the compressed air from the compression chambers, means mechanically interconnecting said power motor, said air motor and said compressor for common mechanical operation, and means for communicating the expanded air from the air motor to the chambers so that both the compressor and air motor can supply the chambers.

4. In a power plant in which gaseous combustion products of a free piston engine are expanded through a power turbine to produce mechanical energy with the engine having a piston reciprocating within a cylinder intermediate compression and combustion chambers defined by the cylinder and piston, an air compressor for compressing ambient air from atmospheric to superatmospheric pressure preparatory to combustion, and means communicating an outlet of the compressor with inlets of the chambers, the improvement comprising a second turbine for expanding the compressed air from the compression chamber to provide supercharge air for the chambers, means communicating an inlet of the second turbine with an outlet of the compression chamber, means communicating the outlet of the second turbine with the inlet of the combustion chamber and with the inlet of the combustion chamber, and means to mechanically interconnect the power turbine, compressor and second turbine for common mechanical operation so that both the power turbine and the second turbine drive the compressor to stabilize the operation for all requirements of mechanical energy.

5. In a power plant utilizing the expansion of combustion products from a free piston engine to drive a power turbine, the power turbine in turn mechanically driving a compressor to compress air preparatory to its combustion in the engine, the engine including a cylinder and a

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pair of free pistons reciprocable in the cylinder, each of the pistons having on its opposite ends defined by the cylinder a compression chamber and a combustion chamber, means including the pistons to communicate the compressed air from the compressor alternately to the chambers, means to ignite the compressed air in the combustion chambers, and means to exhaust the combustion products from the combustion chambers to the power turbine, the improvement comprising an air turbine mechanically interconnected on a common shaft to the power turbine and com-

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pressor for common mechanical operation therewith, means communicating the outlets of the compression chambers with the inlet of the air turbine, and means communicating the inlets of the compression and combustion chambers with the outlet of the air turbine.

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