April 22, 1958 F. C. PERROTT 2,831,626

FREE-PISTON COMPRESSORS OR GAS GENERATORS

Filed Aug. 16, 1954 5 Sheets-Sheet 1

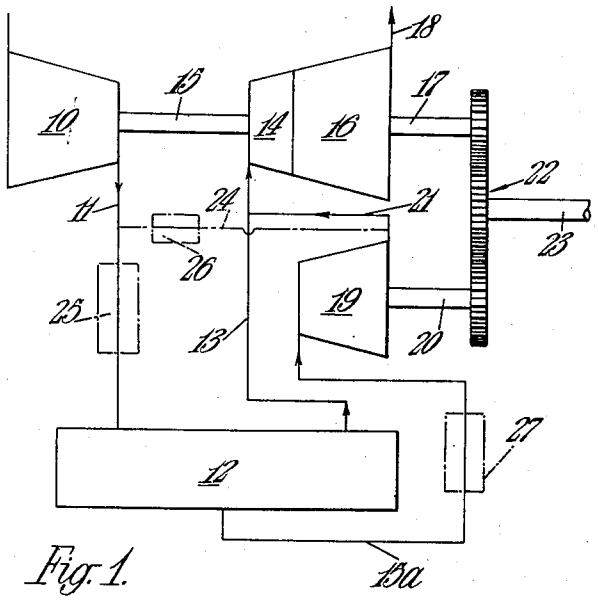
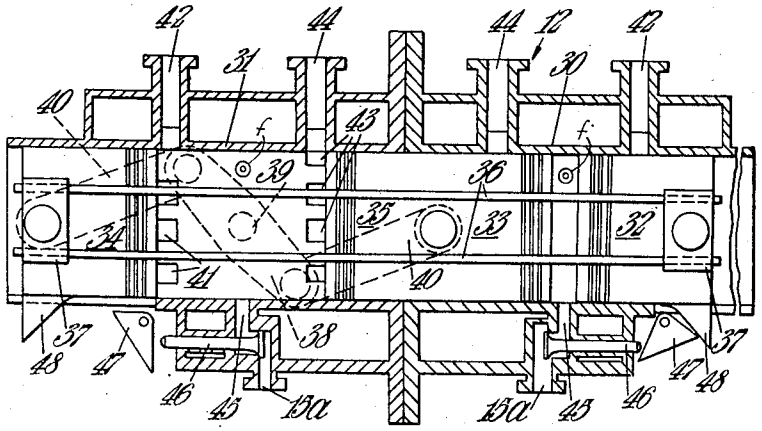


Fig. 1.

Fig. 2.



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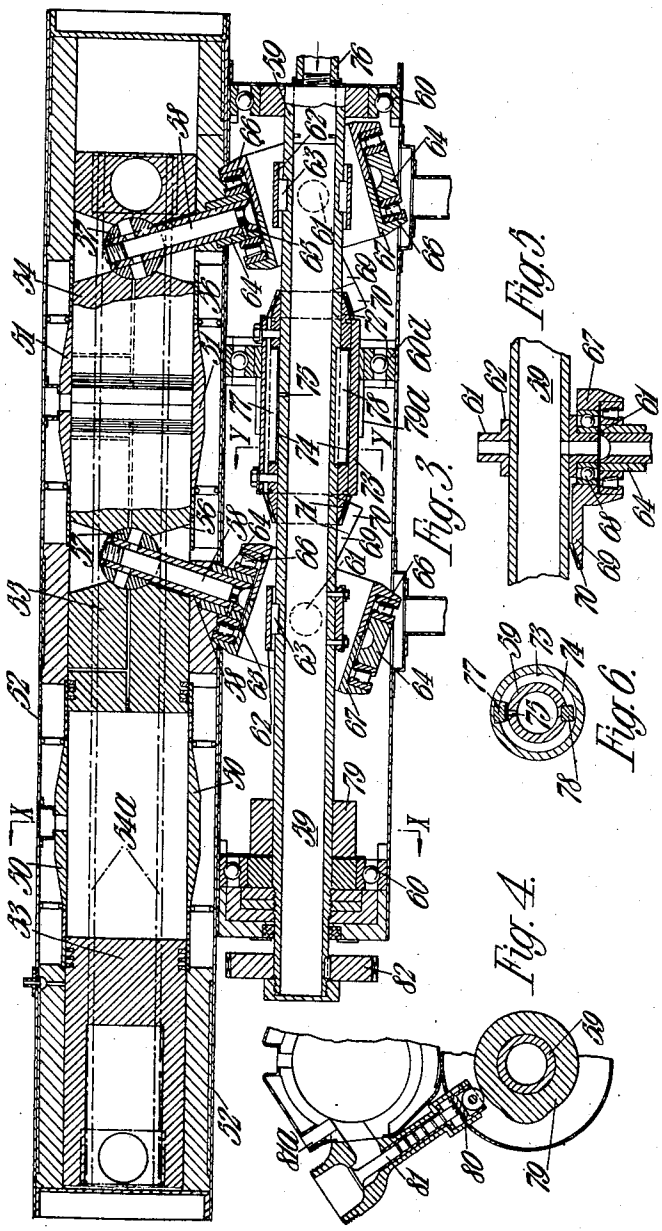


Fig. 3.

Fig. 6.

Fig. 4.

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FREE-PISTON COMPRESSORS OR GAS GENERATORS

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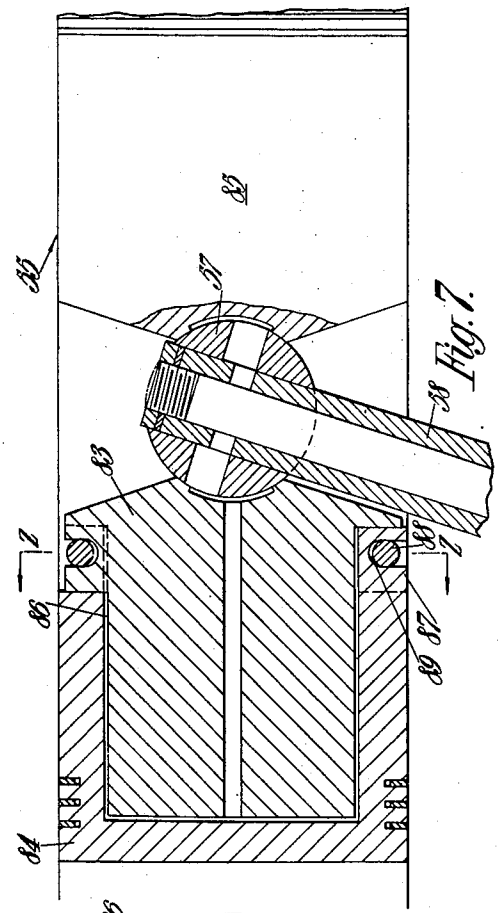


Fig. 7.

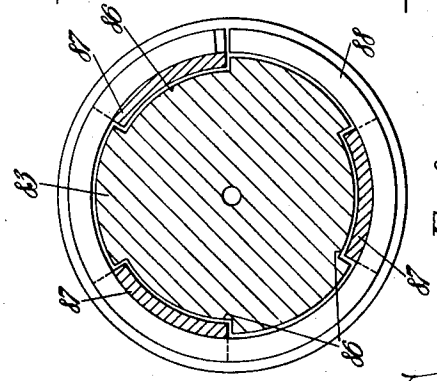


Fig. 8.

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FREE-PISTON COMPRESSORS OR GAS GENERATORS

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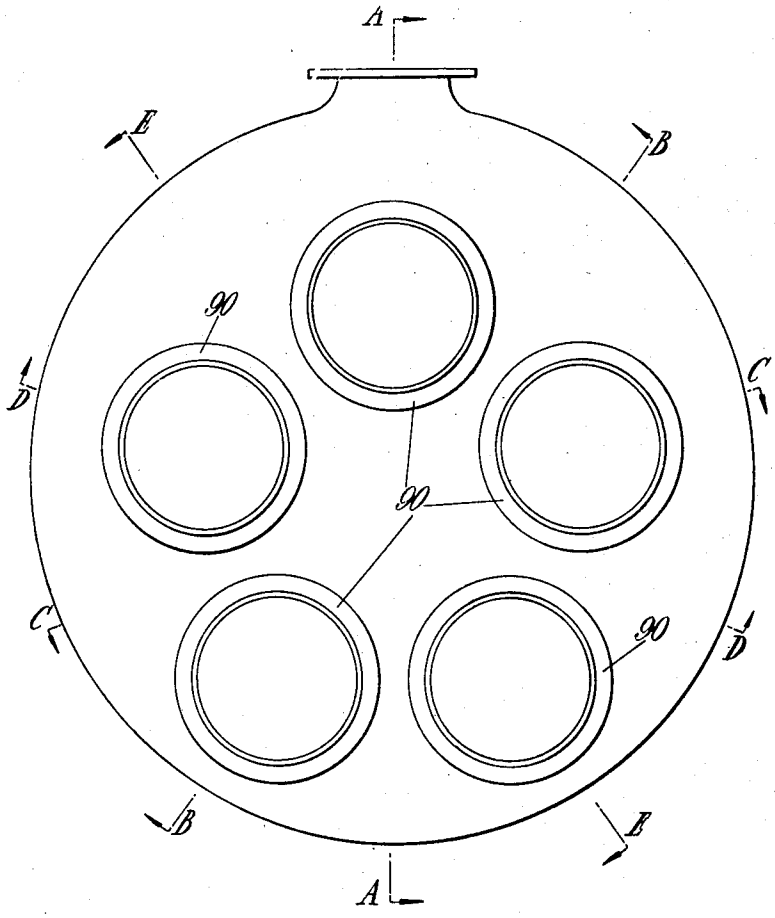


Fig. 9

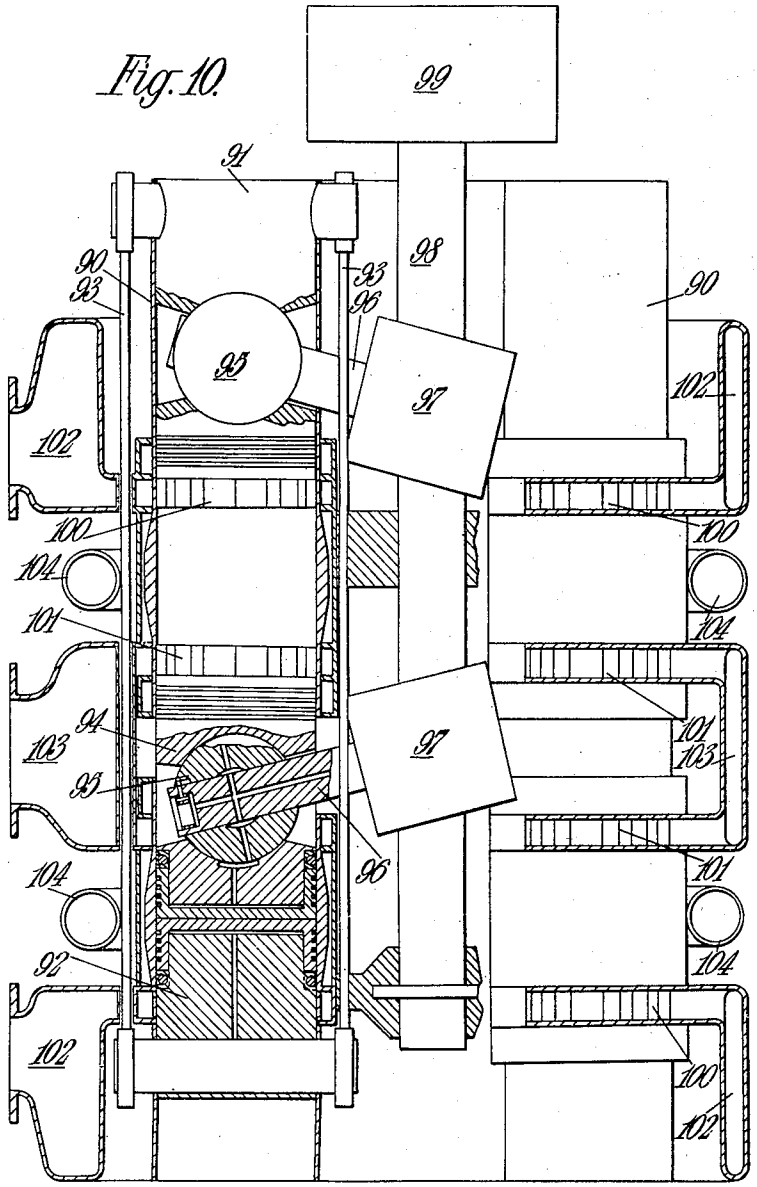
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FREE-PISTON COMPRESSORS OR GAS GENERATORS

Filed Aug. 16, 1954 5 Sheets-Sheet 5

Fig. 10.



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2,831,626

FREE-PISTON COMPRESSORS OR GAS GENERATORS

Francis Cyril Perrott, Edinburgh, Scotland

Application August 16, 1954, Serial No. 450,126

13 Claims. (Cl. 230—56)

This invention relates to free-piston compressors or gas generators of the kind comprising a cylinder and free piston assembly and wherein the compressor or gas generator is driven by the combustion of fuel in said cylinder. A compressor or gas generator in accordance with the present invention has very wide applications but it is particularly adapted to supply compressed air and/or combustion products for use in gas turbines, power plants, or compressed air for compressed air turbines or for a combined steam and gas turbine power plant.

The present invention has for its main object to provide an improved construction of compressor or gas generator of the kind referred to in the preceding paragraph.

The conventional form of air compressor incorporating reciprocating pistons driven by a crank shaft and connecting rods is coupled to a separate source of power so that the installation becomes heavy and costly and suffers from known disadvantages. A further object of the present invention is to provide an improved construction of compressor in which the weight and cost of construction is reduced.

Free piston internal combustion air compressors and gas generators are known which comprise two primary reciprocating assemblies each having a large and small piston rigidly connected together, all four pistons being coaxially arranged. The pair of small pistons operate in a common cylinder and the large pistons operate each in a separate cylinder so that combustion may take place between the small pistons. A small piston is directly coupled to an associated large piston and the pairs of pistons are interconnected so that the piston assemblies reciprocate in opposite directions in phase with each other, being alternately moved apart during combustion and returned by air compressed in the larger cylinder by each of the larger pistons. By a suitable arrangement of non-return valves and ports in the ends of one or both of the larger cylinders compressed air may be discharged therefrom and used externally, or passed through the smaller common cylinder in which combustion takes place.

The known free-piston compressor does not suffer from many of the disadvantages of rotary compressors but it is necessary to make special provisions to ensure steady running. To this end certain relative sizes for the larger and smaller cylinders have to be selected which precludes these being designed solely from considerations of the compression which is required. Another disadvantage associated with this type of compressor is that only one of the larger cylinders can be utilized as a compressor of air to be used externally of the compressor since the other larger cylinder must act as a cushioning member, that is to say for purposes of steadying the operation, or else an additional cushioning cylinder must be provided, the larger cylinder being then not used for cushioning. Again from requirements of steady running it is necessary that the cylinder supplying compressed air externally of the installation shall have a large clearance volume and

although this does not necessarily affect the overall efficiency it does adversely affect the output of the compressor whereby the latter is required to be made correspondingly larger for a given output. Considerations of steady running also restrict the quantity of fuel which may be burned in the small combustion cylinder and so restrict the output of the compressor or generator. These factors in combination with the use of slow-acting non-return valves limit the speed of operation of the compressor so that once more for a given output, the size requires to be increased. The combined result of the several factors referred to is to produce a compressor or gas generator which is heavy and bulky and is accordingly not commonly used for gas turbine engines. In addition, variations of output produce variations of compression ratio in the small cylinder and this is undesirable for efficient combustion.

The present invention provides a compressor or gas generator of the free-piston type in which certain or all of the disadvantages referred to above are reduced or overcome.

According to the present invention, a compressor or gas generator comprises a pair of cylinder and free-piston assemblies of which the pistons are coupled to move together within the cylinders and each piston performs alternate compression and working strokes (the latter involving combustion of fuel in the cylinder), the arrangement being such that while a working stroke takes place in one cylinder a compression stroke takes place in the other, and wherein from each assembly, compressed air and/or combustion products is or are discharged from the cylinder at the end of the compression and/or the start of the working stroke, the energy of the gases remaining in the said cylinder after said discharge being utilized to perform a further compression stroke in the other cylinder.

In a preferred embodiment of the invention, the compressor or gas generator comprises a pair of cylinders arranged end to end, a pair of free pistons movable within each cylinder to perform alternate compression and working strokes therein, a coupling between each of the pistons of one cylinder and its corresponding piston in the other cylinder whereby the corresponding pistons move together, the arrangement being such that while a working stroke takes place in one cylinder a compression stroke takes place in the other, means for supplying fuel and air to, and exhausting combustion products resulting from ignition of the fuel and air from the cylinders, and means for discharging from each cylinder compressed air and/or combustion products at the end of the compression and/or the start of each working stroke.

Other features and advantages of the invention will be described hereinafter.

Practical constructions of the present invention will now be described, merely by way of example, with reference to the accompanying drawings in which:

Figure 1 is a schematic representation of a gas turbine power plant incorporating a gas generator in accordance with the invention;

Figure 2 is a sectional elevation of a gas generator in accordance with the present invention;

Figure 3 is a sectional elevation of an alternative construction of gas generator;

Figure 4 is a cross section taken on the line X—X of Figure 3;

Figure 5 is a longitudinal section of a portion of the swash plate shaft showing the trunnions in line with one of the swash plate arms;

Figure 6 is a cross section taken on the line Y—Y of Figure 3;

Figure 7 is a sectional elevation through one of the

pistons of Figure 3 showing an alternative construction;

Figure 8 is a cross section taken on the line Z—Z of Figure 7;

Figure 9 is a plan in diagrammatic form of a gas generator plant incorporating ten cylinders;

Figure 10 is a cross section taken on the line A—A of Figure 9.

Referring to Figure 1, the power plant comprises a rotary compressor **10** which is indicated as being of the axial flow type. The compressor **10** delivers by the duct **11** to a gas generator which is generally indicated by the reference numeral **12**, and the combustion products are delivered from the generator **12** by way of a duct **13** to a gas turbine **14** which drives the compressor **10** through a shaft **15**. The exhaust gases from the turbine **14** pass through a gas turbine **16** which provides external power at a shaft **17**, the exhaust gases being finally discharged to atmosphere at **18**. Air and/or combustion products also pass from the generator **12** by way of a duct **15***a* to a high pressure turbine **19** which provides external power at a shaft **20**. The exhaust gases from the turbine **19** go by a duct **21** to the duct **13** to assist in driving the turbines **14** and **16**. The shafts **17** and **20** are connected together by gearing **22** to drive a common output shaft **23**. Instead of connecting the turbine **19** to the duct **13** by the duct **21** it may be connected with the duct **11** by a conduit **24** indicated by broken lines.

If desired, an intercooler **25** may be provided between the compressor **10** and the gas generator **12**. Alternatively, or additionally, an intercooler **26** may be provided in the conduit **24** if the latter is provided.

In certain circumstances it may be advantageous to provide a combustion chamber **27** in the duct **15***a*.

Referring now to Figure 2, the gas generator **12** comprises a pair of cylinders **30**, **31** within each of which there is arranged a pair of pistons **32**, **33** and **34**, **35** respectively. The pistons **33** and **35** are coupled together and in the arrangement shown this is achieved by providing a single double ended piston. The pistons **32** and **34** are coupled together by rods **36** secured to each piston at **37**. Finally there is carried by the cylinder assembly a beam **38** which is pivotally mounted at **39**, and has its ends connected by links **40** with the piston **34** and with the double ended piston **33**, **35**. In this way it is arranged that when the pistons **32** and **33** approach one another the pistons **34** and **35** move apart and vice versa, and that such operations are performed in phase.

The wall of cylinder **31** is provided with a plurality of ports **41** which lead to an induction belt communicating with an induction pipe **42**. The cylinder **31** also has a plurality of exhaust ports **43** which communicate with an exhaust belt and exhaust pipe **44**. A similar arrangement is provided for the cylinder **30** and similar references are generally used. The exhaust pipes **44** lead to the duct **13** of Figure 1 supplying the low pressure turbine **14**.

Each cylinder **30** and **31** is provided with a discharge port **45** for supplying gases to the duct **15***a* of Figure 1 leading to the high pressure turbine **19**.

Each port **45** is controlled by a poppet valve **46** having its stem positioned for engagement by a pivoted lever **47** mounted on the cylinder assembly. The gases discharging from the ports **45** act on both sides of the heads of the valves **46** owing to the back pressure in the duct **15***a*, and as the areas on the two sides of the valve head are different there will be a differential pressure on the valve tending to close it. The levers **47** are so positioned that with reciprocation of the pistons **32** and **34**, cam elements **48** having the forms of inclined planes and carried by each piston **32** and **34** will engage the levers **47** to move them and open the valves **46**. The ports **45** lie centrally between the pistons **32**, **33** or **34**, **35**.

The ports **41** and **43** are uncovered when the pairs of pistons have moved fully apart. The ports **41** admit air and the ports **43** discharge products of combustion and air from the cylinders. The inlet ports **41** may, if desired, be so designed as to impart to the entering air a tangential movement so that the air within the cylinder has imparted to it a rotary motion. Air is supplied to these ports by the compressor **10** at slightly above atmospheric pressure and in sufficient quantity to perform the required scavenging.

The rotary compressor **10** may be replaced by a reciprocating compressor driven by the reciprocating movement of the pistons of the gas generator.

In Figure 2 the pistons **34** and **35** are shown as having completed their working stroke and exhaust gases are being discharged through the pipe **44** while scavenge air is being admitted through the induction pipe **42**. The working stroke which has resulted in the pistons **34** and **35** moving apart has also resulted in the pistons **32** and **33** being brought near together. These pistons **32** and **33** are approximately at their nearest when combustion is initiated within the cylinder **30** as later described, whereupon the pistons **32** and **33** will separate and perform the working stroke, at the same time the pistons **34** and **35** being brought together to perform the compression stroke. It will be apparent therefore that the pistons **32**, **33** and the pistons **34**, **35** alternately assume the positions shown in Figure 2.

While each of the pairs of pistons **32**, **33** and **34**, **35** are near together at the end of a compression stroke, the injection and ignition of fuel is effected by any convenient or known means in accordance with diesel engine practice. For example, in Figure 2, the circles "*f*" designate the ends of suitable fuel injector nozzles.

It is arranged that during or immediately prior to the injection of fuel the valve **46** is opened and a certain quantity of compressed air is discharged to the duct **15***a*. This partial discharge is preferably accomplished in part by piston movement and the fuel is injected into the cylinder as soon as the pistons commence to move apart. It is necessary that the energy contained in the gases remaining in the cylinder after the aforesaid partial discharge of air and the combustion of fuel shall be sufficient to accomplish the subsequent working stroke in said cylinder and the simultaneous compression stroke in the other cylinder. In other words the quantity of air which is allowed to escape must not be too great.

It is, moreover, necessary for the steady running of the gas generator that the work output of the working stroke in one cylinder shall substantially equal that required for the compression stroke in the other cylinder, and this may be accomplished in the following manner. The valve **46** is arranged to open at a certain position in the compression stroke and to close at the corresponding point in the working stroke, the opening of the valve being aided by the actuation of the lever **47** as previously described. Discharge of a certain quantity of air will take place at about the conclusion of the compression stroke and combustion is arranged to take place at approximately the beginning of the working stroke. During the start of the working stroke, some air will generally flow back into the cylinder through the open valve, but the quantity of this air will be small owing to the thermal expansion of the air already in the cylinder. Alternatively, it may be arranged that this thermal expansion is sufficient to cause discharge of air through the valve to be continued during this period. In any case the pressure inside the cylinder will be approximately constant while the valve is open, because the valve communicates with the duct **15***a*, which is to be maintained at constant pressure. The arrangement is such that the valve opens at a predetermined position in the compression stroke and closes at the corresponding position in the expansion stroke. Thus the work of the two strokes is equal during the period that the valve is open. Moreover the pressure,

volume, and specific heats of the contents of the cylinder when the valve opens are approximately equal to their pressures, volumes, and specific heats when the valve closes, and thus the energy of the compression stroke before the valve closes is approximately equal to that of the working stroke after the valve has closed. Thus the total energy of the working stroke is automatically made to be approximately equal to that of the compression stroke.

To compensate for the effect of friction, changed specific heats, and other secondary factors which may affect the lengths of stroke of the pistons, suitable means (not shown) may be provided to adjust the time of opening of the valve 46 so that this takes place either shortly before or shortly after the instant when the pressure inside the cylinder becomes equal to that beyond the valve. For example the opening of the valve may be advanced to the instant when the pressure inside the cylinder is somewhat below that in the duct 15*a*. It will then close at the corresponding position in the expansion stroke, leaving the pressure inside the cylinder somewhat higher than it was when the valve opened. This will increase the power of the expansion stroke, as might be required to overcome friction. Similarly, retarding the opening of the valve will have the opposite effect, should this be required to compensate for the changed specific heats of the contents of the cylinder. This adjustment concerns the instant when the valve opens, relative to the instant when the pressure inside the cylinder becomes equal to that in the duct 15*a*, and may be accomplished by adjustment, preferably automatic, either of the valve timing or of the pressure in the duct. Such latter adjustment may be effected for example either by a throttle valve placed in the duct 15*a* or by regulating the amount of fuel injected into the cylinder and thereby the amount of air delivered to the duct.

In case the aforesaid adjustment is imperfect, or if for any other reason the length of stroke of the pistons changes from that required for steady running, the following two further means may be utilized to restore the stroke to its normal length. Some degree of throttling is expected to take place during the passage of air through the valve port 45 and past the valve 46, and this will tend to cause a pressure difference. In the case of an abnormally long compression stroke, this throttling will restrict the return flow into the cylinder, while the valve 46 is open, and consequently the power of the working stroke will be reduced whereby the subsequent compression stroke is also reduced. This throttling effect is inherent in the valve or may be increased by means which restrict the lift of the valve. If the compression stroke is shorter than normal the discharge of air past the valve 46 will continue after the start of the working stroke due to thermal expansion of the air, and the throttling effect referred to above will cause a greater quantity of air than normal to be retained in the cylinder, the result being to afford greater power than normal for the working stroke, thus compensating for the shorter compression stroke. The throttling effect may be provided by the duct through which the air or combustion products are discharged. Thus there is provided means for automatically controlling the length of the piston stroke to maintain it at a mean value required for steady running.

Also means is preferably provided for adjusting the point in the stroke at which fuel injection commences. This point of fuel injection is adjusted so that in the case of the normal stroke length, combustion is completed by the time the valve 46 closes. The effect of combustion is, as previously described, to reduce the air content of the cylinder at the start of the power stroke, and it will thus be seen that if combustion is not completed during the period while the valve 46 is open, this will result in an increased quantity of compressed air being retained in the cylinder and consequently combustion will continue after the valve has closed, automatically increasing the energy for the working stroke.

An additional factor, which in extreme cases is to prevent the pistons from coming into contact, is the tendency of the pistons to close the port 45 as they come together at the end of the compression stroke.

In the arrangement described above the gas generator 12 is adapted mainly to deliver compressed air and this is preferred although combustion products may be admixed with the air. In any event it is desirable that the temperature of the discharging gas shall not be excessive, particularly having regard to the provision of the gas discharge valves 46.

Although the pistons are described as freely movable they may be interconnected when several gas generators are provided. Thus several generators as described with reference to Figure 2 may be grouped together and the pistons of the generators interconnected through a swash plate for operation in a desired sequence, the pistons being otherwise freely movable in the cylinders.

An alternative to the previously described pivoted beam and link for interconnecting the piston assemblies is shown in Figures 3 to 6, and this construction offers further advantages as explained hereinafter. The aligned cylinders 30, 31 of the example of Figure 2 are represented by the liners 50 and 51 of the cylinder assembly which has a jacket 52. The pistons 53 and 54 correspond to the pistons 32 and 34 of the previous example and are connected by rods 54*a*. The double ended piston 55 corresponds to the piston 33, 35 of Figure 2. Each of the pistons 54 and 55 is recessed at 56 and carries an angularly movable cross pin 57 to the centre of which is attached an arm 58 extending outwardly through an opening in the cylinder jacket 52. A hollow shaft 59 is carried in bearings 60 parallel with the axis of the cylinders and to the shaft are secured two pairs of trunnions 61 by means of sleeves 62 and keys 63 engaged with the shaft. The arms 58 are arranged to fulcrum about the trunnions 61, which revolve with the shaft 59 by means of the swash plate construction now to be described. Each of the arms 58 is secured in a wobble plate 64 by a retaining pin 65. Each wobble plate 64 is carried by bearings 66 in a revolving tilt plate 67 which comprises two annular halves as seen in Figure 3. The tilt plates are supported by bearings 68 on the trunnions 61 as shown in Figure 5. Forming part of each tilt plate 67 is an axially extending arm 69 upon which is formed a toothed arc 70. The arcs 70 are respectively meshed with a pair of bevel gears 71 and 72 which are rigidly connected together by a sleeve 73 and are rotatable on the shaft 59 within the middle bearing 60*a*. The arrangement is such that any pivoting of a tilt plate 67 relative to the shaft causes rotation of the sleeve 73 so that the angle of tilt of the two tilt plates is always equal and opposite, and the pistons will move together and apart as described in the previous example. In order to ensure proper operation of the swash plates and to prevent occurrence of high frequency flutter therein while permitting the angles of tilt to adjust themselves as required, it is desirable to provide damping in the rotation of the sleeve 73 relative to the shaft 59. This is afforded by an annular space 74 between the sleeve 73 and the shaft, which space is filled with pressure oil through a passage 75 from the interior of the hollow shaft which is fed from a suitable source through a connection 76. A key 77 is secured to the sleeve 73 and a key 78 is secured to the shaft 59, both these keys extending over the full axial length of the space 74 and having a small radial clearance. For the sleeve 73 to rotate, oil must flow past these keys 77, 78, thus damping is provided, the degree of damping being determined by the radial clearance. By this means the angle of tilt of the swash plates cannot change rapidly, but the provision for relatively slow change permits the length of stroke of the pistons to adjust itself to particular running conditions as required.

Thus the pistons are relieved of undesirable constraint and the swash plates are relieved of any undue stresses which such constraint might impose.

It will be understood that the arrangement of the sleeve **73** which constitutes the interconnection between the swash plates may be varied if desired. Thus a pair of concentric sleeves with concentric bevel wheels could be provided and engaged by toothed arcs on a pair of arms forming part of each tilt plate and disposed on opposite sides of the shaft to balance any side thrust. Also instead of bevel wheels and toothed arcs the ends of the arms **69** could be formed to engage with radial projections on the sleeve or sleeves, the interengagement of the said arms and projections being such that the required motion can be transmitted, e. g. said interengagement could be by a pin and slot type of connection.

The cross pins **57** of the pistons may be axially located by means of spring rings.

The hollow shaft **59** carries a cam **79** arranged to actuate a tappet **80** of a valve **81** which corresponds to one of the valves **46** of the previously described example. A similar valve is provided in the other cylinder, and this is operated by the cam **79***a*. For clarity, parts **80** and **81** are shown only in Figure 4. The shaft **59** also carries a gear wheel **82** adapted for driving pumps (not shown) for effecting the injection of fuel into the cylinder through injection nozzles of any suitable kind (not shown) mounted at **81***a* and the corresponding position on the other cylinder. Additional cams and valves may be provided if required.

The following advantages are obtained with the arrangement described. There is afforded a rotary motion which is in timed relation with the movement of the pistons and can therefore be employed to actuate the fuel injection pumps and the discharge valves, and also the two piston assemblies which comprise a single gas generator arc synchronised in a convenient manner. The rotary motion provided can also be conveniently employed for the driving of auxiliary units or mechanisms, such as cooling and lubricant pumps or scavenging air pumps or blowers for example. It will be appreciated that correct timing of the fuel injection and operation of the discharge valves is important and that these operations must take place when the pistons are nearing the ends of their strokes and their velocity is reduced. Correct timing is not determined by the positions of the pistons relative to their cylinders but by the proportion of their total stroke over which the pistons have travelled at the time in question. Since the length of stroke may vary with the running conditions, a piston position corresponding to the end of the stroke under one set of conditions might be reached before the end of the stroke under other conditions. The arrangement described above therefore affords improved timing of the fuel injection and discharge valve operation under different running conditions.

A preferred construction of the double ended piston **55** of the arrangement described in connection with Figure 3 is shown in Figures 7 and 8, and this construction avoids the necessity for accurate alignment of the liners **50** and **51** in which the piston travels. The cross pin **57** of the arm **58** is mounted in a carrier member **83** which is of double ended piston form and upon the opposite ends of which are mounted two hollow piston heads **84** and **85**. Radial clearance is provided between the carrier **83** and the two piston heads so that the latter can align themselves with their respective cylinder liners **50** and **51**. The carrier **83** has axially extending recesses **86** into which extend corresponding projections **87** on the piston heads and the carrier and piston heads are held together axially by spring rings, one of which is shown at **88**, which lie in annular grooves **89** common to the carrier and head. The axial thrust of the piston heads is transmitted by suitable engaging faces on the head and carrier respectively and the arrangement is such that no thrust either from the operating forces or from thermal expansion is transmitted to the spring rings **88**.

Figures 9 and 10 illustrate the manner in which a number of generator units of the kind shown in Figure 3 may be grouped together and the piston assemblies interconnected by a common swash plate arrangement. The unit comprises five cylinder assemblies **90** each consisting of a pair of cylinders placed end to end. As shown in Figure 10 each cylinder assembly contains a pair of outer pistons **91** and **92** interconnected by rods **93** and a double ended piston **94**. The pistons **91** and **94** have cross pins **95** with arms **96** each connected to swash plate assemblies **97**, which are mounted on a common rotary shaft **98**. The arrangement is generally as shown in Figure 3 for a single generator, but, of course, five arms **96** are connected to each wobble plate (**64** of Figure 3). Sections of Figure 9 at BB, CC, DD and EE would be identical with Figure 10, except for variations in the width of the cylinder exhaust belts **100** and inlet belts **101** and in the axial positions of the pistons in the cylinders. The shaft **98** is supported centrally with respect to the cylinder assemblies **90** in suitable bearings and may be arranged to drive any desired auxiliaries indicated at **99**. Thus all the pistons will operate in synchronism. Cylinder exhaust belts **100** and inlet belts **101** are shown communicating with exhaust ducts **102** and inlet ducts **103** respectively, and the high pressure discharge connections are shown at **104**. It will be appreciated that a considerable saving of space is achieved by this arrangement.

I claim:

1. A free piston compressor or gas generator comprising two axially aligned combustion cylinders, a first pair of axially aligned pistons one in each of said cylinders and rigidly connected in back to back relation, a second pair of axially aligned pistons one in each of said cylinders and arranged so that each piston faces a piston of the first said pair, means rigidly connecting the pistons of said second pair so that they move in unison, whereby movement of the pistons in one cylinder towards each other is accompanied by movement of the pistons in the other cylinder away from each other and vice versa, inlet ports at one end of each cylinder arranged to be uncovered by one of the pistons in said cylinder as the two pistons in said cylinder move away from each other, means for causing scavenging air and air to be compressed to enter said cylinders through said inlet ports, exhaust ports at the other end of each cylinder arranged to be uncovered by the other piston in said cylinder as the two pistons in said cylinder move away from each other to allow said scavenging air and products of combustion to leave said cylinders, means for admitting fuel alternately into each of said cylinders when the two pistons therein are in proximity and for effecting ignition of the fuel to afford an expansion stroke, a discharge valve in communication with the middle region of each of the said cylinders, means for causing each of said valves to open and subsequently close to discharge compressed gas from the cylinder when the pistons therein are near their position of closest proximity, a receiving space for compressed gas, and duct means providing communication between each of said valves and said receiving space.

2. A compressor or generator according to claim 1, further comprising means for limiting the maximum displacement of said piston pairs.

3. A compressor or generator according to claim 1, further comprising a coupling between the first and second pair of pistons and including a double armed lever having a stationary fulcrum and links pivotally connected respectively to the ends of said lever arms and to the pairs of pistons.

4. A compressor or generator according to claim 1, in which said means for causing each of said valves to open comprises a cam mechanism directly driven by one of said pistons in the corresponding cylinder.

5. A compressor or gas generator according to claim 1, in which said valves are normally closed, and said means for causing each of said valves to open comprises cams having inclined faces driven respectively by one of said pistons in each cylinder, and pivoted levers actuated by said cams for opening the valves.

6. A compressor or generator according to claim 1 in which said means for causing said valves to open holds said valves open for a period such that the energy of the gases remaining in the cylinder after such discharge produces a work output from the subsequent working stroke in said cylinder that is substantially equal to the work output required for performing the simultaneous compression stroke in the other cylinder.

7. A compressor or generator according to claim 1, further comprising a coupling between said pairs of pistons including a pair of swash plate assemblies interconnected by a rotary shaft, and means for maintaining the angles of tilt of said swash plate assemblies equal and opposite.

8. A compressor or generator according to claim 1, further comprising a coupling between said pairs of pistons including a pair of swash plate assemblies interconnected by a rotary shaft, and a sleeve rotatable on the shaft, said sleeve being operatively connected to each swash plate assembly whereby any variation in the angle of tilt of one of the assemblies is transmitted to the other assembly and their angles of tilt are maintained equal and opposite.

9. A compressor or generator according to claim 1, further comprising a coupling between said pairs of pistons including a pair of swash plate assemblies interconnected by a rotary shaft, means for maintaining the angles of tilt of said swash plate assemblies equal and opposite, and means for damping the movements of said latter means.

10. A compressor or generator according to claim 1, further comprising a coupling between said pairs of pistons including a pair of swash plate assemblies interconnected by a rotary shaft, a sleeve rotatable on the shaft, said sleeve being operatively connected to each swash plate assembly whereby any variation in the angle

of tilt of one of the assemblies is transmitted to the other assembly and their angles of tilt are maintained equal and opposite, and oil dashpot means opposing rotation of the sleeve relative to the shaft to provide damping in the operative connection between the swash plate assemblies.

11. A compressor or generator according to claim 1, further comprising a coupling between said pairs of pistons including a pair of swash plate assemblies interconnected by a rotary shaft and means for maintaining the angles of tilt of said swash plate assemblies equal and opposite, at least one piston of each pair of pistons comprising a carrier member pivotally connected to a swash plate arm and a piston head mounted with radial clearance on said carrier member.

12. A compressor or generator according to claim 11, in which one of said carrier members carries a pair of piston heads to constitute thereby one of said pairs of pistons.

13. A compressor or generator according to claim 1, comprising a plurality of pairs of end to end cylinders having pairs of pistons as aforesaid, and further comprising coupling means between the pairs of pistons associated with each pair of cylinders having swash plate assemblies interconnected by a common rotary shaft.

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