Aug. 19, 1947. R. J. WELSH 2,425,850

FREE PISTON TYPE INTERNAL-COMBUSTION COMPRESSOR

Filed Sept. 4, 1945 3 Sheets-Sheet 1

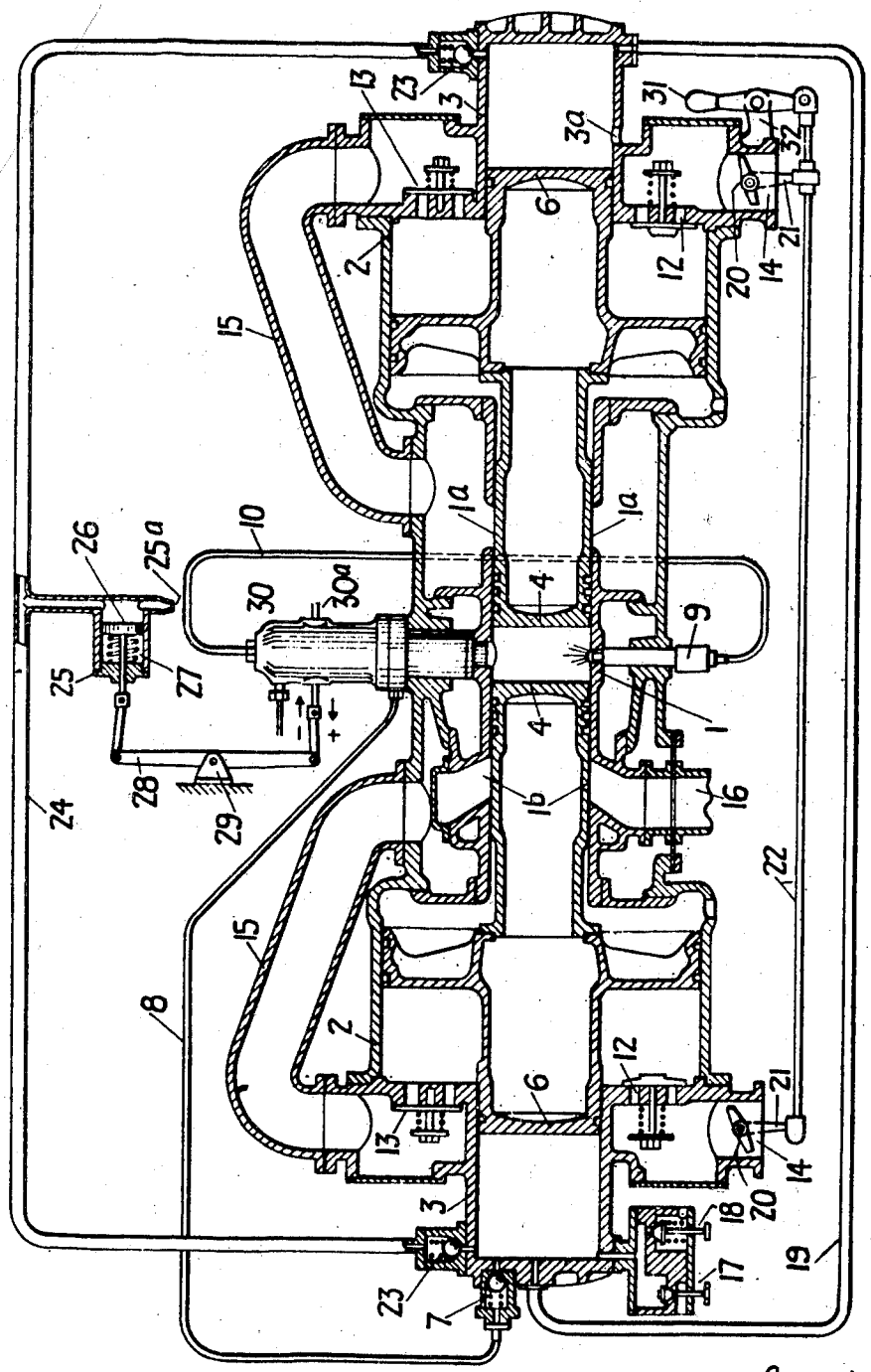


FIG. 1.

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FREE PISTON TYPE INTERNAL-COMBUSTION COMPRESSOR

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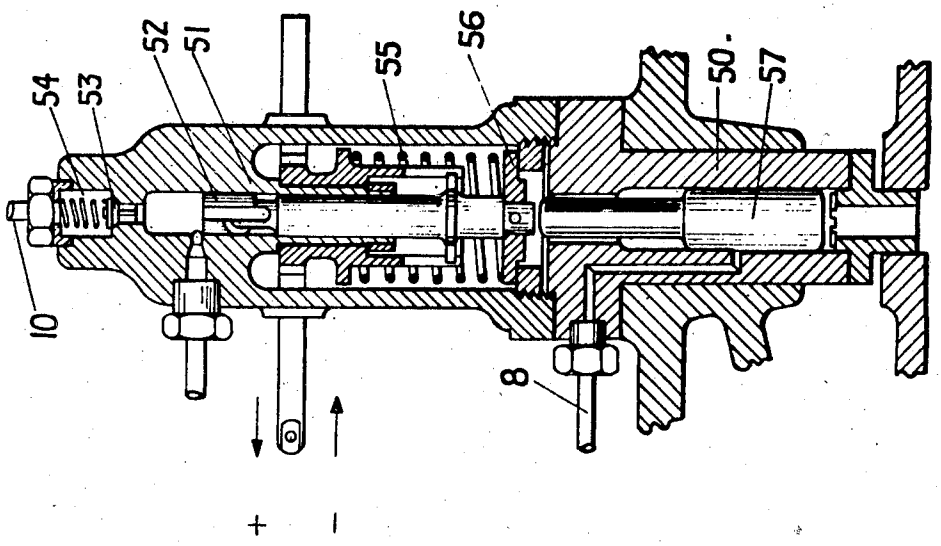


FIG. 5.

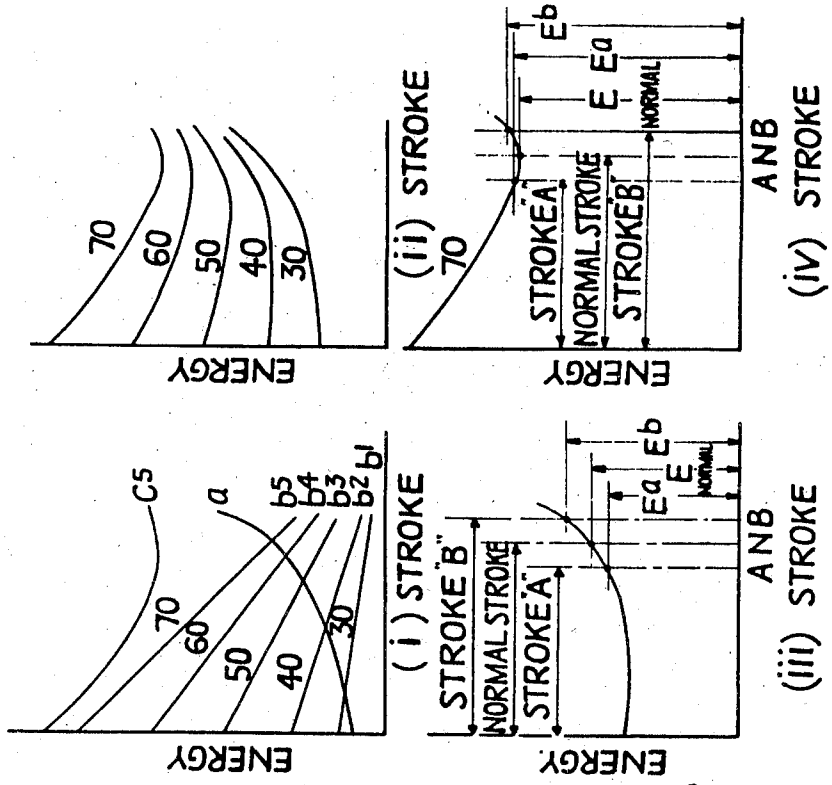


FIG. 2.

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FREE PISTON TYPE INTERNAL-COMBUSTION COMPRESSOR

Filed Sept. 4, 1945 3 Sheets-Sheet 3

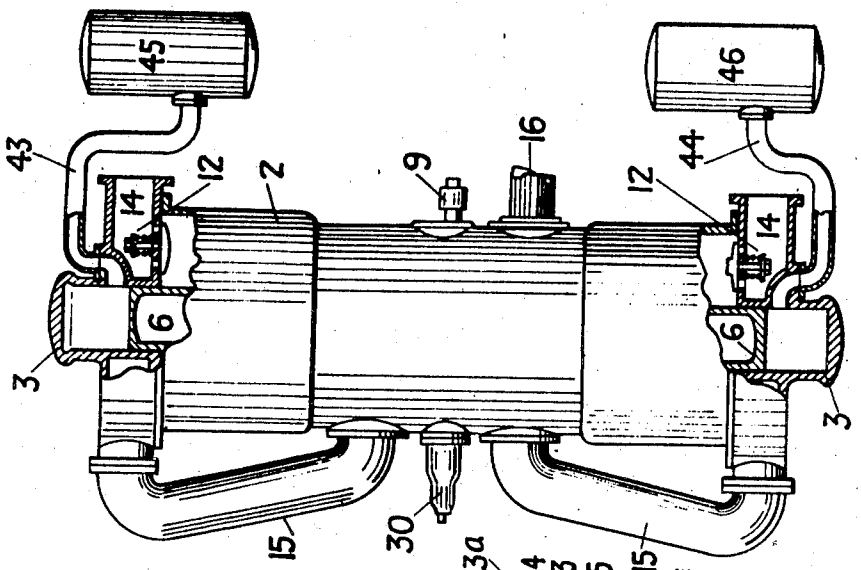


FIG 4.

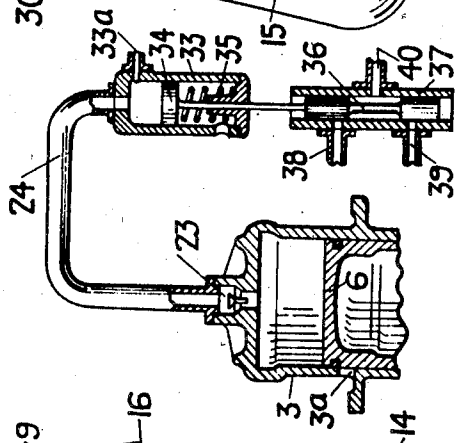


FIG 6.

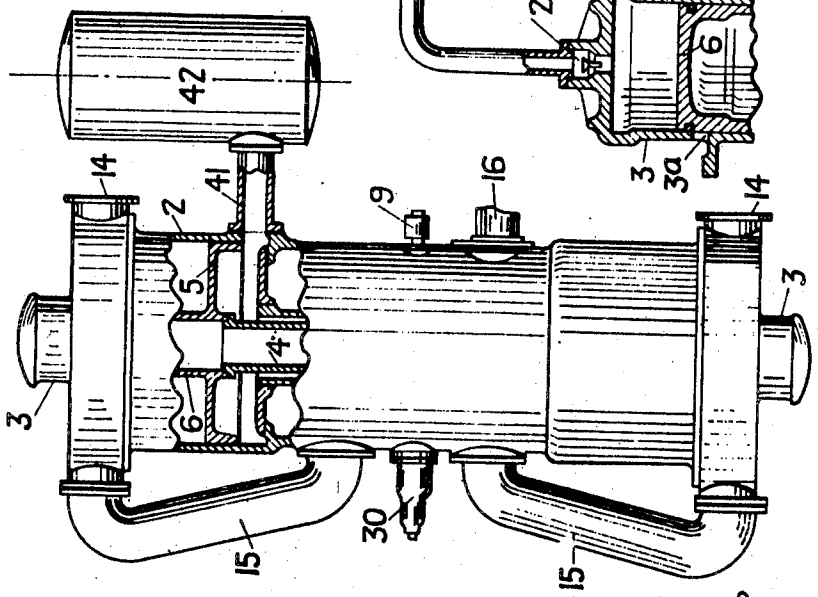


FIG 3.

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Patented Aug. 19, 1947

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UNITED STATES PATENT OFFICE

2,425,850

FREE PISTON TYPE INTERNAL-COMBUSTION COMPRESSOR

Robert James Welsh, Rugby, England, assignor to The English Electric Company Limited, London, England, a British company

Application September 4, 1945, Serial No. 614,306
In Great Britain February 11, 1944

9 Claims. (Cl. 230—56)

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This invention relates to a free piston type internal-combustion compressor—i. e., a device wherein a compressor piston is directly connected to or combined with an internal combustion engine piston instead of being driven thereby through connecting rods, cranks and crankshafts and wherein there may be two opposed engine pistons reciprocable towards and away from each other in one cylinder.

Hitherto in free piston type internal combustion compressors having a pair of opposed engine pistons the latter have been mechanically connected by a synchronising linkage—in the form, for example, of swinging links or of racks and pinions. In a vertically opposed piston arrangement both piston units would tend to fall downwards, but the synchronising linkage hitherto used would cause the weight of one piston to balance the other through this linkage.

It is an object of the present invention to enable mechanical synchronising gear to be omitted and furthermore to allow of an arrangement without such mechanical synchronising linkage to operate vertically. Thus, according to the invention, the general proportions of the compressor are such that the pistons have an inherent tendency to return to inner dead centre positions which are symmetrically disposed about a chosen diametral centre line in the cylinder. If necessary, means may also be provided to damp any tendency for the mid-point of the pistons to oscillate relatively to the said centre line. The fuel supply may be partly under the control of the maximum outstroke made by the pistons so that in the event of a partial seizure of one piston the fuel supply is reduced, thereby preventing a considerably increased impulse being given to the unseized piston.

According to a further feature in a vertical arrangement means are adopted to make the resultant of the forces, including gravity, acting on the pistons, symmetrical about the said chosen centre line which in such case is horizontal.

In carrying out the invention, the parts are so proportioned that over a normal range of scavenge pressure in the combustion cylinder the energy stored in the various cushioning spaces and tending to bring the pistons back to the inner dead centre position increases with outward displacement of the outer dead centre and either

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decreases with inward displacement of the outer dead centre or increases at less rate.

The co-pending patent application No. 512,835 is concerned with an arrangement in which the internal combustion compressor is controlled at least in part in response to a deviation of the outer dead centre of a free piston from a predetermined point so that the compressor device works with substantially a determined outer dead centre. For this purpose a throttle at the inlet to the compressor is opened progressively each time the free piston overruns the said predetermined point and is closed progressively each time the piston stops short of that point.

The co-pending patent application No. 547,592 sets forth an alternative arrangement wherein the motion of the inlet throttle and the fuel supply are so adjusted that the tendency is to maintain substantially a predetermined outward stroke. In both of these cases, however, with a pair of opposed engine pistons, it was contemplated that the latter should be mechanically connected as hitherto by a synchronising linkage.

In carrying out one form of the present invention, the free piston compressor is again designed to tend to maintain substantially a predetermined outer dead centre position. The compressor cylinder diameter and the diameter of any cushion cylinder that may be provided as well as the clearance volumes in these cylinders are so proportioned in relation to the dimensions of the engine cylinder that at all scavenge pressures within the normal working range of the device the total energy of re-expansion contained in the compressed gas trapped in the compressor cylinder or cushion cylinder or both when the pistons are at the said predetermined outer position is always approximately equal to the energy required to bring each piston from the said outer dead centre position to a desired inner dead centre position which will vary in accordance with the scavenge pressure, but will correspond to a substantially constant and preferred compression pressure in the internal combustion cylinder of the device; the proportions are furthermore such that if a piston travels beyond the desired normal outer dead centre, the stored energy tending to return the piston increases, while if the piston stops short of the desired normal outer dead cen-

tre, the said stored energy either decreases or, if it increases, it does so at a less rate than the rate of increase with over travel.

According to a further feature of the invention there are provided one or more cushion cylinders which are arrangned to open to the atmosphere or to any supply of gas at substantially constant pressure when a free piston is at or near its inner dead centre position. The working range of pressure in the cushion cylinder may be maintained within desired limits by the provision therein of at least one automatic valve opening inwards to the cylinder and connected to a supply of gas at low or moderate pressure and alternatively or additionally at least one automatic valve opening outwards from the cylinder and either connected to a supply of gas at a higher pressure or spring-loaded.

The invention is illustrated by the accompanying drawings of which Fig. 1 is a cross section through an opposed piston type horizontal compressor device according to the invention. Fig. 2 is a set of curves illustrating the principle of the invention, Fig. 3 is a vertical section through a vertical opposed free piston compressor device according to the invention, Fig. 4 is a vertical section through a modification of Fig. 3, Fig. 5 is an enlarged section through the fuel pump, while Fig. 6 shows a detailed modification.

Referring to Fig. 1, the body of the free piston type internal-combustion engine-compressor comprises the engine cylinder 1, the pair of compressor cylinders 2 at opposite ends thereof and the pair of buffer cylinders 3 at the extreme outer ends while each free piston unit consists of one of the opposed engine pistons 4 reciprocating in cylinder 1, a compressor piston 5 recipocating in one of the cylinders 2, and a buffer piston 6 reciprocating in one of the cylinders 3. In the head of each compressor cylinder 2 is at least one inlet valve 12 which can put the cylinder 2 in communication with the suction inlet 14 and at least one discharge valve 13 which can put the cylinder 2 in communication with the conduits 15 communicating with the annular space surrounding the inlet ports 1a in the engine cylinder 1. The exhaust ports 1b in the latter open into an annular space communicating with the discharge conduit 16. The engine operates on the compression-ignition principle. Fuel can be injected into cylinder 1 by a fuel injector 9 of conventional form supplied by a fuel pump 30 of the conventional gas-pressure operated type through the fuel pipe 10 and as hereinafter described in more detail with reference to Fig. 5. The arrangement so far described is conventional; the hitherto necessary mechanical synchronising linkage between the free pistons is however omitted.

The air in cylinder 1 has to be compressed by pistons 4 from the scavenge pressure at which it is delivered through conduits 15 to a normal compression-ignition value of pressure on the inward stroke of the pistons. The energy required to effect this compression has to be provided by such air as may be trapped under pressure in the cylinders 2 when the valves 13 close and by the air compressed in the cushion cylinders 3. Thus, the pistons should operate with their desired length of stroke in such a manner that the sum of the energy in the said air in the cylinders 2 and 3 should be equal to the energy required to compress the working charge in the internal combustion cylinder 1 plus the energy required to overcome any pressure which may exist on the front of the compressor pistons 5 plus the energy

required to overcome friction on the inward stroke. Furthermore, the available energy must provide for the work done in moving the opposed piston units from their outer dead centre position to the point at which the scavenge inlet ports 1a and the outlet port 1b are closed. The inward movement of the pistons up to this point does not cause any appreciable compression of the working charge and the energy absorbed in this initial inward movement does not, therefore, contribute towards the attaining of the desired compression pressure in the internal combustion cylinder.

The energy given out by the expansion of the air trapped in any of the clearance spaces may be calculated from the well-known expression

$$E=\frac{P_1V_1-P_2V_2}{n-1}$$

where the value of n is approximately equal to the ratio of the specific heat at constant pressure to the specific heat at constant volume, but will vary somewhat from this ratio depending upon the proportions of the cylinders, the ratio of cylinder surfaces to cylinder volume, the nature of the cylinder cooling and other similar factors. The energy varies with the piston outstroke.

Each of the curves in Fig. 2 represents energy plotted against stroke of one of the free pistons as it approaches the outer dead centre position. The curves are not plotted to show how energy varies over the whole of the stroke, the origin accordingly representing, on the horizontal axis, a position of the piston not far from the outer dead centre. In diagram (i), the curve *a* represents the energy stored in the buffer cylinder 3. This energy increases as the piston moves outwards, and the pressure in the buffer cylinder 3 increases but is of course independent of the scavenging pressure in the internal combustion cylinder 1. The lines *b1*, *b2*, *b3*, *b4* and *b5* represent the energy stored in the air trapped in the clearance space in the compressor cylinder 2 at different working scavenge pressures. This energy decreases with increased outstroke because less air is trapped. The line *b5* represents how in a particular example the energy varies with stroke when the working pressure (i. e., the scavenging pressure in the combustion cylinder) is 70 lbs. per sq. inch absolute; the line *b1* similarly represents the energy variation when the working pressure is, for example, 30 lbs. per sq. inch absolute, while the intermediate lines *b2*, *b3* and *b4* similarly represents intermediate values of the working pressure. The various values for the energy refer to the net energy above atmospheric pressure. The curve *c5* is obtained by adding together the curves *a* and *b5* and therefore shows how in this example the total return energy available from the various cushioning spaces when the device is working with a scavenging pressure of, say, 70 lbs. per sq. inch absolute varies with the stroke within the range of the diagram. Other curves can be obtained by adding together *a* and *b4*, etc., the family of such curves being shown in diagram (ii).

Diagram (iii) shows an analysis of one of the curves of diagram (ii) corresponding to operation at a scavenging pressure in the combustion cylinder of 40 lbs. per sq. inch absolute. If the total return energy at normal stroke (i. e. when the outer dead centre of the piston is in the desired position N) is indicated by "E normal" then E*a* represents the return energy when the outstroke is slightly shorter than normal and E*b* represents

the return energy when the outstroke is a little longer than normal. If the operation is remaining stable at a substantially constant total length of stroke of the two opposed pistons, but if the point exactly midway between the two outer dead centres is displaced from the chosen point on the axis of the cylinder, one piston may be finishing its outward stroke at a point B beyond the normal outer dead centre indicated at N while the other is ending its outward stroke at the other end of the cylinder at a point corresponding to A which is short of the normal outer dead centre position; it will be seen that provided the conditions are such that the return energy varies with stroke as depicted in diagram (iii) the piston which is furthest displaced from the desired mid-position at the end of any one outward stroke will be given a return energy E*b* which is greater than the energy E*a* given to the other piston which is displaced inwards from its normal position, and so there will be an inherent tendency for the two pistons to be returned to inner dead centre positions which are symmetrical about the chosen mid-point.

Diagram (iv) shows an analysis of the curve taken from diagram (ii) corresponding to operation at a scavenge pressure of 70 lbs. per sq. inch absolute. Here it will be seen that an appreciable portion of the slope of the curve on the inward side of the normal outer dead-centre position N is a negative slope as distinct from the positive slope on both sides of point N of the curve in diagram (iii). It will be seen however that even with the curve of diagram (iv), the energy E*a*, when the outer dead centre is at point A, although greater than the energy E normal nevertheless is less than the energy E*b* when the outer dead centre is at the point B, therefore there is still an inherent tendency for the two pistons to be returned to inner dead centre positions which are symmetrical about the chosen mid-point.

These various curves all have shapes depending on various piston and cylinder dimensions which can be altered to give quite different shapes to the curves. The invention is carried out by choice of dimensions and pressures to give the desired curve shapes. The energy stored is made up of a part which increases with outward displacement of the outer dead centre and a part which varies in the reverse way with change of outer dead centre, and accordingly over a considerable energy range it is possible for these two parts of the stored energy to be so relatively proportioned as to give operating conditions represented by curves shaped as diagram (iii) or (iv). Thus, according to this feature of the invention, the various dimensions are so chosen that over a normal working range of scavenge pressures and over the normal range of varying outer dead centre positions the energy stored and tending to move the pistons back to the inner dead centre position increases with outward displacement of the outer dead centre and decreases with inward displacement; or if the energy increases with inward displacement beyond a certain point it does so only at a lesser rate than with outward displacement from that point, there being also means tending to maintain the outer dead centre at that point.

Since the action depends on the proper relationship between the position of a piston **6** and the pressure in the cylinder **3** being maintained, the cushion cylinder **3** is arranged to open to atmosphere through ports **3***a* uncovered by the piston **6** when at or near to the inner dead centre as shown at the right-hand end of Fig. 1. Alternatively,

the pressure in the cushion cylinder **3** is maintained between a desired maximum and a desired minimum by an automatic inlet valve **17** responding to under-pressure in the cylinder **3** and a loaded automatic escape valve **18** responsive to over-pressure as shown at the left-hand end of Fig. 1.

If it be necessary furthermore to provide for damping any oscillations of the piston dead centre positions with respect to the centre line of the compressor device, this may be achieved by connecting the two cushion cylinders **3** by a connecting pipe **19**, the minimum bore of which is chosen so that it is not too large to cause the self-centring effect on the pistons to be lost while being large enough to give a useful damping effect.

Provision may be made to ensure that there is a tendency to maintain the outer dead centre of each piston at the point N indicated in Fig. 2 more especially if on occasions the operating conditions are represented by diagram (iv). This is done by control of the volumetric efficiency of the compressor. For this purpose, the inlets **14** to the compressor are provided in known manner with inlet throttles **20** secured to actuating levers **21** linked together by the actuating link **22**. Each of the cushion cylinders **3** is connected through a non-return valve **23** and pipe **24** to the servo-motor cylinder **25** wherein reciprocates the piston **26** biased by the spring **27** and linked to lever **28** which turns about fixed pivot **29** and is linked to the rack or other fuel adjuster **30***a* of the fuel pump **30**. The cylinder **25** is provided with an appropriate leak **25***a* and the spring **27** acts in a sense tending to increase the fuel supply.

The operation is as follows:

If it be desired to reduce the output the throttles **20** are partially closed. This can be done by movement of regulating lever **31** turning about fixed pivot **32** to move the actuator link **22**. Alternatively, it will be understood that if the outlet **16** provides motive fluid to a gas turbine, the link **22** can be actuated under the control of the turbine speed governor whereby an increase in speed will partially close the throttles **20**. Such closure will have no immediate effect on the output of the device but will cause an increase in piston stroke, and therefore an increased maximum pressure in both cushion cylinders **3**. Provided that at least one of these pressures is sufficient to open one of the non-return valves **23**, this pressure will be applied to the servo-motor piston **26** which will move the fuel adjuster **30***a* in a sense to decrease the fuel supply. This reduction in fuel supply (which will reduce the output of the device) will tend to reduce the stroke of the pistons until the pressure in the cylinder **25** balances the force of spring **27** which force changes only inappreciably over the full operating stroke. The leak **25***a* is so proportioned that when the pistons **6** are operating with an outer dead centre point corresponding to the point N in Fig. 2 the pressure in the cylinders **3** causes the air supply to cylinder **25** to balance the leakage at a pressure in the cylinder **25** which just balances the force of spring **27**. Similarly, on a decreasing stroke (in response, for example, to opening of the inlet throttles **20**) the fall in pressure in the cylinders **3** will result in the non-return valves **23** closing until the spring **27** has forced the servo-motor piston **26** back—by continued leakage—to a position in which the increased fuel supply again produces the correct stroke of the free pistons.

In the modification shown in Fig. 6, the non-

return valve 23 and pipe 24 connect the cushion cylinder 3 to the cylinder 33 of an auxiliary servo-motor having a proportioned leak 33a. Within this cylinder is a piston 34 biased by a spring 35 having a large rate of change of pressure with movement in either direction about a normal intermediate position. The piston 34 is connected to a valve 36 which can move in cylinder 37 to either side of a neutral position to connect either the pressure supply pipe 38 or the drain pipe 39 to the supply pipe 40 going to the servo-motor cylinder 25 which now has no leak 25a. If the outer dead centre deviates from the desired point, the pressure in cylinder 33 at which the supply balances the leakage through 33a will correspond to a position of the spring 35 such that the piston 34 moves the valve 36 to set the servo-motor 25 in motion until the altered fuel supply has restored the dead centre and hence the pressure in cylinder 33, the pressure in cylinder 35 and the position of piston 34 to normal.

In the event of a partial seizure of one free piston assembly an impulse considerably above that normally due to the combustion of the fuel might be given to the other piston unless the fuel supply be promptly reduced. With the arrangements described wherein the control of the fuel supply is dependent at least in part on the outward stroke of each of the opposed free piston units this objection is overcome. Seizure of one free piston unit will tend immediately to cause an increased out-stroke of the other piston unit and an increased pressure in one of the cushion cylinders 3 which will lead to operation of the servo-motor piston 26 to reduce the fuel supply on the next stroke.

It is frequently desirable to provide a free piston internal combustion compressor with what may be called a running indicator—e. g., for such purposes as are set forth in the co-pending patent application No. 546,360; hitherto a running indicator has been actuated by the synchronising linkage but with an arrangement according to the present invention the running indicator actuator is a duplicate of the cylinder 25, piston 26 and spring 27.

In the absence of the synchronising linkage, the fuel pump 30 is operated by a pressure responsive device in communication with the compression space in cylinder 1 in accordance with expired British Patent No. 226,523. As shown in Fig. 1, and to a larger scale in Fig. 5, this device comprises the lower cylinder 50 and the pump cylinder 51. Reciprocating in the latter is the pump plunger 52 which can pump a quantity of liquid fuel through the non-return valve 53 loaded by spring 54, the fuel passing into the pipe 10, the whole of this construction being that of the conventional fuel pump. The plunger 52 is biased to its lower position by the spring 55 acting against the washer 56. Reciprocating in the lower cylinder 50 is the pump-operating piston 57 exposed to the pressure in the combustion cylinder 1. Under the force of this pressure, this piston can move upwards, its upper end engaging the lower end of the pump plunger and moving the latter against the force of spring 55 to operate the pump. According to a feature of the present invention, the lower cylinder 50 is in communication through non-return valves 7 (Fig. 1) and restricted pipe 8 with at least one of the cushion cylinders 3. By this means a small quantity of clean air from cylinder 3 will be introduced around the operating piston 57 at each stroke and there will be a general direction of flow of air from this fuel pump operating device down the communicating passage into the engine cylinder. This tends to reduce the formation of gummy or carbonaceous deposits on the fuel pump operating piston 57.

In Figs. 3 and 4, the cylinders and pistons 1 to 6, the valves 12 and 13, the inlets 14, the scavenging conduits 15, the outlet 16, the fuel pump 30, its means of operation and of control are as described with reference to Fig. 1 although some details are omitted from Figs. 3 and 4 for clarity. In each case the axis of the cylinders is vertical. According to a further feature of the invention, the weight of the free piston units is balanced by connecting compressor and/or cushion cylinders to reservoirs wherein the gas pressure is such as to balance the weight. Thus in Fig. 3, the upper compressor cylinder 2 is connected below the lowest position of the piston 5 and by the pipe 41 of sufficient diameter to form a substantially unrestricted connection to the reservoir 42 wherein the gas pressure is such that when this pressure is applied to the effective area of piston 5 exposed to this pressure the resultant force is equal to twice the weight of the upper piston unit. If the pipe 41 and reservoir 42 are sufficiently large there will be no appreciable change in the pressure in the space at the front of the compressor piston during the upward and downward strokes thereof. The upper piston will thus have a tendency to move to the outer dead centre position under the influence of force equal to its own weight (i. e., the gas pressure force minus the force due to gravity). Provided that the other requirements of synchronisation without synchronising linkage are met, as hereinbefore set forth, the motion of the pistons will be symmetrical about a fixed horizontal centre line.

In Fig. 4, the top and bottom cushion cylinders 3 are caused to communicate through pipes 43 and 44 with reservoirs 45 and 46, respectively, from ports uncovered by the pistons 6 as they approach their inner dead centre positions. If the pressure in the lower reservoir 46 be higher than that in the upper reservoir 45 there will be a force of varying magnitude acting upwards on the lower of the free piston units throughout the whole of each cycle of operation. By suitable choice of the pressure difference the mean upward force on the lower of the pistons 6 may be made equal to the sum of the weights of the two piston units in which case there will be a tendency for the lower piston to move towards the inner dead centre position under the influence of a force substantially equal to its own weight—assuming that the two units are of equal weight. This is precisely the effect which gravity has on the upper piston unit and, other things being equal, the resulting motion of the two pistons will be symmetrical about a fixed centre line.

Clearly the arrangements according to Figs. 3 and 4 can be combined by use of the reservoir 42 of Fig. 3 in combination with the reservoir 46 of Fig. 4, whereby the inward thrust applied to the lower piston 6 by the pressure in reservoir 46 added to the outward thrust applied to the upper piston 5 by the pressure in the reservoir 42 substantially balances the weight of the free piston units.

It will be understood that any of the features described with reference to Figs. 1, 2 and 5 can be used in conjunction with any of the arrangements described with reference to Figs. 3 and 4. The invention is not restricted to the particular method of control of fuel injection in combina-

tion with inlet throttles as described with reference to Fig. 1 but other methods can be used tending to maintain the desired outer dead centre. The invention furthermore is not limited to the particular and symmetrical piston arrangement described but can be adapted to other piston arrangements.

What I claim as my invention and desire to secure by Letters Patent is:

1. A free piston type internal combustion compressor comprising in combination a combustion cylinder, two compressor cylinders and two cushion cylinders, two opposed free piston assemblies each consisting of a power piston, a compressor piston and a cushion piston adapted to reciprocate mechanically independently of one another in the said cylinders; variable throttle means at the inlet to the said compressor cylinders adapted to be controlled by an external regulator and to act as stroke control means; a fuel pump in fuel supply connection with the said combustion cylinder, means for operating said pump, a fuel quantity regulator on said pump and piston stroke responsive means in operable connection with the said cushion cylinders and with the said fuel quantity regulator in a sense to reduce the fuel supply in response to excessive piston outstroke; the piston diameters and lengths of clearance space in the respective cylinders being proportioned in relation to the working pressure to give the said piston assemblies an inherent tendency to return to the inner dead centre positions which are symmetrically disposed about a chosen diametral plane through the cylinder.

2. A free piston type internal combustion compressor comprising in combination a cylinder having combustion, compressor and cushion parts and two opposed composite piston units operating therein mechanically independently from one another, the cushion cylinder portion being closed over part of the length of the piston stroke but including a port uncovered by the piston, a maximum pressure valve and a minimum pressure valve arranged between said port and the open atmosphere adapted to open at predetermined pressures when the piston is adjacent to its respective dead centre position, the compressor cylinder part and the cushion cylinder part and the lengths of clearance space bounded by piston faces and the cylinder ends being proportioned in relation to the working pressure to store energy giving the pistons an inherent tendency to return to inner dead centre positions which are symmetrically disposed about a chosen diametral plane through the cylinder.

3. The combination according to claim 1 including a small-bore oscillation-damper pipe connecting said cushion cylinders.

4. A free piston type internal combustion compressor comprising in combination a combustion cylinder, two opposed piston units operating therein mechanically independently of one another, cushion cylinders formed on said cylinder and parts formed on said piston units which operate in said cushioning cylinders, a fuel pump in fuel supply connection with said combustion cylinder, means for operating said pump, a fuel quantity regulator on said pump, a fluid pressure servo-motor in operable connection with said fuel quantity regulator, and including a cylinder having a proportioned leak, a spring biasing the said fuel quantity regulator in a direction tending to maintain the supply, a pipe connection from said servo-motor cylinder to the said two cushion cylinders putting the servo-motor cylinder in communication with the pressure in the cushion cylinders in a sense opposing the action of the said biasing spring, and non-return valves in said pipe connection.

5. A free piston type internal combustion compressor comprising in combination a combustion cylinder, two compressor cylinders and two cushion cylinders, all of these cylinders being arranged on a common, substantially vertical centre line, two opposed free piston assemblies each consisting of a power piston, a compressor piston and a cushion piston adapted to reciprocate mechanically independently of one another in the said cylinders, fluid pressure means connected to the underside of one of the said pistons in a sense to counterbalance the piston weights, the piston diameters and lengths of clearance space in the respective cylinders being proportioned in relation to the working pressure to give the said piston assemblies an inherent tendency to return to the inner dead centre positions which are symmetrically disposed about a chosen diametral plane through the combustion cylinder.

6. The combination according to claim 5 including an air storage reservoir in communication with said cylinder immediately underneath a compressor portion of the upper one of said pistons, the pressure in said reservoir having such a value that the product of this pressure by the exposed area of the piston is equal to twice the weight of the upper piston.

7. The combination according to claim 5 including an air storage reservoir at one pressure in communication with the upper of said cushion pistons to apply downward air pressure thereto, a second air storage reservoir at a different pressure in communication with an under-face of the lower of said cushion pistons in a sense to provide an upthrust thereto, the differences between the pressures in the two said reservoirs being approximately sufficient to balance the weight of the two opposed free pistons.

8. The combination according to claim 1 including a fuel pump in fuel supply connection with said combustion cylinder, a piston in driving connection with said pump and exposed to the pressure in said combustion cylinder, a cylinder wherein said driving piston reciprocates, and a pipe connection adapted to convey comparatively pure scavenging air from one of the said cushion cylinders to the driving piston of the said fuel pump.

9. The combination according to claim 4 including a cylinder having a proportioned leak, a piston adapted to move in the said cylinder, a spring biasing said piston, a piston rod extending to the outside of said cylinder, a pointer on said piston rod, a scale attached to said cylinder, the said pointer and scale serving as a running indicator, a pipe connection from the said cylinder to the two cushion cylinders putting the said running indicator cylinder in communication with the pressure in the cushion cylinders in a sense opposing the action of the said biasing spring, and non-return valves in said pipe connection.

ROBERT JAMES WELSH.

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