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METHOD AND APPARATUS FOR THE CONTINUOUS REGULATION OF
THE OUTPUT OF PISTON PUMPS AND PISTON COMPRESSORS

3,511,582

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3 Sheets-Sheet 1

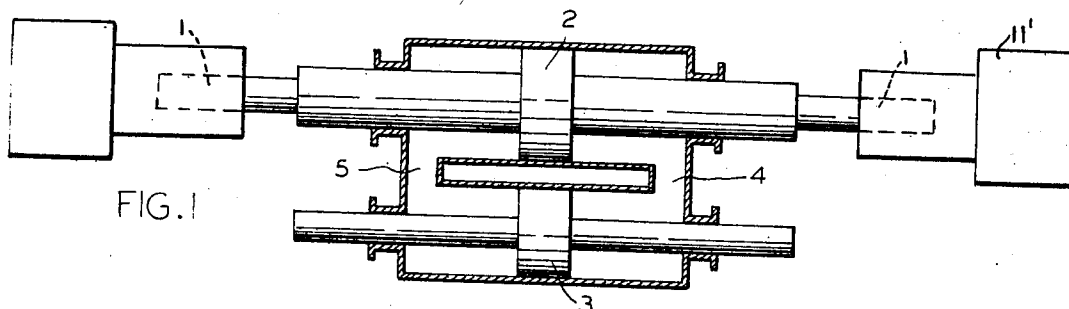


FIG. 1

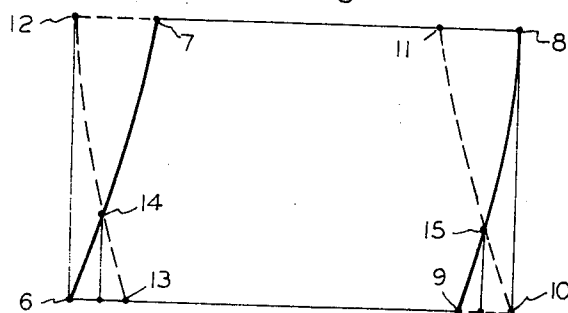


FIG. 2

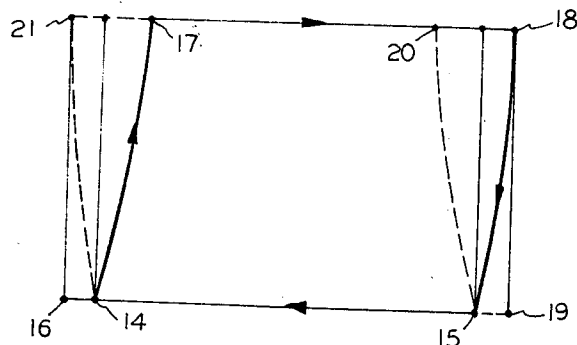


FIG. 3

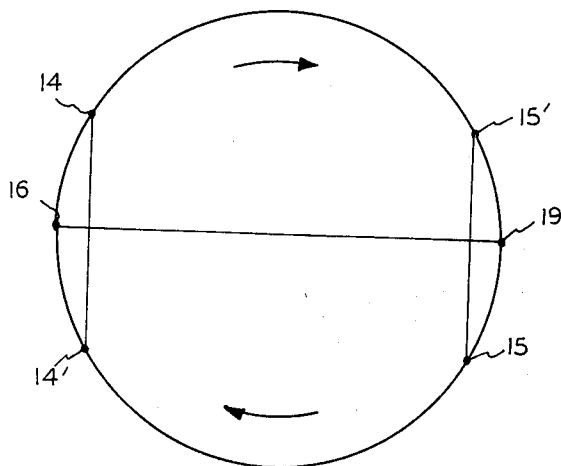


FIG. 4

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FIG. 5

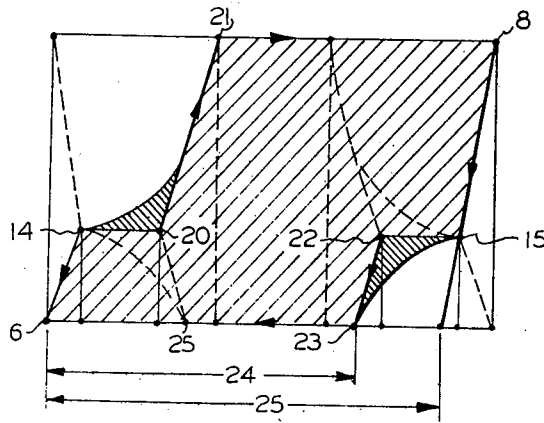


FIG. 6

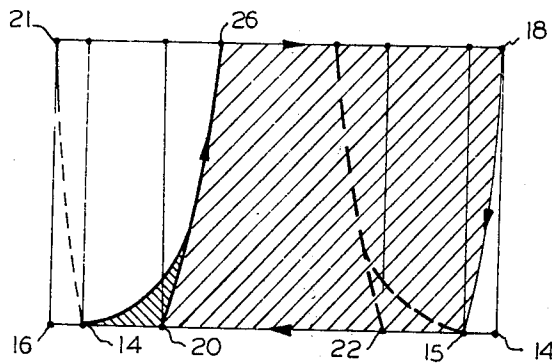
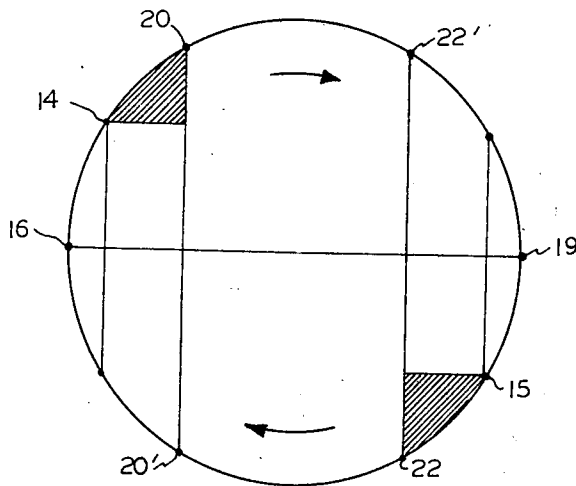


FIG. 7



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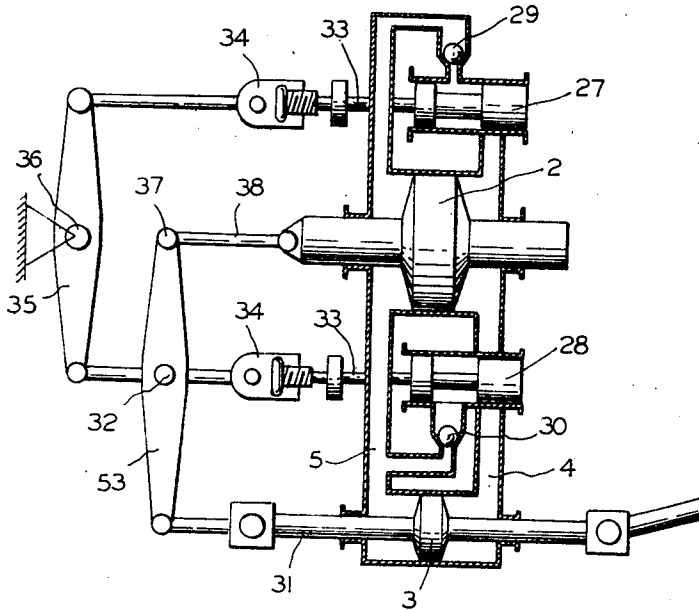


FIG. 8

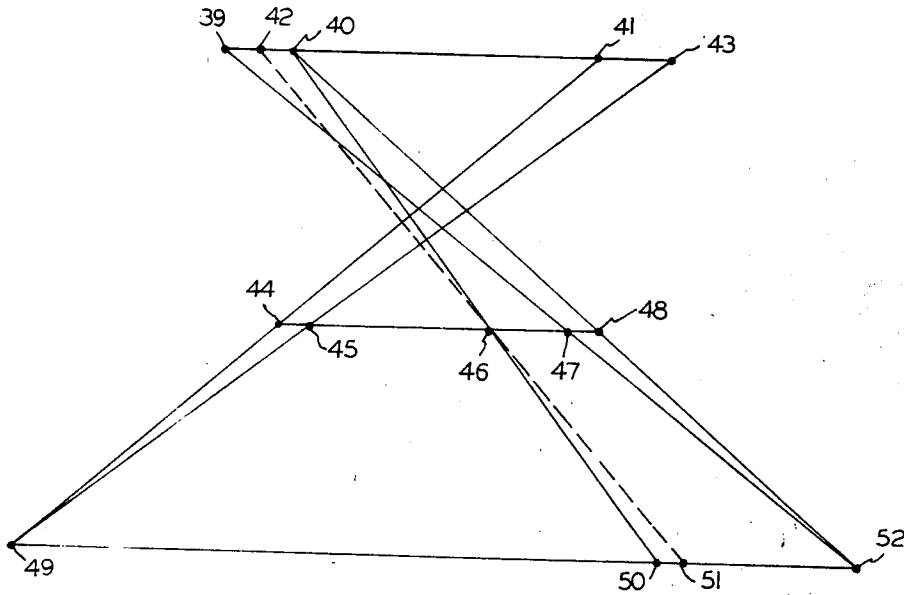


FIG.9

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METHOD AND APPARATUS FOR THE CONTINUOUS REGULATION OF THE OUTPUT OF PISTON PUMPS AND PISTON COMPRESSORS

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7 Claims

ABSTRACT OF THE DISCLOSURE

An apparatus for the continuous regulation of the output of piston pumps and piston type compressors having transmission of the driving power from a mechanically driven driving piston to a working piston rigidly connected with the pump or compressor piston by means of a hydraulic medium filling two cylinder end chambers separated by said driving piston and working piston. This is done without changing the speed of the mechanical drive by making connection between the end chambers in each stroke period from that instant at which the pressure of the hydraulic medium on opposite sides of the driving and working pistons is practically equal and continuing for an adjustable time interval forming a predetermined fraction from zero upwards of the stroke period. The working piston and thereby the high pressure piston are brought to rest during the said time interval in each stroke, and the chambers are disconnected for the remainder of the stroke.

The invention relates to a method and apparatus for the continuous regulation of the output of piston pumps and piston compressors, especially pumps and compressors working at very high pressures, having hydraulic transmission of the driving power from a mechanically driven driving piston, e.g. driven by a crank or the like, to a working piston rigidly connected with the pumps or compressor piston, without changing the speed of the mechanical drive.

With pressures which lie above e.g. 1000 atmospheres, in the present state of the art it is not economic in compressors and pumps of constant speed to use regulating devices of known construction as for example auxiliary chambers of variable size or suction valves which are held open over an adjustable part of the pressure stroke by abutment devices.

A limit is set to the rational use of these modes of regulation because along with the difficulty in coping with the adjusting force necessary at high pressures, the constructional design presents extremely difficult problems.

For these reasons in practice relatively expensive driving units of variable rotational speed have come into use for regulation of the output of such high pressure piston pumps and compressors.

In the method according to the invention from that instant at which the pressure of the hydraulic medium on opposite sides of the driving and working pistons is practically equal, the two end chambers separated from one another by the driving and working pistons are connected together for an adjustable time interval forming a predetermined fraction (which may be zero) of one stroke period, whereby the working piston and thus the high pressure piston are brought to rest during the said time interval during each stroke.

Apparatus serving for carrying out this method comprises a first cut-off valve controlled in dependence on

the movement of the driving piston and arranged in a first transfer duct between said end chambers, a first non-return valve arranged in said first transfer duct, a second cut-off valve also controlled in dependence on the movement of the driving piston, and arranged in a second transfer duct between said end chambers. The movements of the cut-off valves being in opposite phase, and a second non-return valve arranged in said second transfer duct, the non-return valves being so disposed and the cut-off valves so timed that each transfer duct is opened by the non-return valve when the pressure in said end chambers equalise if the cut-off valve is open and is closed by the cut-off valve after a predetermined fraction (which may be zero) of the stroke.

An example of an apparatus according to the invention and its mode of operation are illustrated in the accompanying diagrammatic drawings.

FIG. 1 shows the arrangement of the pistons for the hydraulic power transmission;

FIG. 2 is a diagram showing pressure of the fluid medium being pumped or compressed, plotted against the movement of the two high pressure pistons, at full load;

FIG. 3 is a diagram showing the pressure in the hydraulic driving medium plotted against the movement of the driving piston, at full load;

FIG. 4 is a valve diagram at full load, that is with the cut-off valves closed;

FIG. 5 is a similar diagram to FIG. 2 but at less than full load;

FIG. 6 is a similar diagram to FIG. 3 but at the same fraction of full load as in FIG. 5;

FIG. 7 is a corresponding cut-off valve diagram;

FIG. 8 shows the complete mechanism with cut-off valves, synchronising lever and non-return valves, and

FIG. 9 is a diagram illustrating the mode of operation of the synchronising lever.

In the hydraulic power transmission shown in FIG. 1, two high pressure pistons 1 are in known manner carried and moved by a working piston 2 which is actuated through oil or other hydraulic medium acted upon by a driving piston 3. The pistons 1 are of smaller diameter than the piston 2 so that the pressure in the hydraulic medium is reduced compared with that in the fluid being pumped or compressed. The high pressure cylinders 11' of the compressor or pump in which the pistons or plungers 1 work lie on opposite sides of the machine. The effective surface of the working piston 2 is made greater than that of the driving piston 3 (as more clearly indicated in FIGURE 8) to reduce still further the pressure developed in the hydraulic medium the pressure in which is in the inverse ratio of the effective surface of the working piston 2, to that of the driving piston 3 which is mechanically driven in known manner, the piston strokes occurring in phase but in opposite direction, and their lengths also being in the same inverse ratio.

It is possible to dispense with regulation devices in the high pressure part of the pump or compressor and instead to provide equally effective devices in the hydraulic medium circuit which is under proportionate but lower pressure. Hereinafter oil will usually be referred to as the hydraulic medium.

In the diagram in FIG. 2 the course of the pressure of the fluid being compressed acting on the right hand high pressure piston 1 at full load during one complete stroke cycle is represented by the solid line 6-7-8-9-6. The dash line 12-13-10-11-12 shows the course of the pressure of the fluid medium occurring at the same time and acting on the left hand high pressure piston 1. The diagram in FIG. 3 shows the course of the hydraulic medium pressure in the cylinder end chambers 4 and 5 during power transmission at full load, neglecting the effects produced

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by compressibility of the oil, and inertia of the moving piston in order to simplify the drawing.

In the diagram of FIG. 3 the course of the oil pressure in the cylinder end chamber 5 is shown by the solid line 16-14-17-18-15-16 and the course of the oil pressure at the same time in the cylinder end chamber 4 by the dash line 21-14-19-15-20-21. When the assembly 1, 2 moves to the right with full pressure acting on the right hand high pressure piston 1, the course of the oil pressure at the cylinder end 5 follows 16-14-17-18 and at the same time the pressure at the cylinder end 4 during the suction stroke of the left hand high pressure piston 1 follows 21-14-19.

With these courses of oil pressure, a stroke adjustment of the working piston 2 and thereby of the high pressure pistons 1 for regulation could be achieved in a very simple manner by establishing a constant connection between the two cylinder ends 4, 5 through a regulatable throttle valve. Regulation in this manner however involves very high losses because even at reduced output the total volume of oil would still have to be displaced under full operating pressure. The power consumption would therefore not be reduced with reduction of output. Such throttle regulation would only be reasonable if so little reduction in output needed to be provided for that the power loss would not be worth consideration.

Regulation of the output with low loss is made possible however if by means of a controlled cut-off valve the pressure on both sides of the working piston 2 is equalised and the oil allowed to flow across, the working piston 2 and with it the high pressure pistons mounted on both sides thus being brought to rest in the periods of transfer flow.

In FIG. 2 the line 6-7-8-9-6 indicates the pressure at full load for the right hand high pressure piston 1 and the line 10-11-12-13-10 the pressure for the left hand high pressure piston 1. The compression lines and expansion lines cross at the points 14 and 15. At these two points therefore the pressures acting on the left hand and right hand high pressure pistons 1 are the same and in consequence again ignoring compressibility of the oil and inertia effects at the cylinder end chambers 4 and 5, the oil pressure must be equal at the points 14 and 15 of the diagram of FIG. 3.

If now as indicated in FIG. 6 a connection is established at the point 14 through a cut-off valve and a suitably directed non-return valve between the two cylinder ends 4 and 5, the oil flows from one side of the working piston 2 to the other side, the working piston 2 and the two high pressure pistons 1 coming to rest during this period. As can be seen from FIGS. 5 and 6 the pressure on the working piston 2 and on the high pressure pistons 1 no longer rises during this period. After a certain valve movement set at choice by variation of the valve rod length or of the crank angle in the case of rotary cut-off valves, the controlled cut-off valve which had already opened at the point 14 is again closed at the point 20, FIG. 6.

At the same time at the point 20 (FIG. 5) the compression interrupted over the period 14-20 commences and continues to 21, whereafter the delivery period follows from 21 to 8. In FIG. 6 the oil transfer terminates at point 20. The pressure rises from the point 20 to the point 26 whereafter the pressure continues at the same value until the point 18. From the beginning of the compression at point 20 up to the end of the delivery period at point 8 (according to FIG. 3) or at point 18 (according to FIG. 6) the movement of the working piston 2 and thus of the two high pressure pistons 1 is positively affected. During the suction stroke the non-return valve opens at point 15 at which the oil pressures on the two sides of the working piston 2 are again equal so that the oil can flow from one side of the piston to the other,

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when the working piston 2 and therewith the two high pressure pistons 1, again come to rest.

If further transfer of oil is stopped by the cut-off valve at the point 22, uninterrupted expansion continues to point 23 where suction commences. The diagram of the reduced output according to FIG. 5 is characterised by the hatched area 6-14-20-21-8-15-22-23-6, and according to FIG. 6 by the hatched area 20-26-18-15-20. If the transfer is free of loss these areas of the diagram in FIG. 5 and FIG. 6 are equal and represent the power requirements.

The suction stroke which in the full load diagram is equal to the distance 25 (FIG. 5) is shortened by the regulating operation to the distance 24. It will be seen that the distances 24 and 25 are proportional to the areas of the corresponding diagrams, i.e. the reduction in power requirement is as great as the reduction in the suction load and the regulation is theoretically free of loss.

Practically such regulation cannot be effected completely free of loss. If the cut-off valves were suddenly closed at the points 20 and 22 and the working piston 2 thus had to be accelerated suddenly from rest to full speed, the acceleration would take on the character of a shock. It is therefore necessary to accelerate the working piston 2 gradually from rest which can easily be achieved by gradual closure of the valves. By this measure the sharp corners of the diagrams 20 and 22 are rounded off and losses arise the amounts of which are shown in FIGS. 5 and 6 by the cross hatched areas. These losses are somewhat increased by the fact that in the flowing over of the oil at 14-20 or 15-22 a small increased pressure is necessary to overcome the flow resistance in the valve.

The total amount of losses remains extraordinarily small however and the method permits economical regulation to be effected. In the present method the connection between the cylinder ends 4 and 5 is established by the cut-off valve shortly after the dead point and is closed again by the valve after a period which is adjustable.

The mechanism for regulating the double acting driving cylinder is shown in FIG. 8 which among other things shows a slide cut-off valve and a non-return valve for each end of the cylinders disposed in a corresponding transfer duct connecting the end chambers 4 and 5. The cut-off valves are driven from the driving piston 3 and run synchronously therewith. The mode of operation of the valves is shown in FIG. 7.

During movement of the driving piston 3 to the left the flow of oil to the cylinder end chamber 4 is controlled by a slide valve 27 which at the predetermined setting assumed, opens at the point 20' (FIG. 7) and remains open over the whole period 20'-16-11-20. The valve opening is, however, ineffective up to the point 14 because a non-return valve 29 is provided which is so directed that it is held closed when the pressure in chamber 5 is higher than in chamber 4 and it only permits flow of the oil through the slide valve when the relative values of the oil pressures in the chambers 4 and 5 on opposite sides of the piston 2 change over at point 14. In this way the result is obtained that effective opening starts exactly at the point at which the pressure on opposite sides of the piston 2 equalises and that if the position of the point 14 is changed adaptation takes place automatically. In addition at the point 14 the slide valve is widely open which is desirable to avoid large transfer flow losses. The opening decreases gradually up to the point of closure 20 whereby the desired rounding off in the diagram is achieved.

A second slide valve 28 for the other cylinder end chamber 5 is located in a second transfer duct connecting the chambers 4 and 5 and opens at the point 22' and remains open over the whole period 22'-19-15-22. Opening, however, only becomes effective at point 15 because here again an appropriately directed non-return valve 30 is in-

corporated which only opens with the reversal of the relative values of the pressure on opposite sides of the piston 2.

To adjust the instant at which the valves 27, 28 close and hence the output, it is advantageous to displace the periods 20-20' and 22'-22 parallel to themselves. This is most simply achieved by changing the lengths of the valve rods during operation. If there is no regulation, that is at full load operation, as the diagram according to FIG. 4 shows, the valve rod lengths are so set as described below that the transfer flow opening remains closed at the points 14 and 15.

FIG. 8 shows diagrammatically a driving and adjusting mechanism for the slide valves 27 and 28. The first slide valve 28 is driven from the driving piston 3 through a rocking lever 53. One end point of the lever 53 is connected to the piston rod 31 of the driving piston 3. The other end point 37 of the lever 53 will for the moment be assumed fixed. The valve 28 is driven from the point 32 on the lever 53 and thus makes a movement which is synchronous with that of the driving piston 3 but on a smaller scale. If now the driving piston 3 moves to the right that is to say when pressure acts in the chamber 4, the valve 28 will also move to the right and at a certain instant will open a connection between the chamber 5 and the non-return valve 30. This connection remains ineffective, however, because the non-return valve 30 prevents a transfer flow of the oil into the chamber 5. The non-return valve 30 only opens during the movement of the driving piston 3 to the left when the pressure in the chamber 5 rises above the pressure in the chamber 4, i.e. at the point 15 in the diagram according to FIG. 7. The instant of opening of the valve 28 can be adjusted at will by lengthening or shortening the corresponding valve rod 33. This is indicated diagrammatically in FIG. 8, e.g. by a screw sleeve 38 adjustable during operation. The other chamber 5 is controlled by the slide valve 27 the movement of which must be opposite to that of the valve 28. Accordingly the valve 27 is driven through a reversing lever 35 with fulcrum at 36. So that even with shortened stroke at partial output the mid-point of the stroke may always coincide with the mid-point of the length of the working cylinder, the upper end point 37 of the lever 53 is not in fact fixed as was above assumed but is connected through a link 38 with the working piston 2. The mode of operation of the link 38 is diagrammatically shown in FIG. 9.

First the conditions at full stroke will be considered. At the left hand dead point the driving piston 3 is at the point 49 and at the same time the working piston 2 is at the point 43. Thus the position of the valve 28 indicated by the point 45. At the other dead point the driving piston 3 is at the point 52, the working piston at point 39 and the position of the valve correspondingly at 47. At full load therefore the valve 28 makes the short stroke from 45-47 without opening the duct for oil from the cylinder end chamber 5 to the cylinder end chamber 4.

At part load that is at shortened stroke when for example the working piston 2 is only to move between the points 40 and 41, these points should be at equal distances from the mid point of the length of the cylinder. At the left hand dead point the driving piston 3 is at the point 49, the working piston 2 at the point 41, and the valve 28 at point 44. In order to achieve this regulation of delivery the length of the valve rod 33 is so adjusted that the point 48 corresponds to full opening of the valve 28 and at point 46 the valve 28 closes the connection between the cylinder end chambers 4 and 5. During the movement of the driving piston 3 from points 52 to 50 now following oil flows from the cylinder end chamber 5 to the cylinder end chamber 4 and the working piston 2 remains at the point 40. This continues until the valve 28 is closed i.e. until the valve 28 reaches point 46. At this instant the connection from the cylinder end chamber 5 to the cylinder end chamber 4 is closed and the working piston 2 is

set in movement. It reaches the end of the stroke 41 as soon as the driving piston 3 has reached the point 49. The valve 28 is now at 44. The stroke of the valve 28 is accordingly increased at part load. While at full load it is given by the length 45-47, at part load this length increases to the length 44-48, when 48-46 corresponds to the opening of the valves and 46-44 the closing. The stroke of the valve 28 increases as the stroke of the working piston 2 is set smaller. This condition is desirable because at small outputs larger transfer flow cross sections are necessary.

The diagram of FIG. 9 also shows diagrammatically how by the operation of the synchronising lever 53 the mid-position of the working piston 2 is automatically set with reduction in stroke.

It will be assumed that the length of the valve rods 33 is so set that a stroke of the working piston 2 of length 40-41 is obtained and that closing of the valve 28 takes place at point 46. It will, however, also be assumed that the stroke position of the piston 2 has for some reasons gone wrong and that the point 40 has shifted to 42.

It can immediately be seen from the diagram according to FIG. 9 that the closing point 46 of the valve 28 will then be reached when the driving piston 3 is at point 51. The closing of the valve 28 thus takes place earlier, and the stroke of the working piston 2 will be increased to such an extent that the incorrect stroke position is corrected. It will be understood that the ratio of the lever 53 must take into account the difference between the strokes of the pistons 2 and 3 due to their different diameters. Exactly the same conditions apply to the operation of the valve 27.

What I claim is:

1. In a piston pump or compressor having transmission of the driving power from a mechanically driven driving piston to a working piston rigidly connected with the pump or compressor piston by means of a hydraulic medium filling two cylinder end chambers separated by said driving piston and said working piston, means for continuous regulation of the output of the pump or compressor without changing the speed of the mechanical drive comprising a first cut-off valve controlled in dependence on the movement of the driving piston and arranged in a first transfer duct between said end chambers, a first non-return valve arranged in said first transfer duct, a second cut-off valve also controlled in dependence on the movement of the driving piston and arranged in a second transfer duct between said end chambers, the movements of the cut-off valves being in opposite phase, and a second non-return valve arranged in said second transfer duct, the non-return valves being so disposed and the cut-off valves so timed that each transfer duct is opened by the non-return valve when the pressure in said end chambers equalise if the cut-off valve is open and is closed by the cut-off valve after a predetermined fraction from zero upwards of the stroke.

2. A piston pump or compressor as set forth in claim 1 in which the first cut-off valve is connected to the driving piston through a rocking lever which transmits the movements of the driving piston to the first cut-off valve synchronously but on a smaller scale.

3. A piston pump or compressor as set forth in claim 3 in which the drive of the second cut-off valve is effected from one end of a reversing lever the other end of which is connected to the first cut-off valve and which is fulcrumed between its ends.

4. A piston pump or compressor as set forth in claim 2 in which the drive to the first cut-off valve is through a valve rod the effective length of which can be adjusted, thereby to regulate the time of closing of the cut-off valves.

5. A piston pump or compressor as set forth in claim 4 in which the length of the rod is adjustable by a screw sleeve into which the rod is screwed.

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6. A piston pump or compressor as set forth in claim 2 in which the rocking lever is connected by one end to the piston rod of the driving piston and connected by its other end through a link with the working piston in such a way that even at part-load operation with shortened stroke, the mid-point of the stroke always coincides with the mid-point of the length of the working cylinder.

7. A piston pump or compressor as set forth in claim 3 in which the cut-off valves are slide valves and the drive to both cut-off valves is through respective valve rods the effective length of which can be adjusted thereby to regulate the times of closing of the respective valves.

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