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(54) **Hydraulic cylinder for use in a hydraulic tool**

(57) The invention relates to a hydraulic cylinder, for use in a hydraulic tool, comprising a hollow cylinder body (8) accommodating a first piston, which first piston is composed of a hollow first piston rod (14), which extends from the cylinder body, and a first piston body (20) connected thereto, said cylinder body and said first piston body defining a first cylinder chamber (21) and said cylinder body, said first piston body and said first piston rod defining a second cylinder chamber (22), as well as a second piston accommodated in the hollow first piston rod, which second piston is composed of a second piston rod (26), which extends through the first piston body and which is connected to the cylinder body, and a second piston body (25) connected thereto, said first piston rod and said second piston body defining a third cylinder chamber (23) and said first piston rod, said second piston rod and said second piston body defining a fourth cylinder chamber (24), wherein the second piston rod is at least provided with a bore that terminates in the third cylinder chamber, and wherein the cylinder chambers can be connected to a first (P1) and a second supply line (P2), respectively, for extending and retracting the first piston rod.

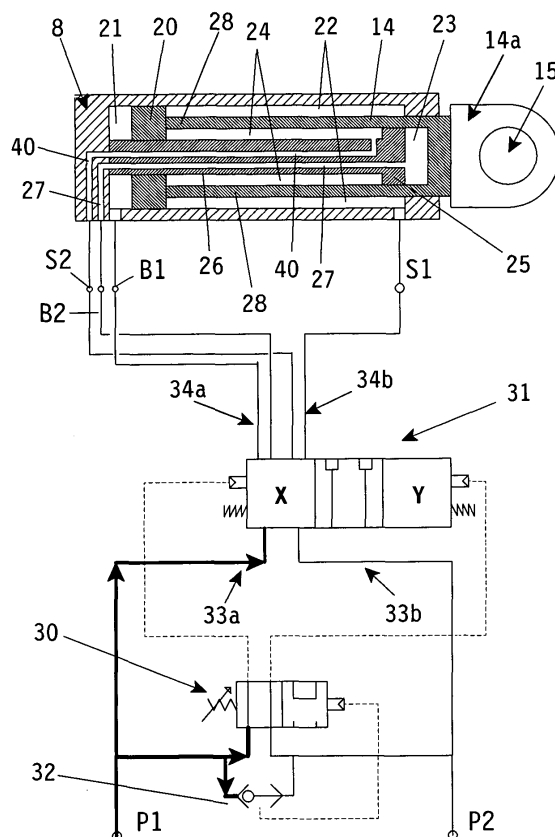


Fig. 2a

Description

[0001] The invention relates to a hydraulic cylinder, for example for use in a hydraulic tool, comprising a hollow cylinder body accommodating a first piston, which first piston is composed of a hollow first piston rod, which extends from the cylinder body, and a first piston body connected thereto, said cylinder body and said first piston body defining a first cylinder chamber and said cylinder body, said first piston body and said first piston rod defining a second cylinder chamber, as well as a second piston accommodated in the hollow first piston rod, which second piston is composed of a second piston rod, which extends through the first piston body and which is connected to the cylinder body, and a second piston body connected thereto, said first piston rod and said second piston body defining a third cylinder chamber and said first piston rod, said second piston rod and said second piston body defining a fourth cylinder chamber, wherein second piston rod is at least provided with a bore that terminates in the third cylinder chamber, and wherein the cylinder chambers can be connected to a first and a second supply line, respectively, for a pressurised fluid for extending and retracting the first piston rod.

[0002] A hydraulic tool that is operated by means of a hydraulic cylinder as described above is known, for example from European patent No. 0 641 618 B1. Said patent discloses a frame that can be coupled to a jib of an excavating machine or the like, to which frame an assembly of two jaws can be coupled. One of the jaws can be pivoted with respect to the other jaw by means of a hydraulic setting cylinder.

[0003] The outward stroke of the piston rod of the setting cylinder causes the pivotable jaw to move towards the other, fixed jaw, whilst the inward stroke of the piston rod causes the pivotable jaw to move away from the fixed jaw.

[0004] A setting cylinder of such a tool is controlled or energized by the hydraulics of the machine in question, whose construction more or less determines the available operating pressure of the fluid as well as the flow of the fluid to be supplied. A large bore diameter of the cylinder body enables high cylinder forces, to be true, but it requires high fluid flows through the hydraulic circuit, which in turn leads to unnecessarily long cycle times.

[0005] On the other hand it is desirable to operate the hydraulic tool with a minimum return fluid flow, because unnecessary continuous pumping of high fluid flows through the hydraulic circuit to and from the setting cylinder on the one hand leads to unnecessary pump losses (pressure loss) due to pipe resistance, whilst furthermore heat development and the additional fuel consumption of the machine in question additionally affects the efficiency.

[0006] A hydraulic cylinder as referred to in the introduction is known from, for example, German patent pub-

lication No. 1,021,612. For the time being, the current setting cylinders are characterized by low cycle times, low return fluid flows and high cylinder forces only during the outward stroke of the piston rod; the known cylinders in particular lack high cylinder forces during the inward stroke, however.

[0007] This latter characteristic is in particular desirable in hydraulic demolition tools, such as scrap cutters and the like, because iron occasionally gets wedged between the jaws and will not come loose unless high cylinder forces are exerted during the inward stroke of the piston rod.

[0008] Consequently it is an object of the invention to provide an improved cylinder of the kind referred to in the introduction, which on the one hand exhibits short cycle times and low return fluid flows, both during the outward stroke and during the inward stroke, but which on the other hand is capable of generating high cylinder forces both during the outward stroke and during the inward stroke.

[0009] According to the invention, the cylinder is to that end characterized in that at least one pressure control valve is provided, which controls the supply of pressurised fluid to the various cylinder chambers in dependence on the pressure difference between the first supply line and the second supply line. The cylinder according to the invention provides a very functional cylinder chamber, thus making it possible to place the various cylinder chambers in communication with the supply lines for pressurised fluid in dependence on the operating conditions.

[0010] More in particular, the pressure control valve controls a pressure-controlled valve on the basis of the pressure difference between the first supply line and the second supply line, in such a manner that when the pressure caused by the load resistance during the outward/inward stroke of the first piston rod is lower than the predetermined pressure level, the pressure-controlled valve will take up a first/second extreme position, and when the pressure caused by the load resistance is higher than the predetermined pressure level, the pressure-controlled valve will take up a central position.

[0011] When the load resistance decreases again, the pressure-controlled valve will take up a first/second extreme position again.

[0012] In a special embodiment, the pressure-controlled valve, in the central position thereof, places the first and the third cylinder chamber into communication with the first supply line and the third and the fourth cylinder chamber with the second supply line, whilst in the first and the second extreme position the first, second, third and fourth cylinder chambers can be placed into communication with the first and/or the second supply line.

[0013] More specifically, when the pressure caused by the load resistance is lower than the predetermined pressure level both during the outward stroke and during the inward stroke of the first piston rod, the speed of movement of the first piston rod will be high and the force

of said piston rod will be low, and when the pressure caused by the load resistance is lower than the predetermined pressure level both during the outward stroke and during the inward stroke of the first piston rod, the speed of movement of the first piston rod will be high and the force of said piston rod will be low.

[0014] In this way high cylinder speeds (and consequently short cycle times) can be realised, whilst on the other hand high cylinder forces can be generated also during the outward stroke and in particular during the inward stroke.

[0015] In its first extreme position, the pressure-controlled valve can place the second, third and fourth cylinder chamber into communication with the second supply line, whilst in its second extreme position the pressure-controlled valve places the third cylinder chamber and at least the second cylinder chamber into communication with the second supply line.

[0016] On the other hand, in a special embodiment the pressure-controlled valve places the fourth cylinder chamber into communication with the second supply line in the second extreme position thereof.

[0017] According to a special aspect of the invention, the second cylinder chamber and the fourth cylinder chamber are in communication with each other; in a first embodiment, said second cylinder chamber and said fourth cylinder chamber are in communication with each other via at least one opening formed in the first piston rod. In another embodiment, the second piston rod is provided with a further bore that terminates in the fourth cylinder chamber, which further bore connects the fourth cylinder chamber to a fourth supply line for the pressurised fluid.

[0018] Thus a fourth cylinder chamber is realised that can be energized independently of the other cylinder chambers. In this way an even more versatile hydraulic cylinder is obtained.

[0019] In a special embodiment, the second and the third cylinder chamber are each in communication with a storage vessel for the fluid via a non-return valve. The non-return valve for the third cylinder chamber may be a pressure-controlled non-return valve, in particular a non-return valve that is controlled by the pressure control valve. As a result, the return fluid flow through the lines is further reduced by temporarily collecting part of the return flow from the second and the third cylinder chamber in a storage vessel during the inward stroke in dependence on the prevailing return pressure of the fluid.

[0020] The stored fluid is delivered to the hydraulic circuit again during the outward stroke of the piston rod. This results in a further reduction of the pump losses, resistance losses, etc.

[0021] The invention will now be explained in more detail with reference to a drawing, in which:

Figs. 1a and 1b are views of an embodiment of a hydraulic tool according to the prior art, which is

coupled to the jib of an excavator;

Figs. 2a-2d show four operating situations of a basic embodiment of a hydraulic cylinder according to the invention;

Fig. 3 shows other configurations of operating situations of a hydraulic cylinder according to the invention;

Figs. 4a-4d show four operating situations of a special embodiment of a hydraulic cylinder according to the invention;

Fig. 5 shows another embodiment of a hydraulic cylinder according to the invention;

Fig. 6 shows yet another embodiment of a hydraulic cylinder according to the invention.

[0022] For a better understanding of the invention, like parts will be indicated by the same numerals in the description of the figures below.

[0023] Figs. 1a and 1b are two views of a hydraulic tool that is driven or energized by a hydraulic setting cylinder. The illustrated prior art tool comprises a frame 1 including a first frame part 2, which frame part 2 is coupled to a second frame part 3 by means of a turntable 2'. The two frame parts 2 and 3 are rotatable with respect to each other by means not shown, for example hydraulically operated setting means, which are known per se.

[0024] The frame part 2 is fitted with coupling means 4, 4' that are known per se, by which the device 1 can be coupled to the end of an arm of an excavator or a similar earthmoving machine, for example.

[0025] A first jaw 12 is connected to the frame part 3 of the frame 1 by means of a pivot pin 10 and a pin 11. The two pins 10 and 11 are accommodated in corresponding openings or bores (not shown) formed in the frame part 3. Furthermore, a second movable jaw 13 can be pivoted about the pivot pin 10.

[0026] The second movable jaw 13 is pivotable with respect to the first jaw 12 by the setting cylinder 8, to which purpose the end 14a of a piston rod 14 of the setting cylinder 8 is coupled to one end of the pivotable jaw 13 by means of a pin 15. The setting cylinder 8 is pivoted in the frame part 3 about pivot point 9 so as to enable outward movement of the piston rod 14.

[0027] Fig. 1a shows the hydraulic tool in the operating situation in which the piston rod 14 is fully retracted (inward stroke), whilst Fig. 1b shows the outward stroke of the piston rod 14, by which the jaw 13 has been moved into contact with the jaw 12.

[0028] Such a hydraulic tool can be used for carrying out demolition, crushing or cutting operations, during which operations large cylinder forces can be transmitted to the jaws 12 and 13. In the case of hydraulic demolition tools, such as scrap cutters and the like, iron occasionally gets wedged between the jaws 12 and 13 (in particular between the cutting edges 16, 16' and 17), which iron will not come loose unless large cylinder forces are exerted during the inward stroke of the piston rod 14.

[0029] It is desirable, therefore, to use a hydraulic setting cylinder 14 which is characterized by high piston rod speeds and consequently short cycle times, low return fluid flows and high cylinder forces not only during the outward stroke of the piston rod 14, but which is capable of generating highest cylinder forces also during the inward stroke (from the position that is shown in Fig. 1b to the position that is shown in Fig. 1a).

[0030] Figs. 2a-2d show various operating situations of a basic embodiment of such a hydraulic setting cylinder according to the invention.

[0031] The hydraulic setting cylinder comprises a hollow cylinder body 8 accommodating a first piston, which piston is composed of a hollow first piston rod 14 extending from the cylinder body 8 and a first piston body 20 that is connected thereto. The external dimension of the first piston body 20 corresponds to the internal dimension of the hollow cylinder body 8.

[0032] The hollow cylinder body 8 and the first piston body 20 define a first cylinder chamber 21, whilst the hollow cylinder body 8, the first piston rod 14 and the first piston body 20 define a second cylinder chamber 22 that surrounds the first piston rod 14.

[0033] Referring to Figs. 1a and 1b, the end 14a of the first piston rod 14 is to be connected by means of a pin 15 to, for example, the pivotable jaw 13 of the cutting and/or crushing tool that is shown in Figs. 1a and 1b.

[0034] Accommodated in the hollow first piston rod 14 is a second piston composed of a second piston rod, which extends through the first piston body 20 and which is connected to the hollow cylinder body 8, and a second piston body 25 connected thereto. The external dimension of the second piston body 25 corresponds to the internal dimension of the hollow first piston rod 14.

[0035] According to the invention, the hydraulic circuit is partially built up of a pressure-controlled valve 31, which is provided with a first supply line P1, which can be placed into communication with the first cylinder chamber 21. When a pressurized fluid (e.g. oil) is led through the first supply line P1 to the first cylinder chamber 21, the piston rod 14 will extend (the outward stroke) under the influence of the pressure increase in the cylinder chamber 21. To that end a flange B1 is provided in the cylinder body 8, to which the first supply line P1 can be connected.

[0036] Furthermore, the second cylinder chamber 22 is provided with a connecting flange S1, which connecting flange can be connected inter alia to the second supply line P2 via a supply line and the pressure-controlled valve 31. The second supply line P2 in particular functions to supply pressurised fluid for retracting the first piston rod 14 (inward stroke).

[0037] The second piston rod 26 is provided with a first through bore 27, which connects the third cylinder chamber 23 to a connecting flange B2, which is in turn connected to the pressure-controlled valve 31 via a fluid line. In this embodiment, the second piston rod 26 is furthermore optionally provided with a second through

bore 40, which connects the fourth cylinder chamber 23 to a connecting flange S2, which is in turn connected to the pressure-controlled valve 31 via a fluid line.

[0038] According to the invention, the pressure-controlled valve 31 can take up three positions, viz. a first extreme position X (as shown in Fig. 2a), a central position (as shown in Figs. 2b and 2d) and a second extreme position Y (as shown in Fig. 2c).

[0039] According to the invention, the pressure-controlled valve 31 is controlled by a pressure control valve 30, which is in turn controlled by a ball valve 32.

[0040] Fig. 2a shows an operating situation of a basic embodiment of the hydraulic cylinder according to the invention, in which an outward stroke is imposed on the piston rod 14 by the hydraulic circuit and in which the piston rod 14 encounters a load resistance (via the first and second jaws 12, 13 (not shown)), which load resistance generates a pressure in the hydraulic circuit which is lower than a predetermined pressure level. To realise a short cycle time, the pressure control valve 30 and the pressure-controlled valve 31 are switched so that the outward stroke of the first piston rod 14 takes place at a high speed.

[0041] During the outward stroke, pressurised fluid is supplied via the first supply line P1, which fluid places the pressure-controlled valve 31 in its first extreme position X as shown in Fig. 2a via the pressure control valve 30. The pressurised fluid is led to the various connecting flanges B1-B2-S1-S2, and consequently to the cylinder chambers 21-22-23-24, via the supply line P1 in dependence on the configuration of the valve position X. The configuration of the first extreme valve position X results in a particular cylinder behaviour during the outward stroke.

[0042] When the piston rod 14 encounters an increasing load resistance during operation, as a result of which the pressure realised by said load resistance in the hydraulic circuit rises above a predetermined pressure level, the ball valve 32 is activated, as shown in Fig. 2b, under the influence of the pressure caused by the load resistance and the pressure difference between the lines P1 and P2.

[0043] In this operating situation, the fluid pressure in the supply line P1 can set the pressure control valve 30 to its other position. This causes the pressure-controlled valve 31 to take up its central position, as shown in Fig. 2b. The valve configuration of the pressure-controlled valve 31 in the central position thereof is such that the first and the third cylinder chamber 21, 23, respectively, are jointly connected to the first supply line P1 via the connecting flanges B1 and B2. Both the third and the first cylinder chamber 23, 21, respectively, are thus fed with the pressurised fluid that is being supplied from the main supply line P1. The second and the fourth cylinder chamber 22, 24, are in communication with the second supply line P2 via the connecting flanges S1 and S2, respectively.

[0044] In this operating situation, the second and the

fourth cylinder chamber 22, 24 are pressureless and the fluid that is present in said chambers is forced out of the cylinder body 8 in the direction of the supply line P2 during the outward stroke of the piston rod 14. In this operating situation, the first and the third cylinder chamber 21, 23 are pressurised via the fluid that is supplied through the main supply line P1. In this situation the setting cylinder 8 is capable of exerting large forces on a hydraulic tool, for example the crushing and cutting tool of Figs. 1a and 1b, via the piston rod 14.

[0045] Figs. 2c and 2d show two operating situations of the basic embodiment of the hydraulic cylinder according to the invention during the inward stroke of the piston rod 14. In this second operating situation, the second supply line P2 is in principle used for supplying pressurised fluid.

[0046] In the operating situation that is shown in Fig. 2c, the pressure created in the hydraulic circuit as a result of the load resistance experienced by the piston rod 14 is lower than a predetermined pressure. The pressure control valve 30 and the ball valve 32 are switched so that the pressurised fluid supplied via the second supply line P2 sets the pressure-controlled valve 31 to its second extreme position Y. The supply of pressurized fluid via the supply line P2 to the cylinder chambers 21-22-23-24 is determined by the valve configuration in the extreme position Y.

[0047] Because iron can get wedged between the cutting faces of a hydraulic cutting and/or crushing tool as shown in Figs. 1a and 1b during operation, the piston rod 14 is preferably capable of transmitting large forces during the inward stroke as well, so as to move the jaws 12 and 13 apart.

[0048] This operating situation is shown in Fig. 2d, in which situation the piston rod 14 experiences such a high load resistance that the pressure that is thus generated in the hydraulic circuit exceeds a predetermined pressure as set by the pressure control valve 30. The increased fluid pressure in the second supply line P2 caused by the increased load resistance switches over the ball valve 32, and consequently also the pressure control valve 30.

[0049] As a result, the pressure-controlled valve 31 takes up its central position (Fig. 2d), thus connecting the second and the fourth cylinder chamber 22, 24 to the second supply line P2 via the connecting flanges S1 and S2, respectively. The first and the third cylinder chamber 21, 23, which are likewise in communication with each other, are pressureless and the fluid that is present in said chambers is forced out of the cylinder body 8 in the direction of the first supply line P1 during the inward stroke of the piston rod 14.

[0050] Fig. 3 shows various possible configurations of the pressure-controlled valve 31, in which the four cylinder chambers 21-22-23-24 (B1-B2-S1-S2) can be placed into communication with the first and the second supply line P1, P2 in various ways both during the outward stroke (position X) and during the inward stroke

(position Y) of the piston rod 14. The valve 31 is in position X when the piston rod 14 moves out quickly and takes up its central position when the cylinder is to deliver the maximum force. During the inward stroke, the valve 31 is in position Y when the piston rod 14 moves in quickly and takes up its central position again when the maximum force is to be delivered.

[0051] There are a number of switching possibilities with regard to the switching symbols X and Y. These valve configurations are shown in Fig. 3. Possibilities X1-X10 are available for the first extreme position X and possibilities Y1-Y5 are available for the second extreme position Y. With these valve configurations, the four cylinder chambers 21-24 are switched in such a manner that application of pressure to the first supply line P1 will result in an outward stroke of the piston rod 14. Application of pressure to the second supply line P2 will result in an inward stroke of the piston rod 14. This is not the case with all the other possibilities.

[0052] Possibilities X1 and Y1 are the same as the configuration of the valve 31 in the central position, i.e. in fact the valve does not switch.

[0053] The valve configurations X1-X10 are so arranged that configuration X1 means the lowest speed of the cylinder 8 and the largest return fluid flow from the cylinder chambers. From configuration X1 to X10 the cylinder speed becomes higher and higher and the return fluid flow becomes lower and lower. With valve configuration X6 (first extreme position) the return fluid flow even equals zero, and with configurations X7-X10 the return fluid flow even becomes negative, i.e. fluid (water, oil, etc) needs to be sucked in.

[0054] This makes it necessary to make a special provision in the hydraulic system, for example in the form of a fluid line that is directly connected to the storage or buffer tank for the fluid, or a storage vessel closer to the cylinder 8, which releases the fluid that has been collected at that moment to the cylinder chambers again. Excessive suction volumes may lead to excessive underpressures being generated in the cylinder lines and chambers, which in turn may lead to cavitation.

[0055] Configurations Y1-Y5 are associated with the second extreme position Y. From Y1 to Y5, the speed of the inward stroke of the piston becomes higher and higher and consequently the return fluid flow becomes lower and lower.

[0056] In theory, 10 x 5 different configurations of the pressure-controlled valve 31 are possible, therefore. A number of variants are special, however:

1) Variant X10-Y5: with this configuration of the valve 31 the cylinder 8 is very fast and the cycle times are shortest. A drawback is the fact that there is a lot of feed-through of oil, so that this configuration is not very practical.

2) Variant X6-Y5: with this configuration of the valve 31 the cylinder 8 has the shortest cycle times and the lowest return fluid flow, and furthermore there is

no feed-through of oil.

3) Variant X6-Y3: with this configuration of the valve 31 an alternative cylinder 8 is obtained, which enables a simpler cylinder construction. The second and the fourth cylinder chamber 22 (S1) and 24 (S2) in the cylinder can be placed into communication with each other, so that one less fluid line in the piston rod 14 as well as one less connecting flange (S4) are needed.

[0057] Also other configurations of the pressure-controlled valve 31 can be readily used, however, depending on the required cylinder behaviour.

[0058] Figs. 4a-4d show operating situations of an embodiment of a hydraulic cylinder according to the invention in which the valve configuration of the pressure-controlled valve 31 in the first extreme position X thereof is the configuration that is indicated X6 in Fig. 3 and the valve configuration of the pressure-controlled valve 31 in the second extreme position Y thereof is the configuration that is indicated Y3 in Fig. 3.

[0059] Fig. 4a shows the operating situation of the hydraulic cylinder according to the invention, in which an outward stroke is imposed on the piston rod 14 by the hydraulic circuit and in which the piston rod 14 experiences a load resistance (via the first and the second jaw 12, 13 (not shown)), which load resistance generates a pressure in the hydraulic circuit which is lower than a predetermined pressure level. To realise a short cycle time, the pressure control valve 30 and the pressure-controlled valve 31 are switched so that the outward stroke of the first piston rod 14 takes place at a high speed.

[0060] During the outward stroke, pressurised fluid is supplied via the first supply line P1, which fluid is branched off via the pressure control valve 30, thus setting the pressure-controlled valve 31 to its first extreme position as shown in Fig. 4a. In this embodiment (in contrast to Fig. 2a, which more or less corresponds thereto), the pressurised fluid is directly introduced into the first cylinder chamber 21 via the first supply line P1 and the connecting flange B1.

[0061] In the first extreme position, pressurised fluid is passed on via the first valve supply line 33a and the pressure-controlled valve 31 to the first and the second valve discharge line 34a, 34b, which connect to the connecting flanges B2 of the bore 27 and the third cylinder chamber 23 and the connecting flange S of the second cylinder chamber 22, respectively.

[0062] In this embodiment, the piston rod 14 is provided with one or more openings 28, via which the second cylinder chamber 22 is in fluid communication with the fourth cylinder chamber 24. In other words, the fluid that is supplied under pressure via the second valve discharge line 34b is introduced both into the second cylinder chamber 22 and into the fourth cylinder chamber 24 via the connecting flange S.

[0063] By setting the pressure-controlled valve 31 to

the first extreme position, as shown in Fig. 4a, the pressurised fluid supplied via the supply line P1 is led to all four cylinder chambers 21, 22, 23 and 24. In combination with the relative proportions of the dimensions of the cylinder chambers and the first and the second piston body 20, 25, this results in the first piston rod 14 being extended at a relatively high speed.

[0064] When the piston rod 14 experiences an increasing load resistance during operation, as a result of which the pressure realised in the hydraulic circuit by said load resistance exceeds a predetermined pressure level, the ball valve 32 is activated under the influence of the pressure caused by the load resistance and the pressure difference between the lines P1 and P2, as is shown in Fig. 4b.

[0065] In this operating situation, the fluid pressure in the supply line P1 can set the pressure control valve 30 to its other position. This causes the pressure-controlled valve 31 to take up its central position, as a result of which the third cylinder chamber 23 is connected to the first valve supply line 33a and the first supply line P1 via the bore 27, the connecting flange B2 and the first valve discharge line 34a. Both the third cylinder chamber 23 and the first cylinder chamber 21 are thus fed with the pressurised fluid that is supplied from the main supply line P1. The second cylinder chamber 22 and the fourth cylinder chamber 24 are in communication with the second valve supply line 33b and the second supply line P2 via the connecting flange S and the second valve discharge line 34b.

[0066] In this operating situation, the second and the fourth cylinder chamber 22, 24 are pressureless and the fluid that is present in said chambers is forced out of the cylinder body 8 in the direction of the supply line P2 as indicated by the arrow during the outward stroke of the piston rod 14. In this operating situation, the first and the third cylinder chamber are furthermore pressurised via the fluid that is supplied by the main supply line P1. In this situation, the setting cylinder 8 is capable of exerting large forces on a hydraulic tool, for example the crushing and cutting tool from Figs. 1a and 1b, via the piston rod 14.

[0067] Figs. 4c and 4d show two operating situations during the inward stroke of the piston rod 14. In these two operating situations, the second supply line P2 is in principle utilised for supplying pressurised fluid.

[0068] In the operating situation as shown in Fig. 4c, the pressure created in the hydraulic circuit as a result of the load resistance experienced by the piston rod 14 is lower than a predetermined pressure. The pressure control valve 30 and the ball valve 32 are switched in such a manner that the pressurised fluid supplied via the second supply line P2 sets the pressure-controlled valve 31 to a second extreme position. The pressurised fluid that is supplied by the pressure-controlled valve 31 via the second valve supply line 33b is led to the connecting flanges B2 and S of the third and the second (and fourth) cylinder chambers 23-22 and 24, respec-

tively, via the first and the second valve discharge line 34a, 34b.

[0069] In this operating situation, the first cylinder chamber 21 is pressureless and the fluid that is present in the first cylinder chamber 21 is returned to the hydraulic circuit via the first supply line P1.

[0070] Because iron can get wedged between the cutting faces of a hydraulic cutting and/or crushing tool as shown in Figs. 1a and 1b, the piston rod 14 must preferably be capable of exerting large forces also during the inward stroke for moving the two jaws 12 and 13 apart.

[0071] This operating situation is shown in Fig. 4d, in which the piston rod 14 experiences such a high load resistance that the pressure thus generated in the hydraulic circuit exceeds a predetermined pressure as set by the pressure control valve 30. The increased fluid pressure in the second supply line P2 as a result of the increased load resistance switches over the ball valve 32, and consequently also the pressure control valve 30. This causes the pressure-controlled valve 31 to take up its central position, as a result of which the second and the fourth cylinder chamber 22, 24 are placed into communication with the second supply line P2 via the second valve supply line 33b and the second valve discharge line 34b.

[0072] The first and the third cylinder chamber 21, 23 are pressureless, and the fluid that is present in said chambers is forced out of the cylinder body 8 during the inward stroke of the piston rod 14 in the direction of the first supply line P1.

[0073] As is shown in Figs. 2a-2d and 4a-4d, a double cylinder action is realised in this manner, which enables the setting cylinder 8 to realise rapid movements of the piston rod 14 during operation (both during the inward stroke and during the outward stroke) but also to generate large cylinder forces both during the inward stroke and in during the outward stroke when the pressure on the setting cylinder 8 that is generated by the load resistance exceeds a predetermined pressure level in the hydraulic circuit.

[0074] According to the invention, the setting cylinder as described herein is in principle characterized by short cycle times and high speeds of movement of the piston rod 14 both during the inward stroke and during the outward stroke.

[0075] Since high return fluid flows have a negative effect on the efficiency of the basic machine that drives the setting cylinder 8 with certain cylinder actions and associated valve configurations of the pressure-controlled valve 31, it is possible to incorporate a storage vessel 35 in the hydraulic circuit in a specific embodiment as shown in Fig. 5, in which storage vessel the return fluid can be stored. Unnecessary fluid flows to be pumped through the hydraulic circuit lead to additional friction resistance in the fluid lines and thus to pump losses.

[0076] Consequently, the return fluid that is forced

from the setting cylinder 8 is temporarily stored near the setting cylinder 8 both during the inward stroke and during the outward stroke and can be directly introduced into the hydraulic circuit again when additional fluid is needed during the outward stroke of the piston rod 14.

[0077] In the embodiment that is shown in Fig. 5, the storage vessel 35 is used for collecting return fluid from the third, the second and the fourth cylinder chamber 23, 22 and 24, respectively, with the connection B2 of the third cylinder chamber 23 to the storage vessel 35 and the connection of the second and the fourth cylinder chamber (connection S) to the storage vessel 35 being closed by ball valves 37 and 36, respectively. The ball valve 37 in the fluid line between the storage vessel 35 and the third cylinder chamber 23 is configured as a pressure-controlled one-way valve (non-return valve), which non-return valve 37 is controlled by the pressure of the fluid in the second supply line P2.

[0078] Especially in the operating situation that is shown in Fig. 4d, a large return fluid flow from the first and the third cylinder chamber 21, 23 is realised during the inward stroke of the piston rod 14. Since the pressure-controlled valve 37 is opened by the pressure in the second supply line P2 in this operating situation, part of the fluid from the third cylinder chamber 23 can be stored in the storage vessel 35, thus reducing the volume of said return fluid, which is discharged from the hydraulic circuit via the pressure-controlled valve 31 and the first supply line P1.

[0079] During the outward stroke of the piston rod 14, on the other hand, the fluid stored in the storage vessel 35 is delivered to the circuit again via the non-return valves 36 and 37, in such a manner that said delivered fluid imparts an additional impulse to said outward stroke and that the storage vessel 35 is completely emptied for the next cycle.

[0080] In the embodiment that is shown in Fig. 6, the non-return valves 37 is controlled by the pressure control valve 30.

[0081] In another embodiment (not shown), on the other hand, in which a setting cylinder 8 according to the invention as shown in Figs. 2a-2d is used, the first cylinder chamber 21 (B1) (instead of the third cylinder chamber 23) and the second cylinder chamber 22 (S1) may be connected to the storage vessel 35 via the non-return valves 36-37. The third cylinder chamber 23 (B2) and the fourth cylinder chamber 24 (S2) are connected to the pressure-controlled valve 31 via their separate fluid lines B2 and S2 in that case.

[0082] In this embodiment, too, the ball valve 37 present in the fluid line between the storage vessel 35 and the first cylinder chamber 21 is configured as a pressure-controlled one-way valve (non-return valve). Analogously to the embodiment is of Figs. 5 and 6, the non-return valve 37 can be controlled by the pressure of the fluid in the second supply line P2 and by the pressure control valve 30.

[0083] Optionally, other configurations of the pres-

sure-controlled valve 31 are possible, so that different switching configurations between the various cylinder chamber's 21-24 can be realised in dependence on the load resistance that the piston rod 14 experiences both during the inward stroke and during the outward stroke, and the setting cylinder 8 can be operated under different operating conditions.

[0084] Characteristic of the setting cylinder 8 described herein, which is provided with four active cylinder chambers 21-24, is the fact that high speeds of movement and thus short cycle times of the piston rod 14 are realised both during the inward stroke and during the outward stroke, and that the piston rod 14 is capable of generating very large cylinder forces both during the inward stroke and during the outward stroke in dependence on the load resistance it experiences. In addition to that, the setting cylinder 8 is characterized by a relatively low return fluid flow, so that pump losses caused by friction resistance, heat development and the like are prevented and consequently the efficiency of the basic machine that drives the hydraulic tool is enhanced.

[0085] Furthermore, a simple, compact and low-weight construction of the setting cylinder 8 is obtained, which can be realised by making use of standard components and seals.

Claims

1. A hydraulic cylinder, for example for use in a hydraulic tool, comprising
 - a hollow cylinder body accommodating
 - a first piston, which first piston is composed of a hollow first piston rod, which extends from the cylinder body, and a first piston body connected thereto, said cylinder body and said first piston body defining a first cylinder chamber and said cylinder body, said first piston body and said first piston rod defining a second cylinder chamber, as well as
 - a second piston accommodated in the hollow first piston rod, which second piston is composed of a second piston rod, which extends through the first piston body and which is connected to the cylinder body, and a second piston body connected thereto, said first piston rod and said second piston body defining a third cylinder chamber and said first piston rod, said second piston rod and said second piston body defining a fourth cylinder chamber,
 - wherein second piston rod is at least provided with a bore that terminates in the third cylinder chamber, and wherein
 - the cylinder chambers can be connected to a first and a second supply line, respectively, for a pressurised fluid for extending and retracting the first piston rod, **characterized in that** at

least one pressure control valve is provided, which controls the supply of pressurised fluid to the various cylinder chambers in dependence on the pressure difference between the first supply line and the second supply line.

2. A hydraulic cylinder according to claim 1, **characterized in that** the pressure control valve controls a pressure-controlled valve on the basis of the pressure difference between the first supply line and the second supply line, in such a manner that when the pressure caused by the load resistance during the outward/inward stroke of the first piston rod is lower than the predetermined pressure level, the pressure-controlled valve will take up a first/second extreme position, and when the pressure caused by the load resistance is higher than the predetermined pressure level, the pressure-controlled valve will take up a central position.
3. A hydraulic cylinder according to claim 2, **characterized in that** the pressure-controlled valve, in the central position thereof, places the first and the third cylinder chamber into communication with the first supply line and the third and the fourth cylinder chamber with the second supply line.
4. A hydraulic cylinder according to claim 2 or 3, **characterized in that** in the first and the second extreme position the first, second, third and fourth cylinder chambers can be placed into communication with the first and/or the second supply line.
5. A hydraulic cylinder according to any one or more of the preceding claims, **characterized in that** when the pressure caused by the load resistance is lower than the predetermined pressure level both during the outward stroke and during the inward stroke of the first piston rod, the speed of movement of the first piston rod will be high and the force of said piston rod will be low, and when the pressure caused by the load resistance is lower than the predetermined pressure level both during the outward stroke and during the inward stroke of the first piston rod, the speed of movement of the first piston rod will be high and the force of said piston rod will be low.
6. A hydraulic cylinder according to claim 5, **characterized in that** the pressure-controlled valve places the second, third and fourth cylinder chamber into communication with the second supply line in the first extreme position thereof.
7. A hydraulic cylinder according to claim 5, **characterized in that** the pressure-controlled valve places the third and at least the second cylinder chamber into communication with the second supply line in

the second extreme position thereof.

8. A hydraulic cylinder according to claim 7, **characterized in that** the pressure-controlled valve places the fourth cylinder chamber into communication with the second supply line in the second extreme position thereof. 5
9. A hydraulic cylinder according to any one or more of the preceding claims, **characterized in that** said second cylinder chamber and said fourth cylinder chamber are in direct communication with each other. 10
10. A hydraulic cylinder according to claim 9, **characterized in that** said second cylinder chamber and said fourth cylinder chamber are in communication with each other via at least one opening formed in the first piston rod. 15
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11. A hydraulic cylinder according to claim 9, **characterized in that** the second piston rod is provided with a further bore that terminates in the fourth cylinder chamber. 25
12. A hydraulic cylinder according to any one or more of the preceding claims, **characterized in that** the second and the third cylinder chamber are each in communication with a storage vessel for the fluid via a non-return valve. 30
13. A hydraulic cylinder according to claim 12, **characterized in that** the non-return valve for the third cylinder chamber is a pressure-controlled non-return valve. 35
14. A hydraulic cylinder according to claim 12, **characterized in that** the non-return valve for the third supply line is a non-return valve that is controlled by the pressure control valve. 40

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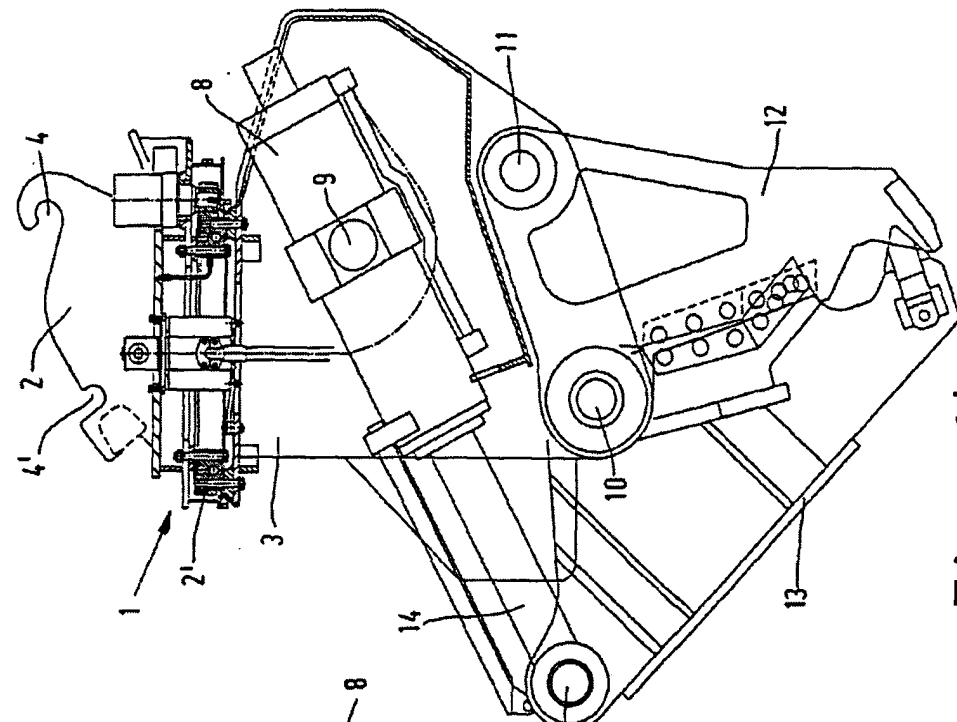


Fig. 1a

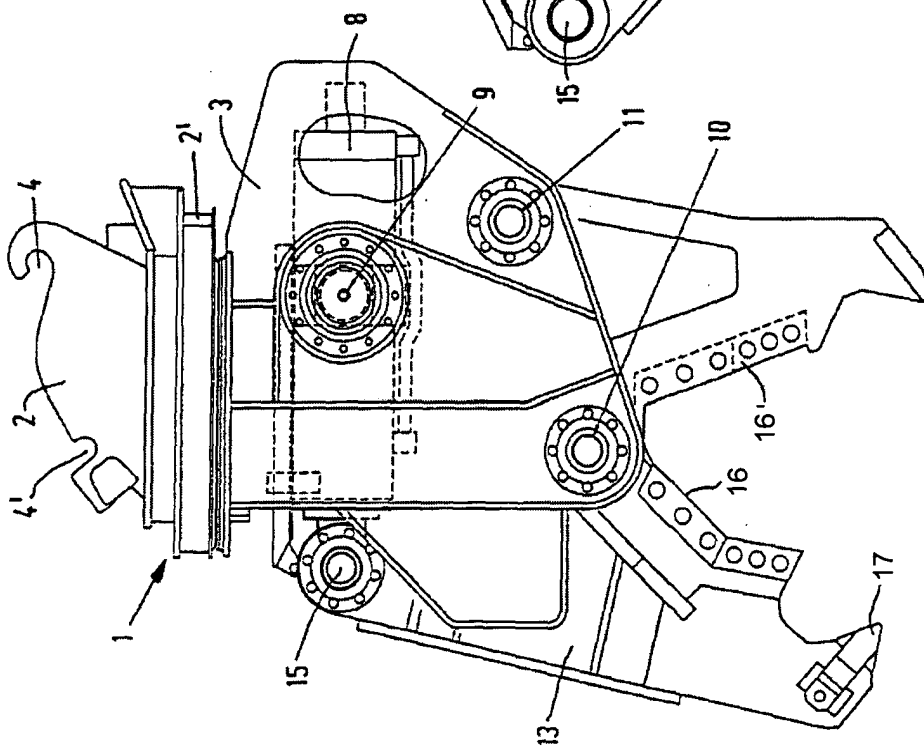


Fig. 1b

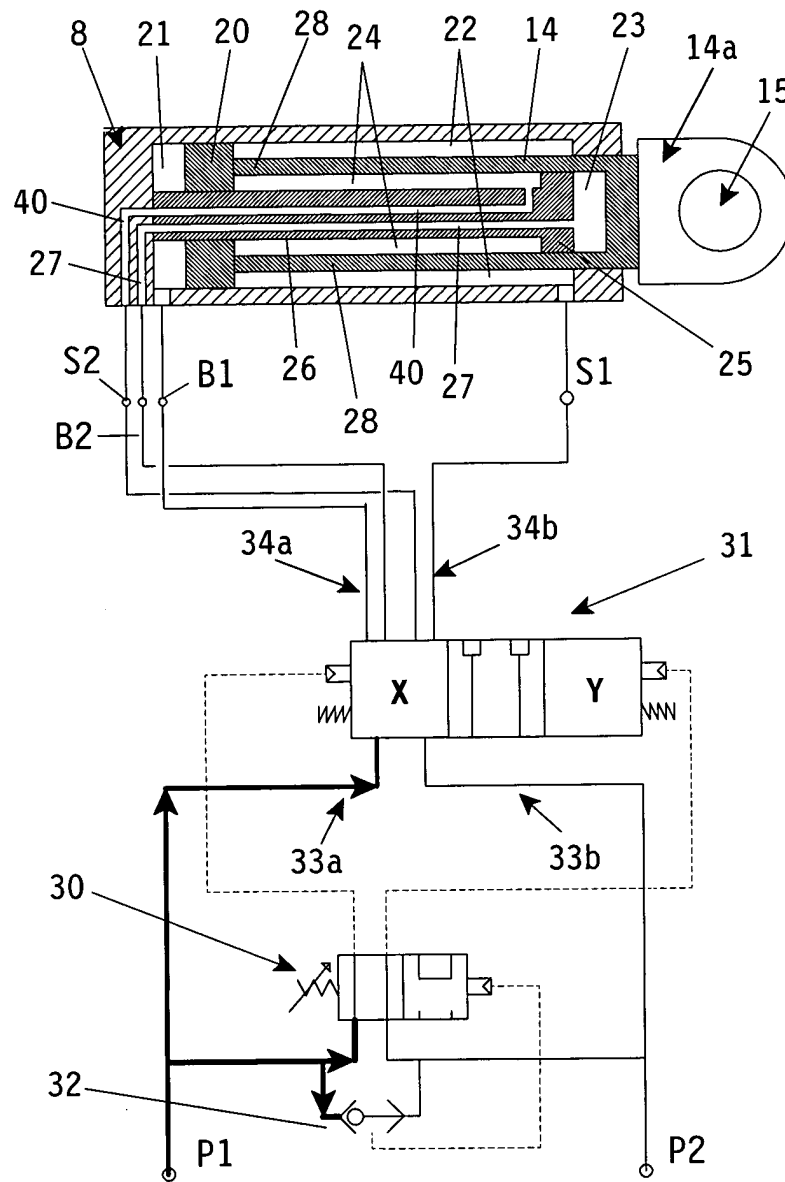


Fig. 2a

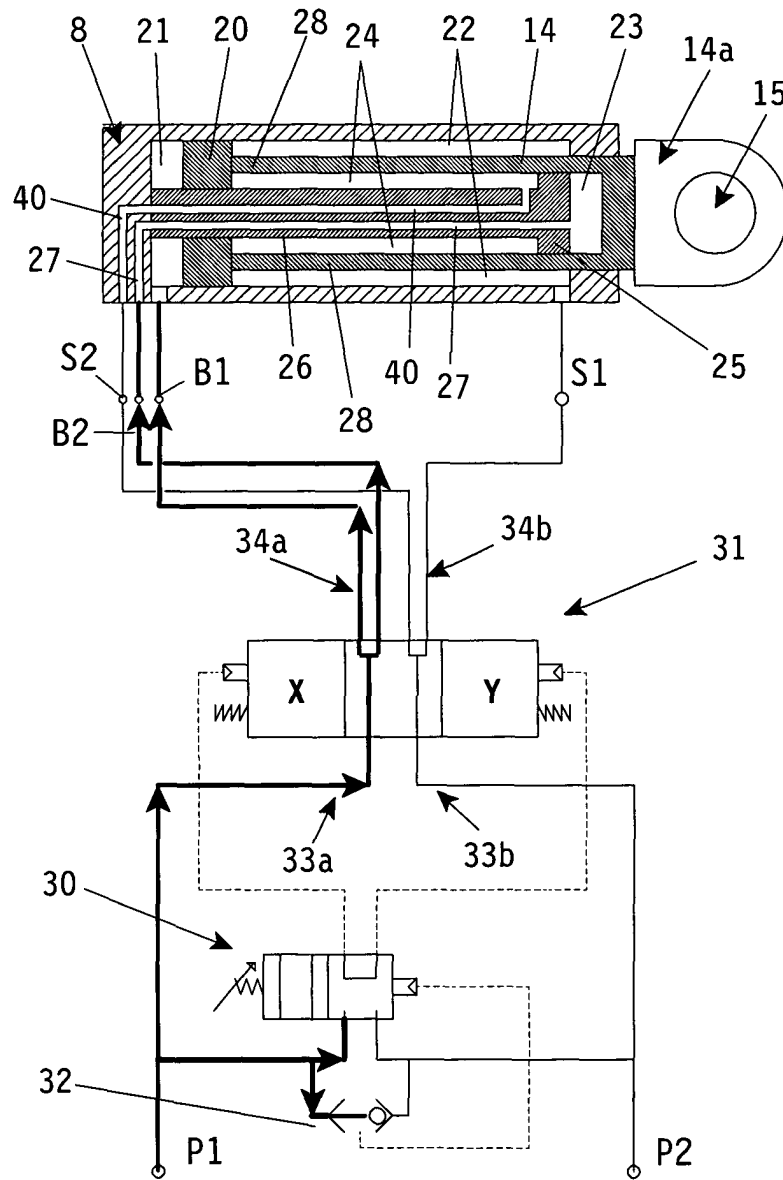


Fig. 2b

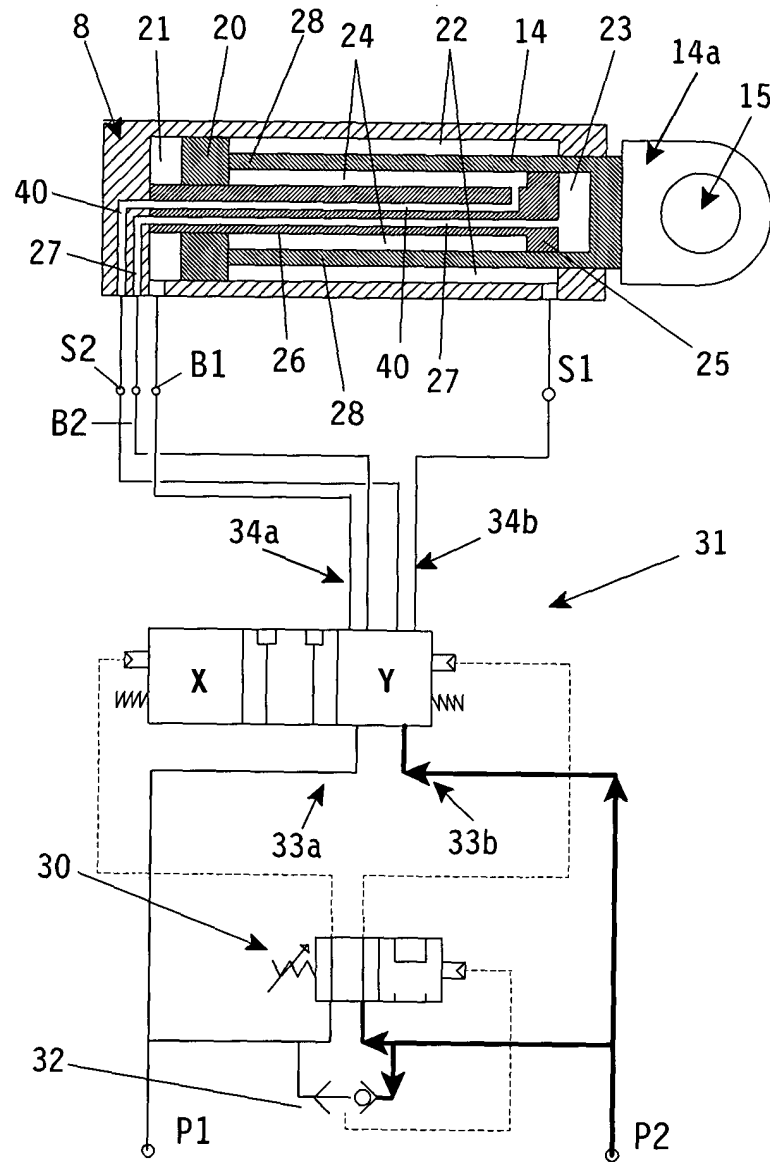


Fig. 2c

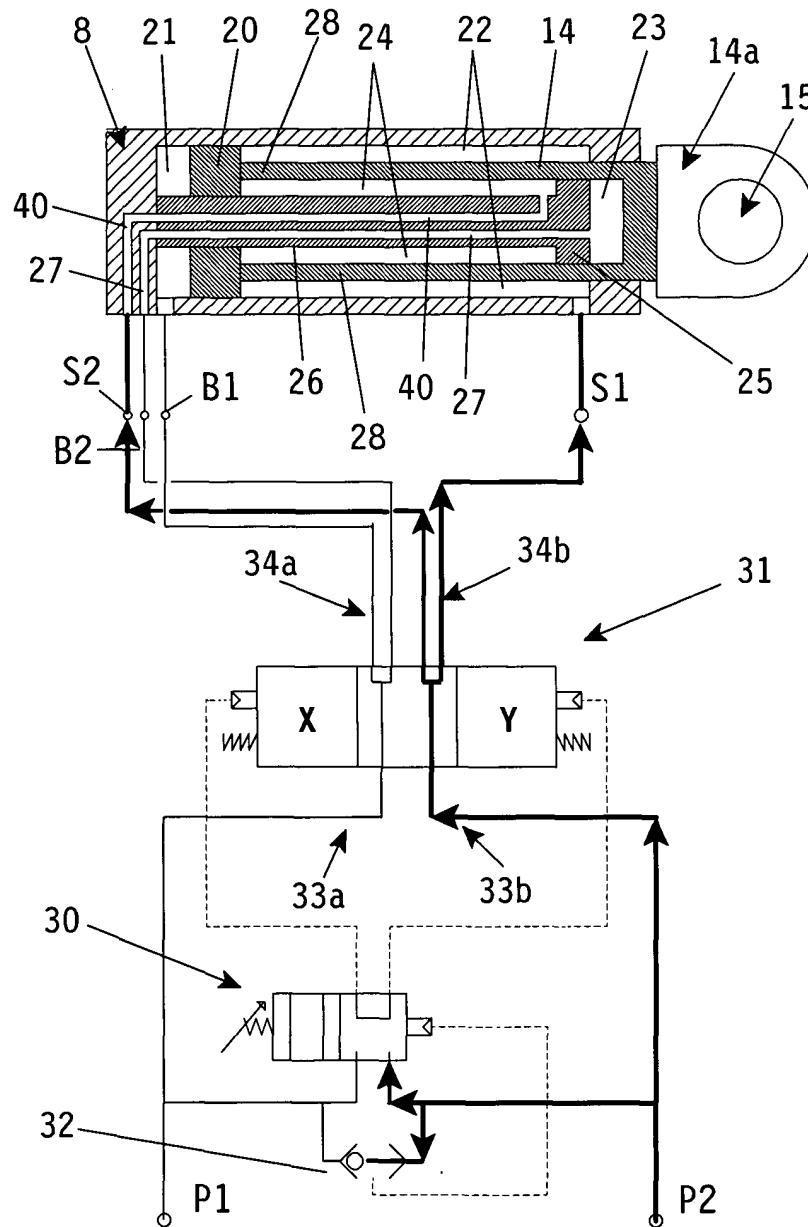


Fig. 2d

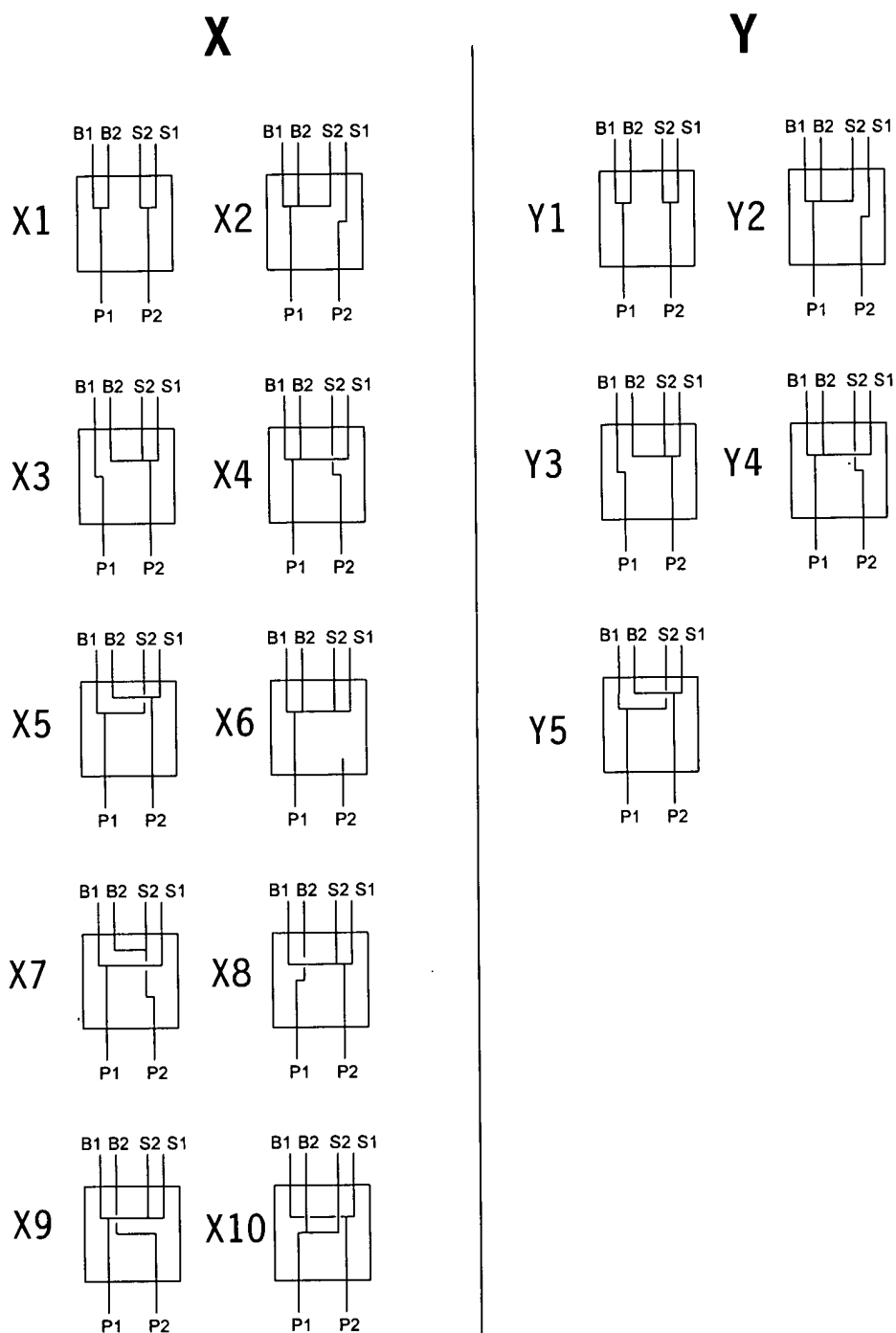


Fig. 3

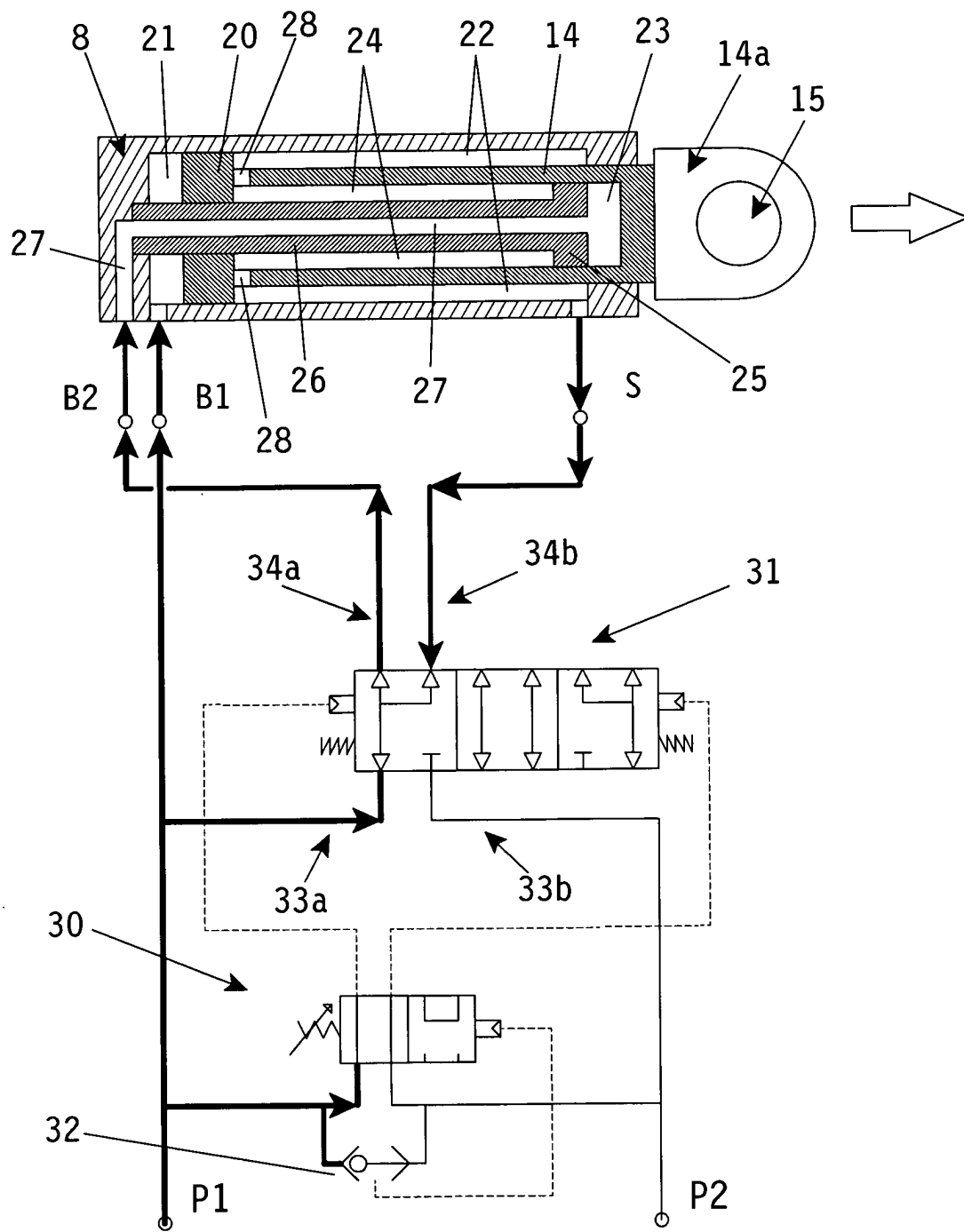


Fig. 4a

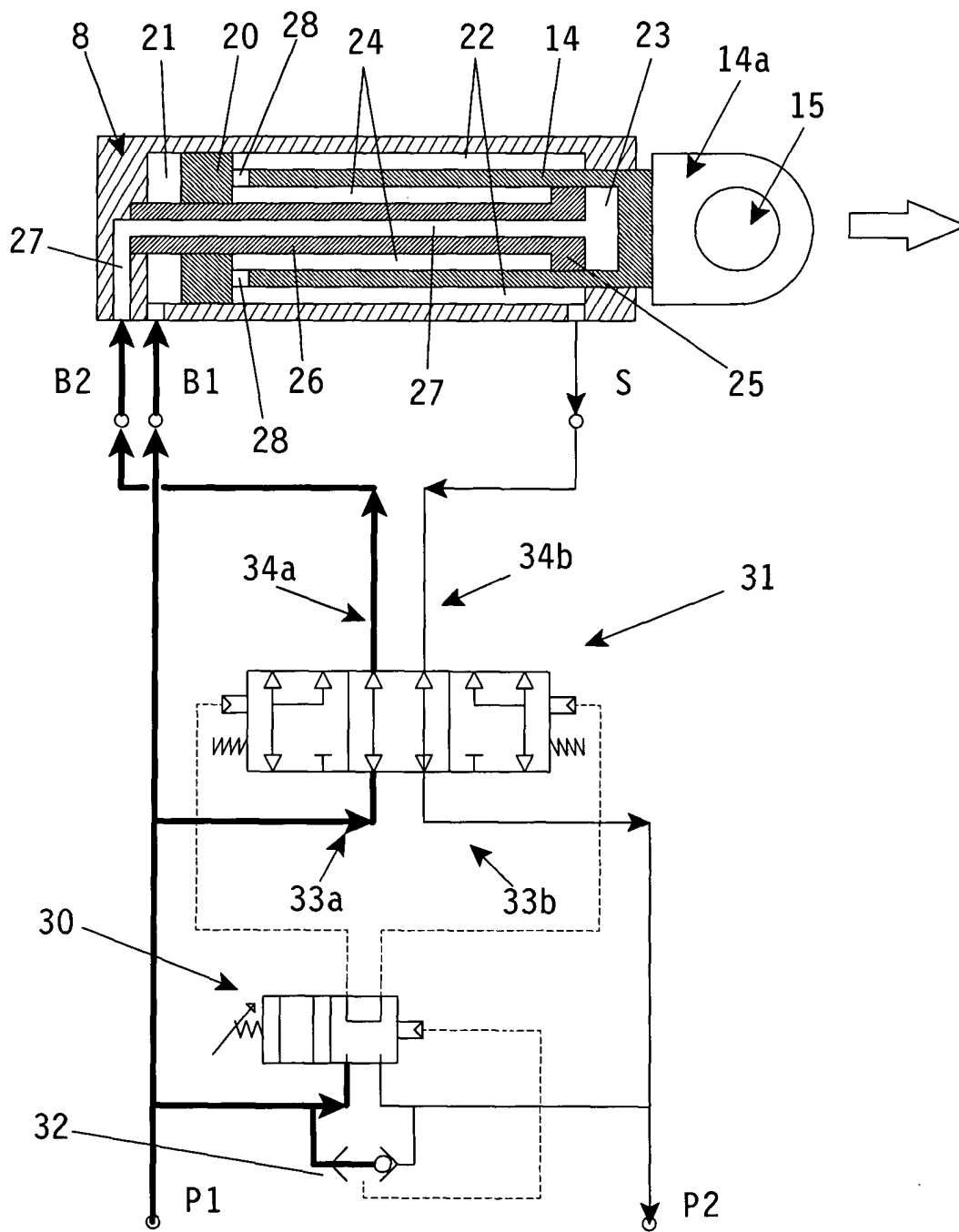


Fig. 4b

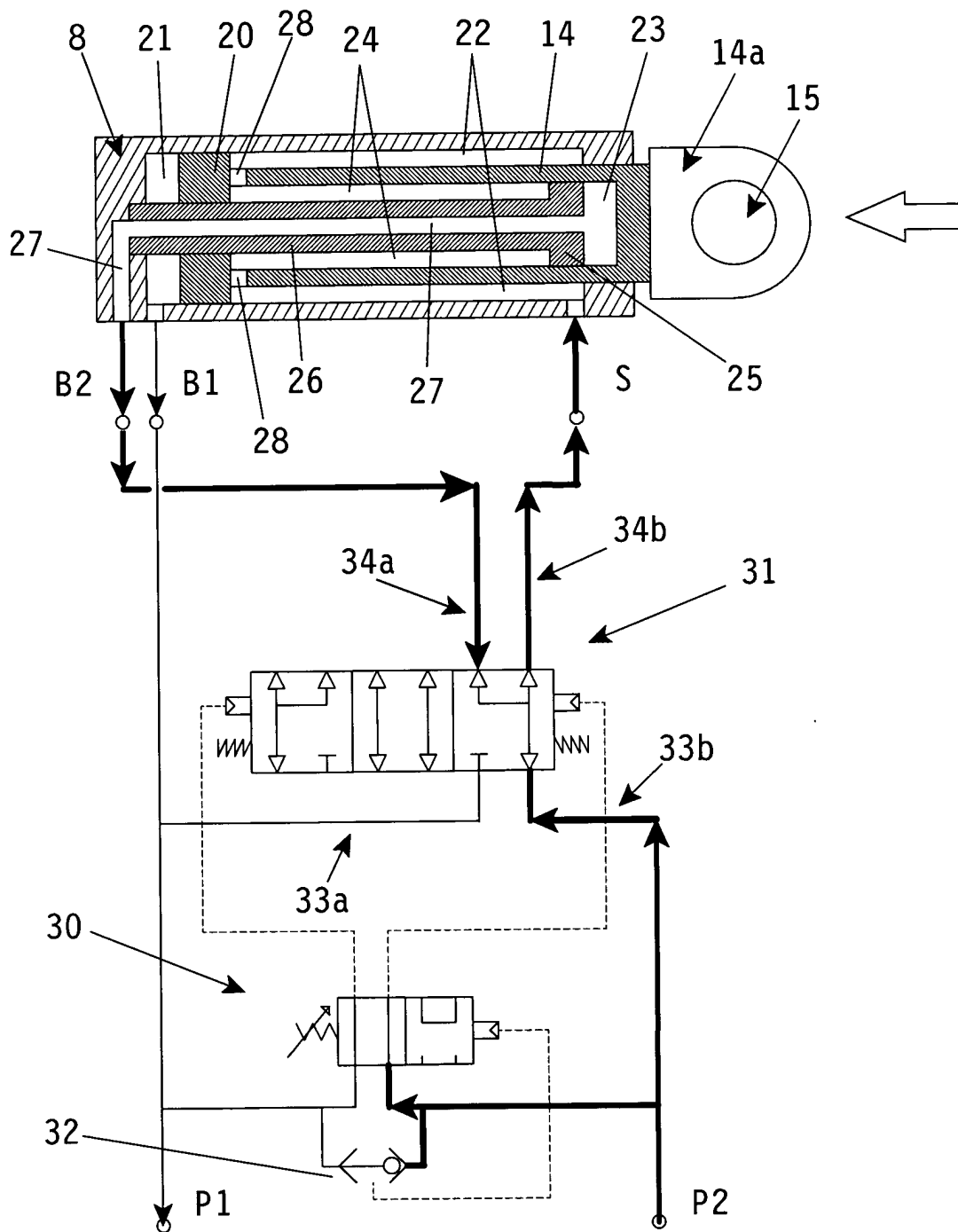


Fig. 4c

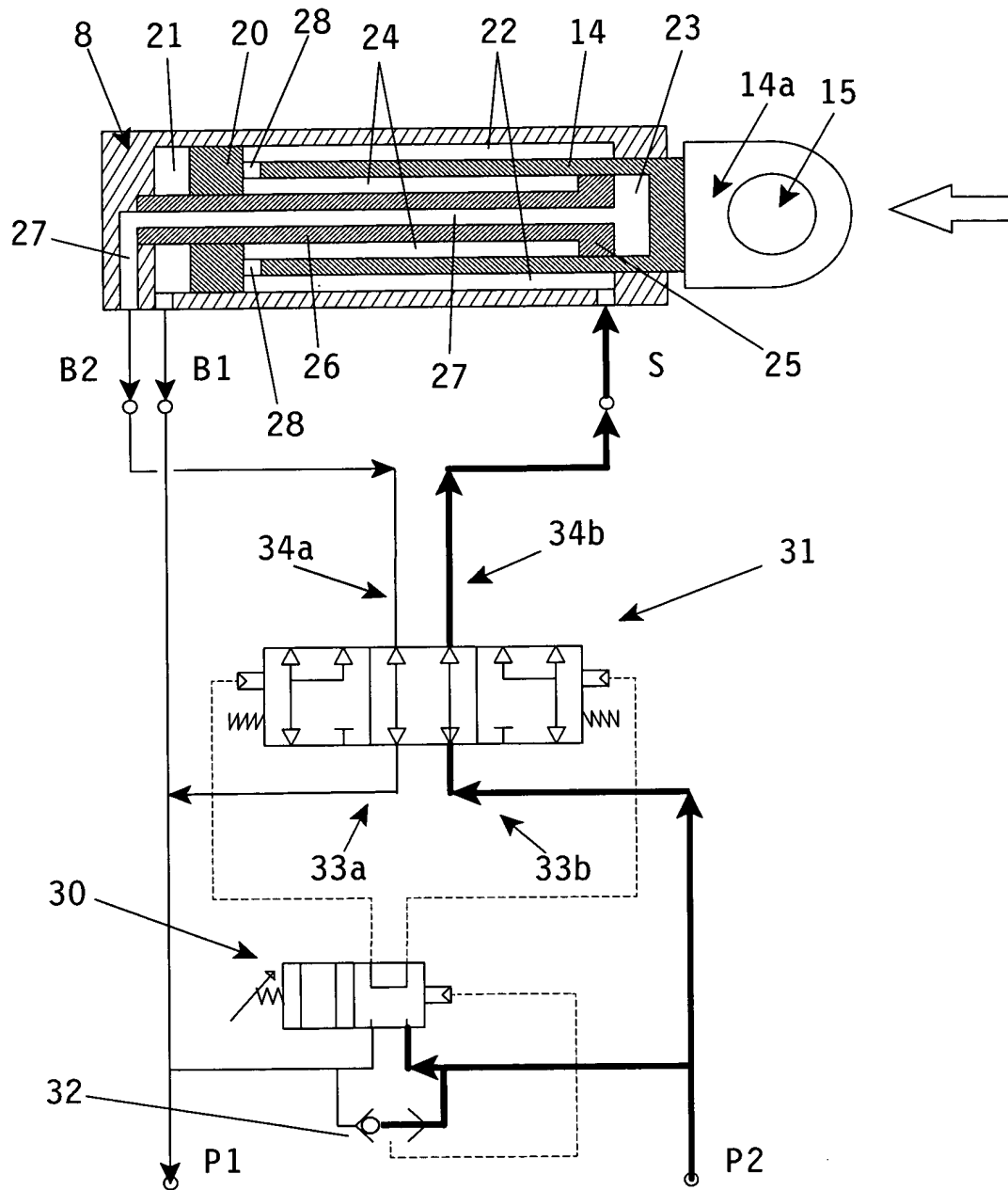


Fig. 4d

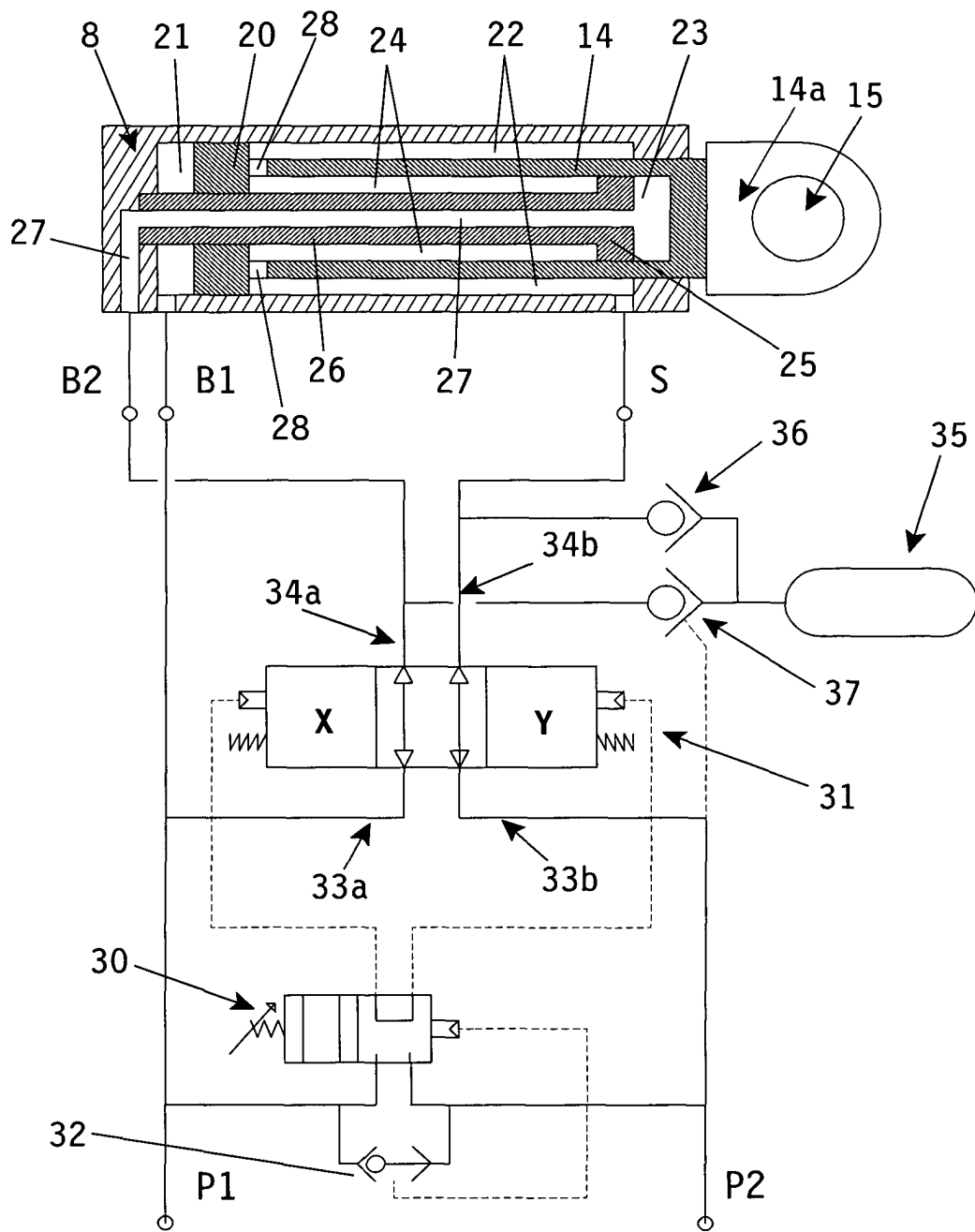


Fig. 5

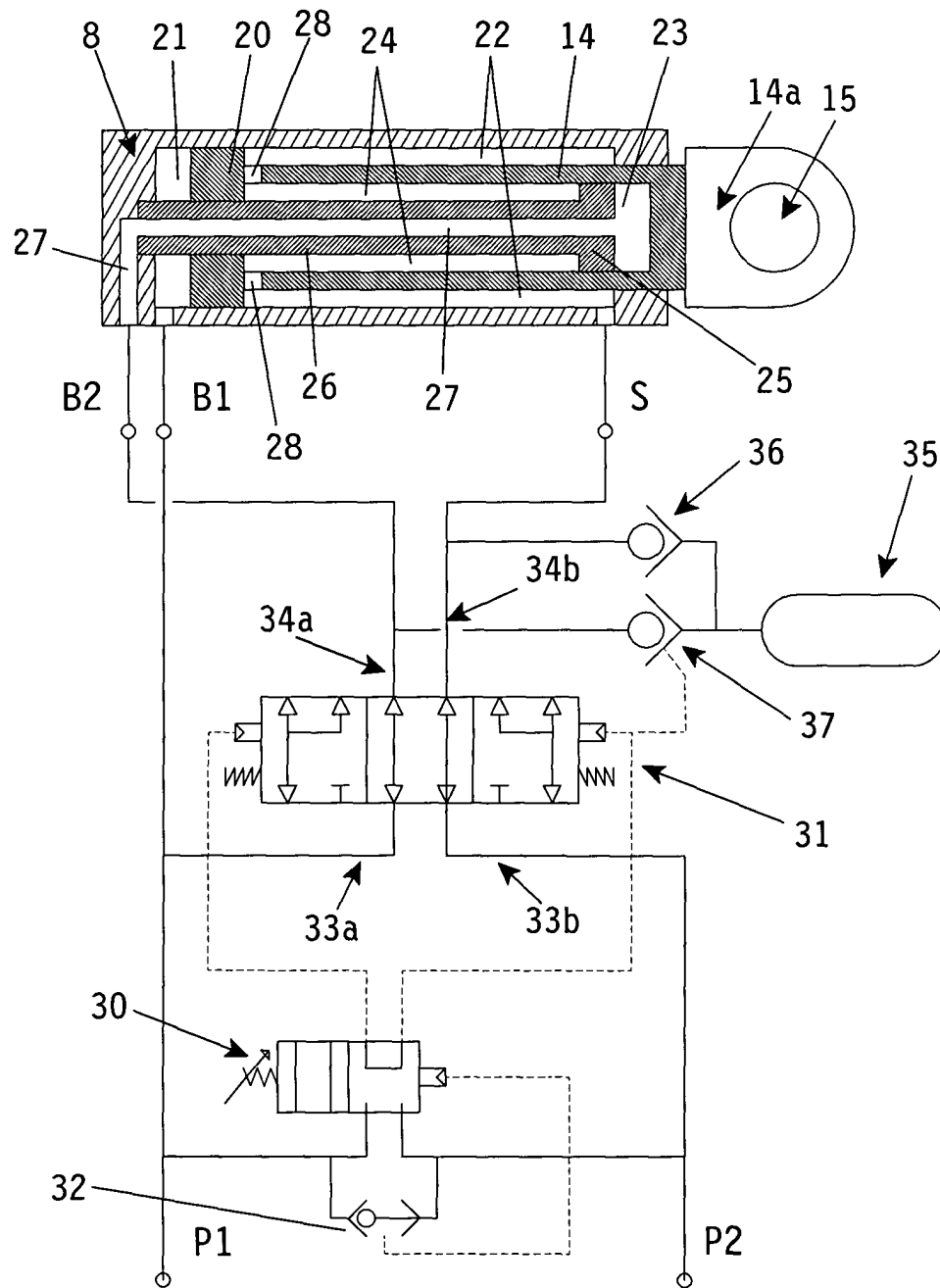


Fig. 6



European Patent
Office

EUROPEAN SEARCH REPORT

Application Number
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Place of search The Hague		Date of completion of the search 9 June 2005	Examiner Rechenmacher, M
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