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PISTON-SLIDE CONTROL VALVE FOR HYDRAULIC PRESSES

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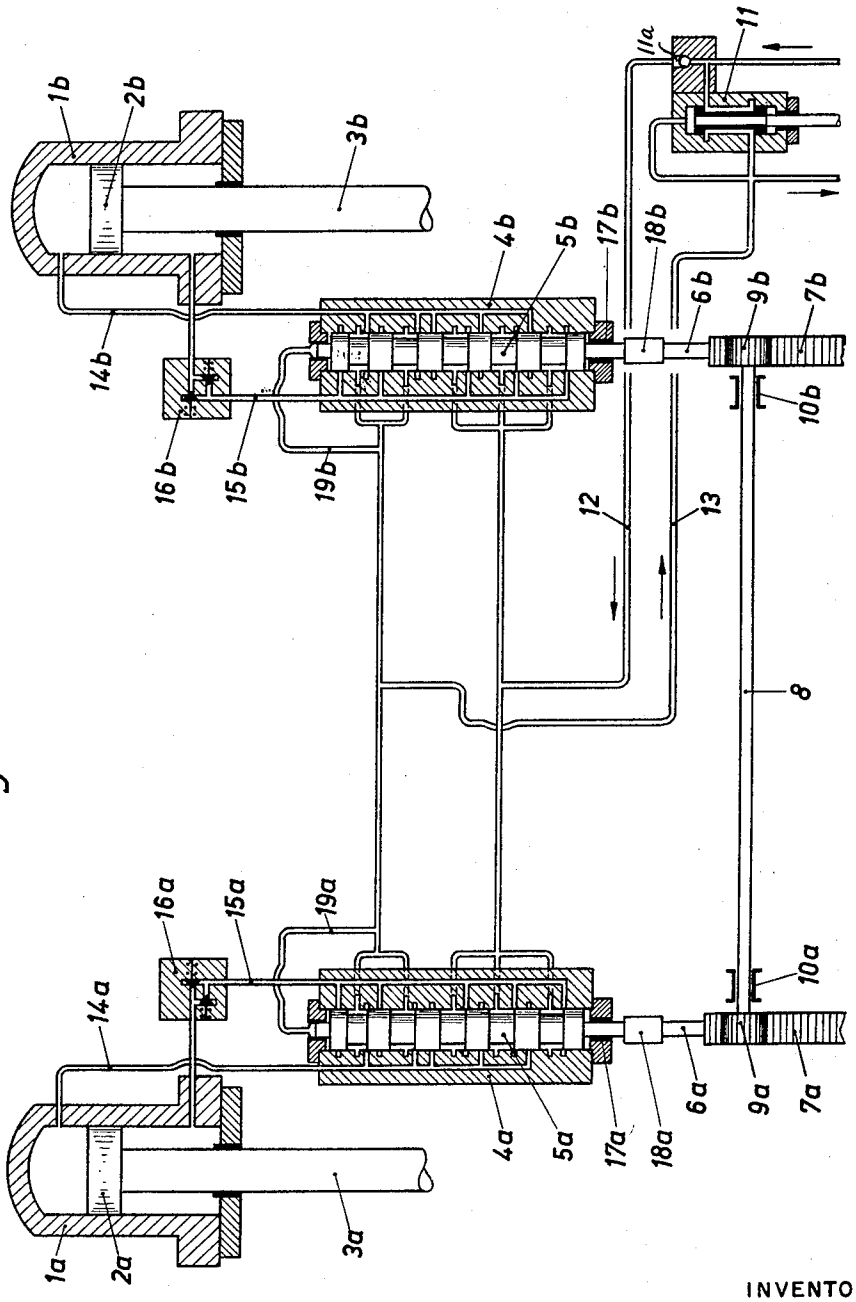


Fig. 1

INVENTOR

HANS SCHARRINGHAUSEN

BY Toulmin & Toulmin

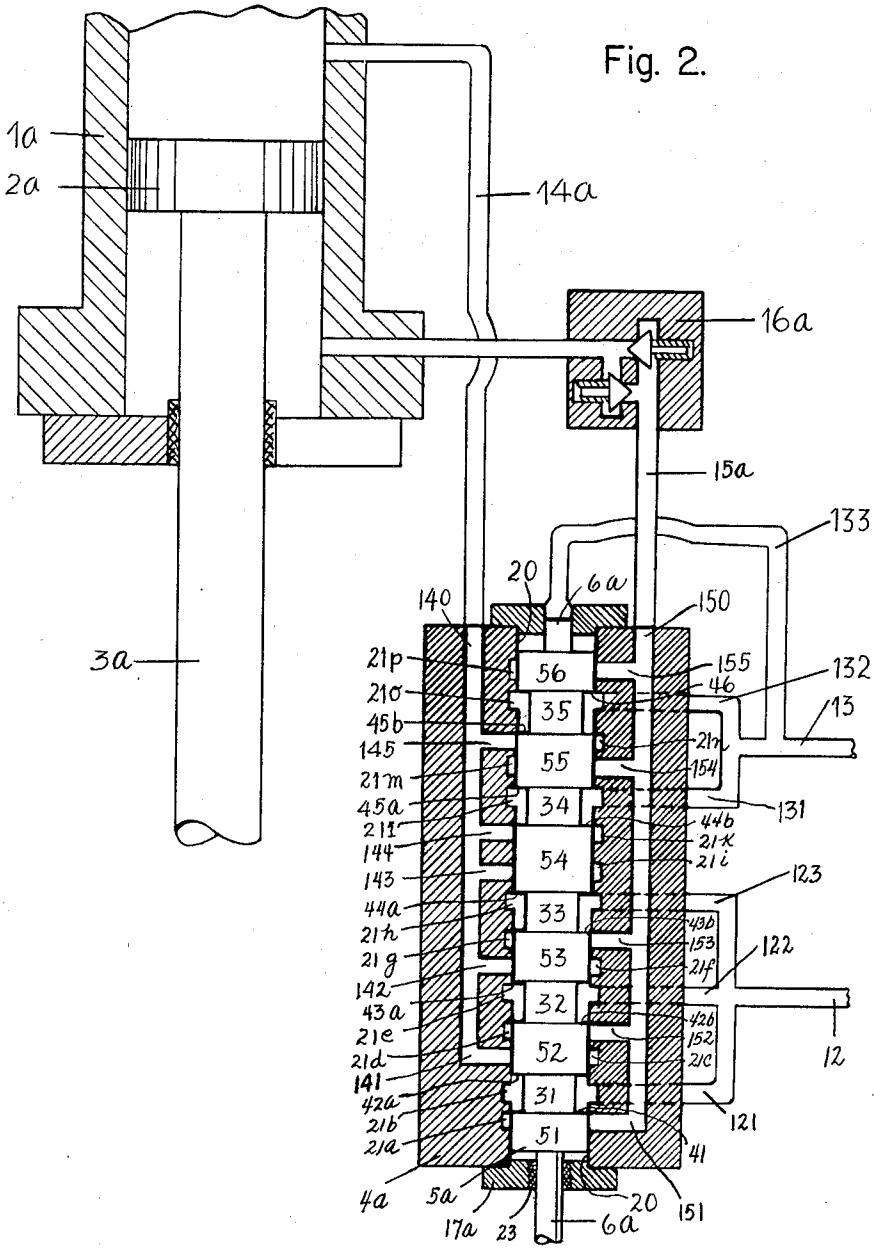
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Fig. 2.



INVENTOR.
HANS SCHARRINGHAUSEN
BY Toulmin & Toulmin

ATTORNEYS

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PISTON-SLIDE CONTROL VALVE FOR HYDRAULIC PRESSES

Hans Scharringhausen, Merklingen (Weser), Germany, assignor to Scharringhausen Maschinenbau Gesellschaft m. b. H., Rheinsheim, Germany

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8 Claims. (Cl. 121—46.5)

This invention relates to improvements in control valves such as piston slide valves and the like for hydraulic presses. It is particularly useful for such valves when applied in plate working machines having several pressing cylinders with double-acting pistons.

It is an object of this invention to provide an improved type of control valve for the aforesaid kind of machines, whose ratio of back stroke velocity to working stroke velocity shall approach the theoretical value.

It is a further object of this invention to provide an improved type of control valve in machines of the aforesaid type, wherein the regulating velocity is increased during the working stroke.

It is another object of the invention to provide for a control valve of the piston-slide valve type which allows for an increased quantity of working fluid to pass therethrough in the time unit without decreasing the sensitiveness of control of the machine with which it is applied, on the one hand, and without increasing the working stroke velocity and the diameter of the control piston on the other hand.

It is already known to use control valves of the piston slide type with hydraulic presses such as plate working machines having several pressing cylinders. These control valves are usually arranged between the supply source for the pressure medium, such as, for instance, a high pressure oil pump having a pressure oil tank connected thereto, and the pressure cylinders carrying out the working operation. Control valves of this kind may control the flow of the pressure medium to and from either one single or several working cylinders, depending on the manner of their being arranged in the flow system. In the first case one single piston edge, or both piston edges, i.e., the upper and the lower edge, and in the second case, usually both piston edges, are utilized as steering edges in the valve cylinder. The control pistons are moved by any kind of steering mechanism suitable for this purpose and free or close the passage for influx or outflow of the pressure medium to a greater or smaller degree, thereby controlling the movement of the pistons in the working cylinders.

The working pistons may either be single acting, in which case the return movement of the pressure tools is effected by strong springs, or by special so-called return pistons, or they may be double acting and may effect the return movement themselves.

My invention is based on observations made with double acting working pistons. The upper piston face of such double acting pistons comprising the surface F_a upon which the pressure medium is acting during the working stroke is known to be substantially larger than the bottom face of the piston having the area F_r which is acted upon during the back stroke, because the face surface of the latter is diminished by the cross section of the piston rod connected with the piston. On the basis of well known hydraulic laws, the piston velocity during the back stroke, V_r, must be greater than the piston velocity during the working stroke, V_a, according to the

reciprocal ratio of the piston surfaces, if the same pressure of, for instance, 100 atmospheres excess pressure, or more, is acting upon the piston during both movements. Thus, if for instance

$$F_a:F_r=2:1$$

then

$$V_r:V_a=2:1$$

In a hydraulic press brake or folding press having this ratio of piston surfaces, the practical values found were

$$V_a=1.9 \text{ and } V_r=2.1$$

meters per minute instead of the theoretical values of

$$V_a=2 \text{ and } V_r=4$$

meters per minute.

In most machine tools having reciprocating tools, it is desirable to have a greater back stroke velocity than the working velocity. Therefore, the fact that the practical value for the back stroke velocity is considerably below the theoretical value represents a serious disadvantage.

When examining the possible causes for this excessive reduction of the practical back stroke velocity, I have found that the influx of the pressure medium is too low. I have further made the unexpected observation that it is also too low during the working stroke. I have discovered in principle a hitherto unknown disadvantage which is common to all piston slide control valves in which the control edges of the piston open annular channels of only very small cross section in the cylinder walls of the valves in order to achieve the highest possible sensitiveness of regulation, which piston edges often open gaps of only a small fraction of a millimeter where extreme sensitiveness of control is required. The hydraulic resistance offered by these narrow annular channels and grooves is, however, so great that it causes a strong braking effect on the flow velocity of the pressure meter. Furthermore, the special kind of pumps which are particularly suitable for use with hydraulically driven plate working machines have unfavorable characteristics insofar as the fed quantity of medium greatly decreases with increased oil pressure. In consequence thereof the flow velocity through this kind of pump is further decreased so that the working piston velocity and together therewith the working velocity of the entire machine remains smaller than is warranted by the construction of the machine.

It must furthermore be avoided to increase either the diameter, or for instance the back stroke velocity for which the control valve pistons have been laid out, although these would be two ways of increasing the flow cross section for some or all working strokes. An increase of the cross section is undesirable since experience has shown that this deteriorates the guidance of the control piston in the valve and makes manufacture of the valve as a whole considerably more expensive. It is, on the other hand, undesirable to increase the back stroke velocity since the control valve pistons are usually guided depending upon the position of the tool so that no differences between the movements and/or velocities of the control valve piston during the working stroke and the back stroke are possible.

Finally, it is necessary to maintain a great sensitiveness of regulation so that it is not possible to cut off the flow of pressure medium completely at certain positions by, for instance, a short travel of the control piston. It is rather to insure such sensitiveness of regulation that grooves and channels of a very small diameter are used in the walls of the control cylinders. For the same reason an increase in the diameter of these grooves and channels

in the direction of the piston movement is not feasible as a means of overcoming the observed disadvantages.

My invention solves this problem by providing the piston slide of the control valves in the aforesaid and similar cases, with a plurality of control edges and with a corresponding plurality of channels and grooves for the influx and outflow of the pressure medium, which channels and grooves are connected with each other by means of annular feeding and outflow channels, thus replacing the hitherto used single influx and outflow channels for the passage of the pressure means, and the single control edge opening or closing them. Thereby I maintain unchanged on the one hand the high sensitiveness of regulation, and on the other hand the velocity as well as the diameter of the control piston, while I nevertheless achieve a considerably greater cross section for the passage of the pressure medium. My invention thus achieves an increased regulating velocity at the working stroke as well as the theoretical value of piston velocity at the back stroke.

This invention will be more clearly understood upon reference to the following specification taken in conjunction with the accompanying drawings which explain my invention by way of example in the case of a piston slide control valve which is, for instance, destined for use with a hydraulic folding press.

In the drawings:

Figure 1 represents a schematic view of the arrangement of a piston slide valve control according to my invention, in partial section; and

Figure 2 shows the left hand control piston valve of Figure 1 in enlarged scale.

Referring to the drawings in detail, the control means comprises as main element (which is usually employed in pairs) two working or pressing cylinders 1*a* and 1*b*, each having a double acting working piston 2*a* and 2*b* respectively, and piston rods 3*a* and 3*b*, as well as two control cylinders 4*a* and 4*b*, each having a control piston 5*a* and 5*b*, together with control piston rods 6*a* and 6*b* respectively. The piston rods 3*a* and 3*b* engage the pressing or pressure platen of the folding press which is not shown. The control rods 6*a* and 6*b* possess at their lower or outer ends racks 7*a* and 7*b* respectively which engage control pinions 9*a* and 9*b* which are in turn mounted on a common control shaft 8. The control shaft 8 is preferably mounted on the displaceable press platen, for instance by means of bearings indicated at 10*a* and 10*b*.

The pressure medium, for instance a pressure oil of 100 atmospheres excess pressure, is supplied from a pressure medium source (not shown) by way of a shiftable main valve 11 and through the conduit 12 to both control cylinders 4*a* and 4*b*, and is returned from the cylinders through the return conduit 13. The main valve 11 is also represented by way of example as a piston valve. Supplying of the pressure medium, for instance from a pressure tank, takes place by way of the check valve 11*a* provided in the conduit 12. When the piston 22 of valve 11 moves upwardly the supply of pressure medium is throttled. If the piston 22 moves downwardly, the conduit of the valve is equally interrupted. In its intermediate position, however, the pressure medium supplied from the tank is returned through the valve and in this position of the main valve 11 the working pistons 2*a* and 2*b* remain stationary in their momentary position.

Conduit 12 ends in their branch conduits 121, 122 and 123 for each control cylinder, and conduit 13 ends in three branch conduits 131, 132 and 133 for each control cylinder.

As will be seen in detail in Figure 2, the central bore 20 of each control cylinder (4*a* or 4*b*) is provided with a number of annular recesses 21*a*—21*p* which are distributed at determined distances over the length of bore 20.

Each of the aforesaid branch conduits 121, 122 and 123, as well as branch conduits 131 and 132 end in one

of these annular recesses, namely conduit 121 in 21*b*, 122 in 21*e*, 123 in 21*h*, 131 in 21*l*, and 132 in 21*o*. Furthermore, two conduits 14*a* and 14*b* are provided leading to the top surfaces of the working cylinders, and two conduits 15*a* and 15*b* lead to the bottom surfaces of the working cylinders in the customary manner. A double check valve 16*a* or 16*b* respectively is arranged in each of the conduits 15*a* and 15*b*.

Referring againt to Figure 2, the aforesaid conduit 14*a* is connected to a channel 140 in the control valve cylinder 4*a* from which branch channels 141, 142, 143, 144 and 145 establish communication with annular recesses 21*c*, 21*f*, 21*i*, 21*k*, and 21*n* respectively. On the other hand, conduit 15*a* is connected to a channel 150 in cylinder 4*a* which communicates by way of branch channels 151, 152, 153, 154 and 155 with annular recesses 21*a*, 21*d*, 21*g*, 21*m* and 21*p* respectively.

Each long control piston, e.g. piston 5*a* in Figure 2, comprises a number of lands 51, 52, 53, 54, 55 and 56 and is provided with grooves 31, 32, 33, 34 and 35 between each pair of adjacent lands, thus defining a plurality of control edges where each land and groove meet, which control edges cooperate with corresponding recesses 21*a*—21*p*.

Thus the upper control edge 41 formed by land 51 and groove 31 cooperates with recess 21*a*; the lower control edge 42*a* between grove 31 and land 52 cooperates with recess 21*c*, the upper control edge 42*b* of land 52 cooperates with recess 21*d*, and so forth.

Branch conduit 133 of return conduit 13 is connected to the top of cylinder bore 20.

The lowermost land 51 in cooperation with the wall portion of bore 20 which it contacts, serves to diminish the pressure on the packing 23 about the control rod 6*a*, and on the guide bush 17*a* for the latter.

The great length and the multiple subdivision of the control piston (5*a* or 5*b*) also offers the added advantage over a conventional increase in diameter of the piston, that it results in a greatly improved guidance of the pistons in the long cylinder bore 20.

In the embodiment of the control valve according to the invention shown in the drawings, each control piston is thus provided with five lands and intermediate annular grooves, and consequently possesses ten control edges 41, 42*a*, 42*b*, 43*a*, 43*b*, 44*a*, 44*b*, 45*a*, 45*b*, and 46; and the cylinder wall of each control valve is provided, again by way of a preferred example only, with fifteen annular grooves and corresponding channels in the cylinder block. Five control edges on the control pistons namely 42*a*, 43*a*, 44*a*, 44*b* and 45*b* are provided for cooperation with the top surface of the working piston 2*a* (or 2*b*) through opening and closing recesses 141 to 145 respectively; and five control edges, namely 41, 42*b*, 43*b*, 45*a* and 46 cooperate with the bottom surface of the working piston 2*a* (or 2*b*) by opening or closing recesses 151 to 155 respectively. Of these five control edges for each end surface of the working pistons three are for instance, used for controlling the influx of the pressure medium to, and two for controlling the outflow of the pressure medium from the working pistons. Thereby the hydraulic cross sectional resistance to the pressure medium flow through each control valve is reduced to one-third for the influx and to one-half for the outflow of the pressure medium.

In order to remove the pressure medium which has been urged upwardly in the cylinder heads along the cylinder walls, the cylinder heads are connected by the aforesaid conduits 133 with the return conduit 13. The three lower branch channels 141, 142 and 143 of channel 140 leading to the conduit 14*a* (and in a similar manner to conduit 14*b*) serve for feeding the pressure medium to the working cylinders 1*a* (and 1*b*) respectively during the working stroke, while the two upper branch channels 144 and 145 serve for withdrawing the pressure medium during the back stroke. Correspondingly the three lower

branch channels **151**, **152** and **153** which lead to the two conduits **15***a* and **15***b* serve for feeding the pressure medium to working cylinder **1***a* (and **1***b*) during the back stroke and the two upper branch channels **154** and **155** serve for withdrawing the pressure medium during the working stroke.

During the first step of operation the cooperating branch channels **141**, **142** and **143** are thus connected in parallel due to a short upward movement of piston **5***a*, whereby control edges **42***a*, **43***a* and **44***a* open coordinated annular recesses **21***c*, **21***f* and **21***i* with which they cooperate, and establish simultaneously communication between conduits **121**, **122**, **123** and channels **141**, **142**, **143**, so as to feed hydraulic medium through conduit **14***a* to the top surface of working piston **2***a*.

During this working stroke the piston **2***a* (or **2***b*) is thus moved downwardly, pressure medium from below the working piston escaping by way of check valve **16***a* and conduit **15***a* and from there through channel **150**, control edges **45***a* and **46** having opened communication from branch channels **154** and **155** through recesses **21***m* and **21***p* to recesses **21***l* and **21***o* and return conduit branches **131** and **132**.

The control edges and annular grooves are so coordinated with each other that (in the case of the embodiment as shown in the drawings) the three parallel feeding recesses **21***c*, **21***f* and **21***i* connected to conduits **14***a* and **14***b*) are opened during the working stroke, i.e., when the pistons **2***a* and **2***b* move downwardly, while the two parallel outflow recesses **21***k* and **21***n* connected to conduits **14***a* (and **14***b*), as well as the three parallel feeding recesses **21***a*, **21***d* and **21***g* connected to conduits **15***a* (and **15***b*) are closed, while simultaneously the two parallel outflow recesses **21***m* and **21***p* connected to the conduits **15***a* (and **15***b*) are open. During the back stroke, when the pistons **2***a* and **2***b* move upwardly, the opening and closing of the annular grooves takes place in inverted order.

It is a further feature of my invention that all parallel connections, i.e., the connecting channels of the annular grooves are located in the cylinder body of the control valves. This requires the casting of a body having the corresponding number of channels, but it has the advantage of great safety against pressure and permits an easy mounting of the control valves in the machine. The internal channels in the control valves are preferably arranged in different planes. In the drawings the channels are all shown as being arranged in the plane of the drawings for drawing purposes only. However, the interruption of the channels leading from the conduits **12** and **13** to the annular grooves is intended to show that the channels are to be arranged in different planes. The last-named channels should be arranged, for instance, in a plane perpendicular to the plane of the drawings. It is also indicated in the drawings that the channels through which the pressure medium passes in the same direction, i.e., on the one side the channel connected to conduits **12** and **13**, should in practice be at a level providing the necessary sealing effects of the lands of the pistons above the level of the channels connecting the conduits **14***a*, **14***b*, **15***a* and **15***b* respectively. Thus in an arrangement of this kind the shortest possible length of the passages through the valves, and correspondingly the lowest possible flow resistances are obtained.

The method of a differentiated feeding of the pressure medium to two or more working cylinders of hydraulic presses and similar machines, which can be carried out with the aid of control valves according to my invention in a particularly advantageous manner, is known as such and need not be described in greater detail. It allows to feed the pressure medium to each working cylinder under such pressure and in such quantity that the resulting factor of pressures exercised by the hydraulic cylinders coincides with the resulting factor of the counterforce exercised by the pressed working piece. This

allows in particular to achieve a parallel and straight guidance of the press platen or other similar tools of hydraulic machines free of binding. The control arrangement according to the embodiment as shown in the drawings provides for the following operation: the piston rods **6***a* and **6***b*, and together therewith the control pistons **5***a* and **5***b* are at first adjusted according to the desired control operation. The pinions **9***a* and **9***b* connected with the control shaft **8** are idle at this stage. However, as soon as the pressed platen, which is assumed to be connected to the control shaft **8** adopts an oblique position, one of the pinions will adopt a twisting angle against the other and will adjust its corresponding rack **7***a* or **7***b* and together therewith the piston rod **6***a* or **6***b* and the control piston **5***a* or **5***b* in such a manner that the excessive advance of the corresponding working piston **2***a* or **2***b* is stopped by a corresponding regulation of the influx of pressure medium, until the pressed platen has been readjusted to a straight position. During this adjustment the annular feeding grooves in the one control valve are opened more, while they are partially or completely closed in the other valve. The operation of the entire control arrangement takes place without any complicated insertion of levers, rod mechanisms, or other organs, thus being automatic and at the same time particularly sensitive. The means indicated at **18***a* and **18***b* serve for the longitudinal adjustment of the control piston rods **6***a* and **6***b* in order to achieve an exact adjustment of the central or starting position of the control pistons **5***a* and **5***b* relative to each other and to their driving means.

Each control piston of the main valve **11** thus reciprocates between two extreme positions. In each of the latter positions, one annular recess is closed entirely by a piston land. In the lowermost position of the piston the upper annular recess (**21***p*) is closed, and in the uppermost piston position the lowest annular recess (**21***a*) is closed.

In both extreme positions the pressure medium rises past the ball in the check valve **11***a* into the conduit **12** and therethrough to the control valves; if, however, the control valve pistons are in central position, the pressure medium can be returned directly to the oil tank (not shown). The check valve **11***a* thus serves to withdraw the pressure medium from the control valves when the control pistons are in central position, and to feed pressure medium to the control valves when the control pistons are in an end position either during upward or downward movement of the working pistons.

A special advantage of my invention resides in the fact that through providing a plurality of control edges on the control valve pistons, heating of the pressure medium is avoided. Such heating occurs frequently through friction of the pressure oil, if only one control edge is available, as is the case in the hitherto known valves. Heat of the pressure oil above 140° F. may cause the oil to be decomposed or saponified and thus to become unsuitable for further use. This will not occur in control valves according to my invention.

The control valves according to my invention may be constructed in a variety of manners and may also be arranged differently from the embodiment shown in the drawings, which has been used as an explanatory example only. In a simplified form, a control valve according to my invention may also be used for controlling machines having only single acting working pistons.

I claim:

1. A piston-slide control valve for use with hydraulic presses and the like, having several pressing cylinders with double-acting working pistons, said valve comprising a cylinder body having a bore therein, a plurality of annular recesses in said bore, a control piston reciprocatingly mounted inside said bore and comprising a plurality of alternate lands and annular grooves forming a first number of control edges on said control piston for

opening simultaneously during a given movement of said control piston certain of said recesses and a second number of control edges for closing the remaining recesses and invertedly closing the former and opening the latter recesses during a movement of the control piston in opposite direction, and a number of channels corresponding to the number of said recesses and connecting the latter with conduits for feeding and withdrawing the pressure medium for said pressing cylinders; each of said opening control edges being disposed with regard to the next adjacent one at a distance equal to the distance between those annular recesses cooperating with said opening control edges, thereby causing all of said opening control edges to cooperate simultaneously with said annular grooves so as to open parallel paths therethrough, and all of said closing control edges cooperating with said annular recesses to close such paths simultaneously.

2. A control valve according to claim 1, wherein several of said annular recesses inside said cylinder bore are connected in parallel at a determined distance from one another for feeding the pressure medium, and that separately therefrom the remaining annular recesses are connected in parallel and at said determined distance from one another for withdrawing the pressure medium.

3. A control valve according to claim 1, wherein all of said annular recesses and channels conducting the flow of pressure medium to and from the head surface of said double-acting working cylinders are separate from all of said annular recesses and channels serving the tail end of said double-acting working pistons.

4. A control valve according to claim 1, wherein said annular recesses and channels are divided into a first and a second group, a first joint conduit being provided for connecting said first group of annular recesses and channels with the head surface of said double-acting working pistons, and a second joint conduit being provided for connecting said second group of annular recesses and channels with the tail surface of said working pistons.

5. A control valve according to claim 4, wherein all said channels are arranged inside said cylinder body of said valve, said channels being provided in different planes through the axis of said cylinder, each pair of a feeding and a corresponding withdrawing channel being arranged on opposite sides of said bore with regard to one another, in planes spaced only by the distance required for sealing by said control piston lands.

6. A control valve according to claim 4, wherein the total number of said control edges is ten and wherein five of said control edges are provided by certain lands on said control piston for controlling the flow of pressure medium to and from the head surfaces of said working pistons, and five other control edges are provided by certain lands on said control piston for controlling the flow of pressure medium to and from the tail surfaces of said working pistons, three of the former and three of the latter control edges controlling the flow of pressure medium to said working pistons, and two of the former as well as two of the latter control edges controlling the flow of pressure medium from the same.

7. A piston slide control valve for use with a hydraulic press having at least one pressing cylinder with a double-acting piston therein, a source of hydraulic pressure fluid, and connecting lines from said control valve to said pressing cylinder and from said valve to said source of

hydraulic pressure fluid for the passage of said fluid therethrough, in either direction, said valve comprising a casing having a central bore being closed at both ends, and a plurality of annular recesses in the wall of said bore, a control piston disposed for reciprocating movement in said bore, said piston comprising several lands and annular grooves therebetween, each land and adjacent groove forming a circumferential control edge of said piston, and a plurality of channels in said casing connecting said connecting lines with said annular recesses so as to establish communication with determined groups of the latter for the flow of pressure fluid in parallel through the same in a given direction of flow, said annular recesses in the wall of said bore, and said annular grooves of said piston being so distributed over the length of the bore and piston respectively, that by the movement of said piston in a given direction, several of said control edges always open simultaneously communication between certain of said annular recesses and certain of said annular grooves so as to allow the flow of pressure fluid through several parallel paths between said valve and said pressing cylinder, and that by the movement of the piston in the opposite direction, the same control edges always interrupt simultaneously the aforesaid communication.

8. A piston slide control valve for use with hydraulic presses and the like which presses have several pressing cylinders with double-acting working pistons and a source of hydraulic fluid therefor, said control valve comprising a cylinder body having a bore therein, a control piston reciprocatingly mounted inside said bore, a plurality of first, second and third annular recesses in the wall of said bore, first conduit means for feeding hydraulic fluid from said fluid source to said valve and comprising a first number of parallel channels opening each into one of said first annular recesses, second conduit means for withdrawing hydraulic fluid from said valve and comprising a second number of parallel channels each opening into one of said second annular recesses, third conduit means for the flow of fluid alternatingly in different direction between said control valve and one of said pressing cylinders, and comprising a plurality of channels each connected to one of said third annular recesses, said control piston comprising a plurality of lands bearing two groups of control edges, one of which groups opens communication between certain of said third annular recesses simultaneously with all of said first and second annular recesses so as to cause fluid flow between said valve and said pressing cylinder in one direction during a given direction of movement of said piston in said valve bore, and the other group opens communication between the other recesses of said third annular recesses simultaneously with all of said first and second annular recesses so as to cause fluid flow between said valve and said pressing cylinder in opposite direction during opposite movement of said piston in said bore.

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