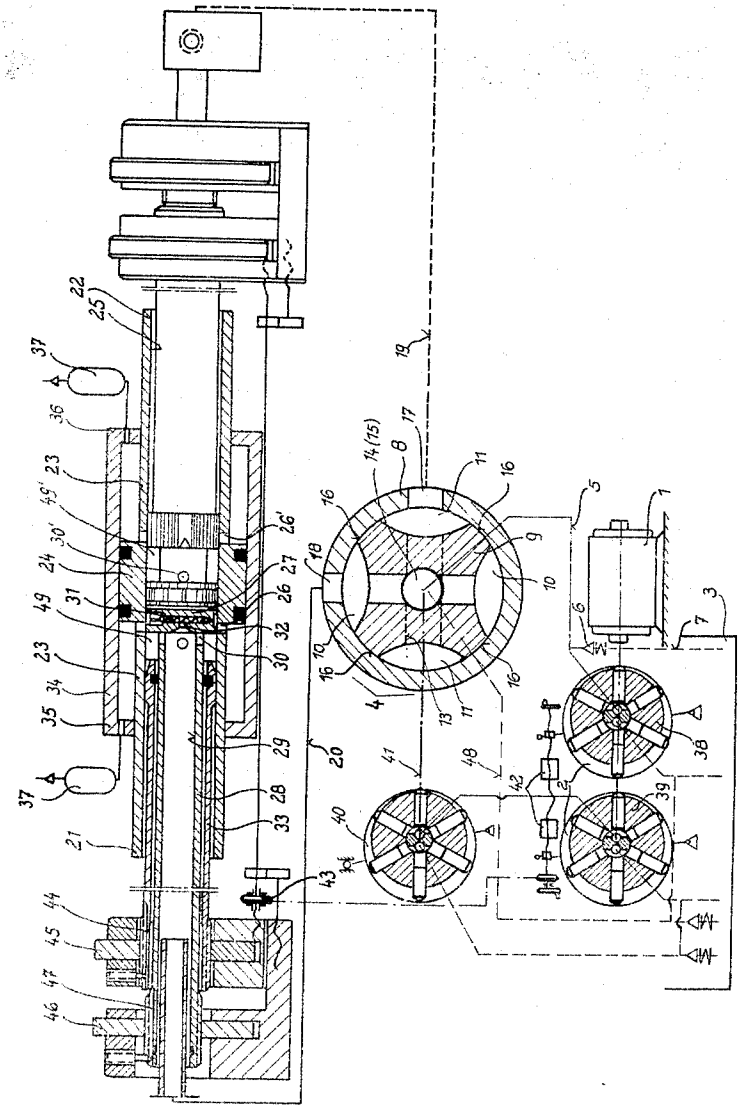
Dec. 15, 1959 M. SCHÖNHERR 2,917,028

CONTROL DEVICES FOR HYDRAULIC DRIVES

Filed April 30, 1958 3 Sheets-Sheet 1

Fig. 1

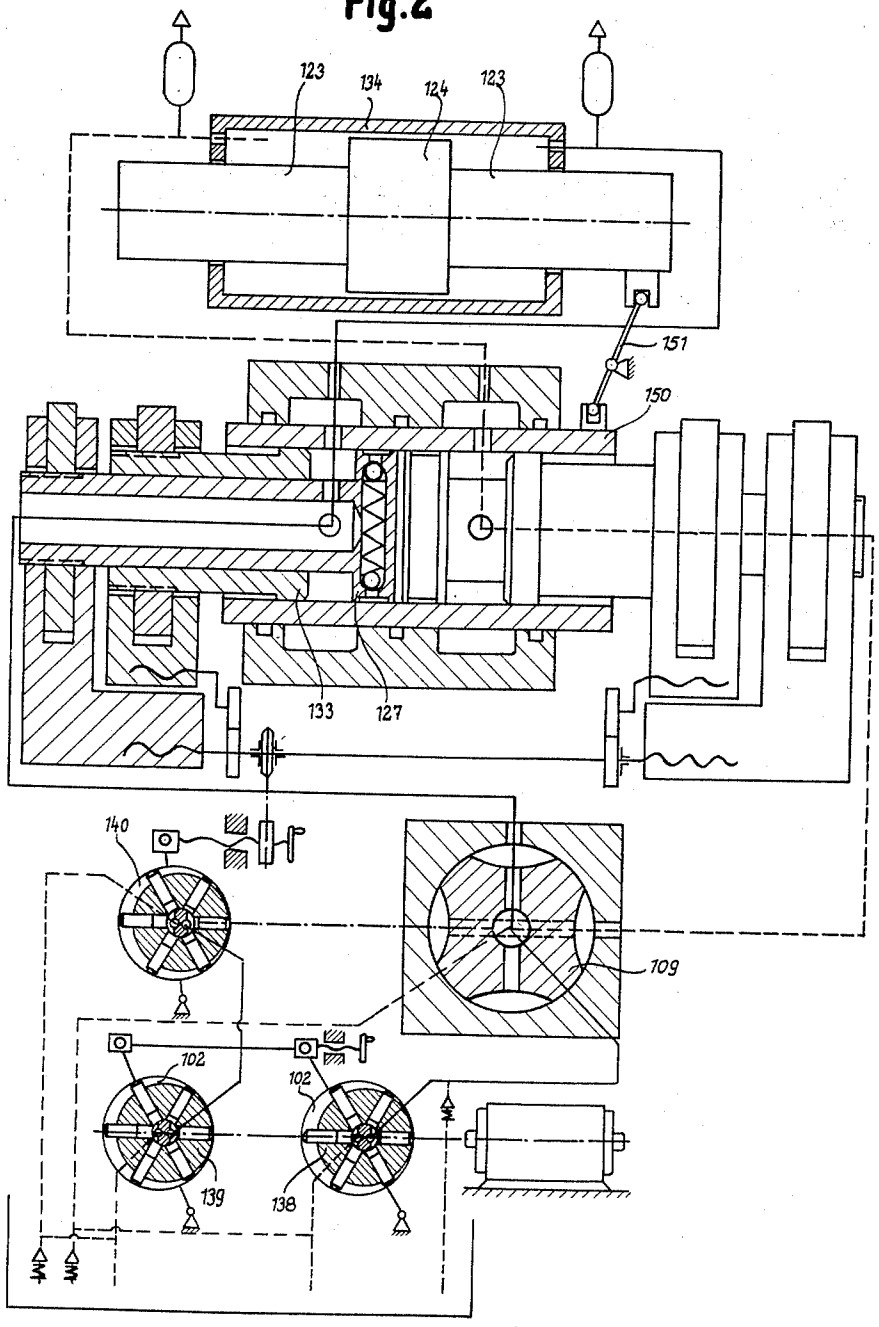


Dec. 15, 1959 M. SCHÖNHERR 2,917,028

CONTROL DEVICES FOR HYDRAULIC DRIVES

Filed April 30, 1958 3 Sheets-Sheet 2

Fig.2



Dec. 15, 1959 M. SCHÖNHERR 2,917,028

CONTROL DEVICES FOR HYDRAULIC DRIVES

Filed April 30, 1958

3 Sheets-Sheet 3

Fig.3

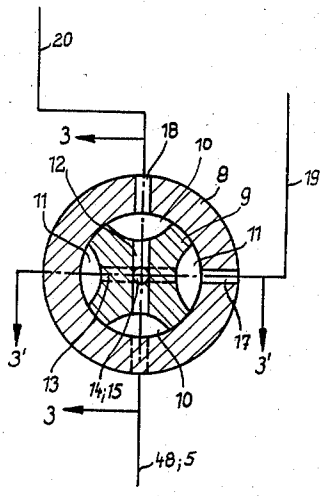


Fig.4

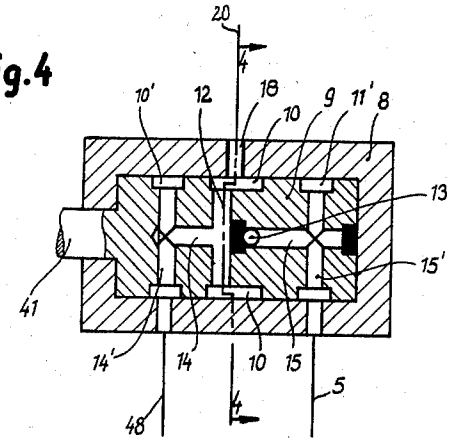
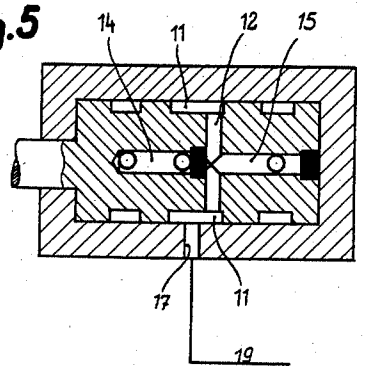


Fig.5



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CONTROL DEVICES FOR HYDRAULIC DRIVES

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Application April 30, 1958, Serial No. 732,030

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5 Claims. (Cl. 121—147)

This invention relates to a method and apparatus for controlling the movement of a double-acting piston actuated by a hydraulic fluid.

Heretofore, in such devices, it has been customary to provide means for causing a stream of hydraulic fluid to flow from a pressure generator through a preliminary control valve to the principal control valve, and from the latter alternatingly to the two ends of the hydraulic cylinder. The control is not actuated directly in such devices by the main control valve, but rather by stops or abutments fastened to the work table which pivot abutting levers, which in turn change the position of the preliminary control slide valve. The supply of hydraulic fluid to the main cylinder is thereby gradually throttled and finally the main control slide valve is hydraulically shifted. Thus, the main control slide valve overcomes its dead center position while the work table is at rest without requiring any additional spring biasing or balanced loading.

It is furthermore known to control reversal of flow by means of a pivotable piston valve. The pivoting movement of such valve is initiated either by stops cooperating with levers or by electrical switches and solenoids.

The prior control methods and devices have a common disadvantage caused by the inertia of the control elements which can only be slowly overcome by forces of acceleration. This inertia limits the number of strokes possible per unit time, which is of particular disadvantage when the length of stroke is short.

Another disadvantage of the control devices of the prior art is that the change in direction is initiated by the abutments. The impulses generated by the abutments are transmitted to the preliminary control valve pistons by extensive lever systems or similar devices. This method of transmission of the reversal of direction causes further time losses because of its inertia. It is also rather difficult to eliminate the source of accidents inherent in abutments. Furthermore, because of such stops or abutments it becomes very costly to produce a machine with a flat plane surface, which is often very desirable.

According to another known method, reversal of direction can be achieved by "over dead center control" of the hydraulic pump. The piston housing of the pump is pivoted in an opposite direction for each reversal of direction in such a manner that, in one position the flow of fluid is directed to one side of the cylinder, and in the other position to the other side of the cylinder. However, in such devices, it is particularly difficult to obtain acceleration of the large masses inherent in the piston housing. Such acceleration consumes a certain amount of time which must be subtracted from the time available for the work stroke.

It is a further disadvantage in these latter devices that the working piston achieves its full speed only when the pump is fully pivoted, because of the large masses which have to be moved. The pivoting movement requires a certain amount of time. This means that at a high reciprocating speed of the hydraulic device or with short

strokes, the portion of the path of the working piston at which its speed is constant becomes relatively small.

Accordingly, an object of the present invention is to provide a novel and improved system for controlling hydraulic drives which obviates the above-mentioned difficulties.

It is a further object of the invention to provide a method and apparatus for controlling the movements of double-acting pistons which reverses the piston stroke direction as a function of time without the use of abutments or switches, using braking energy for the reversal of direction and permitting a high frequency of strokes per unit time.

These objects of the invention are achieved basically by directing the flow of hydraulic fluid from its source to a rotary piston valve and to a hydraulic motor connected with the rotary piston. The constant but controllable rotation of the rotary piston directs the flow of hydraulic fluid alternately into one of the two lines leading to the two opposing sides of the work piston, the corresponding other line acting as a return line, and being connected to the oil sump. In order to assure the position of the work stroke, a hollow double-ended main piston rod and the work piston are provided with a common axial bore in which bore a control piston is guided from each end of the hollow main piston rod. The control piston is provided with a control piston rod having a central bore connected to a line coming from the rotary piston valve, the control piston rod being surrounded by a sleeve separating it from the bore of the main piston rod. The control piston and sleeve are axially adjustable relative to each other and independently from each other.

The hollow control piston rod is provided with radial bores located near the control piston. The control piston also has a radial bore in which are located ball check valves provided with a common spring and which are connected with the bore of the control piston rod.

The work cylinder is provided at its ends with connections for pressure accumulators. These pressure accumulators, on the one hand cause strong damping of vibrations. On the other hand, braking energy is accumulated therein in a simple manner and is utilized again to cause acceleration of the piston in the opposite direction during the return stroke. It is thus possible to design and install a drive with a capacity which is determined almost exclusively by its work output.

The control system of the present invention has the following advantages, in addition to the usual advantages of a hydraulic drive over a mechanical drive, such as stepless control under load, constant speed over a maximum path, simple means for safety devices against overload, possibility of automatic cycling, etc.

These additional advantages include the possibility of achieving a very large number of double strokes per unit time since, aside from the work piston no control or switching elements need to be accelerated when the direction of the piston is reversed. Reversal of direction, therefore, requires the least possible time and the time absolutely available can be utilized almost exclusively for performing the work stroke.

The continuously rotating rotary piston is not accelerated during the reversal operation and, therefore, has to absorb substantially smaller amounts of energy as compared to the control elements of the prior art which have to be accelerated in conformity with the number of strokes. The present inventive control system, therefore, is universal and is applicable to all high speed reciprocatory movements (up to high oscillatory speeds) although fewer numbers of strokes per unit time are also possible.

The compound control system of the working circuit

and the hydraulic control circuit of the present invention permit a control of piston speed at constant length of stroke. The control system also does not require any electrical control or switching device other than those customary for an electric motor. This structure thus substantially increases the freedom of the device for maintenance.

The particular structure of the device of the present invention which comprises a rotary piston valve and a stationary stroke limiting device makes a switching device unnecessary and thus eliminates inaccuracies of stroke normally associated therewith and resulting therefrom.

Other objects and advantages of the invention will hereinafter become more fully apparent from the following description of the annexed drawings, which illustrate preferred embodiments of the invention and wherein:

Fig. 1 is a partially diagrammatic representation of one embodiment of the invention, showing a piston drive with a work cylinder which remains stationary while the piston rod performs stroke. Only one end of the control piston rod is shown in section, although both ends are similar;

Fig. 2 shows a partially diagrammatic view of another embodiment having a piston drive and work cylinder of the same kind as Fig. 1 but with the stroke limiting device located outside of the drive proper;

Fig. 3 is a sectional view of the rotary piston valve taken along line 4—4 of Fig. 4;

Fig. 4 is a longitudinal section taken along line 3—3 of Fig. 3; and

Fig. 5 is a longitudinal section taken along line 3'—3' of Fig. 3.

Referring now to the drawings, there is shown in Fig. 1 an electric motor **1** which drives a radial piston pump **2** to supply hydraulic fluid from oil supply tank **3** to a rotary piston valve **4**. A pressure release valve **6** with return line **7** leading to oil tank **3** is arranged in a conduit line **5** which leads from pump **2** to valve **4**. The rotary piston valve **4** comprises a housing **8** and a rotary piston **9**. The rotary piston **9** has two axially spaced annular grooves **10'**, **11'** and, in the space between these two grooves, two pairs of opposing milled recesses **10**, **11**. The pairs of recesses **10**, **11** are respectively connected with the annular grooves **10'**, **11'** by transverse bores **12**, **13**, internal central bores **14**, **15**, and transverse bores **14'**, **15'**. The incoming supply of hydraulic fluid is provided to rotary valve **4** through one of the two annular grooves and its corresponding bore **12** or **13**, and the efflux occurs through the other annular groove, as the fluid thus flows to and from the rotary piston **9**. The lands **16** located between the recesses **10**, **11** temporarily obstruct or cover two control ports **17**, **18** located in housing **8** and offset by 90° from each other. These control ports **17**, **18** are connected with the two opposite ends **21**, **22** of a double-ended hollow piston rod **23** by means of two telescoping tubes **19** and **20**. The double-ended hollow main piston rod **23** and the work piston **24** mounted midway thereon have a common axial bore **25**. The double-ended main hollow piston rod **23** is provided with four radial bores **26** and four more radial bores **26'**, respectively located on each side of and immediately adjacent work piston **24**.

A control piston **27** is guided within the axial bore **25** of piston **24** and of hollow piston rod **23**. The control piston **27** is axially adjustable from either end. It has two piston rods **28** (control piston rods) which are also hollow, each having an axial bore **29**, and four radial bores **30** located near control piston **27**. Control piston **27** itself is equipped with a double ball check valve **31** transversely located therein and communicating with the cylindrical surface of control piston **27**. The double ball valve **31** is connected by a central channel **32** with the axial bore **29** of the hollow control piston rod **28**. The control piston rod **28** is surrounded by an axially adjustable stepped sleeve or bushing **33** which provides external sealing means against the bore **25** of piston rod **23**.

The work piston **24** and its double-acting hollow main piston rod **23** are axially movable in a stationary cylinder **34**. A pressure accumulator **37** is connected to each of the two ends **35** and **36** of cylinder **34**.

The following elements are specifically provided for controlling the speed of piston **24** and for controlling the magnitude and position of the stroke:

In order to control the speed of piston **24** and the magnitude of the stroke, the radial piston pump **2** which provides pressure of the hydraulic fluid is equipped with two sets of eccentric pump portions **38**, **39** in each of which the cylinders are star-arranged or radially arranged within a cylinder block which rotates eccentrically within a reaction ring. Since this type of pump portion is known, it is indicated only diagrammatically in the drawings as a central circular star cylinder block and external reaction ring. One such pump portion, star of cylinders **38**, provides a flow of working fluid to the rotary piston valve **4**, the flow passing from there through lines **19** and **20**, through the control piston rod **28** to the cylinder ends **35**, **36**. (This hydraulic circuit will henceforth be referred to as the work circuit.) When this star pump portion **38** is adjusted, the reciprocating speed of work piston **24** is changed. For causing a variation in the length of stroke of piston **24**, hydraulic fluid provided by a second star of cylinders or pump portion **39** is directed to hydraulic motor **40** whose shaft **41** drives rotary piston **9** of rotary valve **4**. (This latter hydraulic circuit will henceforth be referred to as the control circuit.)

The eccentric changes of position of the two star cylinder pump portions **38**, **39** of the radial piston pump **2** are preferably connected in a predetermined ratio. It is, however, also possible to control the star of cylinders **38** separately from those of **39** for generating the flow of working fluid. Combined control of both stars of cylinders **38** and **39**, however, leads to an increase of piston speed with uniform length of stroke.

In order to establish the position of the stroke, both of the control piston rods **28** are shifted together with their respective sleeves **33** in the desired direction. In order to permit supplementary control of the position of piston **24** and under certain circumstances also the length of the stroke, control devices **42** of the radial piston pump **2** are coupled with adjusting devices **43** for adjusting sleeve **33** and for control piston rod **28**. The two sleeves **33** on each of the two control piston rods **28** can be thus simultaneously shifted in opposite directions. A screw thread **44** provided on the end of sleeve **33** and a cooperating axially stationary nut **45** provide a supplementary adjustment means. Control pistons **28** also are further adjustable by means of a nut **46** whose axial position is fixed, and by a thread **47** provided at the end of control piston rod **28**.

The control device of the invention operates as follows:

When electric motor **1** is started, the radial piston pump **2** is driven and begins to rotate. Depending on the position of the controls, the working circuit and the control circuit are provided with determined amounts of hydraulic fluid. The star of cylinders **38** of the working circuit provides a stream of hydraulic fluid to the pressure line **5**. The pressure in line **5**, however, is limited by the pressure relief valve **6**. Any excess of fluid at this point is returned to storage tank **3** through valve **6** and line **7**.

The stream of hydraulic fluid passes through hydraulic line **5** to housing **8** of the rotary piston valve **4** and into one of the annular grooves **11'** therein. It is then directed from one side of the annular groove **11'** of piston **9**, it rises above and through the transverse bore **19**, the internal bore **15** and the transverse bore **13**

into the pair of opposite milled recesses 11. The two other milled recesses 10 are connected by the transverse bore 12, arranged at 90° thereto and by the other internal bore 14, the transverse bore 14 and the annular groove 10' towards the other side with the return line 48, which leads to storage tank 3.

The rotary piston 9 of valve 4 is actuated for steplessly controllable rotary movement by the hydraulic motor 40, which is connected to the control circuit. Operation of the above apparatus therefore results in one of the ends, for example 35, of cylinder 34 being filled with hydraulic fluid through lines 5, 20, the axial bore 29, the bores 30, the annular channel 49 and the bores 26, whereas the other end, for example 36, is connected through bores 26', the annular channel 49', the bores 30' and line 19 with the return line 48.

During the forward stroke (for example to the right in Fig. 1) the hydraulic fluid flows through the telescoping conduit or pipe 20 (or 19 when the stroke is to the left) and bores 29 in control piston rods 28 outwardly through bores 30 into the annular channel 49 between control piston 27 and sleeve 33, and from there into the cylinder 34 through bores 26. Shortly before the stroke is completed, the pressure and also the return flow of the hydraulic fluid is throttled at the bores 26 and 26' of the piston rod 23, which thus acts as a blocking or control member, acting in cooperation with the edge of control piston 27 (in the position of Fig. 1) and sleeve 33. The throttling of the incoming and outgoing flow of hydraulic fluid to the piston 24 thus reduces its speed of travel. The edge of sleeve 33 limits the travel of control piston 27. After the bores 26 and 26' have been completely covered over by control piston 27, upon further continuation of the forward stroke, the pressure accumulator 37 becomes charged with the kinetic energy of the main piston 24. Upon completion of the forward stroke of main piston 24, the rotary piston 9 of valve 4 is then turned sufficiently in its continuous rotary travel until the functions of lines 19 and 20 are interchanged, whereby the pressure line 20 assumes the former function of return line 19 and vice versa.

The initial impulse for the return stroke of the main piston 24 (towards the left in Fig. 1) is provided by the pressure energy which has been stored in the pressure accumulator 37 at the right hand end of cylinder 34. This initial impulse moves bores 26 of piston rod 23 leftwards beyond control piston 27 so as to uncover them. The hydraulic fluid present at the opposite end of cylinder 34 (left side of piston 24) flows outward first through uncovered bores 26, through annular channel 49, through radial bores 30 in hollow control piston rod 28, through axial bore 29 into line 20, and from there through control port 18 of valve 4, into return line 48, through which it returns without pressure into oil sump 3. As the return stroke progresses, back pressure will build up in conduit lines 20, 29 between valve 4 and control piston 27, causing the latter to move leftward more slowly than piston rod 23, until the inlets of ball valve 31 are opposite the bores 26. The remainder of the fluid in cylinder 34 at the left side of main piston 24 then flows through ball valve 31, through channel 32, through axial bore 29 in control piston rod 28, through line 20 into port 18 of rotary valve 4 and from there through return line 48 into oil sump 3.

The position and length of the work stroke is determined by adjustment of the positions of control piston 27 and of the sleeves 33, and their positions relative to each other. The number of strokes per unit time is controlled by the rate of revolution of rotary piston 9. In order to obtain a greater frequency of strokes of equal stroke length, the speed of rotary piston 9 must be increased by increasing the output of star of cylinders 38 in the work circuit. To achieve this result, a combined control of both stars of cylinders 38, 39 is possible. An increased output of star of cylinders 39 of the control circuit then increases the rate of revolution of rotary piston 9. Fine adjustment of the number of strokes per unit time is provided by motor 40 of the control circuit.

Another embodiment of the invention is shown in Fig. 2. In this embodiment, the entire stroke limiting device has been removed from piston 24 and from the double-acting piston rod 23, and arranged externally thereof. The function of this embodiment is substantially the same as that described above in connection with Fig. 1. However, in the stroke limiting means, the position of former piston 24 and piston rod 23 of the embodiment of Fig. 1 is now occupied by the sleeve 150 moving in the same manner but in the opposite direction. In Fig. 2, sleeve 150 and piston rod 123 are linked by a two-armed lever 151.

The combined control of the device of Fig. 1 has also been altered somewhat in this embodiment. The adjustment of control piston 127 and of sleeve 134 is performed simultaneously with the adjustment of hydraulic motor 140 in the control circuit. The common adjustment of the two stars of cylinders 138, 139 of pump 102, therefore, no longer has any influence on the adjusment of the stroke limiting device. However, since stars of cylinders 138, 139 are simultaneously controlled by the control circuit and work circuit, the speed of motor 140 in the control circuit is thus adjusted and therewith the rotary speed of rotary piston 109. The speed of piston 124 is thereby changed at constant length of stroke.

It should be understood, of course, that the foregoing disclosure relates to only preferred embodiments of the invention and that numerous modifications or alterations may be made therein without departing from the spirit and the scope of the invention as set forth in the appended claims. For example, it will be apparent to those skilled in the art that radial piston pump 2 or 102 with its two stars of cylinders 38, 39 and 138, 139 may be replaced by two radial piston pumps or other steplessly controllable pressure source generators.

It is furthermore possible to create the control circuit by any mechanical stepless transmission or by a hydraulic transmission.

Having thus described the invention, what is claimed is:

1. A device for controlling the movement of a double-acting hydraulic piston, comprising a working cylinder, a working piston reciprocable within said working cylinder, a double-ended main piston rod, midway on which said working piston is fixedly mounted, said main piston rod being hollow to form inlet and outlet ducts for flow of hydraulic fluid to and from said working piston and having radial bores in the walls thereof adjacent said working piston, said working piston and said main piston rod being provided with a common axial bore, a control piston reciprocable within said axial bore, a hollow control piston rod operably attached to said control piston, conduit means connected to said hollow control piston rod for flow of hydraulic fluid thereto and therefrom, sleeve means partially surrounding said control piston rod for spacing the latter from said axial bore of said main piston rod, and means for axially adjusting said control piston and said sleeve means relative to each other.

2. A device as set forth in claim 1, wherein said hollow control piston rod is provided with radial bores near said control piston, said control piston having a transverse radial bore, a check valve arranged within said transverse radial bore of said control piston, and means forming a channel from said check valve communicating with the axial bore of said hollow control piston rod.

3. A device as set forth in claim 1, including pressure accumulator means operably connected to at least one of the respective ends of said working cylinder.

4. A device for controlling the movement of a double-acting hydraulic piston, comprising a working cylinder, a working piston reciprocable within said working cylin-

der, a double-ended main piston rod, midway on which said working piston is fixedly mounted, said main piston rod being hollow to form inlet and outlet ducts for flow of hydraulic fluid to and from said working piston and having radial bores in the walls thereof adjacent said working piston, said working piston and said main piston rod being provided with a common axial bore, a control piston, a control cylinder sleeve in which said control piston is reciprocable, a hollow control piston rod operably attached to said control piston, conduit means connected to said hollow control piston rod for flow of hydraulic fluid thereto and therefrom, sleeve means partially surrounding said control piston rod for spacing the latter from said control cylinder sleeve, lever means operably connecting said main piston rod with said control cylinder sleeve, and means for axially adjusting said control piston and said sleeve means relative to each other.

5. A method for controlling the movement of a double acting hydraulic piston in a working cylinder comprising the steps of providing two pressure independent fluid streams by a common pressure source, directing one of said streams to a steplessly controllable hydraulic motor which drives a reversing valve, and directing the other stream rhythmically by way of said reversing valve alternately to the two opposed ends of said working cylinder.

References Cited in the file of this patent

UNITED STATES PATENTS

1,032,689	Martens	July 16, 1912
2,079,041	Ryan et al.	May 4, 1937
2,295,880	Valois et al.	Sept. 15, 1942
2,402,300	Shimer	June 18, 1946