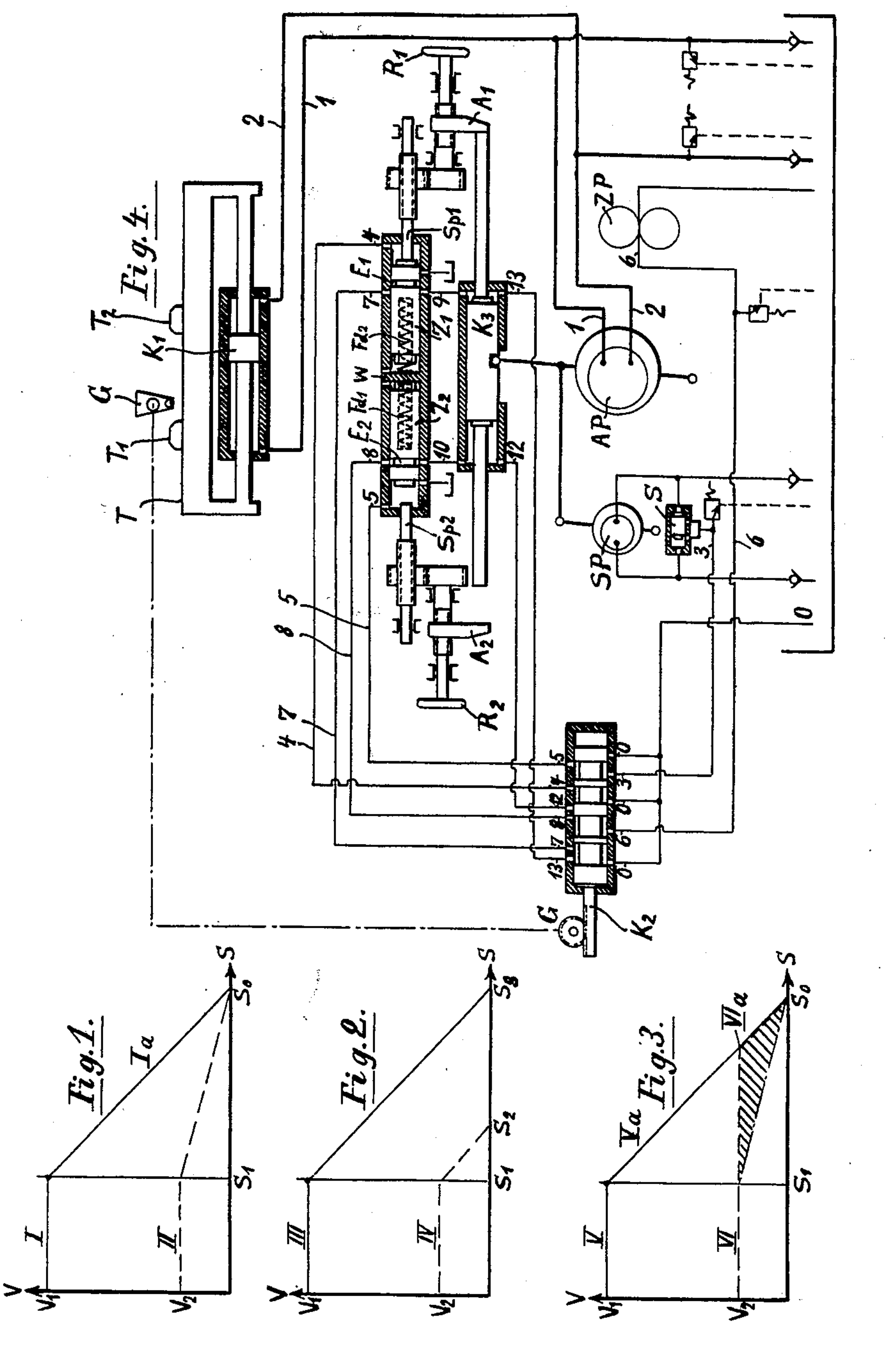
Nov. 3, 1964 F. WEGERDT 3,154,923

HYDRAULIC DRIVE FOR PRODUCING LINEAR INTERMITTENT RECIPROCATORY MOTIONS OF A MACHINE TOOL

Filed July 27, 1962 3 Sheets-Sheet 1



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Filed July 27, 1962 3 Sheets-Sheet 2

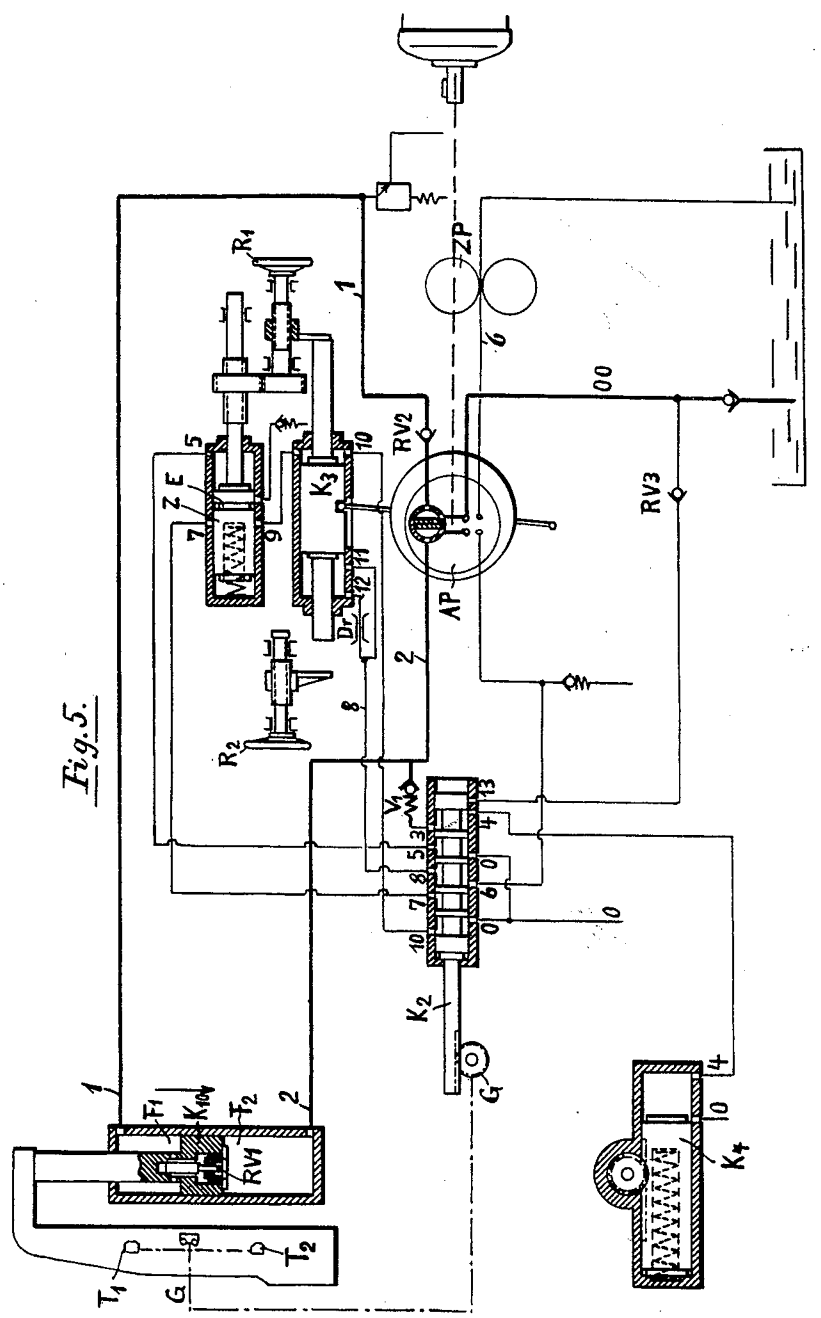


*Fig. 5.*

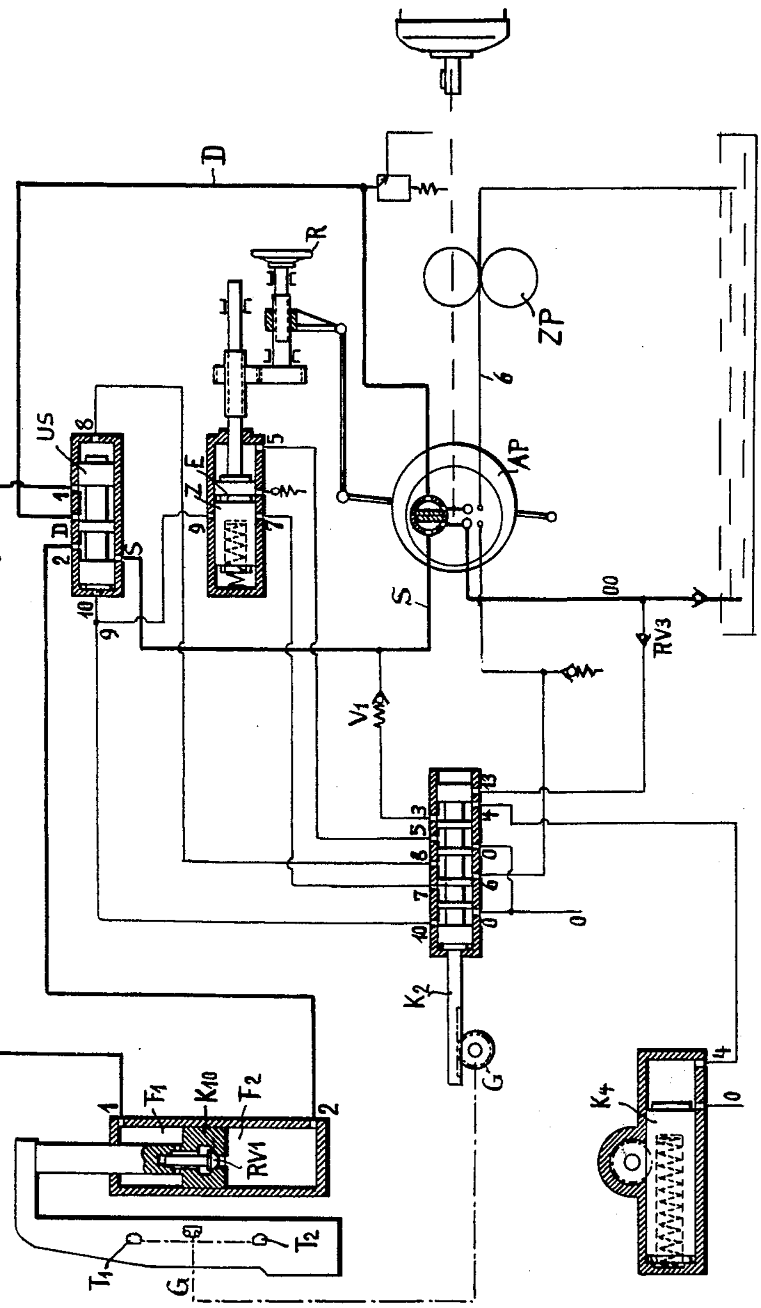


Fig. 6.

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3,154,923

## HYDRAULIC DRIVE FOR PRODUCING LINEAR INTERMITTENT RECIPROCATORY MOTIONS OF A MACHINE TOOL

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This invention relates to hydraulic drives for intermittent motions and is concerned with such a drive having at least one constant reversing point. Such hydraulic drives are used in many applications, more particularly for controlling linear intermittent motions in machine tools or the like. Instead of a working piston carrying out linear movements, however, hydraulic rotary motors carrying out intermittent movements may also be used.

Such hydraulic drives are known. As a rule, they comprise a hydraulic working motor producing the intermittent motions, the pressure medium being supplied alternately to the alternately operative working surfaces of the said motor from a source of working pressure medium with adjustable volumetric output by way of a hydraulic reversing device. As source of working pressure medium, a hydraulic pump with constant working direction and a reversing valve connected in front thereof can be used, or a hydrostatic pump which can be reversed continuously. As a rule, a predistributing slide valve is arranged between the reversing device for the source of working pressure medium and an auxiliary source of pressure medium associated with the said device and usually of constant volumetric output. The slide valve can be adjusted mechanically, electrically or hydraulically by a control impulse generator. The control impulse generator generally comprises corresponding mechanical feeler switches, which are arranged in the path of the machine part being moved by the hydraulic working motor.

In many cases it is desirable to be able to adapt the rate of advance of the hydraulic working motor, preferably variably for both stroke directions, to the particular conditions by varying the volumetric output of the source of working pressure medium. On the other hand, it is necessary in most cases to be able to control the reversing of the working motor at least at one end of the stroke, independently of the particular working speed of the motor, so that the machine part being moved always comes to a standstill exactly at the same predetermined position.

As the releasing of the reversing impulse practically always takes place at the same point of the stroke, this aim can be achieved in the known hydraulic drives only by varying the retarding or acceleration of the working motor during the reversing operation in dependence on the particular normal working speed of the motor. These conditions are shown in FIGURE 1, in which the speed/path curve of the working motor is illustrated in the vicinity of a reversing point. The ordinate of the diagram gives the speed $v$ of the working motor and the abscissa $s$ gives the stroke path of the working motor. $s_0$ is the controlled reversing point, while the releasing of the reversing impulse takes place at $s_1$. In the graph I the working motor has the maximum permissible working speed $v_1$. When the point $s_1$ is reached, the releasing of the reversing operation takes place, so that the working motor is braked with a constant lag according to the line I$a$, until it comes to a standstill at the point $s_0$. From here the reacceleration takes place in the reverse direction. In the graph II it is assumed that the working motor has the substantially lower working speed $v_2$.



In this case also the releasing of the reversing impulse again takes place at the point $s_1$. In order that with this working speed the working motor still comes to a standstill at the point $s_0$, the retarding of the working motor must in this case be considerably less than in the case of the graph I.

If one dispenses with the controlling of an absolutely definite reversing point in the stroke path, then naturally one can make the retarding and accelerating of the working motor constant, regardless of the particular working speed, as is indicated in FIGURE 2 with the graphs III and IV. Here one has to accept the fact that with a varying working speed, the point at which the working motor comes to rest is located at a different position on the stroke path with each working speed, as indicated by the reversing points $s_2$ and $s_3$ in the graphs IV and III of FIGURE 2.

Whereas the known hydraulic drives for intermittent motions operate either according to the diagram of FIGURE 1 or according to the diagram of FIGURE 2, it is the object of the present invention to provide a hydraulic drive of this type, wherein the retarding and accelerating of the working motor is always constant and yet the reversing point is always located at the same position on the stroke path regardless of the particular working speed. These conditions which can be realized by the invention are illustrated diagrammatically in FIGURE 3. The graphs V and VI represent diagrams for the varying working speeds $v_1$ and $v_2$ of the working motor. $s_1$ is the release point for the reversing impulse, while $s_0$ is the controlled, unvarying reversing point. In order to be able to retard or accelerate the working motor independently of the particular working speed, it is necessary to extend the constant movement of the working motor during its normal stroke, at all working speeds less than the maximum working speed according to the graph V, beyond the release point $s_1$ until the working motor has reached the constant retarding line V$a$. For the graph VI this point is shown at VI$a$. FIGURE 3 clearly shows that designing of a hydraulic drive operating according to the diagram shown brings about a considerable mechanical advantage in the vicinity of the reversing phase at all working speeds which are less than the maximum permissible working speed according to the graph V. For the graph VI the mechanical advantage in the reversing phase located between the points $s_1$ and $s_0$ is shown in FIGURE 3 by shading.

The above-mentioned object is achieved according to the invention by a time relay controllable in dependence on the release of impulse, the time constant of which is adjustable in inverse ratio to the volumetric output of the source of working pressure medium, and which relay, when started by the reversing impulse retards the opening of the connection between the auxiliary source of pressure medium and the reversing device, at least during one of the two reversing operations, until the time mechanism has run out. The particular volumetric output of the source of working pressure medium determines the working speed of the working motor. Setting of the time constant in inverse ratio to the volumetric output of the source of working pressure medium therefore means that the time constant equals zero at the maximum working speed (graph V in FIGURE 3), so that directly after release of the reversing impulse at the point $s_1$, the constant retarding of the working motor is instituted, as indicated by the line V$a$. The lower the working speed, the greater is the time constant of the time relay, so that the retarding is always instituted only when the normal working curve intersects the constant retarding curve V$a$ (point VI$a$). The above-mentioned requirements are therefore fulfilled according to the teaching of the present invention.

The retarding times are as a rule very low (less than one second). It has therefore proved particularly advantageous if a time relay with hydraulic time mechanism is used, the supply of pressure medium to the time mechanism being able to be controlled by the predistributing slide valve. The extent of the time constant of the time mechanism can be influenced in two ways: First, a stop for the rest position determining the length of the time stroke of the hydraulic time mechanism may be provided, which can be adjusted in dependence on the volumetric output of the pump for the working pressure medium. Secondly, one may ensure that the time mechanism is fed by an auxiliary source of pressure medium, the volumetric output of which is always proportional to the output of the working pressure medium pump. Both measures may be applied either individually or preferably jointly.

The time mechanism may be associated with only one of the two reversing points. In cases where an exact control is required at both reversing points, it may be advisable to associate a time mechanism with each of the two reversing points, the time constants being able to be different on the working and idle strokes of the working motor according to the variable speed.

The invention is explained in more detail hereinafter with reference to several embodiments illustrated diagrammatically in the accompanying drawings, in which:

FIGURE 4 shows a first embodiment of a hydraulic drive, in which the teaching of the invention is applied.

FIGURE 5 shows another embodiment, in which the working piston acting as a working motor operates with a differential action.

FIGURE 6 finally shows a modified embodiment of a hydraulic drive according to the invention.

The invention is shown here as applied to a working piston carrying out linear movements. However, it is quite evident that it may also be applied to working motors having a reversible rotary movement. In the drawings the same references have been used for identical parts.

The embodiment according to FIGURE 4 shows a hydraulic working motor in the form of a working piston K1 carrying out linear movements, the piston rods of which extend out of the associated cylinder in both directions. In the example illustrated, the piston rods act on a machine table T, on which are fixed two stops T1 and T2, which can cooperate with an impulse generator G, in order to be able to set off the reversing impulse mechanically.

In the example shown the source of working pressure medium is a continuously reversible hydrostatic pump AP. Such pumps are known. The volumetric output of such pumps can be regulated continuously from a maximum value through zero in the opposite direction to the maximum value again. The pump is connected to the working piston by the two main pipes 1 and 2. For the reversing a reversing piston K3 is used in a manner known per se, the piston rods of which extend out of the associated cylinder in both directions. For adjusting the volumetric output in both stroke directions of the working piston, stops A1 and A2 are used, which limit the stroke of the reversing piston K3 in both directions and can be adjusted by setting wheels R1 and R2.

The actuating of the reversing piston K3 is effected by means of an auxiliary source of pressure medium in the form of a gear pump ZP with constant volumetric output. The pressure pipe 6 of this pump leads to a predistributing slide valve K2, which connects the pressure medium from the pipe 6 alternately with the pipes 7 and 8 leading to both sides of the reversing piston. The predistributing slide valve likewise controls the flow of pressure oil from the pipes 12 and 13 of the reversing cylinder. To this end the predistributing valve is connected via the pipes O to the sump of the hydraulic circuit indicated only diagrammatically. The reversing of the predistributing valve K2 from one to the other of its two possible positions is effected by the reversing impulse, which acts from the impulse generator G on the predistributing valve either electrically, hydraulically or, as shown, mechanically.



In the two pressure medium pipes 7 and 8 leading to the reversing piston K3 are arranged valves Z1 and Z2 controlled by a time mechanism and controlling the supply of pressure medium to the reversing piston. In the example shown there is a time relay with a hydraulic time mechanism. The time mechanism has in each case a time piston (Z1 and Z2), which is prestressed by a spring Fd1 or Fd2 in its rest position. This rest position is determined in each case by a stop SP1 or SP2. In the example shown the right-hand time piston Z1 is bearing on its stop SP1 and is hence in its position of rest. Each time piston serves at the same time as a slide valve and for this purpose comprises an annular groove E1 and E2 which, when the end position W of the piston is reached, brings about communication between the supply pipe 7 or 8 and the reversing cylinder, as shown for the piston Z2. The time of travel of the piston from its adjustable rest position to its end position, in which it brings about the communication, constitutes the time constant of the time mechanism.

As the setting of the time constant is in inverse proportion to the setting of the volumetric output of the source of working pressure medium, in order to ensure a positive connection between the two settings, a gear connection is provided in the example shown between the setting mechanism R1 or R2 for the stops A1 or A2 of the reversing piston K3 and the setting mechanism for the stops SP1 or SP2 of the time mechanism.

It is advisable to ensure that the time mechanisms are fed from a source of pressure medium, the volumetric output of which varies proportionally with the volumetric output of the working pump. For this purpose, in the example shown, the time mechanisms are fed by a hydrostatic pump SP with continuously variable volumetric output, the setting mechanism of which is directly connected to the setting mechanism for the working medium pump, as shown in Patent No. 2,928,244, dated March 15, 1960, so that the setting of both pumps always takes place in the same direction and to the same extent. In order to ensure that this control pump SP always delivers to the same pressure medium pipe 3, a floating piston S is arranged between the two outlets of the pump, which together with the associated non-return valves brings about the required manner of operation.

It is quite clear that with maximum output of the working pressure medium pump AP, the associated stops SP1 or SP2 of the time mechanisms may be pushed so far into the cylinder space that the time mechanism pistons Z1 and Z2 are stopped at their end position W. In this position there is not only a free communication between the gear pump ZP and the reversing piston K3, but the pressure medium delivered by the control pump SP can also flow freely through outlet openings exposed by the time piston, so that the control pump is working without pressure.

In the phase of the hydraulic drive illustrated, the working pump AP is operating via the pipe 1 on the left-hand side of the working piston K1. The latter therefore moves at a constant speed towards the right. The time mechanism Z2 has run down and hence maintains the communication with the reversing piston via the pipe 10, so that the reversing piston K3 is held in its right-hand position. The predistributing slide valve assumes the position shown in the figure. As soon as the stop T1 sets off the impulse generator G, the predistributing slide valve K2 is reversed to its other position. Pipe 8 and pipe 12 are then without pressure as they are connected to the outlet in the predistributing slide valve. The pipe 7 is in free communication with the pressure pipe 6, this pressure, however, not yet being able to act on the right-hand side (pipe 9) of the reversing piston K3, as communication between the pipe 7 and the pipe 9 is blocked by the time piston Z1. In spite of the reversing impulse released, the reversing of the working pump does not yet take place, so that the working piston K1 continues to move at constant speed in the existing stroke direction. On the reversal of the predistributing slide valve K2 the pipe 4 for the time mechanism Z1 has also been connected to the pressure pipe 3 of the predistributing pump SP. The time mechanism thus begins to run, as the cylinder space in front of the piston face of the time piston Z1 fills with pressure medium via the pipe 4. After expiry of the time constant, i.e. as soon as the piston Z1 reaches the fixed stop W, the communication between the pipes 7 and 9 is produced by the annular groove E1, while the pressure pipe 4 is connected with the outlet. At this moment the reversing piston K3 begins to reverse the working medium pump, the working piston K1 being braked with constant lag. As soon as the reversing piston has reached its middle position, the working piston K1 has reached the desired reversing point. On a further movement of the reversing piston K3, an increase in the delivery of the working pump and hence a constant acceleration of the working piston K1 takes place as far as the constant working speed determined by the stop A2. The stops A2 and A1 in addition to determining the delivery of the working pump also determine the working capacity of the control pump.

The reversing operation at the other end of the stroke is set off by the stop T2 on the table. As this reversing operation proceeds in the same way as the reversing operation described above in connection with the stop T1, it is not necessary to go into the same here in more detail.

The embodiment according to FIGURE 5 shows certain deviations from the example according to FIGURE 4. First, the working motor is a piston K10 carrying out linear movements, the piston rod of which only extends outwards in one direction. Moreover, the piston carries out a vertical movement. The working surfaces which are operative during both strokes are of different size. In the example shown the actual working stroke is directed downwards, as indicated by the arrow. In order to be able to carry out the return stroke at increased speed, a non-return valve RV1 is provided in the piston K10, which can open in the direction of the smaller piston surface F1. This ensures that when the larger piston surface F2 is energized by the pressure pipe 2, the two sides $F_1$ and $F_2$ of the piston are in communication through the non-return valve RV1, so that only a surface equal to the difference between the two surfaces $F_1$ and $F_2$, i.e. a surface of the cross-sectional dimensions of the plunger of the piston is effective during the return stroke. Such a valve connection may also be provided outside the working piston directly between the two pressure pipes 1 and 2.

It is assumed that an exact reversal is required in this example only at the end of the actual working stroke, so that a time relay Z is associated only with this reversing region. The structure of the time relay is practically the same as in the example according to FIGURE 4, so that detailed explanation is unnecessary. A hydrostatic pump with reversing piston K3 is again provided as source of working pressure medium. Here also the reversing piston K3 is fed by a gear pump ZP. A predistributing slide valve K2 controlled by the impulse generator G again controls the setting off of the time mechanism Z and the communication between the gear pump and the reversing piston K3.

In the present example a separate auxiliary source of pressure medium for the feeding of the time mechanism Z is dispensed with. Instead, the pressure fluid leaving the working cylinder through the pipe 2 during the working stroke is used to feed the time mechanism. As the quantity of pressure fluid accumulating in the pipe 2 is greater than the quantity delivered again by the pump AP to the pressure pipe 1 on the other side of the piston K10, there is an excess of pressure fluid. During the normal working stroke, this pressure fluid is delivered to the cylinder of the feeding piston K4 via the non-return valve V1 and the pipe 3 of the predistributing slide valve K2 as well as the pipe 4. As the working piston K2, at all working speeds lower than the maximum speed, always moves beyond the releasing point $s_1$ (see FIGURES 1 to 3) for the reversing impulse at constant speed, there is still sufficient excess pressure oil available in the pipe 2 even after reversal of the predistributing slide valve K2 from the position illustrated to the other position. This oil, after reversal of the predistributing slide valve K2, is delivered to the time mechanism Z via the non-return valve V1, the pipe 3 and the pipe 5, in order to set off the time mechanism. After the time mechanism has run down, the time relay brings about communication between the gear pump ZP and the reversing piston K3, so that the actual reversing operation is initiated. In order to ensure that on passing through the zero position, the pump can still suck up sufficiently and that there is a steady reacceleration of the working piston, in this position the excess oil is supplied additionally to the feeding piston K4 of the pump AP through the fact that on reversal of the predistributing slide valve K2, the pipe 4 is put into communication with the suction pipe 13.



As in the example shown, the return stroke of the working piston takes place at increased speed, a special measure is provided in order to restrict the acceleration of the working piston up to the increased speed to the same value as that provided for the remaining accelerations and retardations (line V*a* in FIGURE 3). This measure consists in that the pressure pipe leading to the left-hand side of the reversing piston K3 opens into the reversing cylinder at a point 11, which corresponds to the position of the left-hand side of the piston K3 in the zero position of the working pump. Leading to this pipe is a bridging pipe which opens into the end of the cylinder at 12 and which comprises a throttle D*r*, which limits the acceleration or retarding of the working piston K3 in its movement in the left part of the cylinder to the required value.

For the rest, the manner of working of the arrangement according to FIGURE 5 is substantially the same as the arrangement according to FIGURE 4 so that detailed explanation is unnecessary.

The embodiment according to FIGURE 6 differs only slightly from that according to FIGURE 5. In FIGURE 6, for the most part, the same reference numerals are used as in FIGURE 5 when it is a question of the same parts.

In the present example the working pump is a pump AP with variable volumetric output but constant working direction. The output of the pump may again be fixed by a mechanical setting mechanism R, which at the same time serves for setting the time constant of the hydraulic time mechanism Z.

The pressure pipe D of the working pump AP leads to a reversing slide valve US, the freely floating piston of which can connect the two pipes 1 and 2 of the working piston 10 alternately with the pressure pipe D. As the working piston K10 is again, as in the example according to FIGURE 5, designed as a differential piston with intermediate valve RV1, a back-flow into the pipe 1 can be prevented by a non-return valve 1*a*. The cylinder space F1 may empty directly into the other cylinder space F2 during the accelerated return of the working piston K10 only through the intermediate valve RV1. During the working stroke, on the other hand, the suction side of the working pump AP is in communication, in the position of the reversing slide valve illustrated, with the pipe 2 and the cylinder space F2 via the additional suction pipe S.

Reversing of the reversing slide valve US takes place in the same way as that of the reversing piston K3 in FIGURE 5 by means of the pressure oil from the gear pump ZP, in that control is effected by the predistributing slide valve K2. On reversal of the working stroke to the return or idle stroke, the time mechanism is effective through the pipe 7, the annular groove E and the pipe 9. For the rest, the manner of working is the same as in the example according to FIGURE 5.

What is claimed is:

1. A hydraulic drive for producing linear intermittent reciprocatory motions of a machine tool, comprising a reciprocatory hydraulic working motor having opposite working surfaces and said motor being adapted to move the tool, a source of working pressure medium having an adjustable volumetric output, a hydraulically operated reversing device movable between two positions and cooperating with said source to cause flow of said medium from the source to opposite working surfaces of said motor alternately for reciprocating the latter, an auxiliary source of pressure medium of constant volume, means for producing a reversing control impulse at a predetermined position of the machine tool during its movement in each direction, a predistributing slide valve movable between two positions in response to an impulse for directing flow of pressure medium between the auxiliary source and reversing device to cause movement thereof, and a time relay having a time constant rendered operative in response to movement of the slide valve to at least one of the positions for delaying flow of pressure medium between the auxiliary source and reversing device for the duration of the time constant, and means for adjusting the time delay constant of the relay in inverse ratio to the volumetric output of the source of working pressure.

2. The hydraulic drive as claimed in claim 1, in which the time relay includes a normally closed valve interposed in the connection between the predistributing slide valve and the reversing device.

3. A hydraulic drive for producing linear intermittent reciprocatory motions of a machine tool, comprising a reciprocatory hydraulic working motor having opposite working surfaces and said motor being adapted to move the tool, a source of working pressure medium having an adjustable volumetric output, a hydraulic operated reversing device movable between two positions and cooperating with said source to cause flow of working pressure medium from said source to opposite working surfaces of said motor alternately for reciprocating said motor, an auxiliary source of pressure medium of constant volume, means for producing a reversing control impulse at a predetermined position of the machine tool during its movement in each direction, a predistributing slide valve movable between two positions in response to an impulse for directing flow of pressure medium between said auxiliary source and reversing device for causing movement thereof, a time relay having a time constant rendered operative in response to movement of said slide valve to at least one of the positions for delaying flow of pressure medium between the auxiliary source and reversing device for duration of the time constant, said time relay including a hydraulically operated time mechanism defined by a calibrated cylinder, a piston movable therein, spring means for biasing said piston to its rest position for preventing flow of pressure medium between the auxiliary source and reversing device, said piston being hydraulically energized in the opposite direction for permitting flow of the pressure medium between the reversing device and the auxiliary source of pressure medium at the termination of the delay time, and means for adjusting the time delay constant of the relay in inverse ratio to the volumetric output of the source of working pressure.

4. The hydraulic drive as claimed in claim 3, including an adjustable stop for the piston in its rest position, and means for adjusting the stop concurrently with the setting of the volumetric output of the source of working pressure medium for adjusting the time constant of the relay.

5. The hydraulic drive as claimed in claim 3, including a second auxiliary source of pressure medium for supplying the hydraulically operated time mechanism, the volumetric output of said second auxiliary source being adjustable proportionally to the volumetric output of the source of working pressure medium.

6. The hydraulic drive as claimed in claim 5, in which the source of working pressure medium and the second auxiliary source each comprise a hydrostatic reversible pump and the reversing device is a hydraulic reversing piston having adjustable end positions and coupled to both pumps for reversing the flow of pressure medium therefrom.

7. The hydraulic drive as claimed in claim 6, including a second similar hydraulically operated time relay rendered operative in response to movement of the slide valve to the other of its positions, both of said time relays being supplied from said second auxiliary source of pressure medium.

8. The hydraulic drive as claimed in claim 4, in which the motor comprises a cylinder, a piston movable within said cylinder, said piston having two alternately active piston surfaces of different areas, and means for supplying the hydraulically operated time mechanism of the time relay with pressure medium including a connection from the cylinder for the excess quantity of the pressure oil flowing from the larger area side of the piston upon energization of the smaller area piston surface during the working stroke.

9. The hydraulic drive as claimed in claim 8, in which said motor piston is provided with a duct leading through the opposite working surfaces thereof, and a non-return valve for said duct carried by the motor piston and opening only toward the smaller area surface thereof to provide communication between the surfaces during the return stroke of the working piston.

10. The hydraulic device as claimed in claim 8, wherein the reversing device comprises a cylinder having a reciprocating piston, adjustable end stops for the reciprocating piston for limiting its respective end positions, a connecting pipe between the auxiliary source of pressure medium and an intermediate point of the cylinder of the reversing device which is at one side of the reciprocating piston when the latter has completed its movement in one direction, a bridging pipe between the connecting pipe and the one end of the cylinder which is at the one side of the reciprocating piston, and a throttling device associated with the bridging pipe.

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