June 2, 1942. H. F. VICKERS 2,285,069

HYDRAULIC FEED CONTROL SYSTEM

Filed Sept. 29, 1937 2 Sheets-Sheet 1

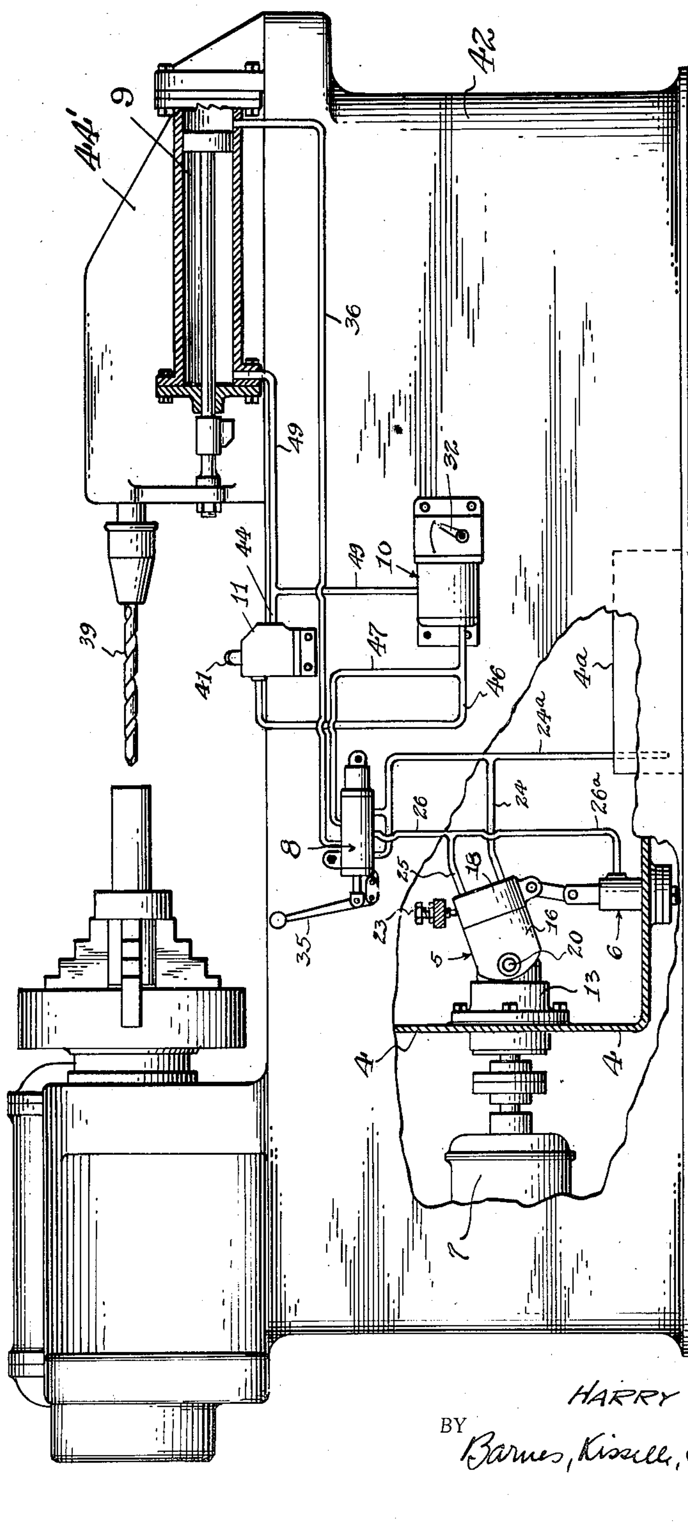

Fig. 1

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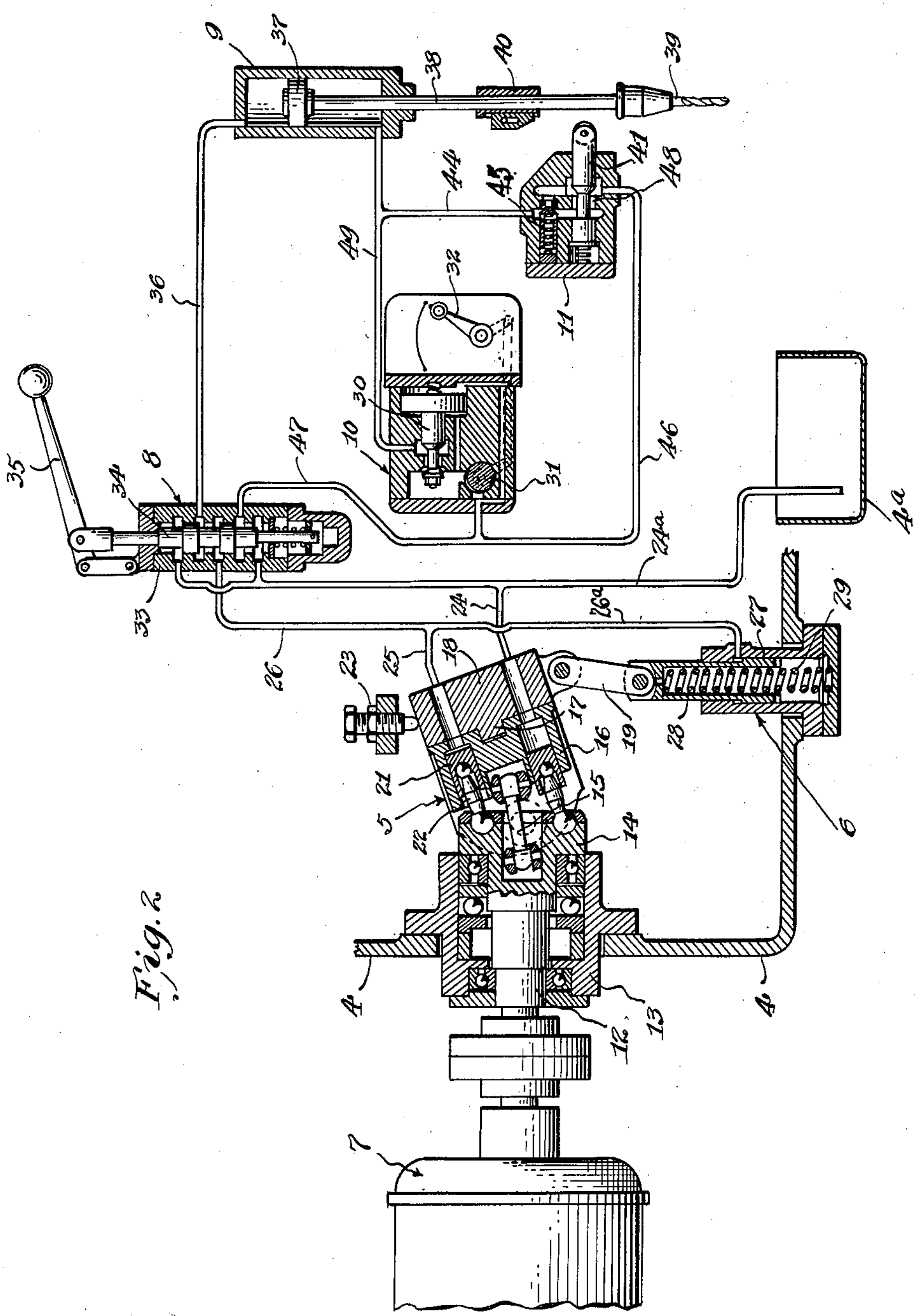

Fig. 2

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HYDRAULIC FEED CONTROL SYSTEM

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2 Claims. (Cl. 60—52)

An object of the present invention is to provide a liquid pressure system utilizing a variable delivery pump, and a pressure compensating device therefor, a liquid pressure motor, a four-way or directional control valve, and an automatic reversing device for said motor. In systems using variable pumps it is a practice to set the pump corresponding to certain load and predetermined speed. Theoretically such a varible pump should maintain a constant speed of the liquid actuated element regardless of a varying load, but practically, there is a change in slippage in said pump which results in a slowing down when an increased load is met or an increase in speed equivalent to the amount of slippage when the load is released. In practice, with systems using a variable pump, fairly good results are often obtained when the work is first set up, but when the oil warms up and thins, and as the tools become dull, there is a noticeable slowing up due to a change in the slippage of the variable pump.

The present invention contemplates a flow control arrangement for a hydraulic system which will provide a constant speed of the hydraulically actuated member regardless of the load conditions and pump slippage. The system, in the main, consists of a combination of a variable delivery pump and a hydraulic speed control member disclosed in my British Patent No. 399,609, dated June 11, 1932, together with an automatic pressure regulator and various control valves.

A further feature of the present invention is that the system will require a constant horsepower consumption. This allows the greatest efficiency in the performance of the power unit.

An illustration of the invention embodied in the hydraulic system having a reversible cylinder motor is set forth in the following description and accompanying drawings.

In the drawings:

Fig. 1 shows the location of the various elements of the system in relation to a table drilling machine.

Fig. 2 is a partially diagrammatic view of the elements of the hydraulic system, said elements being shown in cross-section.

Referring to Fig. 2 the portion of a housing 4 is an enlarged portion of the liquid supply tank shown diagrammatically at 4*a*, in which is mounted a variable delivery pump 5 and a pressure compensating unit 6. A motor 7 is arranged to drive the pump 5 which is hydraulically connected through a four-way valve 8 to a cylinder motor 9. A flow control valve 10 and a control valve 11 are also connected in the hydraulic circuit.

Referring more specifically to the variable delivery pump, the pump part 5 is driven through a drive shaft 12 by the electric motor 7 or any other constant speed power means. The drive shaft 12 runs in ball or roller bearings within a bearing housing 13 and a flange 14, said flange being a part of the drive shaft. The drive shaft carries in a central bore a universal link 15 which drives a cylinder block 16 with the same angular velocity as the drive shaft. The cylinder block 16 bears upon a bearing pin 17 and a swivel yoke 18. The swivel yoke 18 is supported by means of a link 19 which is connected to the pressure compensating unit 6. The cylinder block 16 is pivotally mounted on the bearing housing 13 by pins 20. A plurality of pistons 21 are mounted in axial bores or cylinders of the cylinder block 16, said cylinders have axis of symmetry coincident with axis of the cylinder block. By means of piston rods 22 which are provided with universal spherical end bearings, the pistons 21 are in axial positive connection with the drive shaft flange 14.

The variable delivery pump is of the well known type wherein the volume of delivery of the pump is dependent on the angle between the axis of the drive shaft 12 and the axis of symmetry of the pistons 21. For example, referring to Fig. 2 the pump would be in neutral position when the cylinder block 16 is in horizontal position, while the maximum volume positioned would be as shown with the swivel yoke 18 bearing against an adjustable stop 23. A flexible conduit 24 connects the inlet of the pump with the tank 4*a* and a conduit 24*a*, and a flexible conduit 25 connects the outlet of the pump to a conduit 26 which leads through the four-way valve 8, and conduit 26*a* which leads to the compensating unit 6.

The compensating unit 6 consists of a small cylinder 27, one side of which is in communication with the main pressure line through the conduit 26*a*. A smaller cylinder or plunger 28 is slidably fitted co-axial with the bore of the cylinder 27 and a spring 29 normally urges the cylinder 28 to its extreme position shown in Fig. 2. The flow control valve 10 which consists, in the main, of a balanced orifice valve 30 and an adjustable throttle 31 is adapted to maintain a predetermined differential pressure across the orifice of the throttle 31. A control lever 32 is provided for adjusting the throttle 31. The four-way valve 8 is a standard type of four-way valve, consisting of a housing 33 and a slidable piston 34 operatively connected to an operating lever 35.

The cylinder 9 has one end directly connected to the four-way valve 8 by a conduit 36 and has the other end connected to the valve 8 alternately through the flow control valve 10 and through the control valve 11. A piston 37 is slidably mounted in the cylinder 9 and has a piston rod 38 bearing a drill member 39 and an adjustable cam block 40, said cam block being adapted to contact a slidable plunger 41 in the housing of the valve 11.

In Fig. 1 the hydraulic feed control system is shown on a table drilling machine 42, where it controls cylinder motor 9 and the sliding head 44'.

In the operation: The variable delivery pump is driven through the coupling or universal link 15 by the motor 7. When the system is starting, the pressure in the compensating unit 6 is negligible and the spring 29 will serve to displace the pump housing against the stop 23. Oil under pressure will flow from the pump conduit 25 through conduit 26 to the four way valve 8 where it will be directed to the cylinder 9 through the conduit 36 when the valve piston 34 is in the position shown in Fig. 2. This pressure in the piston end of the cylinder will cause the piston 37 to move downwardly and will thereby force the oil in the rod end of the cylinder through a conduit 44, around a check valve 45, and through the control valve 11 and conduits 46 and 47 to the valve 8 where it will be directed to the tank conduit 24*a*.

This exhaust or tank oil will pass freely through the valve 11 causing rapid traverse of the piston until the cam 40 on the piston rod 38 depresses the plunger 41 thereby causing the closure of a port 48. The exhaust oil from the rod end of the cylinder will then be directed through a conduit 49 to the feed control valve 10 where it will be metered at a constant rate through conduits 47 and 24*a* to the tank. The rate of movement of the piston 37 may be directly controlled by the adjusting of the throttle valve 31.

The return of the piston 37 is accomplished by shifting the handle 35 of the four-way valve 8 to move the valve piston 34 downwardly as shown in Fig. 2. In this position the four-way valve will direct pressure from the pump to the conduits 47 and 46. Oil under pressure will pass around the plunger 41 and will open the check valve 45 to pass through conduit 44 to the rod end of the cylinder 9. The piston 37 will then be returned to starting position at rapid traverse rate. During this traverse the driving piston is locked between two columns of oil under pressure, thus insuring positive feed rates. The oil at the piston end of the cylinder 9 is forced through the four-way valve back to the tank. The rate of travel of the driving piston depends upon the relation of the piston rod area to the area of the cylinder.

It will be understood that during any given feeding stroke of the piston 37 that the flow controlling unit 10 maintains a fixed rate of exit of liquid from the bottom end of cylinder 9. Accordingly, there is also a constant rate of flow of liquid into the upper end of cylinder 9. It follows then that the compensating mechanism 6 will move the yoke 18 into a position where the pump displacement is equal to the rate of flow into the upper end of cylinder 9. The yoke 18 is positively maintained in such a position because if it tended to move toward neutral and thus decrease the rate of delivery into cylinder 9, the pressure would fall, thus permitting spring 29 to overcome the force on piston 27 and move the yoke 18 upwardly. If the yoke 18 tended to move upwardly to a position of greater displacement than that corresponding to the rate of flow out of cylinder 9, the pressure would build up in lines 36 and 26, thus permitting piston 27 to overcome the force of spring 29 and bring the yoke back to its intended position.

The force of the spring 29 and area of piston 27, of course, are so chosen as to maintain in line 36 a pressure greater than that required to overcome the maximum resistance encountered at tool 39 plus the friction of movement of piston 37. Under conditions where the tool resistance is less than maximum, the additional resistance to movement of piston 37 is produced by building up pressure on the liquid in the bottom of cylinder 9 and in pipe 49. It is inherent that the pressure above piston 37 must be equal to the sum of the pressure below the same plus the tool resistance. Thus as the tool resistance decreases, the pressure in line 49 must increase so as to maintain this sum constant.

Since the action of the flow control device 10 is independent of the pressure in line 49, it is obvious that the piston 37 moves at a fixed rate. It is also obvious that since the stroke of the pump is automatically controlled to pump only the quantity of fluid required for driving the motor that no power is wasted in the device except the small quantity which escapes to the suction side of the system by the slippage in the pump itself. Of course, should a variation in slippage occur at the pump 5, the compensating mechanism 6 will automatically increase the stroke setting of yoke 18 slightly and sufficiently to make up for the additional slippage. The rate of flow from pipe 36 into cylinder 9, however, remains constant at any rate of slip in pump 18 because liquid cannot flow into cylinder 9 faster than it can flow out of the same. If the tool 39 meets with any unusual resistance or if the operator fails to reverse the four-way valve when the piston 37 reaches its lowermost position, the pressure in the system will rise and the volume of the pump 5 will decrease. If the pressure reaches a maximum, the pump will deliver no liquid at all thus insuring no rupture of the system.

What I claim is:

1. In combination in a hydraulic system, a hydraulic motor having a piston and cylinder, a variable delivery pump for preloading the inlet side of the motor during the power stroke, said pump having a movable part for varying the delivery capacity, a spring pressed piston arranged to urge the movable part to on-stroke position, means connecting the pressure on the outlet side of the pump between the pump and the motor to said spring pressed piston to act in opposition to the spring whereby the volume output of said pump is proportional to the requirements of the system, and a speed control device including a chamber for receiving the liquid flow from the outlet side of the motor, a variable discharge orifice member, a chamber on the intake side of said orifice member, a valve between said two chambers, and a spring pressed piston operatively connected to said valve for controlling the pressure in said second named chamber to control the pressure differential and the flow across said orifice member, one side of said piston being in direct com-

munication with said second named chamber and the other side of said piston, which is the side upon which the spring acts, being in direct communication with the discharge outlet on the discharge side of the variable orifice member, said speed control device and said means cooperating to effect constant speed movement of said motor piston irrespective of pump slippage and resistance met thereby.

2. In a hydraulic power transmission system the combination of a variable displacement pump, a fluid motor, conduits forming a circuit extending from the pump outlet to the motor and from the motor to the pump inlet, means forming a predetermined restriction in said conduits, valve means responsive solely to the pressure differential across said restriction and connected to maintain such differential substantially constant, and means responsive to slight variations in pump outlet pressure for controlling the pump displacement comprising a spring pressed piston arranged to put said pump on-stroke, and means connecting the pressure on the outlet side of the pump between the pump and the motor to said spring pressed piston to act in opposition to the spring, said means coacting together to maintain the motor speed substantially constant independently of variations in motor load and of variations in pump slippage.

HARRY F. VICKERS.