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[54] **SYSTEMS FOR OPERATING AND CONTROLLING HYDRAULICALLY DRIVEN WINCHES, HOISTS, WINDLASSES AND THE LIKE**

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91/498; 37/135; 254/150 FH

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[57]

ABSTRACT

In cable winding gear driven by a hydraulic motor of the kind having radial pistons driving a surrounding cam ring, a control is provided for applying differential pressure to the motor pistons so that they are disengaged from the cam ring which can then rotate freely. This provision enables the cable to be run out freely, so that, for example, in a crane free fall of the load cab be allowed while avoiding damage to the motor. Pipe-laying equipment including the free-fall provision is described.

2 Claims, 4 Drawing Figures

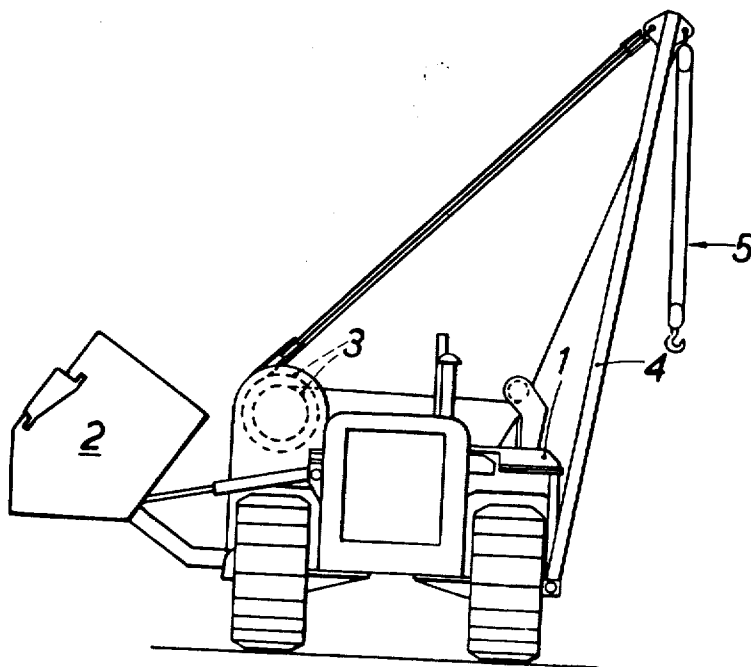


Fig - 1 -

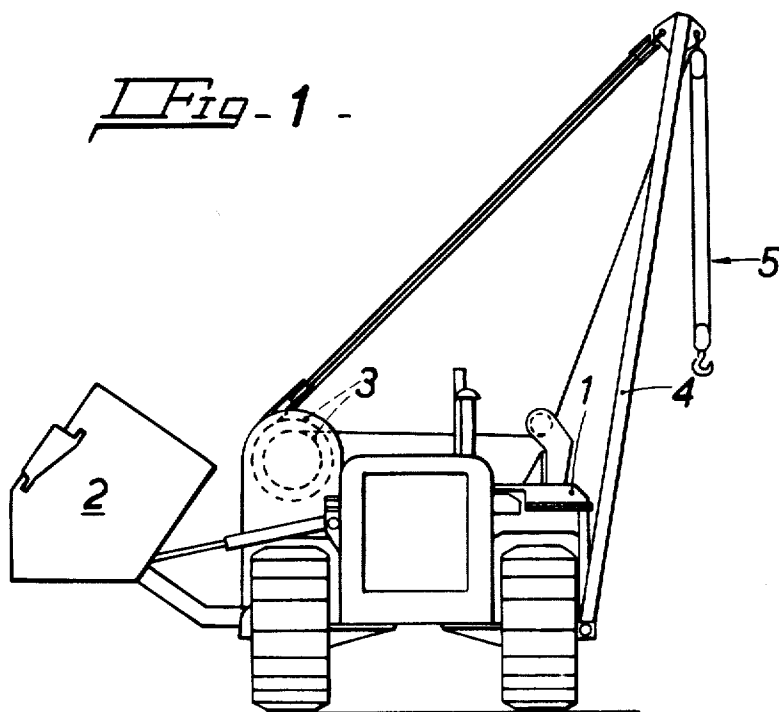
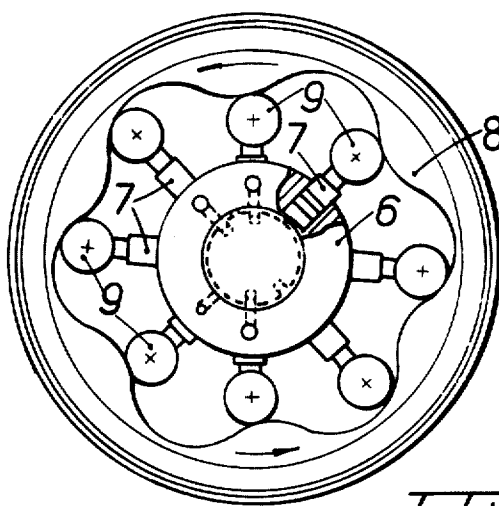
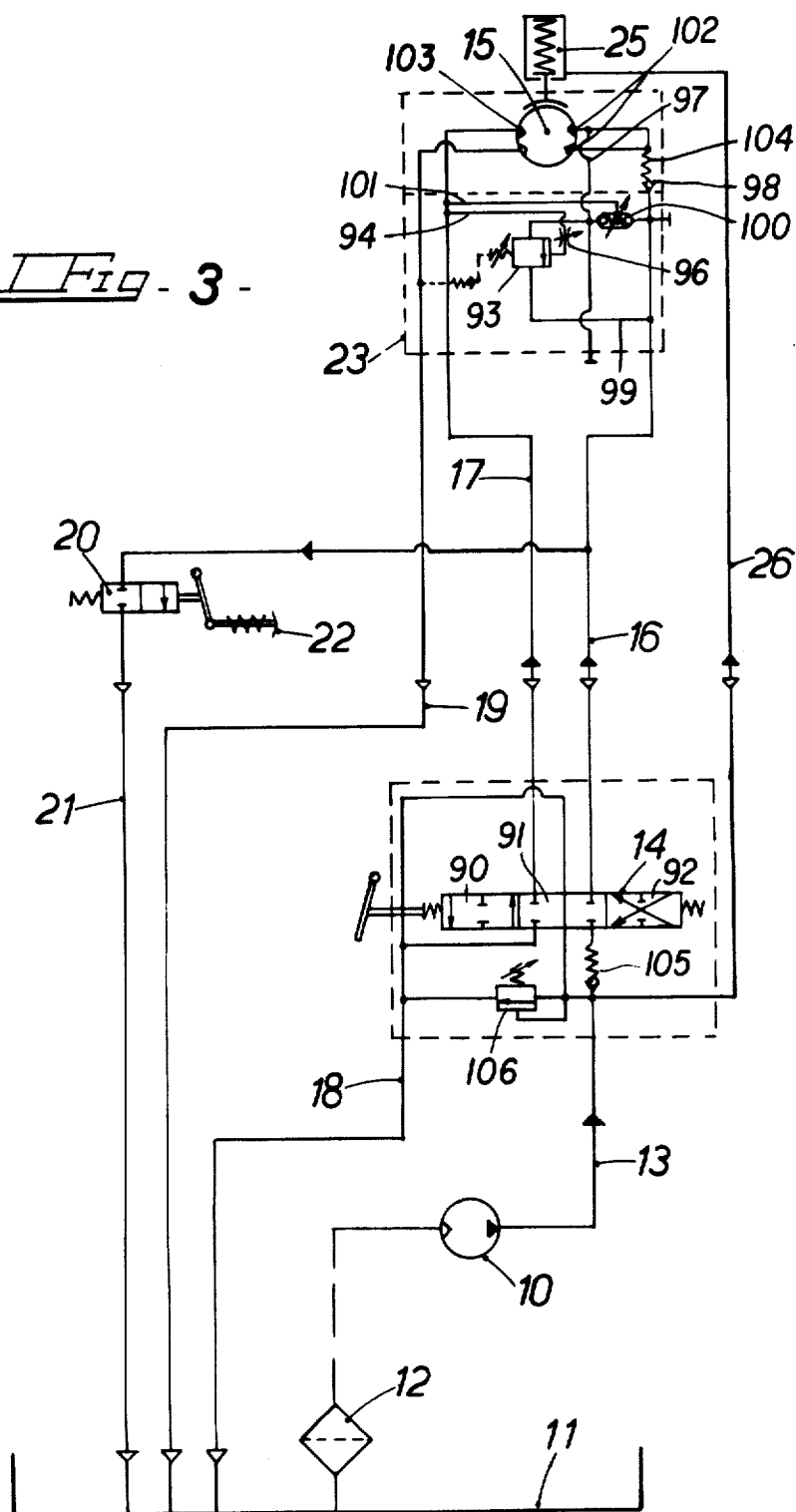
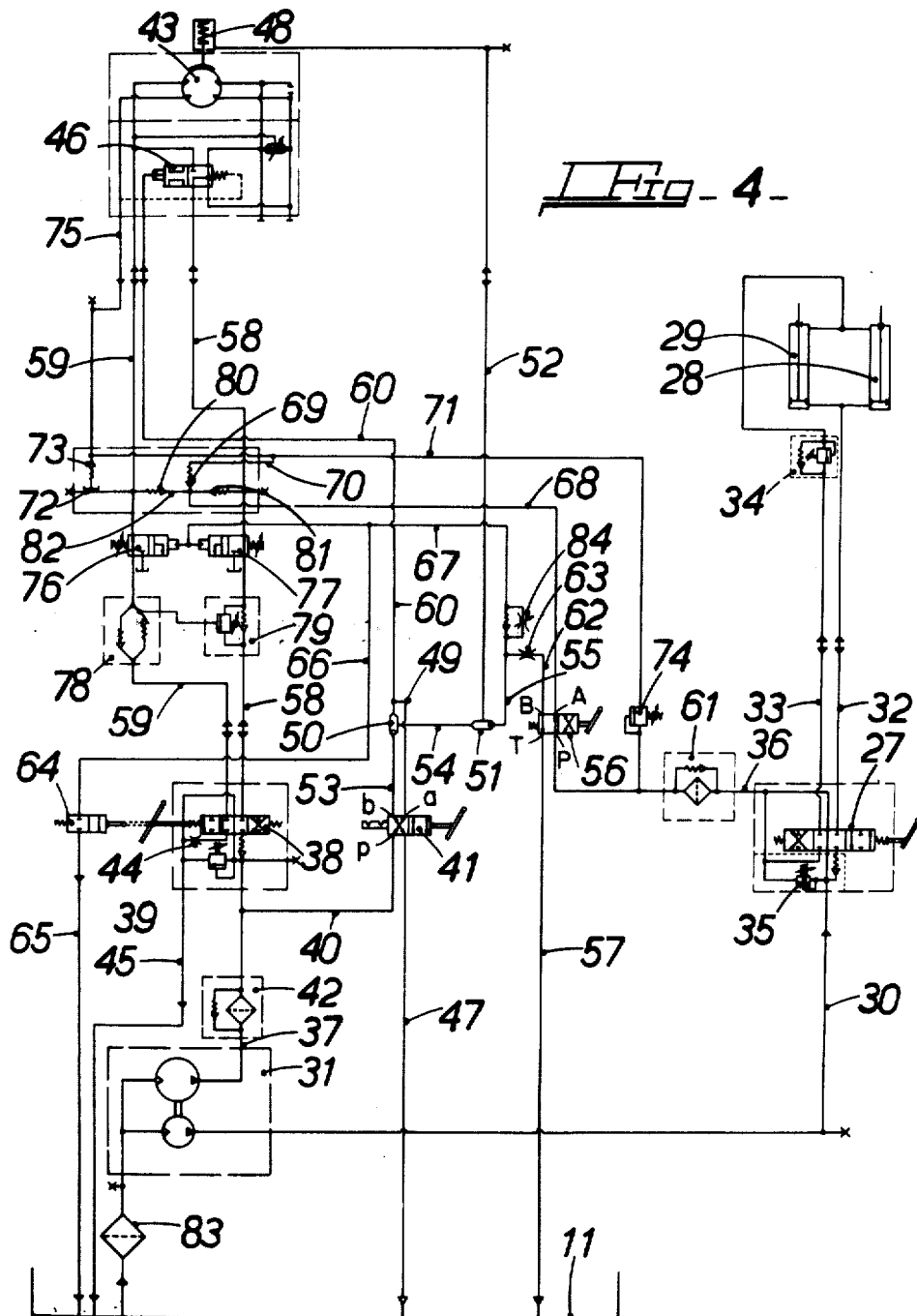


Fig - 2 -

Fig - 3 -





SYSTEMS FOR OPERATING AND CONTROLLING HYDRAULICALLY DRIVEN WINCHES, HOISTS, WINDLASSES AND THE LIKE

This is a division of Ser. No. 84,978, filed Oct. 29, 1970.

This invention is concerned with improvements in or relating to systems for operating and controlling winches, hoists, windlasses or the like which are driven by hydraulic motors.

In particular, this invention is concerned with systems embodying a hydraulic motor of the kind including a stationary cylinder having a number of radial bores in each of which a piston is displaced and a cam ring which is rotatably driven by the pistons. The cam faces of the cam ring are engaged by driving means, such as rollers mounted on the radially outward ends of the pistons, or the outer end faces of the pistons, and hydraulic fluid is selectively supplied to the bores to radially displace the pistons in a predetermined sequence whereby the pressure engagement of the driving means with the cam faces rotates the cam ring from which the drive is obtained in any suitable manner. Such hydraulic motors as above described are herein referred to as of the kind specified.

Hydraulic motors of the kind specified are already well known and they are used for driving winches, hoists, windlasses or the like. Although in such applications hydraulic motors of the kind specified are very satisfactory, it is most desirable to be able to operate the winch or the like in the conventional manner in which the drive can be disengaged so as to allow the winch drum or the like to rotate freely or over-run the drive. For example, it is desirable to provide an operating condition such that the load supported by the winch can fall freely under gravity, or such that the cable or drag line can run out freely.

In systems having a mechanical drive transmission this drive-free condition can easily be achieved by a releasable clutch arrangement. However, with a hydraulic motor of the kind specified direct drive to the winch or the like is utilised. We have found that in a hydraulic system where the source of hydraulic fluid for the motor is derived solely from one or more pumps, and not a pressure accumulator, free-rotation of the cam ring cannot be obtained without risk of damaging the hydraulic motor. The reason for this is that in a hydraulic motor of the kind specified the flow of hydraulic fluid is arranged so that the pistons are urged radially outward whereby the driving means are maintained in engagement with the cam faces. Even if the supply of hydraulic fluid to the bores is interrupted or reduced, then at least some of the pistons will fall radially outwardly of their bores due to gravity so that at least some of the driving means engage with the cam faces. Accordingly, when the cam ring is not being driven by the driving means, but is rotated by the torque applied by the load on the winch cable or the like, the uncontrolled engagement of the driving means with the cam faces of the rotating cam ring causes excessive damage to the motor; in certain instances the cam faces are damaged, or the motor bearing may crack or break up.

It is an object of this invention to provide in a system for operating and controlling a winch hoist or the like driven by a hydraulic motor of the kind specified, an improvement whereby such free fall of the load, or free-running out of the load cable can be achieved

without the aforementioned risk of damage to the hydraulic motor.

Further objects of this invention are to provide an improved hydraulic system for operating and controlling a winch, hoist or the like driven by a hydraulic motor of the kind specified, and to provide a winch, hoist or the like operated and controlled by such improved hydraulic system.

Other objectives will be apparent from the description of preferred embodiment of this invention given later herein.

According to the broadest aspect of this invention, we provide in a system for operating and controlling winding gear, such as a winch, hoist, windlass or the like, driven by a hydraulic motor of the kind specified and to which hydraulic fluid is supplied by pump means, the improvement of valve means for controlling the flow of hydraulic fluid and which valve means on actuation is arranged to produce in the hydraulic motor a hydraulic pressure differential acting on the pistons to displace them radially inwards to an extent such that the driving means are maintained out of contact with the cam faces of the cam ring which can then rotate freely.

In theory there are probably many ways in which such hydraulic pressure differential may be developed by controlling the flow of hydraulic fluid through valve means. However, practical considerations are of paramount importance because of problems that may arise such as, the adaptability of the hydraulic motor, the overheating of the hydraulic fluid, the maximum flow capacity of the pump means, and the overall complexity and cost of providing a special valve arrangement. We believe that we have devised a most practical system which avoids the kinds of problems aforementioned.

One of the preferred ways of obtaining the hydraulic pressure differential in the hydraulic motor is to arrange for the valve means to reduce hydraulic pressure in the cylinder bores and to maintain a low hydraulic pressure in the motor casing so as to develop a pressure differential sufficient to displace the pistons radially inwards. In this manner a relatively low hydraulic pressure in the motor casing can be utilised to displace the pistons, and when the drive is to be restored, hydraulic fluid can be supplied to the cylinder to re-engage the driving means with the cam ring, and then hydraulic fluid for driving the motor may be supplied at the required flow rate.

Although it is envisaged within the scope of this invention to arrange for the valve means merely to control the flow of hydraulic fluid into the motor casing so as to develop therein a hydraulic pressure acting on the radially outer parts of the pistons greater than the hydraulic pressure in the cylinder bores as applied to the radially inner parts of the pistons, this has the disadvantage that the motor casing, including the cam ring will have to be designed to withstand very high bursting loads due to the high hydraulic pressures involved. Additionally, the flow capacity of the pump means and over-heating of the hydraulic fluid may lead to further problems. Usually a hydraulic motor of the kind specified is designed to operate with no or very low hydraulic pressure in the motor casing, and thus where it is desired merely to apply our invention to existing hydraulic motors of the kind specified the aforementioned

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preferred way of developing the hydraulic pressure differential can most conveniently be adopted.

Of course the valve means has to be arranged to act, on actuation, in concert with the other essential operational controls of the winch or the like which must be maintained either inoperative or functional, for instance when the load is released the winch brake must be held off. However, these considerations will be fully understood from an embodiment of the invention which is described later herein.

This invention is also deemed to include a winch, hoist, windlass or the like operated and controlled by a hydraulic system, embodying our improvement as well as the hydraulic system per se.

One application of this invention is in a hydraulic system for operating and controlling pipe-laying apparatus adapted for attachment to, or incorporation in a tractor, preferably of the tracked type.

By way of example an embodiment of this application of this invention will now be described with reference to the accompanying drawings in which:

FIG. 1 is a front-elevation of pipelaying apparatus according to the invention mounted on a tractor;

FIG. 2 is a diagrammatic view of a hydraulic motor of the kind specified;

FIG. 3 is a diagram of the hydraulic operation and control system for the boom of the pipe-laying apparatus of FIG. 1 and;

FIG. 4 is a diagram of the hydraulic operation and control system for counterweights and a hoist of the pipe-laying apparatus.

The principal components of the pipe-laying apparatus which are mounted on the tractor comprise a saddle 1 on one side of which are carried a set of hydraulically controlled counterweights 2 and two winches 3 each adapted to be driven by a hydraulic motor of the kind specified and of which further details will be given later. On the other side of the saddle is mounted the boom 4 which is raised and lowered by one of the winches 3, and the outer end of the boom carries the hoist 5 which is raised and lowered by the other winch 3. The hoist 5 and boom 4 can be operated independently or simultaneously.

The hydraulic motors driving the winches 3 are each of the kind shown in FIG. 2 and including a stationary cylinder block 6 having a number of radial bores in each of which a piston 7 is displaceable, and a cam ring 8 which is rotatably driven by the pistons. The cam faces of the cam ring 8 are engaged by driving means, such as rollers 9 mounted on the radially outward ends of the pistons, or the outer end faces of the pistons, and hydraulic fluid is selectively supplied to the bores to displace the pistons 7 radially in a predetermined sequence whereby the pressure engagement of the driving means with the cam faces rotates the cam ring from which the drive is obtained in any suitable manner.

The hydraulic system depicted in FIGS. 3 and 4 is for operating and controlling the foregoing apparatus and its ancillary parts. The power is derived from the tractor engine independently of drive transmission, and it is arranged to drive the fixed displacement hydraulic pumps which supply hydraulic fluid to the three parts of the hydraulic system for actuating:

- i. the boom 4;
- ii. the counterweights 2; and
- iii. the hoist 5.

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Each of these three parts of the system will now be described with reference to FIGS. 3 and 4 which are in the form of circuit diagrams with all the control valves shown in the neutral position.

i. The Boom System

A fixed displacement hydraulic pump 10 driven from the engine draws hydraulic fluid through a suction filter 12 from an open or low pressure reservoir 11. Hydraulic fluid supply from the output line 13 of the pump 10 is controlled by a three position open-centre valve 14 which is shown in the neutral position. By means of the valve 14 hydraulic fluid may be supplied to a reversible hydraulic motor 15 of the kind specified which is connected directly to the winch drum, and the hydraulic fluid may be returned to the reservoir 11.

For raising the boom 4, the valve 14 is displaced to a second position, which corresponds to moving it to the extreme right in the accompanying drawing. In this second position the motor 15 is rotated in one sense by hydraulic fluid supplied through line 16 to the motor 15 and discharged through the line 17 through the valve 14, to line 18 returning to the reservoir 11. Hydraulic fluid is also drained from the motor casing through line 19 back to the reservoir 11.

Associated with line 16 is a valve 20 which is normally closed, but which on actuation serves to connect the line 16 with line 21 leading to the reservoir 11. The valve 20 is actuated by the engagement of a spring loaded abutment 22 which is arranged to be engaged by the boom if it is raised to an over-centre position, or other predetermined attitude of elevation in relation to the tractor. As will be appreciated, on such actuation of the valve 20, the supply of hydraulic fluid through line 16 to the motor 15 is prevented as the line 16 is connected through valve 20 to the reservoir return line 21, and further elevation of the boom is prevented.

For lowering the boom, the motor 15 is rotated in the opposite sense by changing the direction of flow of the hydraulic fluid supplied to the motor 15. This is achieved by displacing the valve 14 to a third position, corresponding to the extreme left in the accompanying drawing. In this third position, hydraulic fluid is supplied to the motor 15 from the pump output line 13 through line 17 and discharged through line 16, through valves 14 to line 18 which is connected to the reservoir 11.

In order to maintain the boom at a particular attitude, the motor 15 is provided with a counterbalance valve arrangement as depicted in the rectangular outline referenced 23. This counterbalance valve arrangement is adapted only to open when a predetermined high pressure is exceeded so that the motor can operate with full back pressure.

The boom winch is provided with a conventional spring operated band brake that is arranged to be held in the "off" position by a hydraulic brake cylinder 25 which is connected to the valve 14 by a line 26. Hydraulic pressure is maintained in the brake cylinder 25 whilst the motor 15 is driven in either sense. When no hydraulic fluid is supplied to the motor 15, namely when valve 14 is in the neutral position as depicted, the brake line 26 is open to the reservoir return line 18 and the hydraulic cylinder 25 is released so that the brake is applied. Additionally, if there should be a loss of hydraulic fluid with consequent reduction in pressure in

any of the supply lines aforementioned the brake is automatically applied.

The Counterweight System

By way of further explanation of the boom system of FIG. 3, the control valve 14, when in the neutral position shown, blocks lines 16 and 17 from lines 13 and 18 and communicates flow from pump 10 to tank 11 via conduits 13 and 18. In this condition, the brake 25 is in communication with the tank via the conduit 26 which is a branch of conduit 13. Brake 25, which is normally applied by spring force, is allowed to be applied to restrict movement of the boom. Release of the brake is accomplished by pressurization of the brake cylinder to overcome the spring bias when desired.

In order to lower the boom, the selector control valve 14 is shifted to position 92 in which it allows pump flow to be communicated to motor 15 via the conduit 17 and to the counter-balance valve 93 via the conduit 94.

The orifice choke 96 modulates pilot flow to the counter-balance valve 93 and prevents over sensitive operation thereof. When the pilot pressure reaches the range of 300–500 psi, the valve 93 will shift to an open position in which is allowed communication between the motor parts 102 and the tank 11 via conduits the 97, 98, 99, 16 and 18. The counter-balance valve 93 maintains a minimum back pressure range of 300–500 psi in the motor which pressure assures the release of the brake 25 prior to the lowering of the boom and prevents cavitation in the motor 15.

The two-way relief valve 100 protects the motor circuitry by directing flow back to the motor inlet via the conduits 101 and 17 at such times that the load is excessive and back pressure in the motor reaches the 3,000 psi level. This provision also reduces cavitation in the motor by supplying make-up fluid to the inlet ports when the load becomes excessive.

Shifting of the selector control valve 14 to position 90 places the circuit in a boom raise mode. In this condition, flow from pump 10 is communicated through conduits 13 and 16 to motor 15. Conduit 26 also communicates the pressurized pump flow to the brake 25 for the release thereof. The two-way relief valve 100 establishes maximum motor pressure at 3,000 psi and relieves the system by communicating relief flow to the tank 11 via the conduits 101 and 17. Relief valve 106 protects the pump 10 from excessive pressure by communicating conduit 13 with conduit 18 and with tank 11.

Check valves 104 and 105 are provided to block flow in the direction indicated and to maintain fluid in the respective associated conduits and also to block flow to the motor 15 until after the brake spring force is overcome by pressurized fluid and the brake is released.

The lateral position of the counterweights (not shown) relative to the tractor is hydraulically controlled by a three position valve 27 similar to valve 14. Two hydraulic cylinders 28, 29 are arranged to move the counterweights transversely of the tractor centre-line by means of hydraulic fluid supplied through line 30 from the small section of a double hydraulic pump 31 of the fixed displacement type drawing hydraulic fluid through a suction filter 83 from the reservoir 11.

When the valve 27 is in the central neutral position as depicted, the supply of hydraulic fluid is straight through is straight through the open-centre of valve 27 into line 36 connected to the hoist system which is de-

scribed later. If the valve 27 is displaced into either a second or third position, then hydraulic fluid is supplied to one end of the hydraulic cylinders 28, 29 through one of lines 32, 33 and displaced hydraulic fluid flows out of the other end of the hydraulic cylinders 28, 29 through the other one of lines 32, 33. A counterbalance valve arrangement 34 is provided in line 33. A pressure relief valve 35 is associated with valve 27 and is connected to line 36.

The Hoist System

The supply of hydraulic fluid to the hoist system is derived from the large section of the double hydraulic pump 31. Hydraulic fluid in output line 37 is directed through line 39 to a three-position valve 38, through line 40 to a two-position valve 41 with the output line 37 including a by-pass filter 42.

The valve 38 is for controlling a two speed hydraulic motor 43 of the kind specified. Valve 38 is substantially the same as the other three position valves 27 and 14, except that it includes a restrictor 44 which serves to throttle flow to the reservoir 11 through line 45 when the valve 38 is in its second or third position.

The valve 41 is provided for operating the valve 46 of the hydraulic motor 43 that controls, in known manner, the speed of the motor. The valve 41 is connected to a line 47 leading to the reservoir, 11, and in either operative position provides a through passage for hydraulic fluid to the brake cylinder 48 of the spring-operated band type winch brake (not shown) that is held off by the brake cylinder 48.

With reference to the operation of the brake cylinder 48, when the valve 38 is in the neutral position depicted in the drawing there is no hydraulic fluid supplied to the brake cylinder 48 and the brake is in the applied position. The hydraulic fluid in line 39 flows through the open centre of valve 38 to the reservoir 11 via return line 45. The brake cylinder line 52 is also open to the reservoir 11 through shuttle valve 51, line 54 through shuttle valve 50, and the line 49, valve 41 to line 40 and thence to the reservoir 11 through open-centred valve 38.

When the valve 38 is in either its second or third positions, that is when the hydraulic motor 43 is being driven, hydraulic fluid is supplied to the brake cylinder 48 to hold the winch brake in the off position. When the motor speed control valve 41 is in the position depicted, hydraulic fluid flows through line 40 through port p to port a of valve 41. Flow is then through line 49, through shuttle valve 50 into line 54 to actuate shuttle valve 51 and into the brake line 52. The actuation of the shuttle valve 51 prevents the return flow of hydraulic fluid to the reservoir 11 through line 57. When the motor speed control valve 41 is in its other position, the hydraulic fluid flows through line 40, through port p to port b of valve 41, through line 53 to actuate shuttle valve 50 to close line 60. The flow is then to the brake line 52 through line 54 and shuttle valve 51 which is actuated in the same way as just described.

As will be appreciated, when there is a reduction in hydraulic pressure in the brake line 52, the winch brake will be applied. Such reduction in hydraulic pressure will occur when valve 38 is moved to the neutral position, or if a supply line fails, or under other circumstances which will be explained later in connection with

the system for operating the hoist winch with free fall of the load.

The speed of the hydraulic motor 43 is controlled in known manner by the actuation of spool valve 46 to which hydraulic fluid is supplied through line 60 when the control valve 41 is in the position depicted and valve 38 is in its second or third position with hydraulic fluid being supplied through line 40.

The supply of hydraulic fluid for driving the motor 43 is either through line 58, or through line 59 depending on the position of valve 38. The return flow of hydraulic fluid from the motor 43 is through either line 59 or 58. There is also provided an arrangement of valves to maintain a predetermined pressure acting on the motor pistons. This arrangement includes a non-return pressure limiting circuit 78 and a counter-balance valve 79. From this arrangement of valves the hydraulic fluid returns through the valve 38, restrictor 44 to the line 45 leading to the reservoir 11.

The crux of this invention is that the hydraulic motor 43 can be operated so that free fall of the hoist load can be obtained, and controlled without risk of damage to the motor 43. This operation and control will now be explained having regard to the foregoing description and explanation.

The free-fall hydraulic system associated with control valve 56 is arranged so that it can only be effective when the hoist control valve 38 is in the neutral central position, that is when the hydraulic motor is not being driven, such as when a load is suspended on the hoist in mid-air. This is done by providing a drain valve 64 which is mechanically coupled by a suitable linkage to the operating linkage of valve 38. When valve 38 is in either its second or third position, the line 66 leading to valve 64 is open to the reservoir 11 through drain line 65 and thus pressure cannot be developed in line 67 which is in the free fall system. Accordingly, free fall of the load can only be obtained when the hoist valve 38 and drain valve 64 are in the operative positions depicted in the accompanying drawing.

The free fall of the hoist load is controlled by the control valve 56 to which hydraulic fluid flows from the control valve 27 of the counterweight system, through line 36 including by-pass filter 61. The free fall control valve 56 is depicted in the inoperative position whereby the brake line 52 is open to the reservoir 11, flow being through line 55, line 62 including restrictor 63, through port B to port T of the valve 56 to the reservoir return line 57. The hydraulic fluid flows through port P to port A of valve 56, through line 68 to a non-return valve 69 in line 70, to a line 71 and thus to an open reservoir 72 to which flow is controlled by a non-return valve 73 having a predetermined opening pressure. The line 71 also includes a pressure relief valve 74 for relieving the pressure in line 36, but under normal conditions as now being described, this relief valve 74 merely acts as a non-return valve in line 71.

The open reservoir 72 which may be integral with reservoir 11, is also connected to the motor casing by line 75 so that a minimum pressure is maintained in the motor casing by the restricted opening pressure of the non return valve 73. The pressure is selected in accordance with the necessary hydraulic pressure differential which has to be developed before free rotation of the cam ring of the hydraulic motor 43 is allowed.

When the free fall control valve 56 is actuated hydraulic fluid flows through port P to port B into line 62

and through restrictor 63 into line 55, through shuttle valve 51 into brake line 52 so as to actuate the brake cylinder 48. Hydraulic fluid also flows from the line 62 to line 67 connected to a pair of spool valves 76, 77, which are thus actuated. The lower hydraulic pressure which is present in the idle hydraulic motor 43 and which is determined by the valves 78, 79 is vented to an open reservoir through the valves 76, 77 on their actuation, and thus the pressure maintained in the motor casing by the valve 73 is greater than that in the motor cylinder bores and the pistons are displaced radially inwards by the hydraulic pressure in the motor casing so that the driving means as aforesaid is moved out of engagement with the cam ring.

Accordingly, as the spool valves 76, 77 are merely operated by a relatively low pilot pressure, and the brake cylinder 48 has a much higher actuation pressure, the brake is released slightly after the motor pistons have been displaced.

In practice, we have found that if the motor cylinder bore pressure can be reduced to a very low figure, then a motor casing pressure of from 22 p.s.i. to 36 p.s.i. is sufficient in a hydraulic motor of the kind specified developing a maximum torque of 545 ft. lbs. per 100 p.s.i.

To stop the free fall of the load, the free fall control valve 56 is returned into the position as depicted in the drawing. However, now there is no hydraulic pressure in the motor supply and discharge lines 59, 58. A positive pressure, greater than the motor casing pressure has to be developed quickly to render the motor 43 operative again for operating the hoist winch. This development of operative hydraulic pressure in the motor 43 to displace the pistons radially outwards to re-engage the driving means with the cam ring is achieved when the free fall control valve 56 is returned to position depicted in the drawing. When valve 56 is in this position hydraulic fluid from line 36 flowing now through line 68 can overcome the two non-return valves 80, 81 in line 82 connected to motor lines 58, 59. The operating opening pressures of these valves 80, 81 are set lower than the non-return valve 69, thus the minimum operating pressure determined by the arrangement of valves 78 and 79 is quickly restored. Hydraulic fluid also flows to the reservoir 11 from the brake line 52, through shuttle valve 51 through line 55, through the restrictor 63 in line 62, and through valve 56 to line 57. This exhausts the brake cylinder 48 which is released and the brake is applied. The spool valves 76 and 77 are also released again by the pressure drop in line 67, but a throttle non-return valve 84 in line 67 the restrictor 63 provides a restriction which delays the release of the spool valves 76, 77 for a short time to ensure that the brake is applied slightly before the motor pistons are displaced outwards to re-engage the driving means. The hoist winch is then ready for operation under the control of valve 38.

It will be appreciated that the opening and limiting pressures are calculated in relation to a specific application with a particular motor and pump capacity, and without detracting from the scope of this invention, a modified form of hydraulic circuit as just described could be employed to achieve the same result. However, in this embodiment of the invention simple operation and control is achieved, and other advantages are obtained.

It should be noted that the three-position valves 14, 27 and 38 are of the open-centre kind so that when any

of these valves are maintained in the neutral position, such as when the tractor is idling, the hydraulic fluid is returned directly to the reservoir 111 without passing through the associated hydraulic circuits. This arrangement is most advantageous because excessive heating of the hydraulic fluid is prevented and the provision of additional oil coolers can be avoided.

The actuation of the hoist winch brake is synchronised with the readiness of the hydraulic motor, and this provides a fail-safe feature. Additionally, the coupling of the drain valve 164 to the hoist winch motor control valve 138 provides another fail-safe feature. Each of the manually operable control valves are of the "dead-man" type biased to return to the neutral position.

Additionally, if there should be a failure in the hydraulic fluid supply to the hoist winch motor 143 it is still possible to operate the free fall to release, for instance a suspended load. This is because the hydraulic fluid supply to actuate the free fall sub-system is taken from the counterweight system.

With this invented system a luffing crane can be controlled and operated completely by a hydraulic system embodying hydraulic motors of the kind specified. This reduces the overall complexity and weight of hydraulic/mechanical arrangements.

Although the preferred embodiment of this invention as just described is directed to operating and controlling the winches and counterweights of pipe-laying apparatus adapted for attachment to a tractor, it will be appreciated by those familiar with hydraulic engineering that this invention may be embodied in the original equipment of the tractor or embodied in other apparatus such as simple lifting hoists, jib cranes, and winches for trawling or dredging by appropriately modifying the

hydraulic systems.

We claim:

1. In a crane system for operating and controlling a winding gear driven by a hydraulic motor of the kind including a stationary cylinder having a plurality of radial bores, a piston displaceable in each of the bores and each having an associated driving means, a cam ring having cam faces engaged by the driving means, and means for selectively supplying hydraulic fluid to the bores to displace the pistons radially in a predetermined sequence and by pressure engagement of the driving means with the cam faces to rotate the cam ring, the system including operator controlled pump means for supplying hydraulic fluid to the hydraulic motor, the improvement of free-run means for applying to the pistons of the hydraulic motor a hydraulic pressure differential to displace the pistons radially inwards and withdraw the associated driving means from contact with the cam faces of the cam ring whereby the cam ring can rotate freely, the hydraulic pressure for said free-run means being derived from a source separate from the pump means supplying said motor, a hoist including said free-run means, movable counterweight means for balancing the load on the hoist, hydraulic means, including second pump means, for moving the counterweight means, wherein the hydraulic pressure for the free-run means is supplied by the second pump means.

2. A crane according to claim 1 wherein the means for moving the counterweight means includes an operator's second control, and is so adapted that hydraulic pressure for the free-run means is available only when the second control is in an off position.

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