March 8, 1966 JAMES E. WEBB 3,238,730
ADMINISTRATOR OF THE NATIONAL AERONAUTICS
AND SPACE ADMINISTRATION
ANTI-BACKLASH CIRCUIT FOR HYDRAULIC DRIVE SYSTEM
Filed Feb. 5, 1965 4 Sheets-Sheet 1

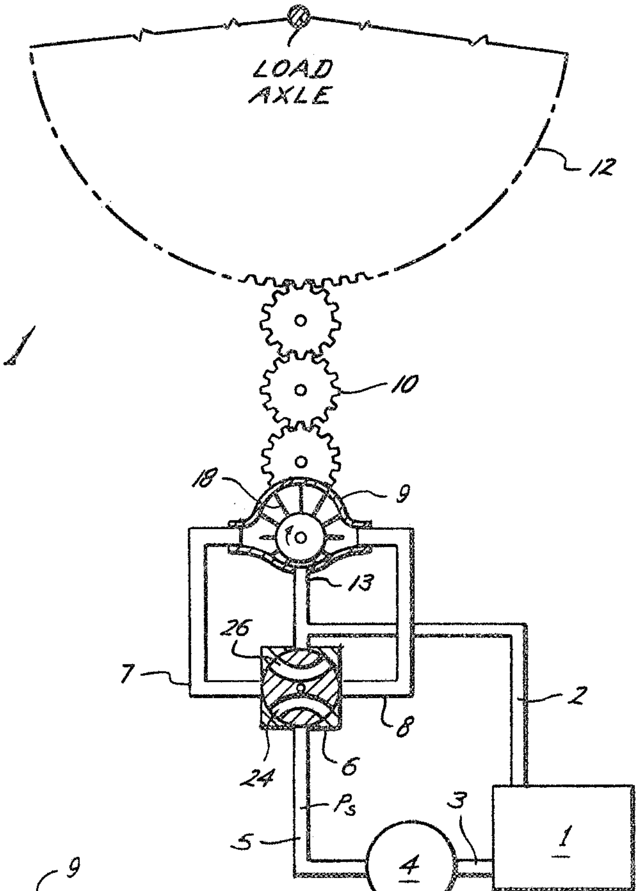

Fig. 1

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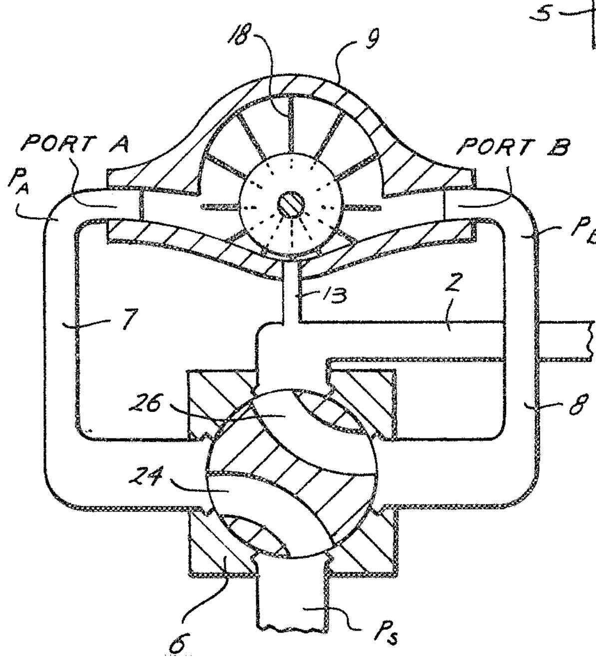

Fig. 2

PRIOR
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Fig. 3

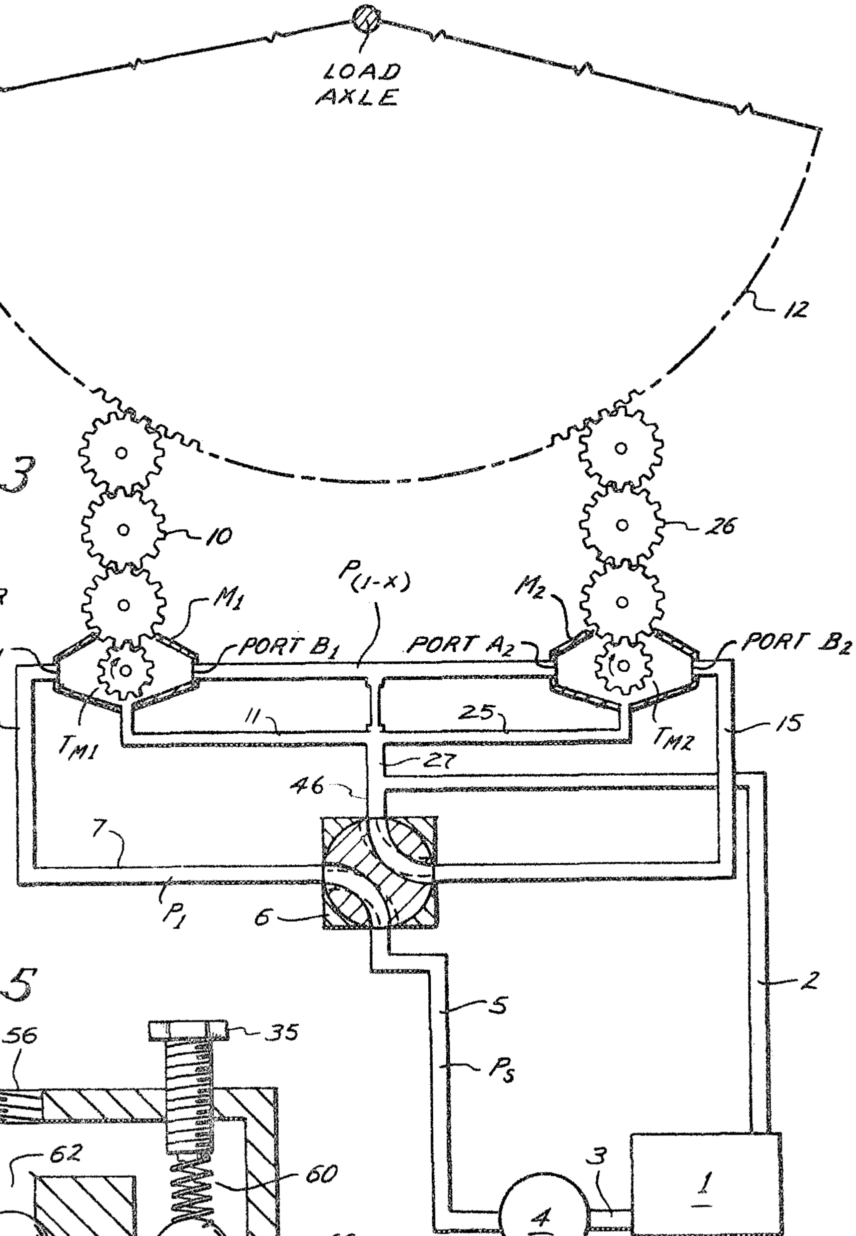

Fig. 5

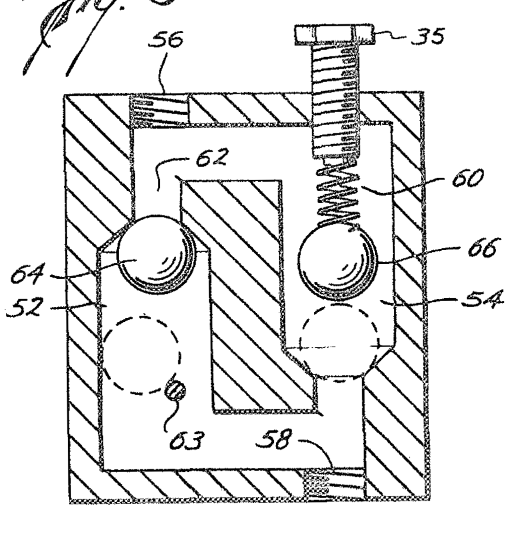

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Fig. 4

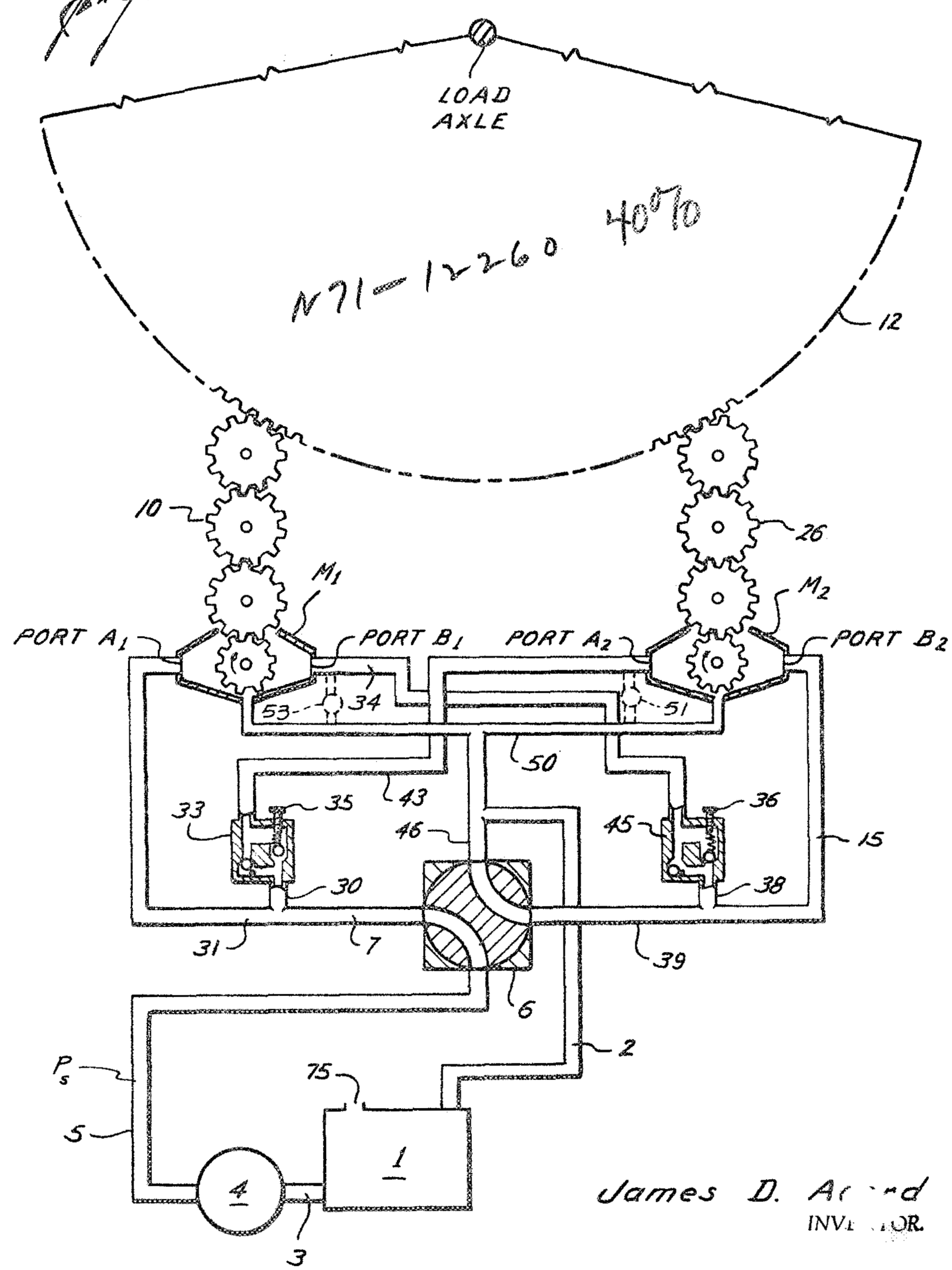

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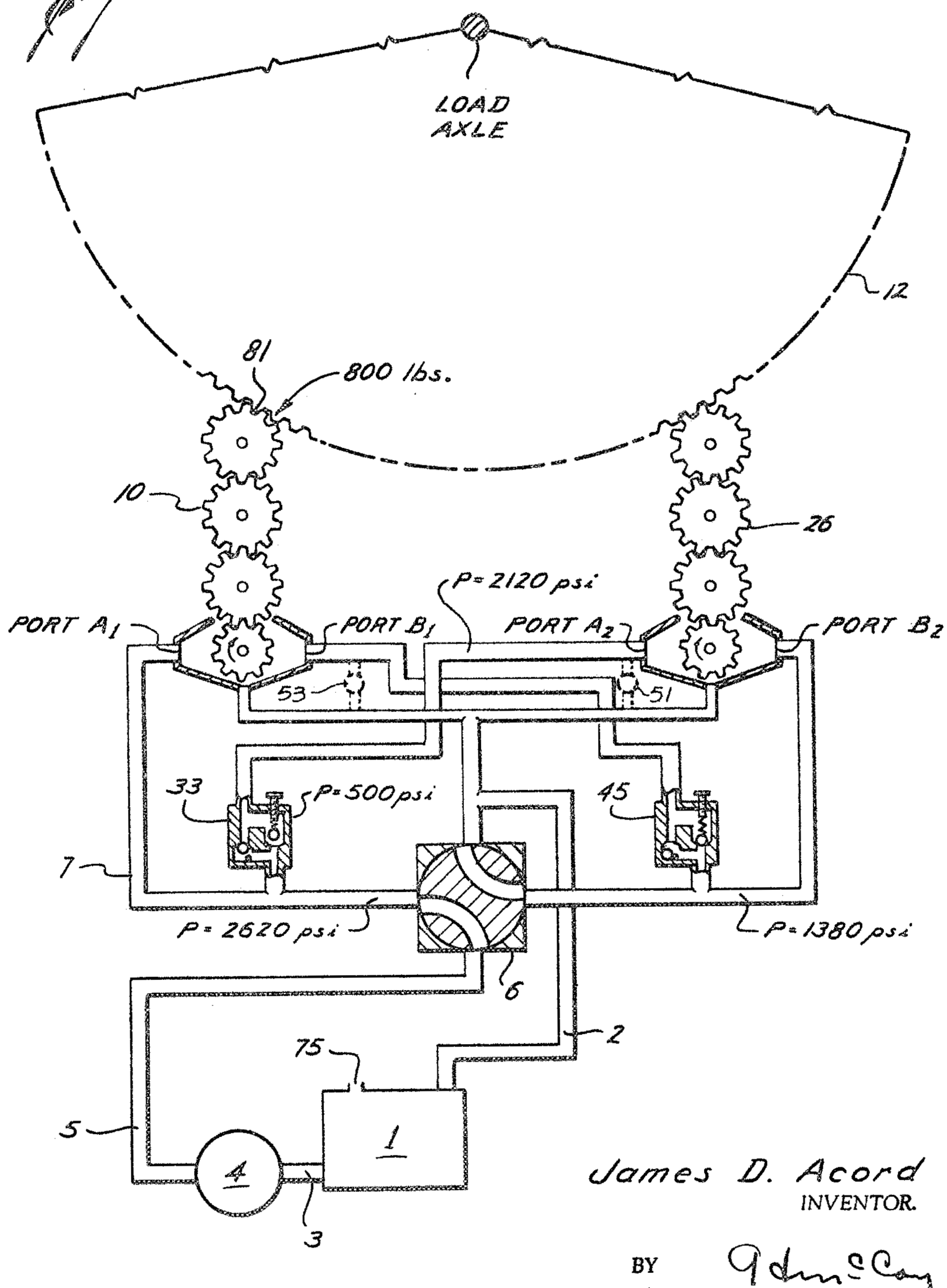
Fig. 6



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3,238,730
ANTI-BACKLASH CIRCUIT FOR HYDRAULIC DRIVE SYSTEM

James E. Webb, Administrator of the National Aeronautics and Space Administration, with respect to an invention of James D. Acord

Filed Feb. 5, 1965, Ser. No. 430,780
10 Claims. (Cl. 60—97)

This invention relates to hydraulic power means for accurately and responsively controlling the movement of large loads such as a deep space tracking antenna. More specifically, it provides for the elimination of backlash in the gear trains of hydraulic motors by the use of an improved hydraulic circuit and alternating valve means therein.

In order to move a body having large weight or mass, it is a common expedient to use a gear train or other similar system which necessarily employs the principle of mechanical advantage. In so doing there is introduced an intermediate power transmitting set of machine elements between the driving body or prime mover and the load being driven. Due to imperfect inter-engagement between these power transmitting elements, the movement of the load is not immediately responsive to the starting of the prime mover. Thus there results a debilitating effect upon load control. This adverse effect is compounded by the well-known problem of gear train backlash which results from the spaced engagement of "play" between inter-engaging teeth of adjacent gears or other drive elements. This may occur when the driven load moves ahead of the drive means, as may occur in a strong wind, when the load is initially moved from a static condition, or when there is a change in the applied power or direction. The spaced engagement of gear teeth may result from several causes such as gear production techniques or normal operational wear. Since the latter is the more predominant cause of gear backlash and non-responsive load movement, certain prior art approaches to these common control problems have embodied means to create a drag on the drive gears and thereby constantly keep them in positive engagement, regardless of their degree of wear. An example of the latter, more fully described herein, is the dual drive, series opposing system in which one motor is back loaded against the other.

In relatively sophisticated machinery, such as large space antennas wherein it is necessary to track the path of probes millions of miles in space, extreme accuracy and control of the antenna is imperative, and thus the problem is even more acute. However, due to the oscillatory type movements of such antennas, it is apparent that the gear and drive systems are subject to severe wear which in turn causes the imperfect fit primarily responsible for the aforementioned backlash. As a result of the backlash, the accuracy of the antenna and the control over it are impaired.

Briefly described, this invention provides for a pair of hydraulic drive motors connected in parallel relation with respect to a common fluid feed line. The two motors are cooperatively connected to a bull or load gear by means of separate power transmitting gear trains. There is additionally provided a hydraulic circuit having valve control and pump means therein so as to selectively vary the pressure and change the direction of the hydraulic fluid flowing therethrough. A pair of adjustable pressure reducing valves are also disposed in the circuit in such a manner as to produce a pressure drop to one motor at a time, thereby causing that motor to have a lower torque output than the primary drive motor. In this manner backlash is not only eliminated from the gear system because of the constant drag applied by the lower torqued motor but, in addition, the magnitude of the drag forces

may be increased or decreased as desired. As a result of the reduced drag forces, there is less stress on machine parts, therefore resulting in longer life and less maintenance. Also, due to the reduced hydraulic pressures in the system, there is more responsive control over the load. For example, in tests of the system disclosed herein, a deep space tracking antenna was moved repeatedly in increments as small as .005 degree, or about 15 seconds of arc in either direction, whereas in prior systems no movement less than .02 to .03 degree could be accomplished.

The foregoing and still other advantages and features of this improved system will become more fully apparent from the following detailed disclosure and drawings wherein like numerals denote like elements in the various views and wherein:

FIG. 1 is a schematic representation in partial cross-section of a conventional prior art hydraulic rotary type drive system having no anti-backlash provision.

FIG. 2 is an enlarged view of the motor and valve portion shown in the hydraulic circuit of FIG. 1.

FIG. 3 shows a schematic representation in partial cross-section of a prior art anti-backlash circuit which incorporates the commonly used series-opposing motor arrangement. Such a system embodies the same general type of servo valve shown in FIG. 2.

FIG. 4 is a schematic representation in partial cross-section of the hydraulic drive circuit embodying the improved anti-backlash provision which is the subject of this invention.

FIG. 5 shows an anlarged view of the simplified counter-balance type valve used in the circuit of FIG. 4.

FIG. 6 shows a schematic representation in partial cross-section of the drive circuit of the invention when under operating conditions.

Reference is made to the conventional hydraulic drive system of FIG. 1. The circuit comprises a fluid reservoir 1 wherein there is stored the hydraulic fluid at atmospheric pressure for operating the system. Since the circuit is a closed loop, the reservoir contains a return line 2 and an out line 3. The out line is connected to pump 4 which may be of conventional variety. Supply line 5 feeds hydraulic fluid from the pump into a four-way servo valve 6 which serves to directionally operate the drive motion of the system. There is provided a pair of fluid lines 7, 8 connected to opposing sides of the servo valve and to a hydraulically driven motor 9.

Mechanically connected and engaged with the hydraulic motor in a well-known manner is the gear train 10 and load carrying bull gear 12. Schematically indicated at the bottom of the motor is a case drain line 13 which communicates with the return line 2 from the servo valve. Case drain 13 serves to prevent internal motor leakage from building up excessive pressure within the case. Lacking such provision, a buildup of pressure could cause external leakage or rupture of the motor casing. There is usually provided a valve of conventional design (not shown) for regulating the leakage flow through this line.

By use of conventional external control, pump 4 creates a system pressure P_s in supply line 5. Likewise, through external controls, servo valve 6 is caused to rotate in the desired direction and thus direct the fluid into the motor. The degree of fluid pressure into motor 9 is dependent on and proportional to the opening resulting from rotation of the valve. If the valve 6 were rotated from its closed position to 22 degrees beyond the horizontal, as shown in FIG. 2, it is apparent that the vanes 18 of the motor would rotate slower and hence result in less torque on the gear train than if the valve were thrown full open—that is, if the valve were rotated sufficiently to present the full cross-sectional area of fluid line 7 to the valve passageway 24. Thus control over the load is affected by appropriate movement of the servo valve. The quality of the control, however, is adversely affected by the presence of backlash in the gear train. As may be visualized by examination of FIG. 1, a backlash situation in gear train 10 may be created upon initial movement of servo valve 6 from the horizontal position because the gears may be in disengagement such as when starting from static condition, or when changing direction. One prior art solution to this problem in relatively small gear trains resulted in splitting the gears at each mesh, and spring loading each split gear against the other. In larger gear trains and heavier loads, such an arrangement is not feasible due to the magnitude of forces involved.

A prior art arrangement designed for the elimination of backlash in the drive system for large loads is referred to as the "dual motor series opposing system." Although the dual motor series opposing drive system presents a limited remedy to the problem of backlash, it has been found to introduce factors which not only detract from optimal control of the load but which will, in some instances to be explained hereinafter, preclude effective control over it. In the series system shown in FIG. 3, servo valve 6, hydraulic motors M_1, M_2, and gear trains 10 and 26 operate in the same manner as the valve, motor, and train described previously.

The dual drive-series opposing system is arranged so that one gear train acts as a drag on the bull gear, while the other gear train is driving the bull gear. When the load direction is changed, these two gear trains exchange functions, that is, the prior drag train now drives, and the prior drive train becomes a drag. In this manner the gear teeth in the driving gear train are always kept in positive engagement and backlash is substantially reduced. An examination of this circuit arrangement shows that with the servo valve 6 in a closed position (as in FIG. 1), the system is in a static condition and a pressure P_s exists in the supply line 5 from the pump. Clockwise movement of the servo valve to a position shown by the dotted lines allows fluid to flow, thus producing an initial pressure P_1 in the fluid line 7. Neglecting frictional losses occurring from fluid flow, the same pressure P_1 is seen by port A_1 going into motor M_1. The pressure P_1 is exerted on the vanes (not shown) of motor M_1, causing it to rotate clockwise and thus causing it to have a clockwise torque TM_1. The torque TM_1 is transmitted to and through gear train 10 to bull gear 12 (or load) which then moves in a clockwise direction. As the bull gear rotates, a torque is applied to and through the train 26 which in turn is applied to motor M_2, causing it to rotate. This causes M_2 to act as a pump with respect to the fluid flowing therethrough since no "effective" pressure exists at its port A_2. Thus, as the bull gear rotates, driving motor M_2, the fluid which exits port B_2 is pumped into conduit 15 and through the servo valve into conduit 46 and then into the return line 2 which leads back to the reservoir. It is seen that if the servo valve is turned to supply fluid into line 15 instead of line 7, that the load will reverse its direction. Otherwise the system will operate as described above.

As with the hydraulic motor described in FIG. 1, there are provided case drain conduits 11, 25 for each of the motors M_1, M_2. These conduits, and connecting conduit 27, provide a path for leakage flow from either motor to reduce internal motor pressure. The connecting conduit transfers the fluid back to the reservoir via return line 2.

With the above circuit arrangement it is seen that the teeth of inter-engaging gears in gear train 10 are kept in positive contact because the bull gear cannot "coast ahead" of the gear train 10 due to the drag exerted on it by the train 26. It is plausible, of course, that some external force acting on the load, such as wind, etc., could exceed the drag forces created by gear train 26 and thus produce a backlash in the drive system of motor M_1. Although the constant drag effect in this system builds an antibacklash provision into its operation, there is created, as a result, a series of operational problems and deficiencies.

One of these problems is the constant high magnitude of stress present at contact points of interengaging gear teeth, the source of which is the added load (or drag forces) which the drive train must move. Another problem is the absence of means for effectively reducing the resistance or drag forces when they are not needed. It is found that the magnitude of these forces causes gear teeth to wear quickly and sometimes even to fracture the gear boxes. Still another problem caused by this antibacklash arrangement is reduction in load control caused by the high system pressures necessary for movement.

The subject invention discloses an improved hydraulic circuit which not only reduces the drive forces required, but which also provides a means for creating an effective adjustable drag-producing antibacklash provision during operation. As a result of this improved antibacklash provision, control over the load is significantly improved and wear on the drive machinery greatly reduced. In this regard reference is made to the improved system of the invention, as shown in FIG. 4, wherein a pair of fluid motors M_1 and M_2 are placed in parallel feed relation with respect to the source of fluid flow of the system by providing a fork in line 7 having two conduits 30, 31 leading to the motors M_2 and M_1 respectively. The pressure in these conduits would thus be the same. In order to create a pressure drop, however, into one of the motors and thus create less torque by that motor, counterbalance valves 33, 45 (i.e. pressure reducing valves) are placed in the lines in such a manner that the lower pressure would be impressed across only one motor at a time, this depending upon which way the valve 6 is turned. Thus if the fluid is moving clockwise in the circuit, a pressure drop would be created in the direction of motor M_2, i.e., across motor M_2, while if it were moving counterclockwise, the pressure drop would exist across motor M_1. Additionally, the pressure reducing valves 33, 45 may be adjusted to vary the extent of the pressure drop, the purpose being to vary the torque on one of the motors. In such an arrangement the drag of the lower torqued motor and its associated gear train may be adjusted to accommodate the conditions under which the machine is operating. This is accomplished by the use of an adjustable setting 35, 36 on each of the valves. Also, the valves are so constructed that fluid may flow therethrough in either direction. Examination of FIG. 5 shows that fluid entering through port 56 may flow freely through the valves with no significant drop in pressure. Should the fluid flow through the valve in the opposite direction however, there is a pressure drop. A detailed explanation of the valve mechanism is set forth hereinafter.

The conduit 31 of FIG. 4 is connected to the port A_1 of motor M_1 so as to constitute either a feed or return line for the motor, depending upon which direction the fluid is moving. To the port B_1 is connected conduit 34 which leads to a counter-balance valve 45. The counterbalance valve in turn is connected by conduits 38, 39 to the servo valve mechanism. The conduits for the motor M_1 is thus of closed loop design with fluid supplied from and returned to the servo valve.

A second closed loop conduit is provided for the motor M_2 and consists of line 39 having a fork therein forming the conduit 15 and the aforementioned conduit 38. Conduit 15 is connected to the port B_2 of motor M_2 and acts as a supply or return line for such motor, depending upon which direction the system is moving. Connected to the port A_2 of the motor is conduit 43 which leads in to the counterbalance valve 33. The valve 33 may work as a pressure reducing valve or as a mere fluid conduit in the same manner as its cooperating valve 45. As previously explained, only one valve acts as a pressure reducer at a given time and this depends upon the direction of system movement. The closed loop conduits for both motors M_1 and M_2 may hereinafter be referred to as a first circuit.

In addition to the ports of the servo valve which receive the conduits 7 and 39 of the first circuit, there is provided a port each for the supply line 5 and return line 2, both of which constitute a portion of a second circuit. These lines conduct fluid to the first circuit and remove it therefrom, respectively, by operation of the conventional servo valve 6. The return line 2 carries hydraulic fluid from the first circuit by means of its connection to an exit conduit 46 leading from the servo valve. The fluid reservoir 1, to which the other end of return line 2 is connected, is at atmospheric pressure by reason of a breather tube 75 therein. Also connected to reservoir 1 is the out line 3 which supplies fluid to the pump 4. This fluid is then delivered from the pump under pressure P_s to the servo valve where it enters the first circuit either through the conduits 7 or 39, depending upon which way the servo valve is turned. It is apparent, of course, that fluid flow is unidirectional in the second circuit, while in the first circuit flow may be either clockwise or counterclockwise.

As noted above, a counterbalance valve is disposed in each loop of the first circuit. These valves 33, 45 are, as shown in FIG. 5, essentially a check valve in parallel relation with a pressure reducing valve, the purpose of the arrangement being to create a pressure drop in one direction through the counterbalance device while effecting flow stoppage in the opposite direction. More specifically, it is seen that the valve consists of a ball-check type arrangement with one ball 66 spring loaded to a closed (dotted) position. The spring constant in the spring 60 may be varied by adjusting the set screw 35 so as to accomplish the desired drop in pressure.

The free flow side 52 comprises a similar ball-check type arrangement in which the ball 64 rests against a shoulder pin 63 when in the free flow position (dotted lines). Thus, when the valve is causing a pressure drop across its ports 58, 56 the flow of fluid into port 58 causes port 52 to be blocked by the ball 64 while the cooperating ball 66 above port 58 is urged upward when fluid pressure exceeds the opposing compressive force of spring 60. The movement of ball 66 is, of course, dependent on the position of the adjustable setting 35 which thus causes the desired pressure drop ΔP.

When fluid is flowing in the opposite direction, that is in through port 56, the ball 66 on the pressure reducing side is forced downward by the spring 60, thus closing off fluid flow to port 58. Ball 64, however, is moved by fluid pressure on to shoulder pin 63, thereby presenting an unobstructed or "free" fluid flow path at the side 52 from port 56 to port 58.

For the purpose of describing the operation of the system shown in FIG. 4 it will be assumed that a system pressure of 4000 p.s.i., exists in the supply line 5 from the pump. Also, it is assumed that the setting P on the counterbalance valves is 500 p.s.i. and that load 12 is a large space probe tracking antenna which is to move initially in a clockwise direction. If the system is to start from a static condition, the servo valve is opened clockwise to a full open position since maximum torque will be required to accelerate the load. Disregarding frictional losses, the pressure a port A_1 and in line 30 will be 4000 p.s.i. The pressure in line 43 and hence at port A_2 of motor M_2 is 3500 p.s.i. In the prior art devices, motor M_1 was required to move the load by itself since motor M_2 was contributing substantially no effective torque, but was instead acting as a drag to prevent backlash. In the subject device, valve 33 is adjusted so that M_2 is acting to lessen or increase the drag of the gear train 26, depending on the needs at the time. If a small drag is required, the valve 33 may be opened wide at setting 35, thereby causing high fluid pressure and applied torque by motor M_2. Oppositely, M_2 could have more drag if the valve opening were reduced by reducing its opening at 35. The adjustable torque feature makes possible the use of smaller motors both at M_1 and M_2 since less torque is required by each motor. Since there is less drag on the main drive motor M_1, the wear and maintenance is substantially reduced. In following the fluid flow through the above system it is seen that fluid leaving port B_1 flows under substantially reduced pressure through the free flow side 52 (FIG. 5) of counterbalance valve 45 and then into line 38. The fluid leaving port B_2 is likewise under substantially reduced pressure as it flows through line 15 where it meets fluid from line 38 and returns to the servo valve via line 39. The fluid then traverses the servo valve and returns to the reservoir 1 through return line 2 of the second circuit. From the reservoir the fluid is recycled into the system by a hydraulic pump 4.

It is clear that should the servo valve be rotated in a counterclockwise direction there will be a pressure drop across valve 45 which causes motor M_1 to take up the anti-backlash function, thus making motor M_2 the primary driver. The flow through the valves, motors, and line circuit may be traced back to the reservoir in a counterclockwise manner which is functionally identical to the clockwise fluid path described above.

When the system is at rest, there wil normally be sufficient leakage flow through motor ports B_1 and A_2 and back through case drain conduit 50 to maintain the pressure differentials across counterbalance valves 33 and 45 and thus maintain opposing torques from motors M_1 and M_2. If this leakage flow is insufficient, small adjustable orifice valves 51 and 53 may be installed as shown to provide the required mount.

Also, as noted above, movement of the servo valve (clockwise) to a position less than full open will cause the pressure in line 7 to increase and the pressure in line 39 to decrease. By controlled movement of the servo valve and controlled counterbalance valve settings it becomes apparent that relatively fine control may be exercised over motor port pressures which in turn provide a more precise control over the system. Once the system is operating under normal loads, that is, no acceleration and no external forces such as winds acting on the main load, the operational torques are significantly reduced below those required for acceleration. These conditions may be exemplified by the pressures shown in FIG. 6, wherein, it is seen that due to the position of the servo valve a pressure in the line and therefore at port A_1 of only 2620 p.s.i. exists. In that the servo valve is not fully open to the return line, port B_1 is at a pressure substantially greater than atmospheric, namely at 1380 p.s.i. The 500 p.s.i. setting on the counterbalance valve 33 causes a pressure at port A_2 of 2120 p.s.i. while the exit pressure (port B_2) is 1380 p.s.i., the same as in the other return lines of the system. The counterbalance valve 45 is operating in free flow position and the main return line 22 is at atmospheric pressure due to breather opening 75 in the reservoir. It is found that the bull gear teeth 81 in a system under the above conditions bear a tangential load of about 800 lbs. due to the low system loading. This compares with loads of between 10 and 15 times as great in prior art systems when under the same operating conditions. It should be noted that although fluid motor M_2 has a net positive differential pressure across it (clockwise drive), the resulting torque is insufficient to overcome both its internal friction and that of gear train 26, and it still places a small net drag on the drive motor M_1 and train 10, thus maintaining a desirable backlash free condition. More importantly, however, is the ability to regulate by means of the counterbalance valves the magnitude of drag which this motor exerts. Thus if the motor M_1 is the prime driver, the load is moving clockwise and M_2 is exerting an anti-backlash effect. If a prevailing wind exerts a counterclockwise force against the load movement, there could result a backlash in the system since the wind force in combination with drag force of M_2 may overcome the effective torque of the prime driver M_1. However, with the ability to vary the drag torque (by use of the settings ΔP), there results a change in the "effective" drive torque. By increasing the pressure drop across the counterbalance valve associated with the "drag train" the probability of setting up a backlash condition is substantially reduced. In the opposite sense it is seen that if a prevailing wind exists in the same direction as the load is moving, then it may be necessary to decrease the pressure drop into the "drag motor" since less "drag" will be required to maintain a backlash free condition. This may, of course, require a simultaneous increase in pressure to motor M_1. It will be recognized by those skilled in the art that these counterbalance valves may be interconnected with each other and with external sensing means which control them, in accordance with the conditions present.

It is thus seen that the subject invention provides improved operating characteristics regardless of whether it is under maximum torque or normal torque requirements. During conditions of low torque operation, such as that which is encountered during the tracking of deep space probes, the gear teeth and structure loads are subjected to less than 1000 lbs., or about 5% of the maximum which the drive system can repeatedly endure. In the prior art devices the gear teeth and support structure were subjected to loads approaching 75% of maximum. As a result of the lower forces it is apparent, as initially pointed out, that wear on moving parts is greatly reduced. Likewise, it is apparent that due to lower friction and loading forces the accuracy and sensitivity of control of the antenna is improved and it can be moved with much less signal to the servo valve than was possible before.

Having thus described the nature and operation of this invention, it will be understood that said invention permits of various modifications, alterations, and substitutions, particularly with respect to individual components and their design, and also it is to be recognized that the drive system disclosed herein is applicable to and may be embodied in apparatus other than the exemplary antenna, all without departing from the essence of the invention and within the scope of the claims following hereafter.

Therefore what is claimed is:

1. A reversible drive system substantially free of backlash for accurately controlling movement of a load driven thereby comprising:
  - dual hydraulically operated drive motors operatively engaged with the load;
  - means for selectively varying the torque of one of said motors with respect to the other so that the lower torqued motor acts as a variable drag on the driven load, said means including a hydraulic circuit having a primary and secondary control means therein, said primary control means operatively disposed in the circuit for selectively controlling the direction and quantity of fluid flow, and said secondary control means includes an adjustable fluid pressure reduction means disposed in the supply line of the lower torqued motor.

2. The reversible drive system of claim 1 wherein the supply line of either of said motors serves as a return line for its respective motor when the direction of fluid flow is reversed by selective control of said primary control means.

3. The drive system as recited in claim 2 wherein the supply line of the lower torqued motor further includes means for allowing the free flow of hydraulic fluid in a direction opposite to the direction of the supply fluid when said line is performing the function of a return line.

4. The reversible drive system of claim 3 wherein the means providing for free flow of hydraulic fluid in a direction opposite to said supply fluid direction also precludes fluid flow in the latter direction.

5. The reversible drive system of claim 3 wherein said means providing for free flow of hydraulic fluid and said fluid pressure adjustment means comprises an integral counterbalance valve.

6. The reversible drive system of claim 5 wherein said counterbalance valve has dual passages with a check valve in one passage and a variable spring-loaded valve in the other.

7. A pair of reversible, hydraulic drive motors operatively engaged with a common load and having a fluid transmission and control system connected thereto for effecting improved control over movement of said load, said fluid transmission and control system comprising:
  - a first closed loop conduit having one of said motors hydraulically connected therein;
  - a second closed loop conduit having the other of said motors hydraulically connected therein;
  - control means common to both said conduits for selectively directing fluid through each in either of two directions;
    - said control means operatively connected to said conduits to drive said motors in the same relative direction;
  - an adjustable pressure reducing means disposed in each closed loop circuit;
    - one of said pressure reducing means disposed to create a pressure drop into one of said motors when fluid flows in a first direction, and the other of said pressure reducing means disposed to create a pressure drop into the other of said motors when fluid flows in the other direction; and
  - means for supplying fluid from a reservoir to said control means and for returning fluid from said motors to the reservoir.

8. The control system of claim 7 including means connecting said motors for selectively adjusting the pressure differential between them when said motors are in a static condition.

9. A reversible drive system substantially free of backlash for accurately controlling movement of a load driven thereby comprising:
  - dual hydraulically operated drive motors operatively engaged with the load;
  - means for selectively varying the torque of one of said motors with respect to the other so that the lower torqued motor acts as a variable drag on the driven load, said means including a hydraulic circuit having a primary control means operatively disposed in the circuit for controlling the direction and quantity of fluid flow, said hydraulic circuit including a fluid reservoir at a pressure substantially below that in the supply lines to said motors;
  - supply and return lines, each connected to said primary control means for conveying hydraulic fluid to and from said hydraulic circuit;
  - pump means in said supply line for providing pressurized fluid to said primary control means;
  - a common case drain line connecting each of said motors to the return line of said hydraulic circuit; and
  - regulating means in said case drain line of each motor for independently adjusting the leakage of pressure from said motors.

10. A pair of reversible, hydraulic drive motors operatively engaged with a common load and having a fluid transmission and control system connected thereto for effecting improved control over movement of said load, said fluid transmission and control system comprising:
  - a first closed loop conduit having one of said motors hydraulically connected therein;
  - a second closed loop conduit having the other of said motors hydraulically connected therein;
  - primary control means having a pair of fluid ports one of said ports common to each of said closed loop conduits and having a pair of other ports connected to supply and return conduits respectively;

said pair of fluid ports operatively connected to said closed loop conduits in a manner enabling said motors to be driven in the same relative direction;

an adjustable pressure reducing means disposed in each said closed loop conduits;

one of said pressure reducing means disposed to create a pressure drop into one of said motors when fluid flows in a first direction, and the other of said pressure reducing means disposed to create a pressure drop into the other of said motors when fluid flows in the other direction;

a fluid reservoir having said return conduit and said supply conduit connected thereto; and

a hydraulic pump disposed in said supply conduit for providing fluid to said primary control means under pressure to thereby drive said motors.

References Cited by the Examiner

UNITED STATES PATENTS

2,541,292	2/1951	Robinson	60—97 X
3,006,215	10/1961	Musser	74—665
3,057,161	10/1962	Henke et al.	60—53
3,166,952	1/1965	Lang	74—665

References Cited by the Applicant

UNITED STATES PATENTS

1,918,180	7/1933	Carter.
1,956,369	4/1934	Woods.
2,461,877	2/1949	Brereton.
2,931,035	3/1960	Reinhard et al.

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