COUNTERBALANCE SYSTEM FOR A MARINE LOADING ARM

Filed Jan. 17, 1967

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ABSTRACT OF THE DISCLOSURE

A marine loading arm having an adjustable counterweight is provided with a hydraulic control circuit and power cylinders which automatically adjust the counterweight to attain weight balance at any attitude of the loading arm when the arm is either empty, partially full, or full.

Background of the invention

The general field of art to which the present invention pertains is in marine loading arms for conveying fluids to and from vessels and, more particularly, concerns counterbalance, power-operated marine loading arms.

Marine loading arms for transferring liquid cargo to and from vessels are now widely used, primarily because they are more efficient and safe than flexible hoses, and can be installed in compact batteries of multiple loading arm units. Various systems for remotely positioning loading arms have been devised, one of which employs a movable counterweight that balances the considerable weight of the extended arm so that power cylinders of reasonable size can be used, and so that the pivotally interconnected conduits are for the most part self-supporting and require only minimum bracing. One difficulty with this type of loading arm is that the balancing force is not constant to provide the optimum counterbalancing action, because the conduits not only assume different attitudes during positioning, but may be full, partially full, or empty. These conditions are obviously most acute in loading arms constructed of large-diameter conduits having a large extended length.

Summary of the invention

The inner conduit section of a marine loading arm projects from a horizontal pivot axis and is provided with a lever arm extending in the opposite direction from the pivot axis. Movable along the lever arm is a counterweight. Individual power means are provided for pivoting the conduit section and for moving the counterweight, the control system for the power means being so arranged that the moment of the conduit section, and its associated parts including the manifold coupling on the terminal conduit section and various swivel joint couplings, automatically adjusts the counterweight along the lever arm to provide a proportional counterbalance force. By this arrangement, a near optimum counterbalance is attained irrespective of the attitude of the loading arm or whether it is full, partially full, or empty.

FIGURE 1 is a diagrammatic side elevation of the loading arm of the present invention.

FIGURE 2 is a diagrammatic rear elevation of the structure shown in FIGURE 1.

FIGURE 3 is an enlarged, diagrammatic elevation, partially broken away, of the general area indicated by the arrow 3 on FIGURE 2.

FIGURE 4A and 4B are schematic hydraulic control and power circuits for governing extension, retraction and the counterbalancing force of the loading arm.

FIGURE 4C is a schematic circuit similar to FIGURE 4B, but illustrating a different operational condition.

Description of the preferred embodiment

With reference to FIGURES 1 and 2, the conventional features of the marine loading arm 10 include a riser assembly 12 which is mounted upon a deck or the like adjacent a berth or mooring location for a tank vessel. The riser assembly is provided with a flanged inlet elbow 13 which by means of a pipeline (not shown) coupled thereto transfers fluid to or from the loading arm 10 through a vertically disposed riser pipe 14.

A swivel pipe coupling 16 is mounted atop the riser pipe 14 and is connected to a 90 degree elbow 18. The elbow 18 supports the later mentioned components of the loading arm 10 for rotation about the vertical axis of the riser pipe 14. In order to prevent excessive loads upon the swivel pipe coupling 16, a vertical strut 20 is connected between a flange 22 (FIG. 3) on the elbow 18, and a lateral yoke 24 (FIG. 2) which embraces a sleeve 26 on the riser pipe 14.

A swivel pipe coupling 28 (FIG. 3) and an elbow 29 interconnect the elbow 18 with an inner pipe section 30, thus mounting the pipe section for pivotal movement about the horizontal axis of the coupling 28 in the usual manner. Similarly, a 90 degree elbow 32 on the inner pipe section 30 at the outer end of the inner pipe section 30 carries a swivel pipe coupling 34 which by means of an elbow 36 is connected to an outer pipe section 38 and mounts the outer pipe section for movement relative to the pipe section 30 about a second horizontal axis including the swivel pipe coupling 34.

The free end of the outer pipe section 38 is provided with a manifold coupling assembly 40 (FIG. 1) which includes a flange valve 42 that is bolted to the manifold of a tank vessel when the loading arm 10 is transferring fluid. The manifold coupling assembly 40 also includes a horizontally disposed swivel pipe coupling 44 (FIG. 2) and an adjacent sheave 46 that mount the coupling assembly 40 for remotely actuated powered movement about the turning axis of the swivel pipe coupling 44. Thus, the sheave 46 is secured to the coupling assembly 40 and one end of a cable 48 is secured to the sheave. The cable 48 is trained around a freely rotatable sheave 50 which is associated with the swivel pipe coupling at 34 (FIG. 2), and extends alongside the inner pipe section 30 to and partially around a lower sheave 52 (FIG. 3). The lower sheave 52 surrounds the pipe section 30 and is rotated with the pipe section when the pipe section is pivoted to pay out or pull in the cable 48 and thus, in a known manner, automatically keep the bolting flange of the valve 42 upright.

Pivotal movement of the outer pipe section 38 relative to the inner pipe section 30 is also effected in a known manner. For this purpose, a sheave 54 is anchored to the inner end portion of the outer pipe section 38, and a sheave 56 (FIG. 3) is secured to the flange 22 adjacent the lower sheave 52. A cable 58 is trained around the sheaves 54 and 56, is anchored to the latter sheave, and includes a single-acting hydraulic cylinder 60 in each run of the cable intermediate the sheaves. By means of a later described hydraulic control circuit, the cylinders 60 are alternately energized so that one cable run or the other is pulled toward a cylinder 60 to swing the outer pipe section 38 relative to the inner pipe section 30.

Power means for pivoting the inner pipe section 30 is generally conventional insofar as the power source and the driving method is concerned, and includes a sprocket 62 (FIG. 3) which is non-rotatable relative to the pipe section 30 and is bolted to the lower sheave 52. A chain 64 is wrapped around the sprocket 62, and each flight of the chain extends outward along the inner pipe section 30 toward a crossbar 66 that is welded to the pipe section. Connected between the crossbar 66 and each
chain flight 64 is a single-acting hydraulic cylinder 68. The cylinders 68, by means of the later described hydraulic control circuit shown in FIGURES 4B and 4C, are alternately energized in the manner of the cylinders 60 and thus in conventional manner rotate the inner pipe section 30 about the horizontal axis of the swivel pipe coupling 28 (FIG. 3).

Further details which are known from prior art structures include the diagonal tension strut 70 (FIG. 2) between the crossbar 66 and the vertical strut 20, a similar strut 72 (FIG. 1) between the sheave 54 and the outer pipe section 38, and other reinforcement means such as various ribs and flanges at 74 (FIGS. 1 and 2) which are welded together to the inner pipe section 30.

Extending rearward of the inner end portion of the inner pipe section 30, and in general alignment with thereof, is a rigid beam 78 which slidably mounts a group of heavy cast plates comprising a counterweight 80. As in similar marine loading arms, the counterweight 80 is adjustable with respect to the beam assembly 78. Effective lever arm including the inner and outer pipe sections 30 and 38 varies according to the relative attitude of the pipe sections, and whether or not the pipe sections are full, partially full, or empty. Although ordinary marine loading arms may employ means for shifting a counterweight to balance the system throughout a range of operational attitudes, the presence or absence of fluid in the loading arm will unbalance the arm according to whether or not the loading arm is designed to balance a full, empty, or partially empty arm assembly. In other words, ordinary loading arms achieve optimum balance under only one of many different conditions resulting from the combination of attitude and presence or absence of fluid in the arm. In the present invention, a near perfect balance is automatically achieved for all of the conditions noted, by means of the movable counterweight 80 (FIGS. 1 and 2) and the specific hydraulic circuit shown in FIGURES 4B and 4C.

The hydraulic circuit of FIGURE 4A controls the power movement of the outer pipe section 38 through a main control valve 82 which, together with the other illustrated control components, may be removed located from the loading arm. The valve 82 is shown in an actuated position in which the parallel flow passages at 84 have directed fluid into the left hand hydraulic cylinder 60 to lower the outer pipe section 38 from a raised position.

Valve 82 has a dead center 86 and crossed passages 88 which, when respectively aligned with a pressure line 90 and a return line 92 to a reservoir 94, block the flow of fluid into the circuit from the main supply line 96. A 360 degree rotation of the valve 82 by means of a knob 98 will change the flow from the reservoir 94 to the left hand side of the cylinder 60.

In the actual instance, fluid under 1200 p.s.i. has been supplied from the supply line 96 by the passages 84 to the left hand cylinder 60 through a line 98, and through a conduit 100 from the right hand cylinder 60 back to the reservoir 94.

The raising of the outer pipe section 38 is accomplished by actuating the control valve 82 to position the crossed passages 88 in place of the passages 84, whereby fluid under pressure is transmitted through a needle valve 102 which restricts the flow of fluid into (and from) the right hand cylinder 60. When the outer pipe section 38 is attained, the control valve 82 is released and its dead center 86 is spring urged into flow-blocking relation to the hydraulic lines 98 and 100.

Interconnecting the conduits 98 and 100 is a conduit 104 that is controlled by a solenoid operated, normally open valve 106. The solenoid 108 of the valve 106 is automatically energized when the power unit, not shown, is turned on to supply the 1200 p.s.i. line pressure prior to the raised or lowered pipe section 38 as described. Accordingly, the elevation position of the raised outer pipe section is maintained as long as the solenoid 108 is energized to close the valve 106, and as long as the pressure in the lines 98 and 100 does not exceed the 1250 p.s.i. pressure setting of the usual pressure relief valves 110 which are associated check valve 112 and are connected between the lines 98 and 100. If such pressure setting should be exceeded, for example as might occur if the pipe section strikes an unyielding obstruction while being power driven, the lines 98 and 100 become interconnected through one of the check valves 110 and its associated check valve 112 so that the loading arm will not be damaged.

With the outer pipe section 38 positioned in a desired location and the coupling assembly 40 connected to the manifold of a vessel, the operator of the loading arm turns off the power unit to deenergize the solenoid 108, whereby the valve 106 returns to its normally open position and the outer pipe section 38 is free to move up or down with the vessel. At the same time, of course, the inner pipe section 30 is similarly free to move.

As thus far described, the structure of the loading arm 10 and the hydraulic system 78 is substantially independent of the loading arm 10 and the hydraulic system 78, as described in accord with known practice in prior art loading arms.

With primary reference to FIGURES 4B and 4C, a LOWERING valve 114 and a RAISING valve 116 respectively control downward and upward movement of the inner pipe section 30. Each valve is manually operated. If the counterweight 80 is in an unbalanced position, the actuation of either control valve to pivot the inner pipe section 30 will actuate the counterweight cylinder 81 and cause a disproportionate movement of the counterweight 80 (FIGS. 1 and 2) along the beam 78 to achieve a near optimum counterbalancing action. Thus, the hydraulic circuit will adjust the counterweight to correct imbalance caused by the presence or absence of fluid in the loading arm before the loading arm is moved toward or from its rest position or its operative position.

As a direct result of this improved counterbalance arrangement, the loading arm pipe sections 30 and 38 can have a large capacity (a large diameter and length) and yet have the compactness and mobility formerly attainable only with loading arms of smaller capacity.

Beginning with an assumed rest position (FIG. 4B) in which the inner pipe section 30 is upright and the hydraulic circuit has not been previously actuated, one possible position of the counterweight 80 is in its maximum extended position away from the pivot axis of the pipe section 30. Another possible position for the counterweight is the maximum retracted position shown in FIGURES 1 and 2. When the power unit is energized to provide the control circuit with fluid under pressure, the actuating solenoid 117 of a normally open valve 118 is energized to block flow communication between a line 120 and a line 122 which are individually connected to the hydraulic cylinders 68 that control powered movement of the inner pipe section 30.

To initiate lowering movement of the pipe section 30, the control valve 114 is actuated so that its straight passages at 124 respectively transmit fluid under pressure from the supply line 96 into a delivery line 126, and complete a flow return path in a return line 128 to the reservoir 94. Part of the fluid in the delivery line 126 passes through a check valve 130 and a flow control valve 132, the latter valve having a bypass check valve 134 into the line 120 which powers the left hand lowering cylinder 68 to lower the pipe section 30. However, because the counterweight 80 is in a maximum extended position, the available pressure in the cylinder 68 is insufficient to retract the piston rod of said cylinder.

Consequently, the operating fluid (at 1200 p.s.i.) is
diverted through an adjustable resistance valve 136, which is set to open at a pressure in excess of 300 p.s.i., and is transmitted through a line 138 into the counterweight cylinder 140. The counter weight rods 80 is thus moved inward or toward the pivot axis of the inner pipe section 30. Only the near cylinder 81 is shown in FIGURE 4B, and the other cylinder 81 is connected in parallel with the cylinder which is illustrated.

After the counterweight 80 is moved sufficiently toward the pivot axis of the pipe section 30 to counterbalance the loading arm to the extent that the pipe section 30 can be lowered with a pressure of 300 p.s.i. exerted against the piston in the LOWERING cylinder 68, the valve 136 closes and the line pressure is then directed entirely to the LOWERING cylinder 68. When the counterweight cylinder is moving the counterweights, the fluid exhausted from behind the piston is transferred through a line 140, a check valve 142, a line 144, and another check valve 146, a line 144, and another check valve 146 into the reservoir 94 via the return line 128 and one of the parallel passages at 124 of the control valve 114. When the LOWERING cylinder 68 is energized, the RAISING cylinder 68 exhausts fluid to the reservoir 94 through the conduit 122, a flow control valve 150 having a bypass check valve 152, and a line 154 which is connected to the line 144.

Thus the setting of the resistance valve 136 causes the valve to resist just enough pressure (300 p.s.i.) for the LOWERING cylinder 68 to lower the pipe section 30, when the loading arm is in a balanced condition, due to the automatic positioning of the counterweight 80. As the pipe section 30 is lowered, its moment increases, but the elevation of the moment of the counterweight beam 78 causes the moment of the counterweight to likewise increase, and the two moments are, and remain, substantially equal. It is believed evident from the preceding description that the control circuit provides means for setting the moment differential between the loading arm 10 and the counterweight 80, relative to the pivot axis of the pipe section 30, and also provides means for bringing the two moments into close balance.

While the pipe section 30 is being lowered, flow from the exhausting RAISING cylinder 68 is resisted by the flow control valve 150 to cushion the movement of the arm. When the appropriate lowered position is attained and the flanged valve 42 (FIG. 1) is maneuvered with the outer pipe section 38 until the valve can be brought to the actuator 108, the actuator of the LOWERING valve 114 is released and hydraulic pressure in the main supply conduit 96 can be shut off. This causes deenergization of the solenoid 117 whereby the valve 118 returns to its normally open position to provide flow communication between the lines 120 and 122. Since the LOWERING valve 114 is now in its FIGURE 4C “off” position, line pressure to the system is blocked off. An open hydraulic circuit exists between the LOWERING and RAISING cylinders 68 through the now open valve 118, and any drifting, rising or falling of line 124, and which may cause the inner pipe section 30 to change its attitude is thus to a large degree unopposed by the cylinders 68.

The valve 42 (FIG. 1) may then be opened to start a fluid transfer operation through the loading arm 10. If sudden vessel movement should pull the loading arm and cause movement of the pipe section 30 sufficient to develop a pressure in excess of 1500 p.s.i. in either cylinder 68, or if the loading arm when being maneuvered into coupling condition strikes a moving obstacle causing the same condition, one of two pressure relief valves 160 will transmit the fluid through an associated check valve 162 into one or the other of the lines 120 and 122. At the end of a fluid transfer operation, the valve 42 is manually closed preparatory to uncoupling the loading arm and retracting it to its rest position shown.

The loading arm in its presently described condition is full of fluid and is consequently unbalanced, because the counterweight 80 is in a position to balance the arm when empty. The power is turned on to restore the main supply line 96, and the solenoid 117 (FIG. 4C) is thus energized to close the valve 118. Because the valve 118 and the LOWERING and RAISING control valves 114 and 116 are closed, there is no exhaust path for the fluid in the RAISING cylinder 68. The pressure of this fluid when the valve 42 (FIG. 1) is uncoupled rises above line pressure since the arm is unbalanced, but is less than the 1500 p.s.i. setting of the relief valves 160. Consequently, even though the arm is unbalanced, it remains in position while the valve 42 is uncoupled from the ship’s manifold.

The counterweight 80 must now be moved away from the pivot axis of the pipe section 30 to restore proper balance, and this is automatically effected, in a manner similar to the previously described counterweight movement, when the RAISING control valve 116 is actuated to the position shown in FIGURE 4C. The parallel passages at 164 (FIG. 4C) are aligned with a pressure line 166 and a return line 168 to transmit line pressure into a line 170 for energizing the RAISING cylinder 68 and the counterweight cylinders 81, and for opening an exhaust path for the LOWERING cylinder 68 to the reservoir 94 through a line 172.

The line pressure of 1200 p.s.i. is transmitted through a check valve 174 and the line 122 into the RAISING cylinder 68, but due to the unbalanced condition of the loading arm, the pressure is not sufficient to retract the piston rod of the cylinder. Consequently, the fluid is diverted through a resistance valve 176, which is set to open at 300 p.s.i. and, into the line 160 which is connected to the counterweight cylinder 81. After the counterweight 80 is moved outward along the beam 78 to a position where the loading arm is so balanced that a pressure of 300 p.s.i. will raise the arm by retracting the piston of the RAISING cylinder 68, the valve 176 closes and all of the line pressure is transmitted to the RAISING cylinder 68.

In rising from its lowered position, the effective moment of the loading arm decreases, and the moment of the counterweight decreases proportionately so that a near perfect balance is maintained until (and after) the pipe section 30 reaches it FIGURE 1 rest position. Unless the fluid in the loading arm is drained, the loading arm is in balanced condition for a subsequent lowering movement of the counterweight 80 because the actuator for the counterweight cylinders 81 is isolated by the closed LOWERING and RAISING control valves 114 and 116. However, if the loading arm is to remain in a rest position for any appreciable time, it is desirable to open the RAISING valve 116 to bleed fluid from the counterweight cylinders 81 to lower the counterweight 80. If the counterweight is lowered, the fluid in the loading arm should also be drained off so that the next operation of the loading arm begins under the same conditions outlined for the first described operation.

When the pipe section 30 is raised as last described, the LOWERING cylinder 68 is exhausted through the line 120, the flow control valve 132, a line 178 and a check valve 180 to the reservoir 94 via the lines 172, 168 and one of the parallel flow passages at 164 in the RAISING control valve 116. Similarly, when the counterweight 80 is moved outward to effect the initial balance as described, the line 138 carries the exhaust fluid to a check valve 182 which communicates with a flow control valve 184. Since the counterweight 80 might be moved downward for a balancing operation or for storing the arm when the pipe section 30 is upright, gravity will assist such movement and the flow control valve 184 restricts the speed of downward movement of the counterweight.

If the loading arm is lowered, empty and unbalanced with the counterweight fully extended a subsequent rais-
ing of the arm after it has been uncoupled from the manifold of the vessel does not require any rebalancing of the counterweight because it already overbalances the empty loading arm and thus tends to automatically return the loading arm to its rest position even though the control circuit is not actuated. Under these conditions a fail-safe condition exists and the hydraulic system acts as a brake to damp the return movement of the loading arm by exhausting the LOWERING cylinder 68 through the flow control valve 132 to reservoir.

The various pressures mentioned in the preceding description are, of course, design considerations related to the physical size of the various components of the loading arm. It should be noted, for example, that the loading arm can be designed so that a pressure above or below the pressure settings of the resistance valves 136 and 176 will effect a pressure transfer from the counterweight cylinders 81 to the LOWERING and RAISING cylinders 68 in order to sequentially sense the moment differential, balance the moments by adjusting the moment of the counterweight, and then raise or lower the pipe section 30.

It will be apparent that the herein disclosed automatic balancing operations do no require that the loading arm be either full or empty of fluid, but that optimum balance is achieved for the loading arm in any partially filled condition.

Although the best mode contemplated for carrying out the present invention has been herein shown and described, it will be apparent that modification and variation may be made without departing from what is regarded to be subject matter of the invention as set forth in the appended claims.

Having completed a detailed description of the invention so that those skilled in the art could practice the same, I claim:

1. A marine loading arm comprising a rigid pipe section, means pivotally mounting said pipe section for movement about a substantially horizontal pivot axis, a counterweight carried by said pipe section and movable toward and away from said pivot axis, first hydraulic power means coupled to said pipe section for regulating the elevational position of the free end of the pipe section, second hydraulic power means coupled to said counterweight for adjusting the position of said counterweight relative to said pivot axis, and an hydraulic control circuit coupled to said first and second power means for automatically sensing the moment differential, as indicated by hydraulic fluid pressure, between said pipe section and said counterweight relative to said pivot axis.

2. Apparatus according to claim 1 wherein said second power means adjusts said counterweight relative to said pivot axis in proportion to said moment differential and reduces said differential to a predetermined value.

3. Apparatus according to claim 2 wherein said counterweight in adjusted position produces a moment substantially equal to the moment of said pipe section, said wherein said control circuit includes means for locking said counterweight in said adjusted position.

4. Apparatus according to claim 1 including a second rigid pipe section, means pivotally interconnecting said first and second pipe sections for fluid intercommunication and for relative pivotal movement about a substantially horizontal axis, third power means coupled to said second pipe section for effecting said relative pivotal movement, said counterweight in said adjusted position producing a moment substantially equal to the effective moment of said first and second pipe sections,

5. Apparatus according to claim 1 wherein said first power means comprises a pair of hydraulically actuated cylinders, said control circuit including a conduit selectively pressurized with hydraulic fluid and connected to each cylinder for transmitting fluid under pressure to effect the power stroke of the cylinder, said second power means including a double-acting hydraulic cylinder connected to said counterweight, a bypass conduit connected to each of said selectively pressurized conduits, said bypass conduits being connected to opposite ends of said double-acting cylinder, and a resistance valve arranged for one direction flow only in each bypass conduit, said resistance valves each opening at a predetermined pressure existing in its associated pressurized conduit to transmit said fluid into said double-acting cylinder and thereby adjust said counterweight.

6. Apparatus according to claim 5 in which said pipe section is pivotable about said axis when the pressure differential between one of said pressurized conduits is pressurized condition, and the associated one of said bypass conduits in pressurized condition, approximates said predetermined opening pressure for the associated one of said resistance valves in order to effect a pressure transfer from said bypass conduit to said pressurized conduit.

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U.S. Cl. X.R.

141—387; 212—49