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[54] **TOWING WINCH CONTROL SYSTEM**

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Related U.S. Application Data

[63] Continuation of Ser. No. 110,695, Jan. 28, 1971, abandoned.

[52] **U.S. Cl.** ..... **254/173 R,** 254/187 R

[51] **Int. Cl.** **B66d 1/48**

[58] **Field of Search** ..... 254/172 R, 173 B, 185 R, 254/173 R, 187 R, 175.7

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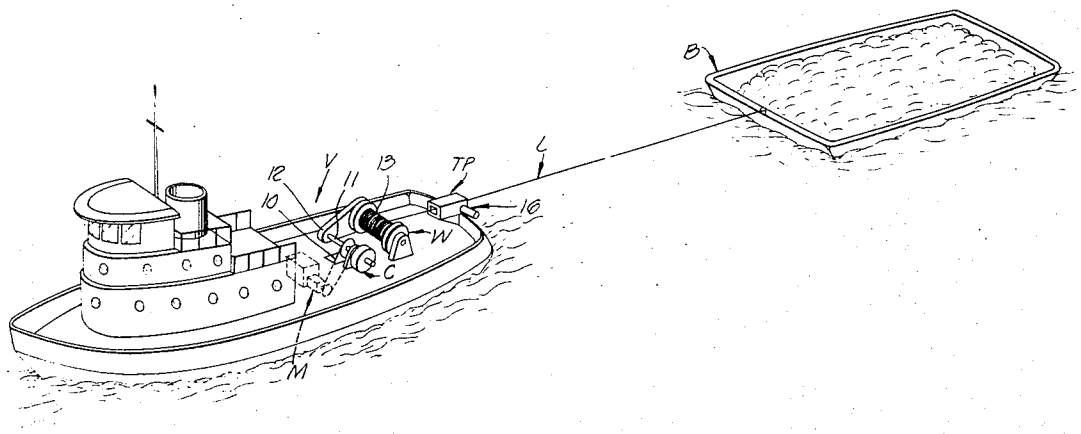
[57] **ABSTRACT**

A towing winch control system in which the winch for the towing cable is driven through a slipping clutch to apply a substantially constant tension to the towing cable, and a position sensor and a tensiometer are operated to control the length of the tow cable between the towing winch and the object being towed and to limit the load on the cable to prevent parting of the cable.

15 Claims, 6 Drawing Figures



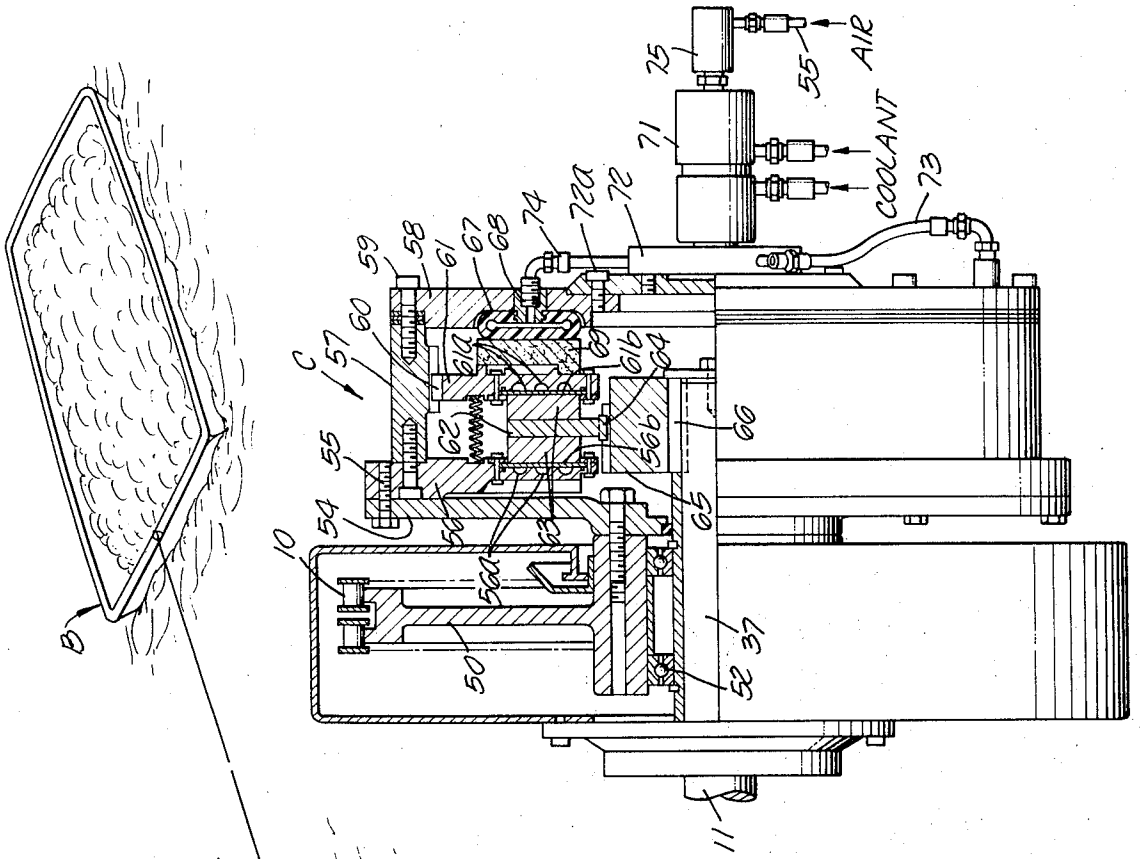


FIG. 5.

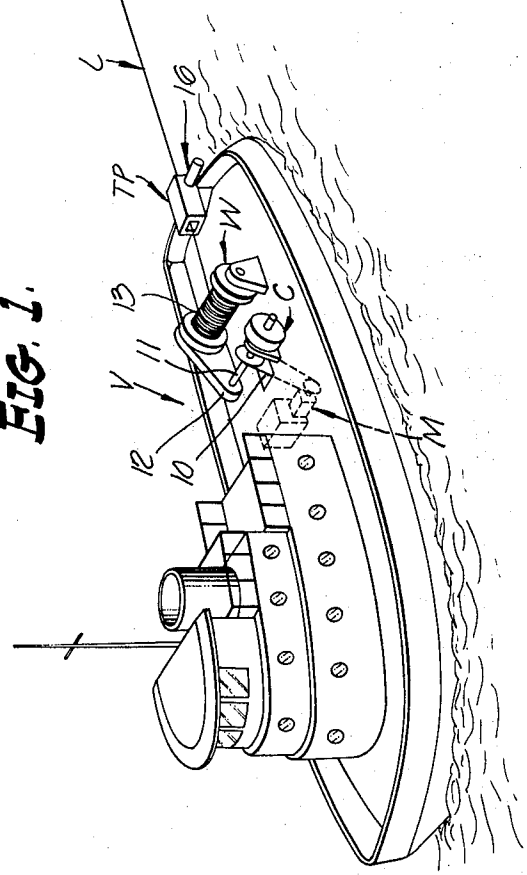


FIG. 1.

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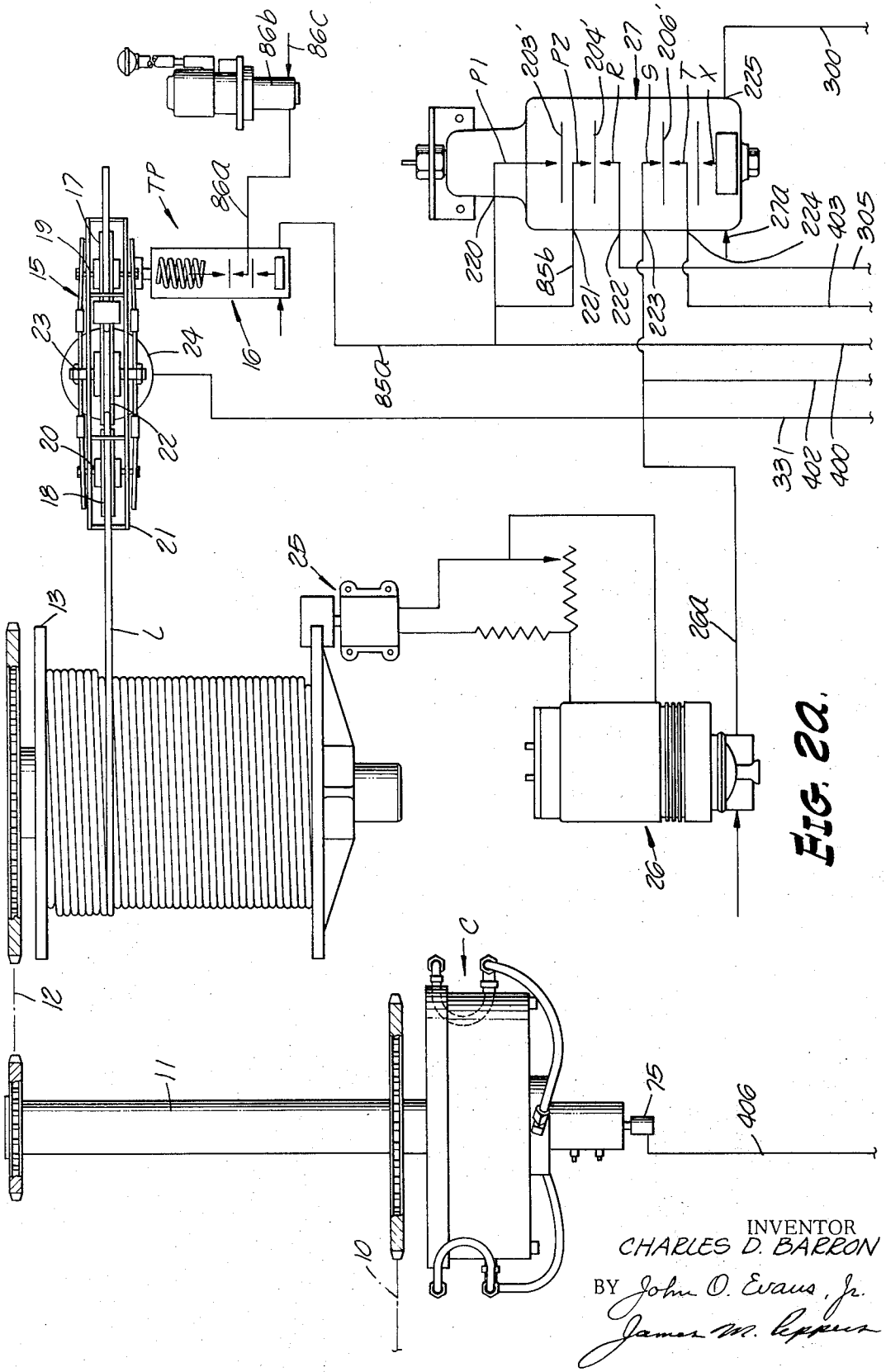

Fig. 2a.

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FIG. 2b.

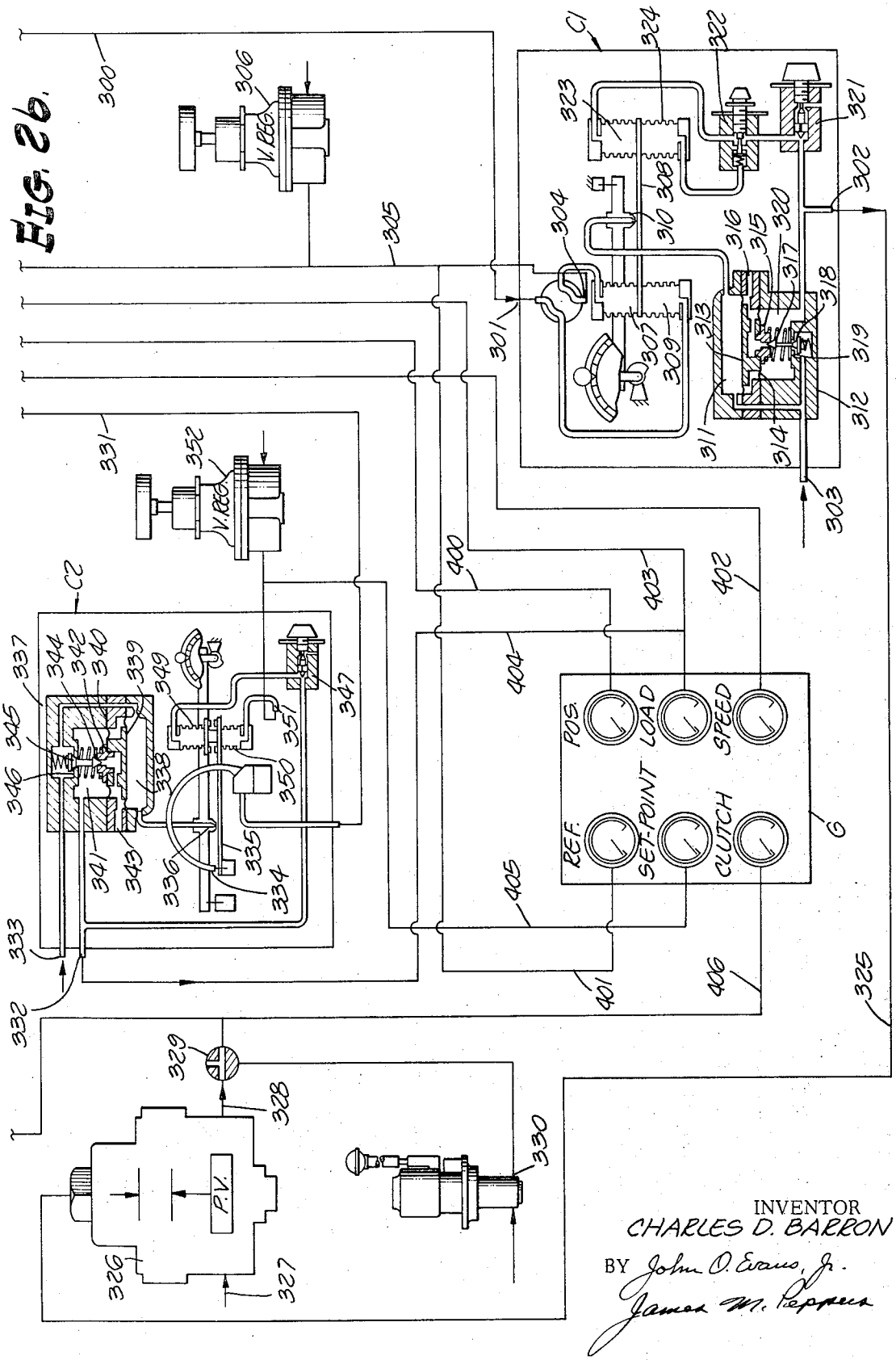

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FIG. 4.

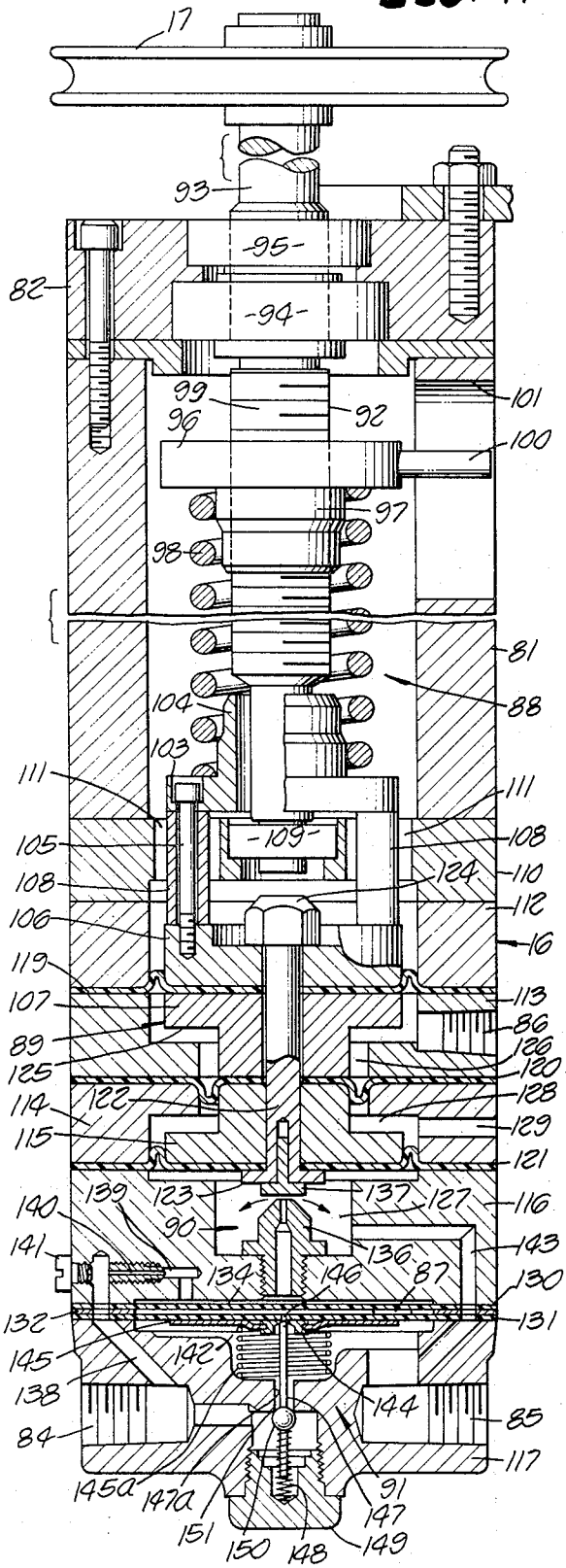
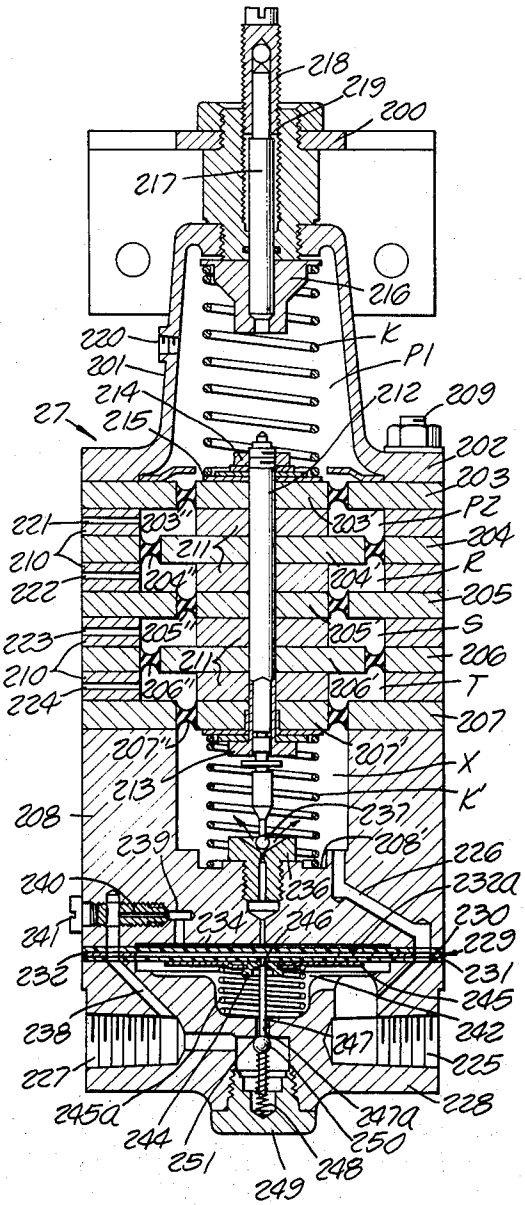


FIG. 3.



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TOWING WINCH CONTROL SYSTEM

This application is a continuation of application Ser. No. 110,695, filed Jan. 28, 1971, now abandoned.

BACKGROUND OF THE INVENTION

The towing of a vessel to sea, such as well drilling barges, cargo barges, or, indeed any vessel, by a tug or other towing vessel, poses problems which are difficult of solution. Heretofore, the towing cable by which the towing vessel tows the towed vessel has been relied upon to compensate for changes in load on the cable due to changes in tidal, wave, or wind action on the vessels, and, as a result, an extremely long run of cable or line has been employed. The cable sags between the vessels, generally beneath the water and provides a long arcuate cable section which can be more or less tensioned as the towing vessel and the towed vessel experience different influences of tide, waves, or wind, or combinations thereof.

Such long tow lines or cables make control of the towed vessel very difficult, particularly when entering or navigating narrow channels or waterways. A heavy barge may require a length of tow line or cable on the order of one quarter mile, for example, to afford a safe bow in the line or cable, whereby the towing and towed vessels may react differently to different influences without parting the tow line or cable. A typical problem in this connection involves the opposite reactions caused by the towed vessel's tending to slow down when on the trailing slope of a wave and the towing vessel's tending to accelerate when picked up by the forward slope of a wave. Such long tow lines or cables also produce a navigational hazard, particularly, when visibility is poor, and the pilot of another vessel may not be aware of the existence of the tow line, due to his inability to see the towed vessel, or, perhaps, either vessel.

Ordinarily, the towing line or cable is on a towing winch and is played off as may be required by the relative conditions of the sea and the weight of the towed vessel. When desired, the winch is operated to pull in the line or cable to shorten the distance between the two vessels. However, if, due to wave, tide or wind influence on the vessels, the line is over loaded it may part, allowing the towed vessel to then drift free, with potentially dangerous results.

SUMMARY OF THE INVENTION

The present invention provides a towing winch system which obviates the problems and hazards of the above-described towing practices.

More particularly, the present invention is a towing winch system which may be employed on a towing vessel to tow another vessel, or string of vessels, in a more closely adjacent relation than has been practical heretofore.

The towing winch system, in general, involves a winch adapted to be driven by a power source through a clutch which is capable of constantly slipping to apply a substantially constant tension on the towing line or cable. When the load on the line or cable increases above a pre-set capacity, the line or cable will be stripped from the winch, but when the tension is thereafter reduced the line or cable will be rewound by the winch until the towed vessel is in the pre-selected spaced relation to the towing vessel.

In accomplishing the foregoing, a position sensor and a tensiometer are engaged with and operated by the tow line or cable to control the slip clutch so as to maintain the tension on the line or cable and to maintain the relative positions of the vessel, so long as different influences on the vessels permit the maintenance of the relative positions of the vessels, and to re-establish the tension on the line or cable and the relative positions of the vessels when the influences on the vessels, or either of them, permit.

Constant tension winches, driven by a fluid pressure actuated slip clutch are well known, as exemplified in United States Letters Patent No. 3,373,972, dated March 19, 1968. Preferably, however, to effect better cooling of the slip clutch with resultant improved efficiency, the clutch may be cooled as disclosed in the pending application for United States Letters Patent of C. D. Barron, Ser. No. 19,601, filed Mar. 16, 1970. The position sensor referred to above is adapted to cause a change in the fluid pressure acting on the slip clutch, and is preferably made in accordance with the disclosure of the pending application for United States Letters Patent of C. D. Barron, Ser. No. 19,564, filed Mar. 16, 1970.

This invention possesses many other advantages, and has other purposes which may be made more clearly apparent from a consideration of a form in which it may be embodied. This form is shown in the drawings accompanying and forming part of the present specification. It will now be described in detail, for the purpose of illustrating the general principles of the invention; but it is to be understood that such detailed description is not to be taken in a limiting sense, since the scope of the invention is best defined by the appended claims. In this connection, while the invention is herein disclosed as being incorporated in a towing winch system, it will be understood that the invention is applicable to other dynamic systems in which the tension on a line and the position of a line must be controlled or adjusted to prevent overloading of the line and to reposition a load in response to changes in load condition.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a perspective, generally illustrating a towing vessel and a towed barge to which the towing winch control system of the invention is applied;

FIGS. 2*a* and 2*b*, together constitute a diagrammatic illustration of the towing winch control system, FIG. 2*b* constituting a downward continuation of FIG. 2*a*;

FIG. 3 is a longitudinal section through a computing relay or pressure transmitter employed to vary the drive of a slip clutch in response to variations in the load conditions on the towing winch line;

FIG. 4 is a longitudinal section through a line position sensing device for varying a pressure signal supplied to the computer of FIG. 3, in response to changes in position of the towing winch line; and

FIG. 5 is a view partly in elevation and partly in longitudinal section, showing a slip clutch assembly for driving the towing winch.

DESCRIPTION OF THE PREFERRED EMBODIMENT

As seen in the drawings, with reference first to FIG. 1, a towing vessel or tug boat V is shown as having a towing winch W from which a tow cable or line L ex-

tends sternward for connection to a towed vessel or barge B.

While a rock barge is shown as illustrative, it will be understood that the problem of overload on a towing line L exists in any similar situation involving a towed vessel or barge, such as, for examples, an offshore well drilling platform, a disabled ship, or the like.

The towing vessel or tug is shown as having the winch W disposed above-deck, for clarity, and a suitable drive or motor source M is shown below-deck, but as a practical matter, it may be preferred that the entire apparatus or any portion thereof be located below deck, with the cable or line L passing through a fairlead or guide at the stern of the vessel V.

More particularly, the illustrative drive M includes a suitable drive connection in the form of a chain **10** adapted to effect rotation of a counter shaft **11** through a slip clutch assembly C, which, as is more particularly seen in FIG. **5** and described hereinafter, is adapted to transmit torque to the shaft **11** under the control of the control system of the invention, whereby to limit the tension on the line L and to adjust the relative positions of the towing vessel V and the barge B. Thus, a drive connection, say the illustrative chain drive **12** provides means for applying torque to the drum **13** of the winch W, whereby the line L will be pulled in if the selected line tension overcomes the load of the barge B, but the line L may be stripped from the drum **13** if the barge exceeds the pre-set line tension. Associated with the line L are sensing means TP, incorporated in the system and adapted, as will be later described, to sense the load or tension on the line L and to sense changes in position of the line L so as to effect control of the line.

Referring to FIG. **2***a*, the sensing means TP is generally shown to comprise a typical hydraulic tensiometer **15** and a position senser **16.** The tensiometer **15** is adapted to be applied to the line L in such a manner that the line engages and extends between a pair of spaced rollers **17** and **19** which are rotatable on shafts **19** and **20,** respectively, supported in a frame **21.** A third roller **22** engages the line L between the rollers **17** and **18** to deflect the intermediate portion of the line, the roller **22** being carried on a shaft **23** which is mounted for movement relative to the frame **21,** whereby such movement is applied to the hydraulic sensing unit **24,** to produce a hydraulic signal the magnitude of which is a function of tension on the line L. Such devices **15** are well known and require no further description herein. An example of such a line tension no responsive device is the Tensiometer Model UD12 of Martin-Decker Corp., of Santa Ana, Calif. The position sensor unit **16** of the sensing means TP is more particularly illustrated in FIG. **4** and will be latter described.

In order to control the speed at which the line L may change position, speed control means including a tachometer signal generator **25** is adapted to be driven by the winch drum **13** to supply a variable signal to an electro-pneumatic transducer **26,** say the Model T-10 electro-pneumatic, transducer of Conoflow Corporation of Blackwood, N.J. The output from the hydraulic unit **15,** the position sensor **16** and the transducer **26** are compared by computer means **27,** the details of which are shown in FIG. **3,** to control the clutch C, as will be more fully described below.

The clutch C, as best seen in FIG. **5,** is associated with an end **37** of the counter shaft **11,** and the drive chain **10** engages a sprocket **50** which is revolvable relative to the shaft end **37** on bearings **52.** Affixed to the sprocket **50,** is a disc **54** which is in turn affixed by fasteners **55** to the outer periphery of the back-up plate **56** of the slip clutch means C.

This slip clutch means C includes an outer annular body **57** to which an annular flange **58** is connected by fasteners **59** in opposed relation to the plate **56.** Internally thereof, the body **57** has a splined connection **60** with the outer periphery of an axially shiftable clutch pressure plate **61.** Between the clutch plates **56** and **61** is a clutch friction disc **62** having friction facing **63** on opposite sides thereof and having, as at **64,** a splined connection with a hub **65** which is disposed upon the shaft end **37** and is keyed thereto by a key **66.** Thus, rotation from the sprocket **50** will be transmitted to the tensioning hoist shaft **11** when the slip clutch means C is engaged to transmit rotation from the clutch body **57** and its plates **56** and **61** to the friction disc **62.**

Engagement of the slip clutch means C is accomplished by an annular expansible actuator tube **67** having an air inlet **68.** The actuator tube **67** engages an annular body of insulating material **69** interposed between the tube **67** and the clutch pressure plate **61.** Each of the clutch plates **56** and **61** has a number of annular, radially spaced and concentric coolant passages **56***a* and **61***a* to which a coolant is supplied to dissipate the heat of friction caused by slippage of the clutch C. These passages **56***a* and **61***a* are defined respectively between the clutch plates and a wear disc **56***b* carried by the plate **56** and a wear disc **61***b* carried by the plate **61,** the friction material on the friction disc **62** being engaged with the wear discs **56***b*, **61***b*.

Such cooled, slip clutches are well known, and generally are provided with a coolant circulating system including a stationary coolant connector **71** through which coolant flows to and from a rotary connector **72** which is connected, as by fasteners **72***a*, to the clutch flange **58** and which has conduit means **73** for supplying coolant to the passages **56***a* and **61***a*, as well as conduit means for the return flow of coolant to the connector **71** and thence to a heat exchanger. Preferably, in order to more effectively cool the clutch, it is constructed in accordance with the aforementioned application for patent. In addition, the rotary connector **72** provides a connection for air conduit means **74** which leads to the air inlet **68** for the clutch actuator tube **67** from a stationary air inlet fitting **75.** As is well known, the torque transmitting capacity of such slip clutches varies with the pressure of air in the actuator tube **67.**

Thus, the tension applied to the line L will be determined by the magnitude of the air pressure supplied to the actuator tube **67** through the coupling **75** under the control of the computing relay **27,** as will be more fully described below.

The preferred line position sensing unit **16,** of the sensing meant TP, is shown in greater detail in FIG. **4,** and is more specifically disclosed in the aforementioned pending patent application. It comprises an elongated housing **81** having at one end a closure or cap **82** and having at the other end an assembly which provides an air inlet or supply port **84,** an outlet **85** for a controlled air pressure signal, a port **86** for bias pres-

sure fluid, and a port 87 communicating with the atmosphere.

Included within the assembly are actuator means generally denoted at 88, fluid pressure responsive piston means 89 operatively connected to the actuator means 88, orifice means 90 operable in response to the application of fluid pressure to the piston means 89 and to the application of force from the actuator means 88 for opening and closing the orifice means 90, and combined inlet and outlet valve means 91 for controlling the flow of air from the supply port 84 to the output port 85 and for controlling the exhaust of air from the outlet port to the atmosphere through the port 87.

In general, it is the purpose of the position sensing pneumatic control device to regulate the output signal pressure to a constant value which is determined by the net force applied to the piston means 89, whereby the orifice means 90 is either opened or closed for a period sufficient to balance the piston means 89, so that the pressure drop through the orifice means 90 remains constant, resulting in a constant output signal pressure at the port 85 which leads to the computer relay 27, as will be later described.

More particularly, the actuator means 88 comprises a shaft 92 which extends longitudinally of the housing 81 and has an end 93 which extends axially from the end cap 82 through a suitable bearing 94 and a suitable seal 95. Disposed upon the shaft 92 within the housing 81 is a spring seat 96 having a reduced central section 97 on which is piloted the upper end of a coiled compression spring 98. The shaft 92 is threaded as at 99, and the spring seat 96 and the reduced pilot portion 97 thereof are complementally threaded, whereby rotation of the shaft 92 will effect longitudinal movement of the spring seat 96 on the shaft, since the seat 96 is held against rotation by a key 100 carried thereby and extending into a lateral slot 101 in the housing 81. At its inner end the spring 98 seats on a spring seat 103 having a reduced pilot portion 104. This spring seat 103 is connected by fasteners 105 to the circular upper body portion 106 of the piston 107 of the piston means 89, the seat 103 and the body 106 being held in axially spaced relation by tubular spacers 108 interposed therebetween and through which the fasteners 105 extend. The lower end of the shaft 92 extends through the seat 103 and is journalled in a bearing 109 which is mounted in a supporting spider 110 having circumferentially spaced openings 111 to accommodate the spacers 108, whereby the piston means 89 is axially movable.

The assembly also comprises, in addition to the spider 110, an annular spacer 112, an annular cylinder 113 for the piston 107 and an annular cylinder 114 which houses a piston 115, an annular body 116 containing the nozzle means 90, and an end member 117. The spider 110, spacer 112, cylinders 113 and 114, annular body 116, and end member 117 are interconnected together and to the cylindrical body 81 by tiebolts or the like, requiring no illustration.

Between the spacer 112 and the cylinder 113 is clamped the outer marginal portion of a diaphragm 119, and between the cylinder 113 and the cylinder 114 is clamped the outer marginal portion of another diaphragm 120. Still another diaphragm 121 has its outer marginal portion clamped between the cylinder 114 and the annular body 116. In the illustrative embodiment, the body portion 106 of the piston 107, as well as the piston 107 and the piston 115 are interconnected by a stem 122 having an enlarged head 123 at one end which clamps the inner periphery of the diaphragm 121 against the adjacent portions of the piston 115, and a nut 124 is threaded onto the other end of the stem 122 to effectively clamp the piston 117, including its upper body portion 106, and the piston 107 together. The inner periphery of the diaphragm 119 is like-wise clamped between the upper body portion 106 and the piston 107, and the inner periphery of the diaphragm 120 is clamped between the piston 107 and the piston 115. Thus, the piston means 89 comprises both the piston 107 and the piston 115. More particularly, the piston 107 has an enlarged portion 125 which is exposed to the pressure in a chamber 126 provided in the cylinder 113 between the diaphragms 119 and 120. The piston 115 is exposed to the pressure in a chamber 127 provided in the annular body 116 across substantially the entire cross-sectional area of the piston 115. Within the cylinder 114, the piston 115 is disposed in a chamber 128 which is vented to the atmosphere through radial ports 129.

Interposed between the annular body 116 and the end member 117, and clamped at its outer margin is a double diaphragm assembly including an upper diaphragm 130 and a lower diaphragm 131 spaced apart by an outer material spacer 132 in which is formed one or more of the radial outlet ports 87, previously referred to, which communicate the space between the diaphragms 130 and 131 with the atmosphere. In the annular body 116 above the upper diaphragm 130 is a chamber 134 and centrally of the body 116 is a threaded bore having therein a nozzle 136, the port through which communicates with the chamber 134, and the outlet of which is opposed by a nozzle seat 137 suitably carried by the lower end of the stem 122. Air is supplied to the chamber 134 and thence to the nozzle 136 from the supply port 84 through a passage 138 which extends through the margin of the diaphragms 130 and 131 and the spacer 132 and connects with a passage 139 leading into the chamber 134. Disposed in the passage 139 is a flow restrictor 140 having a reduced passage therethrough. This flow restrictor is replaceable through an opening in the body 116 which is closed by a threaded closure plug 141. At the outer side of the diaphragm 131 in the end member 127 is a chamber 142 which communicates with the output port 85. The output port 85 also communicates through a passage 143 with the chamber 127 in the body 116 below the diaphragm 121.

The inlet and outlet valve means 91, previously referred to, includes a valve seat 144 carried by a plate 145 below the diaphragm 131 and having a valve port 146 leading from the outlet chamber 142 into the space between the diaphragms 130 and 131. A coiled compression spring 145*a* is provided beneath the plate 145 which applies a normal inward bias to the diaphragm 131 and to the outlet valve seat 144. A valve stem 147 is reciprocably mounted in a port 147*a* in the end member 117, which port leads from the inlet 84 to the outlet 85. The stem is normally biased inwardly by a coiled compression spring 148 which seats in a plug 149 in the end member 117 and acts inwardly on a spherical valve head 150 to bias the same against an inlet valve seat 151.

As previously indicated, the position sensing pneumatic control device functions to regulate the output

signal pressure at the port **85** to a value which is proportional to sensed movement. Accordingly, the outer end **93** of the shaft **92**, in the illustrative embodiment, has mounted thereon the sheave or roller **17** of the tensiometer **15** which is engaged by the cable or line L, to effect rotation of the shaft **92** in response to relative movement between the line L and the sensing device. Movement of the line L is transmitted to the shaft **92** to effect rotation of the latter in one direction or the other depending upon the direction of movement of the line L. It is apparent that rotation of the shaft **92** in one direction or the other will impose more or less compression on the spring **98** to provide more or less force acting on the piston means **89** which will either cause the nozzle seat **137** to close the nozzle **136** or to open the nozzle **136** for communication with the chamber **127**, and hence the discharge or signal output port **85**. Such spring force is opposed by the pressure of air in the chamber **127** acting on the cross-sectional area of the piston **115** and the pressure of fluid in the chamber **126** acting on the effective area of the piston **107**. Thus, the fluid admitted through the port **86** to the chamber **126** may be supplied from a remote set point to quickly and easily calibrate the motion sensor, so that the signal output pressure is at a desired level, less than the input pressure, under the conditions that the shaft **92** is in the neutral or non-moving condition.

With the foregoing details in mind, the operation of the motion sensor is such that the spring **98** is operative to apply a variable force in a direction tending to move the piston means **89** variable force in a direction tending to move the piston means **89** downwardly. Opossing the force derived from the actuator means is the force derived from the application of pressure from a remote set point to the bias or calibrating chamber **126**, which pressure is effective over the area of the enlargement **125** of the piston means **89** to provide a force tending to move the piston means **89** upwardly. Also providing a force tending to move the piston means **89** upwardly is the pressure in the piston chamber **127** which acts upon the piston **115** of the piston means **89**, the other side of the piston **115** being exposed to the atmosphere in the chamber **128**.

The effective signal outlet pressure in the piston chamber **127** is a function of the reduction in the inlet pressure caused by the passage of air from the inlet **84** through the flow restrictor **140** into the pilot pressure chamber **134** and the reduction in pressure resulting from the passage of air from the pilot pressure chamber **134** through the orifice means **136**, as indicated by the arrows, into the piston chamber **127**. When the device is in the condition shown in FIG. **4**, the effective signal outlet pressure at the outlet **85** is the same as that in the piston chamber **127**, and under the condition shown the pressure drop from the inlet **84** to the outlet **85** will remain constant, unless the force derived from the actuator means **88** is varied or the force derived from the remote set point pressure is varied.

Assuming that the force derived from the actuator means tending to shift the piston means **89** downwardly is reduced, the net force acting on the piston means will cause the piston means to move upwardly, allowing greater flow from the pilot pressure chamber **134** into the piston chamber **127**. Such action will result in an instantaneous decrease in the pilot pressure in the chamber **134**. As a consequence, pressure applied to the diaphragm **131** and the force of the spring **145***a* will move the exhaust valve seat **144** upwardly and off of the end of the valve stem **147**, to allow the exhaust of fluid pressure from the outlet chamber **142** and the piston chamber **127** through exhaust port **87**, until the device again assumes the condition shown in FIG. **4** at which the exhaust valve port **146** is again closed. At this time, the pressure at the outlet **85** will again be stabilized at a value determined by the fluid pressures acting on the actuator means **88** and the decreased spring force of the spring **98**. A larger volume of air will flow past the orifice closure **137**, and the signal outlet pressure will be at a lower value.

Assuming that the force derived from the actuator means tending to move the piston means **89** downwardly is increased, overcoming the effect of the signal outlet pressure in the chamber **127**, then the orifice closure disc **137** will engage the end of the orifice means **136**, thereby shutting off the passage of air from the pilot pressure chamber **134** into the piston chamber **127**. Under these circumstances, the pilot pressure in the pilot chamber **134** will build up, forcing the diaphragm **130** and the diaphragm **131** downwardly, thereby unseating the valve **150**, so that inlet pressure will transfer through port **147***a* of the inlet-outlet valve **91**, resulting in an increase in the signal outlet pressure in the outlet chamber **142** and in the piston chamber **127** which will be effective to again condition the apparatus as shown in FIG. **4**, so that the pressure at the outlet **85** again remains constant, but greater.

It will now be understood that variation of the remote set point pressure in the chamber **126** will have the same effect as variation of force derived from the actuator means. In other words, as the remote set point pressure is increased, the force tending to move the piston means **89** upwardly will also be increased, but if the remote set point pressure is decreased, the force tending to move the piston means **89** upwardly will be decreased. The supply of air to the chamber **126** through the port **86** is shown in FIG. **2***a*, as being via a conduit **86***a* leading from a suitable valve **86***b* which controls the pressure derived from a source conduit **86***c* which leads from a suitable pressure source, not shown. The outlet port **85** is in communication with a conduit **85***a* which leads to the computer means **27**, now to be described.

This computer means **27**, as seen in FIG. **3**, comprises a support **200** adapted to be mounted at a suitable location. Carried by the support **200** is an end cup **201** having a marginal flange **202** for connecting the cup **201** with an assembly which comprises a stack of discs **203**, **204**, **205**, **206** and **207** and a body **208**, all connected at the outer peripheries by a suitable number of tie bolts, one of which is shown at **209**. The disc **203** includes a rigid central section **203'** and a flexible annular diaphragm **203''** supporting the central section **203'** and a flexible annular diaphragm **203''** supporting the central section **203'** within the disc **203**. Similarly, each of the discs **204**, **205**, **206** and **207** comprises a rigid central section **204'** to **207'** and an annular diaphragm **204''** to **207''**. Intermediate, the discs **203** to **207** are annular, outer peripheral spacers **210** and central spacers **211**. The outer spacers **210** are connected in the assembly by the tie bolts **209**. The central spacers **211** are interconnected at the respective central sections **203'** to **207'** by a pin **212** having a head **213** at its lower end and a nut **214** at its upper end for clamping the piston sections and central spacers together.

Fluid under pressure, say air, is supplied to the computer means **27** above and below the stack of diaphragms and between the diagrams from various sources, whereby to provide an output pressure signal which is a function of the various input signals and the constant force of an adjustable coiled spring K which is disposed in the cap **201** and seats, at one end, on a seat **215** above the disc section **203'** and at the other end on a spring seat **216** carried by an axially shiftable adjuster pin **217.** The pin **217** is shiftable by an adjuster screw **218** threaded in a nut **219** which is suitably affixed to the support **200.** Below the disc **207** is another coiled spring K' which seats at one end in a seat **208'** and engages at its other end beneath the disc section **207'** in opposition to the spring K. Thus, the spring K is adjustable to provide a selected force on the stacked disc sections **203'** to **207'**, determined by the relationship between springs K and K'.

Air pressure is supplied to a chamber **P1** in the cup **201,** from the position sensor **16** via conduit **85***a*, by suitable means, such as an inlet fitting **220,** to provide a downward force on the effective piston area of the central section **203'** of disc **203.** In order to increase the magnitude of the force derived from air pressure supplied to the computer **27** from the position sensor **16** via conduit **85***a*, a branch conduit **85***b* leads to a chamber **P2** defined between the diaphragms **203''** and **204''**, say through a pressure inlet **221,** so that such pressure also acts downwardly on the effective annular piston area of the central section **204'** of the disc **204,** which extends radially beyond the spacer **211** thereabove.

Below the annular piston area of the disc section **204'** is a chamber R having an inlet **222** to which pressure fluid is supplied, as will be later described, at a value determined by the computer **27,** the pressure in chamber R acting upwardly on the effective annular piston area of the disc section **204'** in opposition to the downward force derived from pressure in the chambers **P1** and **P2.**

Between the discs **205** and **206** is defined a pressure chamber S to which air is supplied through an inlet **223** via a conduit **26***a*, at a pressure determined by the speed of rotation of the drum **13,** under the control of the electro-pneumatic transducer **26,** previously referred to. The disc section **206'** provides an annular piston area projecting radially outwardly of the spacer **211** thereabove, this piston area being responsive to pressure in chamber S to provide a downward force. Below the disc section **206'** is another chamber T to which air is supplied via a port **224** at a pressure determined, as will be later described, by the tension on the line L. Such pressure acts upwardly on the effective annular piston area of the disc section **206'.**

Below the disc **207,** and in the body **208,** is a chamber X which constitutes an output chamber communicating with an outlet port **225** via porting **226.** The pressure in the chamber acts upwardly on the lower disc section **207'**, and this pressure is derived from on inlet conduit **27***a* (FIG. **2***a*) connected to an inlet port **227,** under the control of the computer.

Interposed between the body **208** and an end member **228** having the ports **225** and **227** therein, and clamped at its outer margin is a double diaphragm assembly **229** including an upper diaphragm **230** and a lower diaphragm **231** spaced apart by an outer marginal spacer **232** in which is formed one or more radial outlet ports **232***a*, which communicate the space between the diaphragms **230** and **231** with the atmosphere. In the body **208** above the upper diaphragm **230** is a threaded bore having therein a nozzle **236,** the port through which communicates with the chamber X, and the outlet of which is opposed by a valve head **237** suitably carried by the lower end of the stem **212.** Air is supplied to the chamber **234** and thence to the nozzle **236** from the supply port **227** through a passage **238** which extends through the margin of the diaphragms **230** and **231** and the spacer **232** and connects with a passage **239** leading into the chamber X. Disposed in the passage **239** is a flow restrictor **240** having a reduced passage therethrough. This flow restrictor is replaceable through an opening in the body **208** which is closed by a threaded closure plug **241.** At the underside of diaphragm **231** in the end member **228** is a chamber **241** which communicates with the output port **225.** The output port **225** also communicates through the passage **226,** previously referred to, with the chamber X in the body **208** below the piston or disc section **207'.**

Inlet and outlet valve means are provided to control the admission of fluid from the inlet **227** to the chamber **242** and the exhaust of such fluid through the vent port **232***a*. This valve means includes a valve seat **244** carried by a plate **245** below the diaphragm **231** and having a valve port **246** leading from the outlet chamber **242** into the space between the diaphragms **230** and **231.** A coiled compression spring **245***a* is provided beneath the plate **245** and applies a normal upward bias to the diaphragm **231** and to the outlet valve seat **244.** A valve stem **247** is reciprocably mounted in a port **247***a* in the end member **228,** which port leads from the inlet **227** to the outlet chamber **242.** The stem is normally biased inwardly by a coiled compression spring **248** which seats in a plug **249** in the end member **228** and acts inwardly on a spherical valve head **250** to bias the same against an inlet valve seat **251.**

As previously indicated, the computer means functions to regulate the output signal pressure at the port **225** to a value which is proportional to load on or position of the line L, as well as speed of the drum **13,** and in addition, the computer may be adjusted to modify the output pressure by varying either the effective constant force of spring K or the reference set point pressure in the chamber R. Thus, as will be understood, the output pressure in chamber X is determined by the various pressures in the various chambers **P1, P2,** R, S and T, acting on the various piston areas of the discs **203** to **207.** The equation may be stated:

$$X = P1 + P2 - R + S - T + \text{or} - K,$$

where **P1** and **P2** are the pressure derived from the position sensor **16** tending to close the nozzle **236,** R is the pressure derived from a reference pressure source tending to open the nozzle **236,** S is the pressure derived from the speed of the drum **13** tending to close the nozzle **236,** T is the pressure derived from the tension sensing means **15** tending to open the nozzle **236,** and K is the spring constant.

The effective signal outlet pressure in the outlet chamber **242** is a function of the reduction in the inlet pressure caused by the passage of air from the inlet **227** through the flow restrictor **240** into the pilot pressure chamber **234,** and the reduction in pressure resulting from the passage of air from the pilot pressure cham-

ber 234 through the orifice means 236, as indicated by the arrows, into the pressure chamber X. When the device is in the condition shown in FIG. 3, the effective signal outlet pressure at the outlet 225 is the same as that in the chamber X, and, under the condition shown, the pressure drop from the inlet 227 to the outlet 225 will remain constant, unless the force derived from any of the position sensing means TP, the tachometer 25, or the reference pressure is varied, and, accordingly, as will be later more fully described, the actuating pressure supplied to the clutch C will remain constant.

Assuming that the force tending to shift the stacked disc sections causes the valve head 237 to move upwardly allowing greater flow from the pilot pressure chamber 234 into the chamber X, such action will result in a decrease in the pilot pressure in the chamber 234. As a consequence, pressure applied to the diaphragm 231 and the force of the spring 145*a* will move the exhaust valve seat 244 upwardly and off of the end of the valve stem 247 to allow the exhaust of fluid pressure from the outlet chamber 242 and the chamber X through exhaust port 232*a* between the diaphragms 230 and 231, until the device again assumes the condition shown in FIG. 3 at which the exhaust valve port 246 is again closed. At this time, the pressure at the outlet 255 will again be stabilized at a lower value, determined by the change in forces acting on the stack of disc sections 203′ to 207′.

Assuming that the net force tending to move the stacked discs 203′ to 207′ downwardly is increased, overcoming the effect of the signal outlet pressure in the chamber X, then the orifice valve 237 will close the orifice means 236, thereby shutting off the passage of air from the pilot pressure chamber 234 into the chamber X. Under these circumstances, the pilot pressure in the pilot chamber 234 will build up, forcing the diaphragm 230 and the diaphragm 231 downwardly, thereby unseating the valve 250, so that inlet pressure will transfer through port 247*a* of the inlet-outlet valve means, resulting in an increase in the signal outlet pressure in the outlet chamber 241 and in the chamber X, which will be effective to again condition the apparatus as shown in FIG. 3, so that the pressure drop therethrough again remains constant, but greater, since there is less flow through the orifice means 236.

The conduit 300 leads from the outlet port 225 of the computing relay 27 to the control pressure inlet 301 of a pressure controller C1 of a conventional type adapted to control the pneumatic pressure at an outlet 302 supplied from a suitable source (not shown) through an inlet 303, whereby, as will be later described the slip clutch C is adapted to apply a controlled constant torque to the drum 13 which is a function of the output signal pressure of the computer or transmitter means 27.

More particularly, the controller C1 may be Model 50 Controller of Moore Products Co., of Spring House, Pennsylvania or a Model 2516 Controller of Fisher Governer Company of Marshalltown, Iowa, as examples, the controller, generally shown in FIG. 2*b*, being the latter and more specifically illustrated in Bulletin D–2506A of that company.

The controller C1 is supplied a reference pressure at an inlet 304 from a conduit 305 connected to a source (not shown) by a regulator valve 306, the same reference pressure being supplied to the reference chamber R of the computer or transmitter means 27. This reference pressure is admitted to a bellows 307 of the controller which cts downwardly on a plate 308. Pressure supplied to the inlet 301 from the computer 27 causes an increase in acts in a bellows 309 which is opposed to the bellows 307 and acts upwardly on the plate 308. The position of the plate 308 relative to a nozzle 310 to which controlled fluid pressure is supplied from the source inlet 303, is determined by the difference in pressures in the bellows 307 and 309. If the output pressure from the computer means 27 increases, the pressure increases in bellows 309 causing the plate 308 to move closer to the nozzle 310, restricting flow through the nozzle to cause an increase in the pressure in a chamber 311 of a control valve 312, causing an increased downward force on a diaphragm assembly comprising spaced diaphragms 313 and 314, which carries a valve seat 315, the passage through which communicates with the atmosphere through a port 316 between the diaphragms 313 and 314. The valve seat 315 engages and pushes downwardly, under the circumstances now being described, on an inlet and outlet valve having a head 317 for closing the exhaust passage through the valve seat 315 and a head 318 which is moved away from a seat 319 to allow increased supply pressure into the valve outlet chamber 320 which acts on the diaphragm 314 until the valve seat 315 is again moved upwardly to allow return upward movement of the inlet-outlet valve head 318 towards its seat.

During the same time, pressure is increasing in the chamber 320, such pressure is supplied to the outlet 302, and, thus, to the clutch C, as well as to an adjustable proportioning valve 321 and, depending on the adjustment of the latter, to an adjustable re-set control valve 322 which controls the build up of pressure in a bellows 323. This bellows 323 acts downwardly on the plate 308 tending to move the latter away from the nozzle 310 to decrease pressure at the outlet 302 and in control valve chamber 320, and is opposed by the upward action of a bellows 324 to which pressure is supplied from the valve 322 at a slower rate, depending on the adjustment of the valve 322, until the plate 308 is moved toward the nozzle to again increase pressure at the outlet 302 and in the valve chamber 320.

If a chamge in the system causes a decrease in pressure at the inlet 301 to the controller C1, then, the reverse action will occur in the controller, the tendency being in either case to attempt to return to a preestablished, constant pressure at the outlet 302 which pressure is a function of the outlet pressure from chamber X of the above-described computer means.

The outlet pressure from the controller means C1 is supplied via a conduit 325 to cause actuation of the clutch C, but preferably a typical booster 326 is employed, whereby the actual pressure source (not shown) for the clutch includes an inlet to the booster 326 from a relatively high pressure source, and the pressure in conduit 325 acts on the usual pilot valve of the booster, so that the outlet 328 of the booster is at a greater pressure than the signal pressure from the controller C1. In addition, is preferred that a selector valve 329 be provided, so that the air connector 75 of the clutch C may be connected either to the booster outlet 328 or, alternatively, to a separate source conduit 330 including a manual control valve for operating the clutch C independently of the control system.

A second controller means C2 is employed, as previously indicated, to vary the pressure supplied to the

chamber T of the computer or transmitter means **27**, as a function of line tension, and, thus, to vary the output signal pressure from the computer according to the above equation for the pressure of the chamber X.

Accordingly, leading from the hydraulic load cell **24** of the tensiometer **15** of the sensing means TP, to the controller means C2 is a conduit **331** through which a hydraulic pressure signal depending upon load or tension of the line L is transmitted to the controller C2 to control the pneumatic pressure supplied to the controller outlet **332** from an inlet **333**. This controller C2, for example, may be the Model **4151** remote set proportional controller or transmitter of the Fisher Governor Company, as illustrated in Bulletin D–4150C.

In the illustrative controller C2 is a hydraulic pressure responsive means in the form of a Bourdon tube **334**, an increase in pressure in which forces a plate **335** toward an exhaust nozzle **336** of the control valve means **337**, and a decrease in pressure in which moves the plate **335** away from the nozzle **336** to vary the pressure in the valve chamber **338** as the sensed hydraulic pressure signal is varied to cause a decrease or increase in the outlet pressure at outlet **333** and in the chamber T of the computer means **27**, whereby the net result is the application of a pressure to clutch C dependent upon the tension on line L.

The control valve means **337** is similar to the control valve means **312** previously described, and includes a diaphragm assembly comprising a diaphragm **339** exposed to pressure in the chamber **338** derived from the inlet **333**, and a diaphragm **340** exposed to pressure in the valve chamber **341** which communicates with the outlet **332**. This diaphragm assembly carries a valve seat **342**, the passage through which communicates with the atmosphere through a port **343**. A combined inlet and outlet valve has a head **344** engageable with the seat **342** to prevent exhaust of pressure from chamber **341** and a head **345** engageable with a seat **346**, the passage through the latter communicating between the inlet **333** and the chamber **341**.

Thus, if pressure in the Bourdon tube **334** is increased, due to an increase in tension on line L, the exhaust of pressure from chamber **338** will be restricted, causing an increase in pressure acting on the diaphragm **339**, so that the valve seat **342** will engage valve head **344**, preventing exhaust of air from the chamber **341** through the exhaust port **343**, and, at the same time, the valve head **345** will be moved off of its seat, allowing an increase in pressure in chamber **341** which tends to return the diaphragm to its original position.

Such increased pressure in the chamber **341** is conducted to an adjustable valve **347**, and thence to a bellows **349** which acts on the plate **335** to move the same away from the nozzle **336** to effectively reduce the pressure in the chamber **338**. Resisting such movement of the plate **335** is a bellows **350** having an inlet **351** to which a set point pressure is supplied from a remote point, such as a regulator valve **352** to which fluid is supplied from a suitable source (not shown), as for example, the same source as supplies reference pressure to the controller C1 and the computer or transmitter means **27**.

Reduction in hydraulic signal pressure from the tensiometer **15** to the controller C2 will cause the plate **335** to move away from the nozzle **334** and a reduction in pressure in the control valve chamber **338**, resulting in opening of the passage through valve seat **342** and reduction in the signal at outlet **332** and computer chamber T. This is to say that in the latter case, the operation of the controller is the reverse of that described above, as will be understood without need for further explanation.

For convenience, a gauge panel G is preferably provided, as seen in FIG. 2*b*, whereby to indicate the effective pressures determined by the position sensing means **16**, the speed responsive means **26**, and the line tension or load responsive means **15**, as well as the reference pressure, the set-point pressure for controller C2, and the ultimate clutch actuating pressure supplied to the slip clutch C, respectively, such gauges being designated by the legends "POS.", "REF.", "SPEED", "LOAD", SET-POINT", and "CLUTCH".

The "POS," gauge is connected to the output of the position sensing means **16** by a conduit **400** which joins with the conduit **85***a* leading to the computer chambers P1 and P2. The "REF." gauge is connected by a conduit **401** to the conduit **305** which leads to both the inlet **304** of controller C1 and the chamber R of the computer **27**. The "SPEED" gauge is connected by a conduit **402** with the conduit **26***a* leading to the chamber S of the computer **27** from the drum speed responsive electro-pneumatic transducer means **26**. The "LOAD" gauge is connected by a conduit **403** and a conduit **404** with the outlet **332** from the load or tension responsive controller means C2 and the chamber T of the computer **27**. The "SET-POINT" gauge is connected by a conduit **405** between the set-point inlet **351** of the load responsive controller C2 and the supply regulator valve **352**. The "CLUTCH" gauge is connected by a conduit **406** with the air inlet connector **75** of the clutch C and the outlet of the selector valve **329**.

From the foregoing, it is believed that the operation of the present invention clearly involves the controlling of the slip clutch drive means for the drum **13** to apply a tension to the line L controlled such that the tension is maintained substantially constant below a value established by the load "SET-POINT" pressure, but the air pressure supply to the clutch is, in the automatic mode, controlled by changes in load or position sensed by the sensing means TP, whereby under added load, the line will be stripped from the drum in a predetermined ratio of feet of line to increased load until the line tension resumes the established value, and the line will then be wound on the drum to reposition the load relative to the towing vessel. The speed and extent of line movement in either direction is sensed by the speed responsive means **25, 26** and the position sensing means **16** to vary the clutch operation to maintain a pre-established tension on the line to an established position at a controlled rate.

Thus, under all conditions encountered by a towed vessel or barge B being towed by a towing vessel V, such as waves and wind or other forces, the barge may be positioned closely to the towing vessel without fear of parting the line L.

If it is assumed that a speed reference signal of 9 p.s.i. is supplied to the computer chamber S when the drum is static, the reference pressure of 5 p.s.i. is supplied to the inlet **304** of controller C1 and to the chamber R of the computing relay **27**, the controller C2 supplies a load or tension reference pressure to the chamber T of 9 p.s.i. at a predetermined line tension, and the position sensor **16** supplies a position signal of 5 p.s.i. to the

chambers P1 and P2 of the computer 27, then the equation representing the computer output pressure is

$X = P1 + P2 - R + S - T$
$X = 5 + 5 - 5 + 9 - 9$
$X = 5$

Under these conditions, the output signal to the inlet 301 of controller C1 from the computing relay is the same as the reference input pressure signal at the inlet 304, and, therefore, the clutch actuating signal pressure transmitted to the booster 326 remains constant, and the line tension caused by the torque transmitted by the clutch C remains constant.

Now, if the load on line L decreases, due to the tendency of the towed vessel and the towing vessel to move towards one another, for example, so that the load signal pressure derived from the load sensor 15 and the controller C2 is reduced to 8 p.s.i., then the equation reads:

$X = P1 + P2 - R + S - T$
$X = 5 + 5 - 5 + 9 - 8$
$X = 6$

Under these conditions, the output signal to the inlet 301 of the controller C1 is 6 p.s.i., but the reference pressure is 5 p.s.i., resulting in an increase in the pressure in the chamber 311 of the control valve 312 and a resultant increase in the pressure in the outlet 302 of the controller C1 and in the pressure applicable to the clutch actuator to tend to wind in the line. Conversely, if the load on the line increases, the clutch actuating signal pressure will be decreased.

Under these conditions, however, line position changes are reflected in the pressure signal derived from the position sensor operated controller C2, so that as the line moves in or out the effective pressure in the chamber P1 and P2 of the computer 27 is varied by the position sensing means 16, and the effective pressure in the chamber S of the computer is varied by the speed of drum movement, resulting in a control system which is capable of maintaining position of the towed vessel, under constant line tension conditions, but enables the line to be stripped from the drum in a predetermined ratio of feet of line to increased load on the line above the set-point. Indeed, the system can be at equilibrium when subjected to loads on the line different than the load which the system is adjusted for by the set-point. The system automatically returns the line to a pre-established position when the line is subjected to the pre-established tension.

I claim:

1. In a winch system including a winch drum having a line thereon connectable to a load, drive means for driving said drum to wind said line on said drum, to hold said line, and to allow said line to be stripped away from said drum, said drive means including a continuously operable slip clutch having continuously variable actuator means operable to adjust the torque transmitted to said drum by said drive means, settable control means for establishing a predetermined line position at a preselected line tension, said control means comprising: line position sensing means responsive to the deviation of said line away from said predetermined line position for producing an output signal proportional to said deviation, line tension sensing means responsive to the tension in aid line for producing an output signal proportional to the deviation of said tension from said preselected tension, said control means including computer means comprising means for receiving both said output signals and producing a continuously variable clutch control signal which is a function of the integration of said output signals, and means for varying said actuator means as a function of said clutch control signal to adjust said actuator means for causing said line to be stripped away from, held on, or returned to said drum in response to said clutch control signal to continuously control the deviation of said line from said predetermined line position in proportion to said deviation of the tension in said line.

2. In a winch system as defined in claim 1, speed sensing means operable by said drum, said control means including means operable by said speed sensing means to adjust said actuator means and control the speed of movement of said drum.

3. In a winch system as defined in claim 1, said actuator means comprising an air pressure responsive device, a source of air connected to said device, and said control means varying the supply of air to said device from said source.

4. In a winch system as defined in claim 1, said control means comprising means for producing a variable air signal pressure to adjust said actuator means.

5. In a winch system as defined in claim 4, speed responsive means operable by said drum to adjust said air signal pressure to limit the speed of movement of said drum.

6. In a winch system as defined in claim 1, said line tension sensing means comprising tensiometer means engaged with said line, said position sensing means engaged with said line, and said control means comprising computer means for receiving said output signals and producing a clutch control signal at a constant value determined by the difference between said output signals.

7. In a winch system as defined in claim 6, speed responsive means to produce an output signal determined by the speed of movement of said line, said computer means receiving said output signal of said speed responsive means and being operable to produce a clutch control signal determined by the difference between the sum of said output signal of said position sensing means and said output signals of said speed responsive means, and the output signal of said tensiometer means.

8. In a winch system as defined in claim 1, said line tension sensing means comprising tensiometer means engaged with said line, said position sensing means engaged with said line, said control means comprising computer means for receiving said output signals and producing a clutch control signal at a constant value determined by the difference between said output signals, and speed responsive means operable upon winding of line on said drum and stripping of line from said drum to produce a speed responsive output signal, said computer means receiving the latter output signal and adding the same to said position control signal in producing said clutch control signal.

9. In a winch system as defined in claim 1, said line tension sensing means comprising tensiometer means engaged with said line, said position sensing means engaged with said line, and said control means comprising computer means for receiving said output signals and producing a clutch control signal determined by the difference between said output signals, said control means including a pressure controller operable in re-

sponse to said output signal of said tensiometer means to maintain said output signal determined by the load on said line at a constant value.

10. In a winch system as defined in claim **1**, said line tension sensing means comprising tensiometer means engaged with said line, said position sensing means engaged with said line, and said control means comprising computer means for receiving said output signals and producing a clutch control signal determined by the difference between said output signals, said control means including a pressure controller operable in response to said output signal of said tensiometer means to maintain said output signal determined by the load on said line at a constant value, and speed responsive means operable upon winding and unwinding of line on said drum to produce a speed responsive output signal, said computer means receiving the latter output signal, said speed responsive output signal being added to said output signal from said position sensing means by said computer means in producing said clutch control signal.

11. In a winch system as defined in claim **1**, said actuator means comprising an air pressure responsive device, a source of air connected to said device, said control means varying the supply of air to said device from said source, said control means comprising a pressure computer for receiving variable input pressure signals representative of load on said line, the deviation of said line, and a reference pressure, and said computer producing an output pressure signal determined by the equation

$$X = 2P - R - T,$$

where X is the computer output signal pressure, P is the input pressure signal determined by the deviation of said line, R is the reference input pressure signal, and T is the input signal pressure determined by load on said line, said tension sensing means sensing the load on said line and said position sensing means sensing the deviation of said line and supplying said variable input pressure signals to said computer means representative of load on said line and the position of said line, and means for supplying a selected reference signal pressure to said computer means.

12. In a winch system as defined in claim **11**, said computer means comprising a control valve assembly having an air inlet, an air outlet, and an exhaust port, and valve means for controlling the flow of air from said inlet to said outlet and said exhaust port to maintain said output pressure signal substantially constant at a selected pressure.

13. In a winch system as defined in claim **1**, said clutch actuator means comprising an air pressure responsive device, a source of air connected to said device, said control means varying the supply of air to said device from said source, said control means comprising a pressure computer for receiving variable input pressure signals representative of load on said line, the deviation of said line, and a reference pressure, and said computer producing an output pressure signal determined by the equation

$$X = 2P - R - T$$

where X is the computer output signal pressure, P is the input pressure signal determined by the deviation of said line, R is the reference input pressure signal, and T is the input signal pressure determined by load on said line, said tension sensing means sensing the load on said line and said position sensing means sensing the deviation of said line and supplying said variable input pressure signals to said computer means representative of load on said line and the position of said line, means for supplying a selected reference signal pressure to said computer means, and controller means for receiving the computer output signal and variably transmitting a constant output signal to control said clutch of a magnitude determined by said computer output signal.

14. In a winch system as defined in claim **1**, said clutch actuator means being pneumatically operable, said line tension sensing means comprising hydraulic load sensing means for producing a hydraulic pressure signal determined by the load on said line, controller means having an air inlet, an air outlet, and control valve means operable by said hydraulic pressure signal to cause a constant output pressure signal at said outlet, said sensing means also comprising pneumatic sensing means operable in response to deviation of said line and including an air inlet, an air outlet, and control valve means operable to cause a constant output pressure signal at the latter outlet, additional controller means having an air inlet, an air outlet, and control valve means operable to cause a constant output pressure signal at the last-mentioned outlet, pneumatic computer means having an air inlet, an air outlet and control valve means for maintaining a constant output pressure signal at the outlet of said computer means at a pressure determined by a comparison of said output pressure signals from said controllers, and means for supplying air to said clutch actuator means at a pressure determined by said output pressure signal from said computer means.

15. In a winch system as defined in claim **14**, speed responsive means operable in response to rotation of said drum to produce an air pressure signal determined by the speed of said drum, said computer means receiving the last-mentioned pressure signal to adjust the output pressure signal from said computer means.

* * * * *

UNITED STATES PATENT OFFICE

CERTIFICATE OF CORRECTION

Patent No. 3,801,071 Dated April 2, 1974

Inventor(s) Charles D. Barron

It is certified that error appears in the above-identified patent and that said Letters Patent are hereby corrected as shown below:

Column 1, line 7, "to" should be --at--.
Column 1, line 40, "played" should be --payed--.
Column 2, line 18, after "1970" insert --, now U.S. Patent No. 3,648,814, issued March 14, 1972--.
Column 6, line 6, "117" should be --107--.
Column 6, line 7, "107" should be --115--.
Column 10, line 5, "X" should be --234--.
Column 11, line 18, "145a" should be --245a--.
Column 13, line 23, "333" should be --332--.

Signed and Sealed this

Tenth Day of *October 1978*

[SEAL]

Attest:

RUTH C. MASON
Attesting Officer

DONALD W. BANNER
Commissioner of Patents and Trademarks

UNITED STATES PATENT OFFICE

CERTIFICATE OF CORRECTION

Patent No. 3,801,071 Dated April 2, 1974

Inventor(s) Charles D. Barron

It is certified that error appears in the above-identified patent and that said Letters Patent are hereby corrected as shown below:

Front Page, Assignee: "Byran Jackson, Inc." should read -- Byron Jackson Inc., Long Beach, Calif. --.

Signed and Sealed this

Fifteenth Day of February 1977

[SEAL]

Attest:

RUTH C. MASON
Attesting Officer

C. MARSHALL DANN
Commissioner of Patents and Trademarks