April 11, 1967 W. FISCHER 3,313,345

METHOD AND APPARATUS FOR OFFSHORE DRILLING AND WELL COMPLETION

Filed June 2, 1964 4 Sheets-Sheet 1

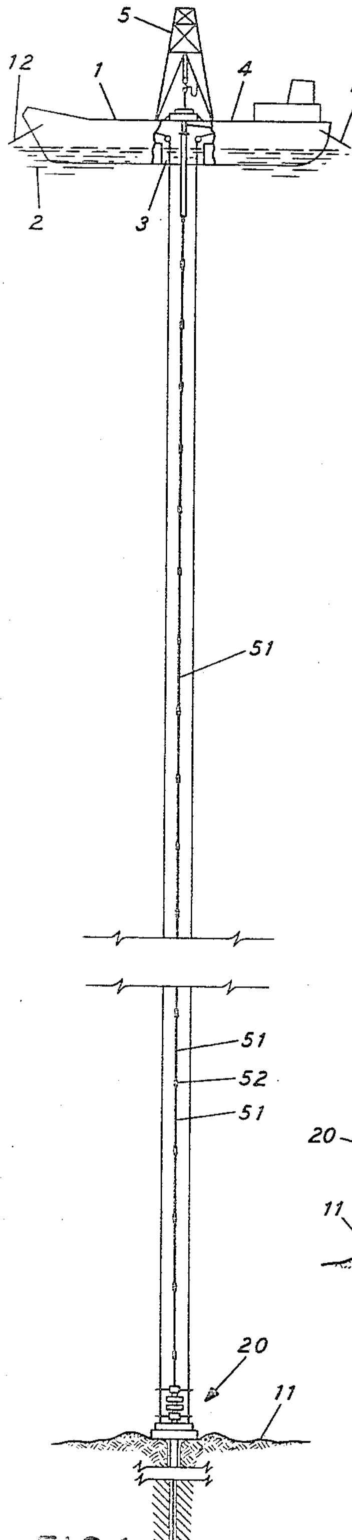

FIG. 1

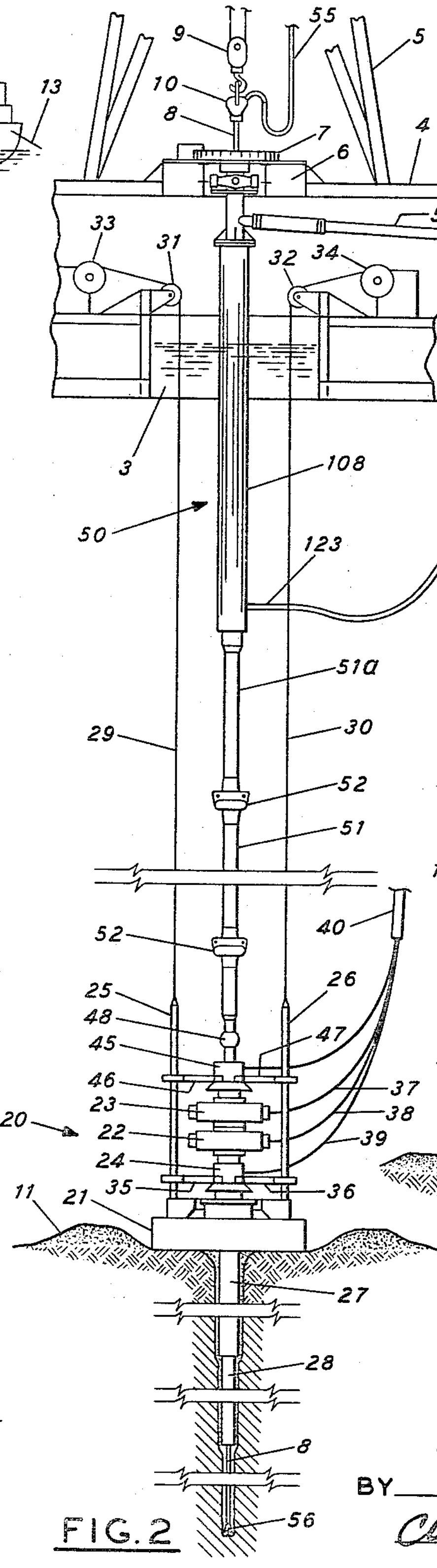

FIG. 2

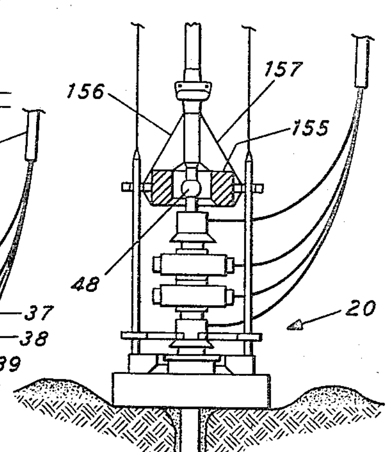

FIG. 2A

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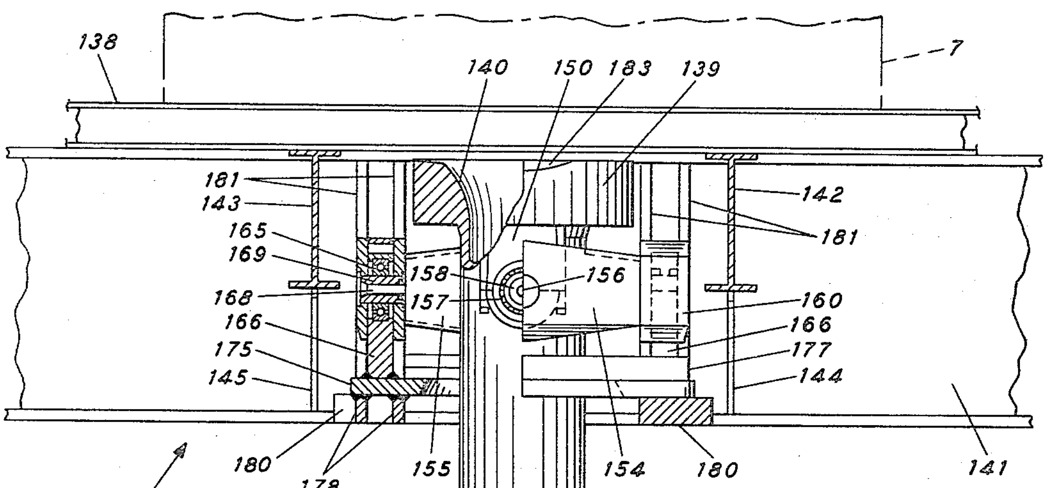

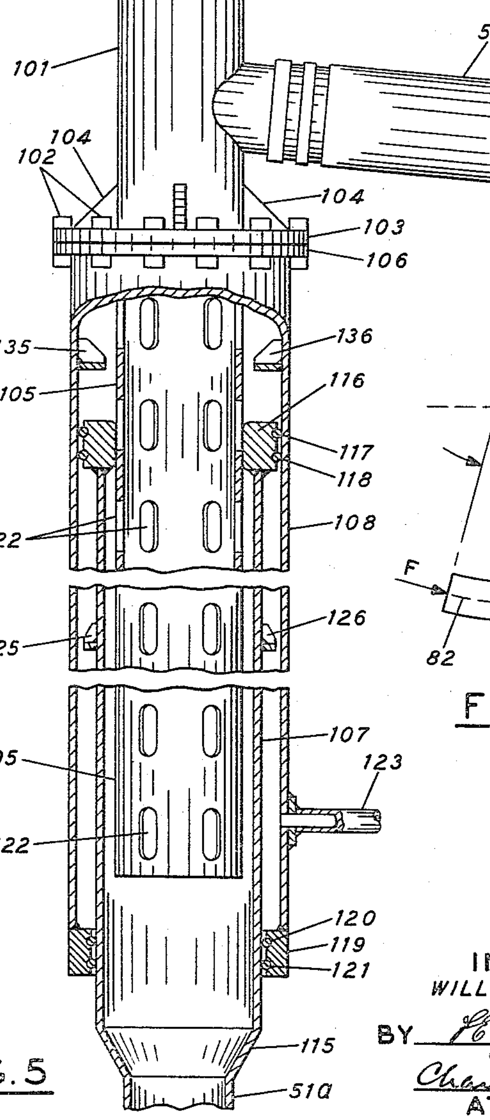

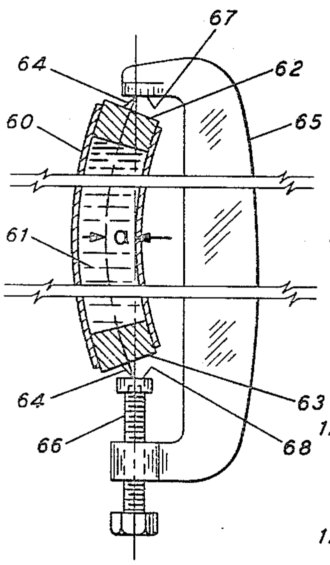

FIG. 4

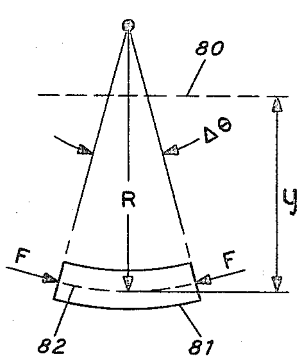

FIG. 3

FIG. 5

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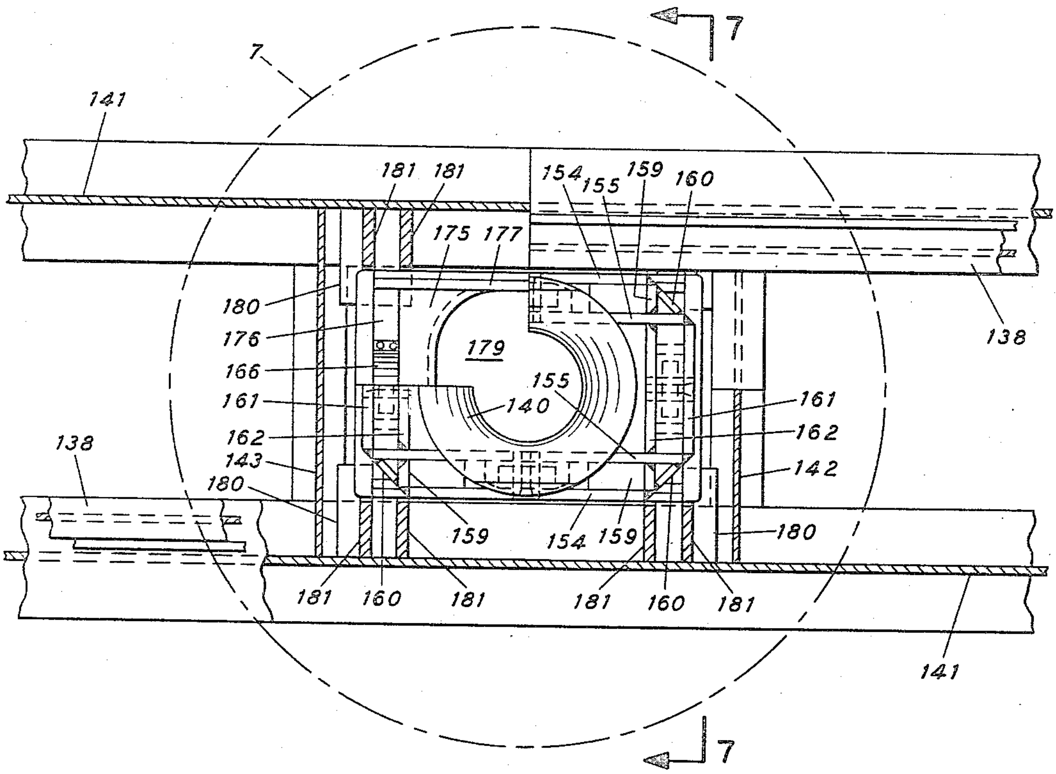

FIG. 6

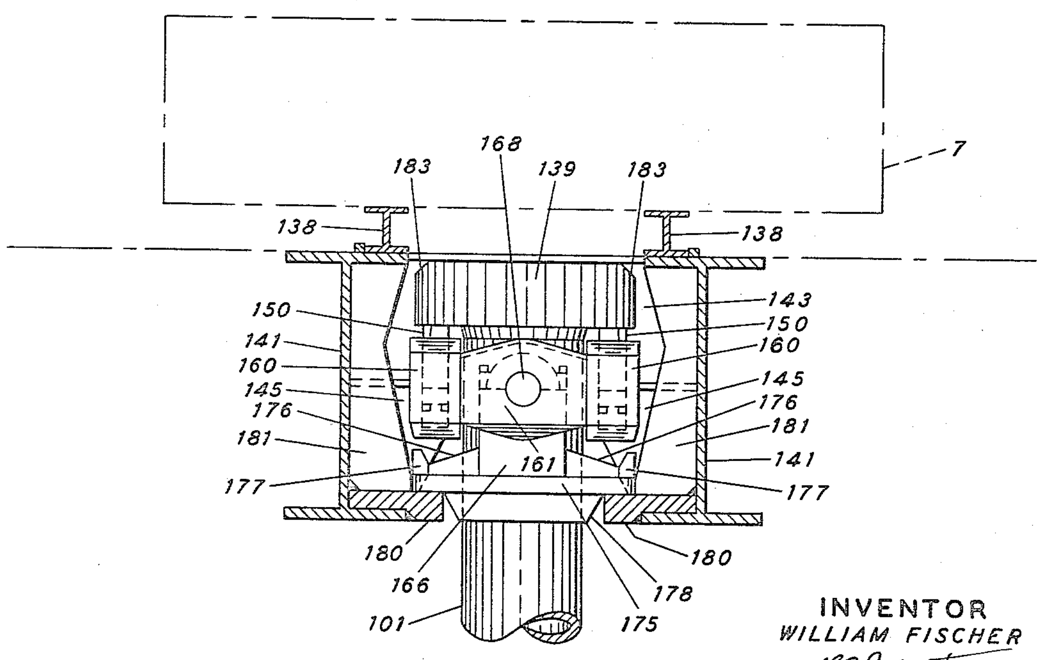

FIG. 7

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METHOD AND APPARATUS FOR OFFSHORE DRILLING AND WELL COMPLETION

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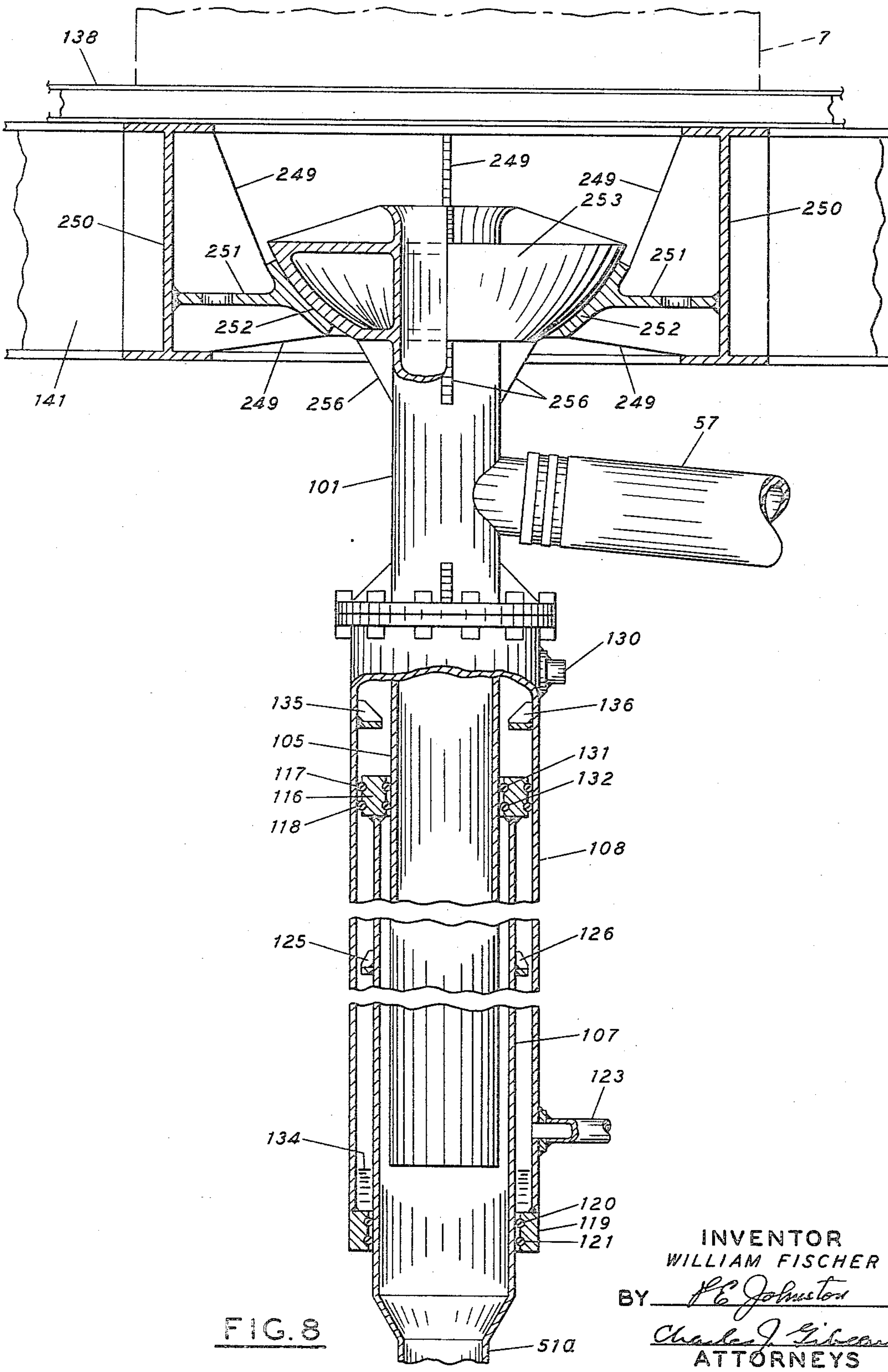

FIG. 8

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METHOD AND APPARATUS FOR OFFSHORE DRILLING AND WELL COMPLETION

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Filed June 2, 1964, Ser. No. 372,061
6 Claims. (Cl. 166—.5)

This invention relates to drilling in earth formations located beneath a body of water, particularly deep, open water such as an ocean. More specifically, this invention relates to an apparatus for use in offshore drilling and well completion to prevent bending and buckling of an elongated tubular member which extends from the ocean floor to a drilling platform at the ocean surface.

In recent years it has become desirable in some circumstances to conduct offshore drilling operations from a floating vessel rather than from a rigid structure or a platform supported from the ocean bottom. In such operations the floating vessel is sometimes connected to the well bore in the submerged formation by a long tubular member through which drilling and well working tools, drilling fluid, etc., pass between the vessel and the well bore. This long tubular member is often called a marine conductor or riser.

In one procedure for offshore drilling the lower end of the riser is connected to a wellhead, which includes blowout preventers and control equipment, while the upper end of the riser is connected to the drilling vessel. The wellhead is designed to remain stationary on the ocean floor. The drilling vessel, on the other hand, is continuously moving under the action of tides, currents, waves and wind. Movement of the vessel is somewhat restricted by anchoring and by special positioning systems. But, the violent and constantly changing forces acting on the vessel often shift it from its position over the well bore by lateral distances in excess of four percent of the water depth. Vertical heaving movements due to wave action can be expected in addition to the relatively slower vertical movement effected by tides. These heaving movements may be very substantial. One accepted method of allowing for the relative lateral and vertical movement between the vessel and the wellhead is to place laterally flexible joints and telescopic joints in the riser. This permits the riser to accommodate the movement of the vessel within design limits.

As drilling operations were carried on in deeper water, it was found that the riser itself tended to deflect or bend along its length between the flexible joints. This bending has often been so great that structural failure of the riser column has resulted. Even when failure does not occur, bending is undesirable if it results in a sharp angle in the riser at any point along its length, thereby causing difficulty in passing drill pipe, well casing, etc. This problem is particularly acute where the flexible or universal joint must be included in the riser near its lower end to allow for vessel movement. Since bending moments are not transmitted through such a joint, any bending of the riser tends to create a sharp angle at this joint. When using a ball and socket joint, for example, an angle greater than five degrees at the joint makes it very difficult to pass the drill string through the joint. The tendency toward angularity at the lower universal joint is accentuated by vessel movement laterally from its position over the wellhead.

In analyzing the causes of the extreme lateral deflection of the riser, it is appropriate to consider the riser as a long tubular column. When drilling in deep water, the length of this column becomes so great as compared to the moment of inertia of its cross sectional area that the riser has virtually no column stiffness. Thus any axial

compression or lateral forces will cause severe deflection or bending of the riser.

If the riser is supported solely at its lower end, its own effective weight, i.e., weight in water, causes it to be in axial compression, increasing from zero at the top to a maximum at the bottom support. When drilling in deep water the compressive stress in the riser wall from this source alone is sufficient to buckle the riser. Friction in the telescopic joint as the vessel moves vertically toward the ocean bottom as well as drilling or well working tools which may bind on and be supported by the riser as they are passed through it add compressive stress to the riser wall and thereby increase the deflection of the riser.

In addition to the buckling effect of the above-mentioned axial compressive forces which create net compressive stress in the riser wall, current and wave forces acting transversely along the length of the riser tend to bend it. Although wave forces decrease exponentially with depth below the surface, recent submarine measurements have shown that they extend deep enough to be a substantial factor in bending the riser. Also strong current forces have been found at substantial depths. For convenience in discussion these current and wave forces will be called environmental forces. It should be noted that since they act generally transverse to the riser length, they do not create a net axial compression in the riser wall at any cross-section but create a tensile force in one diametrical half of the section which is equal and opposite to the compressive force in the other half.

The types and magnitudes of the various forces, such as those discussed above, which create net axial compression in the riser wall as well as the environmental forces which act generally transverse to the riser length vary under different operating conditions. The prior art recognizes the existence of these forces and has sought to overcome their effect by applying enough axial tension to the riser to compensate for the weight of it and to counteract the bending caused by the transverse environmental forces.

In water depths up to approximately 250 feet satisfactory results have usually been obtained by applying tension sufficient only to compensate for the types of forces discussed above. However, in greater depths buckling and excessive bending still occur when this method is employed. Failure to determine the cause of this has resulted in large financial losses.

It has now been discovered that the buckling that has been repeatedly experienced with deep water risers is a result of the high density of the drilling fluid in the riser. Although the difference in density between the drilling fluid and the surrounding sea water does not cause a net compressive force to be applied directly to the riser, it has a buckling effect on the riser. The proof of this, both analytical and experimental, will be discussed below in the detailed description of this invention.

It also will be proven that in a long riser transmitting dense drilling fluid, this effect may be compensated for by applying to the riser, in addition to the tensile forces applied to compensate for axial compressive forces acting directly on the riser and the transverse environmental forces as shown by the prior art, an extra tensile force in an amount calculated in accordance with formulas developed in this specification to compensate for the bending effect of the dense drilling fluid.

When drilling in deep water, the magnitude of the tensile forces required to obtain the desired results becomes extremely large. The apparatus of this invention is capable of applying tensile forces of large magnitude to the riser in a manner such that the amount of tension applied remains constant regardless of movement of the vessel relative to the wellhead. The large upward force is applied to the riser by the apparatus of this invention in a manner such that the upward force will be dissipated in the event of structural failure of the riser. Thus the riser will not shoot from the water endangering the safety of nearby personnel. The apparatus of this invention also provides for adjustment of the magnitude of the tension applied to the riser by personnel on the drilling vessel as operating conditions and the magnitude of tension required vary.

More specifically, the apparatus of this invention includes a piston and cylinder arrangement interposed in the riser to apply longitudinal force to the riser and at the same time to serve as a telescopic joint to permit relative motion between the drilling vessel and the ocean floor, which are connected together by the riser.

In a preferred embodiment of this apparatus the riser extends from a connection to the wellhead at the lower end of the riser up to a point near the water surface. A piston is formed at the upper end of this portion of the riser. A second portion of the riser is connected at its upper end to the drilling vessel, and extends downward beneath the water surface. The lower end of the second portion of the riser forms a cylinder which surrounds the piston formed at the upper end of the first portion. High pressure fluid is applied to the cylinder below the piston from a source on the drilling vessel. The high pressure fluid exerts an upward force on the piston, and thereby on the upper end of the first portion of the riser. This upward force is counterbalanced by a downward force applied to the lower end of the riser either by the wellhead equipment or by a large mass connected to the lower end of the riser.

As the vessel moves relative to the wellhead the piston moves relative to the cylinder to act as a telescopic joint. The fluid pressure source maintains the fluid pressure at a substantially constant magnitude regardless of the position of the piston relative to the cylinder, and therefore the amount of tension applied to the riser is substantially independent of vessel movement. The fluid pressure is adjustable so that the amount of tension applied may be varied as required.

In case of failure of the riser below the piston and cylinder but above the point of application of the lower downward force, the piston will move to the upper end of the cylinder where it will be stopped and the piston and cylinder device will be in equilibrium. Thus in case of riser failure the portion of the riser above the point of failure will merely be suspended from the vessel in equilibrium and will present no hazardous condition.

This invention also includes novel means for supporting the upper end of the riser from the vessel. The support means is designed to permit free pivotal movement of the vessel relative to the riser as the vessel pitches and rolls. In one embodiment the support includes a gimbal with two horizontal pivotal axes at right angles to each other. In another embodiment the support includes a spherical ball which seats in a spherical bearing socket.

The operation and advantages of this invention will be clear from the following detailed description. Throughout this description reference will be made to the accompanying drawings in which:

FIGURE 1 is an overall elevation view of the apparatus of this invention in operating position.

FIGURE 2 is an enlarged view of portions of FIGURE 1.

FIGURE 2*a* is an alternative arrangement for the lower end of the riser.

FIGURE 3 is a schematic diagram of the effect of axial compression on a beam section.

FIGURE 4 is a schematic illustration of the effect of axial compression on a confined fluid.

FIGURE 5 is an elevation view, partly in section, of the tensioner and of the gimbal support for the upper end of the riser.

FIGURE 6 is a plan view of the gimbal support of FIGURE 5.

FIGURE 7 is a sectional view along lines 7—7 of FIGURE 6.

FIGURE 8 is an elevational view similar to FIGURE 5 illustrating a modification of the tensioner and of the top support.

Referring to FIGURES 1 and 2, vessel **1** is floating on a body of water **2** such as an ocean. The vessel includes a vertical opening **3** through its hull near the longitudinal and transverse center of the vessel. Supported on the upper deck **4** of the vessel and approximately centered over the opening **3** is a derrick structure **5** from which the upper end of drill pipe **8** is supported by a traveling block **9** and swivel **10**. The derrick structure and much of the associated equipment are of a type commonly used in offshore rotary drilling and are not shown in detail. Approximately centered in the base of the derrick are a support platform **6** and rotary table **7**. The drill pipe **8** passes vertically through aligned openings in the platform and rotary table and is rotated by the rotary table **7** in a standard manner. Anchors connected to anchor chains **12** and **13** limit the movement of the vessel from its normal position over the well.

A wellhead **20** is located on the submerged formation **11** in which the hole is being drilled. The wellhead includes a base **21**, and stacked blowout preventers **22** and **23** which are releasably connected to the base by coupling **24**. Several lengths of well casing **27** and **28** extend beneath the wellhead into the well. At least two guideposts **25** and **26** extend vertically from spaced points near the circumference of base **21**. Guide lines **29** and **30** extending from guideposts **25** and **26**, respectively, to the vessel **1** are used to guide equipment as it is lowered from the vessel to the wellhead. These guide lines extend upward through the opening **3** in the bottom of the vessel, over pulleys **31** and **32**, and are connected at their upper ends to tensioning devices such as constant tension winches **33** and **34** respectively. These winches maintain the guide lines taut in spite of relative movement between vessel **1** and wellhead **20**. Guide arms **35** and **36** extend outwardly from coupling **24** and include guide sleeves near their outer ends. After the base **21** with its guideposts **25** and **26** and guide lines **29** and **30** are in place, the blowout preventer assembly including coupling **24** may be lowered from the vessel with a guide line passing through each guide sleeve to position the assembly as shown. The coupling and blowout preventers are remotely actuable from the vessel by fluid under pressure from lines **37**, **38**, and **39** which extend to the vessel through tubing **40**. Alternatively steel lines running along the riser can be used for actuation.

With the wellhead in place on the ocean bottom the riser may be lowered to connect the floating vessel to the wellhead. This riser is releasably connected at its lower end to the wellhead by coupling **45** and is connected at its upper end to the vessel in a manner which will be described below. The riser includes a ball and socket joint **48** near its lower end to reduce the torsional stress transmitted to the wellhead as the vessel shifts laterally from its position over the well under the action of wind, tides, waves, and currents. The heavy coupler **45** is pendant below the universal ball joint **48**. Guide arms **46** and **47** are aligned on the center of **48** so that the connection at coupler **45** is assured and is independent of the angularity of the riser.

If the blowout preventer assembly is to be lowered into place at the same time as the riser, the guide arms **35** and **36** may be omitted.

The major portion of the riser consists of a series of elongated tubular members **51** connected end to end by couplings **52** to extend from telescopic joint **50** to ball and socket joint **48**. High density drilling fluid is fed to the well from a pump on the vessel. The drilling fluid passes through flexible hose **55**, down through a passageway in the axis of the drill pipe, and out bit **56** at the bottom of the well. The drilling fluid is returned to the sump by passing upwardly around the outside of the drill pipe through well casings **28** and **27**, through wellhead **20** and then upward through the riser. The drilling fluid is returned to the mud system on the vessel through pipe **57** which is flexibly connected to the riser near its upper end.

As was stated in the introduction to this specification, it has now been determined that the density of the drilling fluid tends to cause the riser to deflect. An analysis of the phenomenon reveals that the axial force due to fluid pressure inside a cylindrical tube has exactly the same effect on lateral deflection, or on buckling, as would an equal axial compressive force in the pipe wall. This can be demonstrated by development of the differential equation commonly used for calculation of the bending and buckling of a beam due to axial loads whereby it will be shown that the bending effect is the same whether a given axial force is applied in the internal fluid or in the pipe wall. The following notations will be used in this derivation:

x=distance along the axis of the beam in inches from the nearest support
Δx=length of a small increment of the beam in inches
y=lateral deflection of the beam in inches
$\Delta\theta$=small angle along arc of slightly curved beam in radians
q=lateral force per unit of beam length in pounds per inch
F=axial compressive force in pounds
R=radius of curvature of the bent beam in inches.

For small deflections,

$$\frac{1}{R}=-\frac{d^2y}{dx^2} \tag{1}$$

as is demonstrated in textbooks such as Timoshenko, Strength of Materials, 2nd ed., vol. 1, p. 135 (1940). The fundamental deflection equation is

$$EI\frac{d^4y}{dy^4}=q \tag{2}$$

as is also shown in Timoshenko, supra, at p. 137.

Referring now to FIGURE 3, **80** is the undeflected position of the axis of beam portion **81**. The axis has been deflected to position **82** under the axial compressive forces F.

The vertical component of forces F is $2F \sin(\Delta\theta/2)$ which for small angles is $F\Delta\theta$. Since q is defined as the lateral force per unit of beam length

$$q\Delta x=F\Delta\theta=F\frac{\Delta x}{R} \tag{3}$$

Therefore

$$q=\frac{F}{R} \tag{4}$$

Combining Equations 1, 2, and 4

$$EI\frac{d^4y}{dx^4}=-F\frac{d^2y}{dx^2} \tag{5}$$

Integrating this twice and assuming hinged ends (moment=0 at each end)

$$EI\frac{d^2y}{dx^2}=-Fy \tag{6}$$

This is the well known differential equation for calculation of the axial deflection force for a beam (see Timoshenko, supra, vol. 2, p. 25).

It should be noted that this derivation made no assumption that the beam was solid, and if the beam is a tubular pipe the derivation applies equally well whether the axial force is in the internal fluid or in the pipe wall. Thus it may be concluded that it is the algebraic sum of the axial forces applied to the internal fluid and to the pipe wall that must be considered in calculating the pipe deflection. Thus buckling can occur with a high fluid pressure even though the axial force in the pipe wall is tensile if the compressive force in the fluid exceeds the tensile force in the pipe wall so that the net axial force is compressive.

A less analytical approach has sometimes been helpful in showing that axial forces applied to a confined liquid tend to deflect the confining pipe. Referring to the example in FIGURE 4 there is shown a longitudinal cross-section through a length of tubular steel pipe **60** filled with a liquid **61**. The pipe is extremely long in comparison with its diameter so that it acts as a long column with a large slenderness ratio. Portions of its length have been omitted for convenience in the drawings. The ends of the pipe **60** are sealed by slidable pistons **62** and **63** which are free to move axially along the pipe **60**. Each piston includes a pivot bearing **64** on its outer face. A clamp **65** including a screw member **66** is designed to exert pressure between stationary face **67** of the clamp and movable face **68** of the screw. The pressure is transmitted through the pistons **62** and **63** to the fluid **61**, which is substantially incompressible. Hence the effect is similar to placing end loading on a solid column. A sufficiently great force placed on bearing **64** by tightening of the screw **66** will cause the pipe **60** to buckle from its original straight position. The deflection of the pipe is indicated by the letter a. It is apparent that as the slenderness ratio of the pipe **60** becomes greater, less force is required to buckle the pipe since it acts as a long column. Thus it can be seen that compressive force applied to a liquid confined in an elongated tubular member causes flexing and buckling of the tubular member in the same manner as compressive forces applied to a solid long column.

Again referring to FIGURE 3, the value of lateral component of axial force

$$F\frac{\Delta x}{R}$$

as set forth in Equation 3, above, can be obtained by integrating the lateral component of fluid pressure over the length Δx of the column. Since the radially outward arc length is greater than the radially inward arc length in FIGURE 3, the net force would be radially outward. Similarly fluid pressure outside of the pipe will produce a net radially inward force.

The net effective axial compressive force for consideration of lateral deflection, and buckling, at any section of a pipe such as shown in FIGURE 3 is then:

$$F=A_1P_1-A_2P_2+(A_2-A_1)S \tag{7}$$

in which:

A_1=internal cross-sectional area of pipe in square inches
A_2=external cross-sectional area of pipe in square inches
P_1=internal fluid pressure in pounds per square inch
P_2=external fluid pressure in pounds per square inch
S=average compressive stress in the pipe wall section in pounds per square inch

In applying Equation 7 to the riser of FIGURES 1 and 2, the riser is considered to be a long tubular column laterally unsupported between its ends. P_2, the pressure of the sea water, will be zero at the water surface and will increase linearly with water depth to a maximum at the ocean bottom. P_1, the pressure of the drilling fluid, will be equal to some substantially constant value at the top of the riser due to pump pressure and friction in the return pipe **57**, and will increase linearly with depth to a maximum at the bottom of the well bore. The average compressive stress S across any transverse section of the riser wall will include the net stress in the riser from all sources such as vessel movement, equipment bearing on the riser, and the weight of the riser itself. From these quantities the net axial compressive force F acting on any section through the riser may be calculated by use of Equation 7.

The above analysis has been repeatedly substantiated by actual instances of riser failure where the effect of the drilling fluid was not accounted for. Also experiments have been conducted using a small scale model of a riser. In these experiments a long steel tube of small diameter was supported only at its top (to remove any compressive stress from the weight of the tube itself) and filled with mercury. The weight of the mercury was supported by a plug at the bottom. The plug was slidable axially along the tube. Under these conditions the loading of the mercury imposed on the tube was similar to that of drilling fluid on a riser. The deflection of the tube under such loading was completely consistent with the above analysis.

Practical application of the above analysis to a riser in offshore drilling requires consideration of the conditions to which the riser is subjected. As was pointed out above, the wave and current forces as well as the forces created by vessel movement are continuously varying within large parameters. While the magnitude of these forces may be calculated and determined experimentally, any such determination is necessarily only an approximation due to the naturally varying and complex nature of these forces. Therefore it is desirable to add a safety factor in the form of extra tension.

Another practical consideration is that in offshore drilling operations in deep water, the riser is extremely long in comparison with the moment of inertia of its cross-section so that it may be considered to have virtually no column stiffness. Thus, even in the absence of lateral environmental forces, it will not withstand any substantial net axial compressive force throughout a substantial portion of its length without undesirably large deflection and possible structural failure. While in the absence of lateral environmental forces, the riser could theoretically withstand some axial compressive force over relatively short increments of its length, for practical purposes, and to insure safe operation, it is preferred to assume that the net axial compressive force F in Equation 7 must equal zero or preferably be negative, i.e., tensile rather than compressive, throughout the length of the riser, and then to add an overpull, or additional tensile force, to limit the deflection from the environmental forces.

For convenience it may be preferable to express Equation 7 as it applies to risers in offshore drilling as: the net axial compressive force acting on any transverse cross-section is equal to the weight in water of the column of drilling fluid above that section, plus the weight in water of the riser column above that section, plus any other axial compressive forces applied directly to the riser above that section. This relationship is based on the assumption that the pressure of the drilling fluid is zero at the top of the column and that its pressure increases linearly with depth in proportion to its weight. While not precisely true in all cases this form of the equation is generally adequate to arrive at a practical determination of the forces involved in a riser used for offshore drilling.

Still another force which may cause buckling of the riser during drilling and well working operations is created by compressive stress incurred by material such as well casing which is being passed axially through the riser. If the well casing becomes bound against the interior wall of the riser so that the well casing weight is supported on the riser, the compressive stress S in the riser wall will be increased. If, instead, the well casing becomes supported from below, such as against the sides of the well bore, the well casing will develop compressive stress in its own walls. The well casing, if of great length, will have virtually no column stiffness. Thus the transverse force of the buckled well casing will be imposed on the wall of the riser. A drill pipe in axial compression would have a similar effect. In order to design for the worst conditions which may be encountered, it is desirable to incorporate these forces in the design. This may be accounted for by revising Equation 7 to:

$$F=A_3P_4+(A_1-A_4)P_3+(A_2-A_1)S+(A_4-A_3)S_2-A_2P_2 \tag{8}$$

in which:

A_3=internal cross-sectional area of the well casing or drill pipe in square inches.

A_4=external cross-sectional area of the well casing or drill pipe in square inches.

P_3=pressure of fluid in space between interior wall of riser and exterior wall of well casing or drill pipe in pounds per square inch.

P_4=pressure of fluid inside of well casing or drill pipe in pounds per square inch.

S_2=average compressive stress in the well casing or drill pipe in pounds per square inch.

The other symbols are the same as in Equation 7.

Other varying conditions may similarly be compensated for by varying the basic Equation 7 in accordance with the above teachings. The generic formula may be conveniently stated as: the net axial compressive force tending to bend or buckle the riser at any transverse cross section is equal to the algebraic sum of all the axial forces acting on the walls of the riser at that section and all the axial forces acting on all materials within the riser at that section minus the product of pressure of the sea water outside that section times the external area of the riser.

As pointed out heretofore, it is undesirable to have a sharp angle at the ball joint located at the bottom of the riser. As the vessel moves laterally from a position over the wellhead, the weight of the materials contained in the riser accentuates this angle because the weight develops a moment about the ball joint. This angle can be controlled by the amount of overpull added to the riser.

The relationship between the amount of overpull and the angle at any point along the riser may be determined from

$$EI\frac{d^4y}{dx^4}=q+w\frac{dy}{dx}+F\frac{d^2y}{dx^2} \tag{9}$$

wherein

E=the modulus of elasticity for the riser, in pounds per square inch

I=the cross-sectional moment of inertia for the riser in inches fourth

x=the vertical distance from the bottom of the riser in inches

y=the lateral offset of the riser from vertical in inches

dy/dx=tangent of angle between the riser and the vertical

q=the lateral load in pounds per inch of length

w=the vertical component of the change of tension in pounds per inch of riser length

F=the vertical component of net axial force as determined from Equation 7 or 8.

Since the lateral movement of the vessel is small relative to the length of the riser, the axial component of the vertical force F is substantially equal to the vertical force F and may be assumed to be the same.

As a specific example of application of the above analysis to a riser, it will be assumed that operations are to be carried on in water 600 feet deep with a 13⅜" O.D. x ⅜" pipe as the riser, and drilling fluid weighing 120 pounds per cubic foot being the only material in the riser.

The maximum force F from Equation 7 will occur at the lower end of the riser. It will be approximately equal to the weight in water of the entire riser column (27,100 pounds), plus the weight in water of the entire volume of drilling fluid contained in the riser (29,200 pounds), giving a total F of 56,300 pounds. Thus an upward force in excess of this amount is required.

If the operations are to be carried on from a floating vessel which will be displaced laterally from a position over the wellhead, for example, by twenty-four feet, and assuming normal current forces, the angle of the riser at the ball joint can be limited to five degrees by an overpull of 20,300 pounds (Equation 9). This overpull is an upward force in addition to the 56,300 pounds required to reduce F to zero. Further overpull can be added as a safety factor and/or to account for friction in the telescopic joint.

In view of the magnitude of the upward force imposed on the riser to achieve the result described above, a counteracting downward force is necessary to prevent the riser from being pulled out of the water. The weight in water of the riser opposes the upward force but is insufficient to completely counteract it. The wellhead and associated equipment may be relied on to either completely or partially counteract the upward force, either by its weight, or by weights added to the wellhead, or by anchoring the wellhead or the well casing which is suspended from it to the submerged formation. Another manner of providing part or all of the downward force is by connecting a large weighted mass or counterweight to the bottom end of the riser.

In applying upward force of the large magnitude required to maintain the necessary tension in the riser wall, many problems arise. The force must be substantially constant even while the vessel is moving vertically relative to the ocean bottom at a rapid rate. It is undesirable to attach bulky mechanisms to the submerged portions of the riser, both because of the difficulty in handling them and because of the increased area subjected to wave and current forces.

Another very substantial problem is the safety hazard to the nearby person and equipment which may result upon failure of the riser. Thus if it breaks at a point between the location where the upward force is applied and that where the downward force is applied, the upward force may accelerate the upper fragment of the riser out of the water at a rapid rate. For this reason it is essential that the upward force be quickly dissipated upon failure of the riser.

To overcome these problems the apparatus of this invention uses fluid pressure to provide the upward force. The preferred embodiment of this invention applies the fluid pressure to a piston and cylinder combination which is incorporated in the riser and forms a part of it. The piston and cylinder arrangement also serves as a telescopic joint to permit elongation and contraction of the riser as the vessel moves relative to the ocean bottom.

The combination telescopic joint and tensioner, designated generally as **50** in FIGURES 1 and 2, will be described with particular reference to FIGURE 5.

The upper portion of the riser includes a tubular pitcher nipple **101** which is connected to the vessel below rotary table **7** in a manner which will be described later. A flange **103** is rigidly connected at the lower end of pitcher nipple **101** and is braced thereto by members **104**. Tubular member **105** extends below pitcher nipple **101** and is aligned therewith to form a continuous conduit. Flange **106** at the upper end of tubular member **105** is connected to flange **103** in any suitable manner such as by bolts **102**.

The uppermost one of the tubular conductor members **51** is designated **51*a***. It includes an enlarged diameter portion **107** at its upper end above transition **115**. This enlarged diameter portion **107** receives tubular member **105** in telescopic relationship. Since tubular member **105** will move with the vessel while enlarged diameter portion **107** is axially fixed relative to the ocean bottom, the overlap between members **105** and **107** is such that they will form a continuous conduit at all positions of the vessel relative to the wellhead within selected design limits.

Cylinder **108** surrounds tubular member **105** and the overlapped portion of **107**. It is connected at its upper end to flange **106**. Flange **106** thus serves as a cap to seal the upper end of the space between tubular member **105** and cylinder **108**. The lower end of cylinder **108** is sealed by cap **119**. Rings **120** and **121** provide a sliding seal between enlarged diameter portion **107** of upper conductor member **51*a*** and cap **119**. An annular piston **116** is connected to the upper end of enlarged diameter portion **107** and surrounds tubular member **105** in axially sliding relationship. Rings **117** and **118** provide a sliding seal between piston **116** and cylinder **108**.

High pressure fluid, either liquid or gas, is supplied to cylinder **108** near its lower end through conduit **123** from a source on the drilling vessel. This high pressure fluid acts on the lower face of piston **116** to apply an upward force to the upper end of the lower portion of the riser. The magnitude of the upward force applied to the riser may be varied by adjusting the pressure of the fluid supplied by the source. Such adjustment may be made by any known device, and the adjusting means will not be described in detail herein. As the drilling vessel **1** moves relative to the wellhead **20**, cylinder **108** and tubular member **105** will move axially relative to annular piston **116**. The pressure of the fluid supplied to the cylinder through conduit **123** is regulated so that the pressure remains nearly constant in spite of such relative movement, and therefore the upward force on the upper end of the riser remains relatively constant. The regulation may be by any of several well-known devices.

In order to maintain the back-pressure above annular piston **116** relatively constant as the piston moves relative to cylinder **108**, perforations **122** in tubular member **105** serve as vents. These perforations permit the drilling fluid conveyed through the riser to pass through the perforations into the portion of the cylinder above piston **116** as the piston moves downwardly relative to the cylinder; and permit the drilling fluid to pass out of this space as the piston moves upward relative to cylinder **108**. The perforations **122** are preferably sufficiently large to permit cuttings which may be suspended in the drilling fluid to flow readily out of the space above piston **116** rather than to accumulate therein.

Stops **125** and **126** attached to the outer surface of enlarged diameter portion **107** limit downward movement of piston **116** relative to cylinder **108**, while stops **135** and **136** attached to cylinder **108** limit upward movement.

In the event of failure of the riser at some point above the wellhead but below the tensioning device, piston **116** will rise under the influence of the high pressure fluid until the piston reaches the upper limit of its stroke. At that point the relative movement between the upper and lower portions of the riser will be stopped and the riser will remain suspended from the drilling vessel. The upward force exerted on piston **116** by the high pressure fluid will be transmitted to cylinder **108** where it will be opposed by the equal and opposite downward force exerted on cap **119** by the fluid. Thus, the upward force is counteracted immediately on failure of the riser, and no hazard is created.

The opposing downward pull at the lower end of the riser may be provided by the wellhead and associated equipment itself, as shown in FIGURE 2. Alternatively, a concentrated counterweight **155** may be suspended from the riser by rigid gusset plates **156** and **157** as shown in FIGURE 2*a*. The counterweight is suspended around ball joint **48** so that the center of gravity of the counterweight is no higher than the center of rotation of the ball joint. Thus, as the riser moves relative to the vertical, counterweight **155** exerts no adverse moment about ball joint **48**.

A modified form of the tensioner of this invention is illustrated in FIGURE 8. In this modification, tubular member **105** is not perforated. The cylinder **108** includes an outlet **130** near its upper end which vents the upper portion of the cylinder to the atmosphere. Rings **131** and **132** in piston **116** provide a sliding seal between the interior space of the piston and the exterior surface of tubular member **105**. Alternatively, a seal can be provided between tublar members **105** and enlarged diameter portion **107** of uppermost conductor member **51***a*, at the lower end of tubular member **105**.

If a gas rather than a liquid is used for high pressure fluid, it is preferable to provide an oil bath **134** in the lower end of the cylinder to lubricate the contact between portion **107** and rings **120** and **121**.

It is sometimes desirable to limit the rate at which the pistons **116** can move relative to cylinder **108**. Thus, in case of riser failure, the impact of piston **116** against stops **135** and **136** is reduced. If liquid is used as the high pressure fluid, the maximum rate at which the liquid can pass through the conduit **123** sufficiently restricts the rate of movement of piston **116** relative to cylinder **108**. Additional restriction can be provided in the form of a properly sized orifice, if desired. If gas is used as the high pressure fluid, dampening may be provided either by properly sizing perforations **122** in the embodiment shown in FIGURE 5 so as to resist relative piston movement above a certain rate, or an orifice may be placed in outlet **130** of the embodiment FIGURE 8. Outlet **130** may then either be connected to a source of liquid, or in some instances may be located so as to be submerged below the ocean surface.

The means for connecting the upper end of the riser to the floating vessel will be described with reference to FIGURES 5, 6 and 7.

Rotary table **7** is supported on skid beams **138**, which are supported on the main support platform **6**. Support platform **6** includes main longitudinal beams **141** and short transverse beams **142** and **143** extending between adjacent longitudinal beams **141** beneath the rotary table. Beams **142** and **143** are strengthened by plates **144** and **145** respectively. The upper end of the riser, including upper tubular member **101**, extends to a point slightly below the rotary table and is approximately centered between the longitudinal and transverse beams. It is desirable that the upper end of the riser be supported from the above described structure at a position closely adjacent to the rotary table so that the bore of the pitcher nipple **101** will not pass from beneath the bore of the rotary table **7** as the vessel **1** pitches and rolls relative to the upper end of the riser.

Pitcher nipple **101** includes a special collar **139** at its upper end. The internal surface of collar **139** is funneled, as is shown at **140**, to aid in passing drill string and other materials into the upper end of the riser, particularly when the rotary platform is at an angle to the horizontal because of pitching and rolling of the vessel. The upper end of the riser is supported from platform **6** by a gimbal with two mutually perpendicular pivotal axes to permit universal angular movement of the vessel relative to the upper end of the riser.

One pivotal axis of the gimbal comprises a pair of hollow spindles **158**, each of which is mounted for rotation relative to the riser in one of a pair of bearings **157**. Each bearing is enclosed in one of a pair of housings **150** which are attached to special collar **139** to extend downward on diametrically opposed sides of pitcher nipple **101**. The axis of spindles **158** intersects the axis of the riser at a right angle.

An elongated plate **154** is attached to each spindle **158** adjacent to the outer face of the associated housing **150**. Each plate **154** rotates with its associated spindle in a plane parallel to the riser axis. An elongated plate **155** is attached to each spindle **158** for rotation therewith adjacent to the inner face of the associated housing **150**. A retaining bolt **156** serves to connect each spindle **158** to its associated plates **154** and **155**. Each of the plates **154** is also connected to its associated plate **155** by small transverse plates **159** and **160**. Thus plates **154** and **155** are mounted for pivotal movement about an axis at right angles to the longitudinal axis of conductor member **101** in a plane parallel to main beams **141**.

Connected betwen plates **155** near each end thereof is a spaced pair of parallel plates **161** and **162**. A housing **166** enclosing a bearing **165** extends upwardly between each pair of plates **161** and **162**. Each pair of plates **161** and **162** is pivotally connected to its associated housing through spindle retaining bolt **168**, hollow spindle **169** and bearing **165**. Each housing **166** is fixed relative to the vessel through plate **175**, which rests on supports **180** which, in turn, are connected to the flanges of beams **141**. Members **176**, **177**, **178** and plates **181** rigidify the structure. Plate **175** includes an opening **179** through which tubular member **101** extends.

The axes of spindles **169** are in longitudinal alignment with each other and are perpendicular to the axes of spindles **158**. Thus, as the vessel pitches and rolls, the gimbal structure supports the upper end of the riser while permitting the vessel to move angularly relative thereto about two mutually transverse axes. Both axes preferably lie in a common plane so that both may be placed at the highest possible position relative to the upper end of the riser, thereby reducing the lateral misalignment of the upper end of the riser relative to the central passage through the rotary table, as the vessel pitches and rolls.

Bearings **165** are preferably thrust bearings since the pull of the riser will at times have a component along the axis of bolts **168**, or thrust bearings may be installed between **161** and beams **142** and **143**. Bearings **157** may also be thrust bearings although there is little likelihood of their receiving anything but right angle loads.

The special collar **139** of the riser is chamfered at **183** so it will not strike the upper flanges of beams **141**.

An alternative arrangement for connecting the upper end of the riser to the floating vesesl is shown in FIGURE 8. In this arrangement,, spaced transverse beams **250** extend between adjacent longitudinal beams of support platform **6**. Horizontal plate **251** extends radially inward from the beams **250** and **141**. The plate **251** terminates in a spherical bearing surface **252**. Plate **251** is rigidly connected to and braced from the structural beams by plates such as **249**.

Seated in rotatably sliding relationship on bearing surface **252** is a spherical ball **253**. Tubular member **101** is connected to ball **253** and suitably braced by plates **256**. Ball **253** and bearing **252** serve to support the riser while, at the same time, permitting the vessel to move angularly relative to the riser in a manner similar to the gimbal of FIGURES 5 to 7.

It is clear, of course, that the tensioner shown in FIGURE 5 may be used in combination with the top support means of FIGURE 8, and vice versa. These and other modifications of the disclosed preferred embodiments will be obvious from the teaching of the above disclosure. It is intended that such obvious modifications be included within the scope of this invention and that the invention be limited only by the following claims and not by the details of the above described embodiments.

I claim:

1. For use offshore, an elongated tubular conduit including an elongated lower tubular member and an elongated upper tubular member connected to said lower member in axially aligned telescopic relationship, means for fixing the lower end of said lower member against axial movement relative to the ocean bottom, means for attaching the upper end of said upper member to a platform for axial movement therewith relative to said lower member, and adjustably variable fluid pressure means on said platform operatively connected to exert a regulated substantially constant upward preselected axial force on the upper portion of said lower member as said upper member moves axially relative to said lower member within selected design limits.

2. An elongated tubular conduit for connecting a floating vessel to a subaqueous formation comprising an elongated lower tubular member, means for fixing the lower end of said lower member relative to and adjacent said subaqueous formation, an elongated upper tubular member telescopically connected to said lower tubular member, means for attaching the upper end of said upper member to said vessel for random movement relative to said subaqueous formation under the influence of water and wind forces, a piston and cylinder operatively connected to exert an upward axial preselected force on the upper portion of said lower member, said piston and cylinder being proportioned to permit said upper member to move axially relative to said lower member as said vessel moves relative to said subaqueous formation within selected design limits, and adjustably variable means for supplying fluid under a regulated substantially constant preselected pressure to said cylinder for maintaining said upward axial force substantially constant at a predetermined magnitude as said vessel moves relative to said subaqueous formation.

3. A tensioning device for an offshore riser, which riser comprises a pair of telescopically connected elongated tubular members for connecting a floating vessel to a subaqueous well as said vessel moves randomly relative to said well, said tensioning device comprising: a first annular surface attached to the telescoped end of one of said tubular members for movement therewith, a second annular surface attached to the other of said tubular members for movement therewith, said first and second annular surfaces forming the ends of a sealed chamber which extends as said riser elongates and contracts as said riser retracts, and adjustably variable means for supplying fluid under a regulated substantially constant preselected pressure to said chamber for applying substantially constant preselected axial force to said members as said vessel moves relative to said subaqueous well.

4. For use in offshore drilling, an elongated tubular conduit for connecting a floating vessel to a well bore to pass dense drilling fluid and drilling tools between said vessel and said well bore, comprising: an elongated lower tubular member and an elongated upper tubular member, means for attaching the lower end of said lower member to said well bore, means for attaching the upper end of said upper member to said floating vessel, the upper end of said lower member and the lower end of said upper member being axially aligned and overlapping in telescopic relationship at all positions of said vessel relative to said well bore within selected design limits, an annular piston attached around one of said telescoped ends, a cylinder surrounding said piston in axial slideable relationship and attached to the other of said telescoped ends, said cylinder being of sufficient length to permit axial extension and retraction of said conduit sufficient to accommodate vessel movement relative to said well bore within said design limits, adjustably variable means for supplying a pressure fluid under a regulated substantially constant preselected pressure to said cylinder for applying substantially continuously during said offshore drilling a substantially constant preselected upward axial force to the upper end of said lower member, said pressure fluid being separate from said drilling fluid, the magnitude of said force exceeding the weight in the surrounding medium of said tubular conduit plus the weight in the surrounding medium of the materials contained therein.

5. For use in offshore drilling from a floating vessel, a riser attached at its lower end to a stationary wellhead below the ocean surface and at its upper end to a floating vessel, means in the lower portion of said riser permitting angular movement of said riser as said vessel moves laterally relative to said wellhead, and means in said riser near its upper end for permitting axial extension and retraction of said riser and for maintaining a constant tension in said riser, substantially continuously during said offshore drilling said last recited means comprising a telescopic connection in said riser, an annular piston connected around one member of said telescopic connection, a cylinder surrounding said piston in axially slideable relationship and connected to the other member of said telescopic connection, said piston and cylinder permitting extension and retraction of said riser sufficient to compensate for a vessel movement relative to said wellhead within selected design limits, adjustably variable means for supplying fluid under a regulated substantially constant preselected pressure to said cylinder for application of a preselected substantially constant upward force to the lower one of said telescoping members independent of the axial position of said piston relative to said cylinder.

6. A gimbal for connecting the upper end of a riser to a floating drilling vessel adjacent the underside of a rotary table on said vessel, said gimbal comprising: a first pair of plates connected to the riser near its upper end for rotation in a pair of parallel planes on diametrically opposed sides of said riser about a first axis which intersects the longitudinal axis of the upper end of said riser at a right angle, a second pair of plates attached between said first pair of plates to form a rectangle around the upper end of said riser, and means connecting said second pair of plates to said vessel for rotation about a second axis which intersects said first axis at a right angle at the point of intersection of said first axis with the longitudinal axis of said riser.

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