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TRANSACTIONS (TM)

DRILL STRING COMPENSATORS AND RISER TENSIONING SYSTEMS FOR OFFSHORE DRILLING

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Drill String Compensators and Riser Tensioning Systems for Offshore Drilling

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INTRODUCTION

There are two basic types of floating drilling vessels: ships and semi-submersibles (Figs 1 and 2). Most drillships are ships that have been modified for floating drilling duty, whereas semi-submersibles are unique, having been developed specifically for drilling in choppy seas.

Drilling vessels are normally self-propelled when in transit and may be either moored or dynamically positioned over the well when drilling.

In general, drillships are noted for their mobility, high storage capacity and, usually, lower day rates. Semi-submersibles are noted for their motion characteristics and are usually capable of drilling in rougher seas than drillships.

A vessel at sea has six types of motion, referred to as degrees of freedom. Three of the motions are translational and three are rotational. Motions restricted to the horizontal plane can be controlled by the station-keeping system, whereas heave, pitch and roll are features that are a direct result of the vessel's design.

Severe pitch and roll of the vessel can impede drilling, while severe heave can over-tension or buckle the riser, thus causing it to fail. These motions increase as their 'natural' periods of motion are approached by the wave incidence. The natural periods of pitch and roll tend to decrease with increasing mass and extend along a line normal to the axis of rotation. For example, the period of pitch of a ship-like vessel will be greater than its roll because the vessel is longer than it is wide. Increasing the beam width will decrease the amplitude of roll in choppy seas as the natural period of roll increases above the period of the waves.

The advantages of one type of drilling vessel over another may be anticipated under certain conditions. For the purpose of this discussion, two types will be considered: a drillship and a semi-submersible, referred to here as, respectively, a 'ship' and a 'semi'.

The major difference between the motions of a ship and of a semi is the roll in beam waves (Fig. 3). In this case, the ship, with waves acting on the quarter, has a much lower roll period than the semi. Pitch is also worse for the ship than for a semi.

The heave of the ship is higher than the heave of the semi for periods up to the natural period for the semi (Fig. 4). For storm waves of less than this natural frequency, the semi may drill when the ship is shut down whilst waiting on weather. The long-period waves that follow a storm, however, can also force drilling operations to be suspended on the semi.

For ships, both roll and heave are worse in beam waves than in head waves. Thus, one way to reduce these motions is to turn the vessel into the prevailing waves. Conventionally moored ships are limited to holding a specified heading while drilling; however, turret-moored and dynamically positioned vessels can assume any heading without having to disconnect from the wellhead. This manoeuvrability greatly improves the performance of such ships.

Various operational limits due to vessel motion have been set out by different operators; however, the limits given in Table I can be considered typical for most designs of vessel. However, the tolerance level of personnel on the vessel must be taken into account when considering the vessel's motions, in order to assure the safe performance of personnel in any particular environment. Such information can be used in conjunction with data on the vessel's motion and distribution of wave height to compare the downtime due to waiting on weather for the ship and semi.

Drilling while heaving, for example, 10 ft would require a drill string compensator. This heave corresponds to a significant wave height of about 20 ft for the ship and 30 ft for the semi (Fig. 5). Heave is the limiting factor in this case because roll and pitch for higher significant wave heights can be tolerated (Fig. 6).

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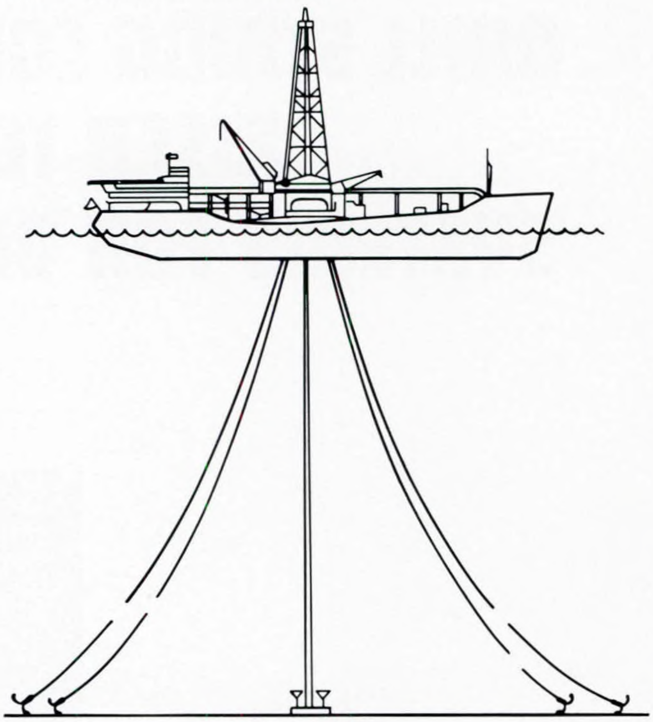


FIG. 1 Drillship

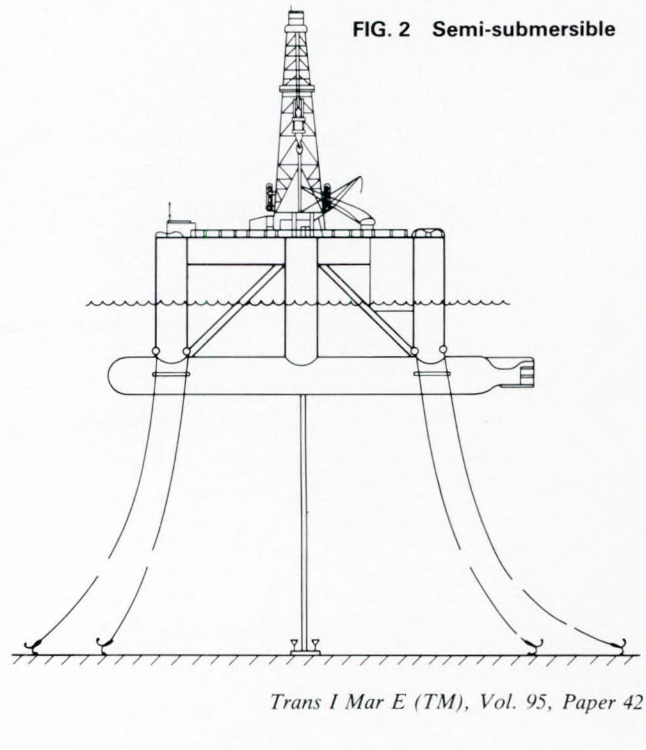


FIG. 2 Semi-submersible

By using a drill string compensator the motion limits, or shutdown condition, for drilling can be extended. For example, when drilling using bumper-subs, the shutdown condition would be reached at a heave of 8 ft. Using a compensator, drilling can continue up to a heave of 12 ft. By the latter means, downtime can be shown to decrease by a factor of four, depending on the average weather situation (Fig. 7).

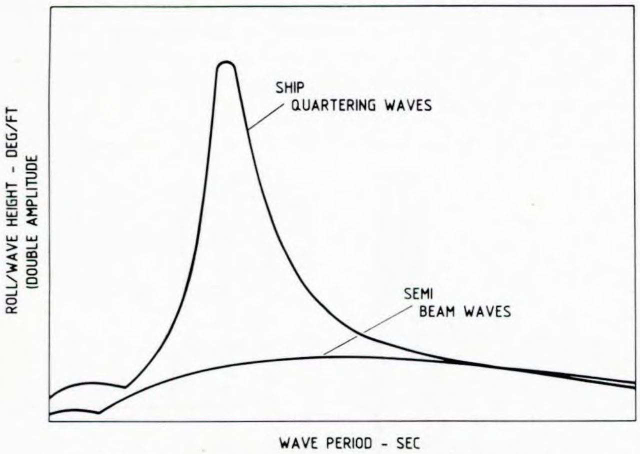


FIG. 3 Vessel motion: roll in regular waves

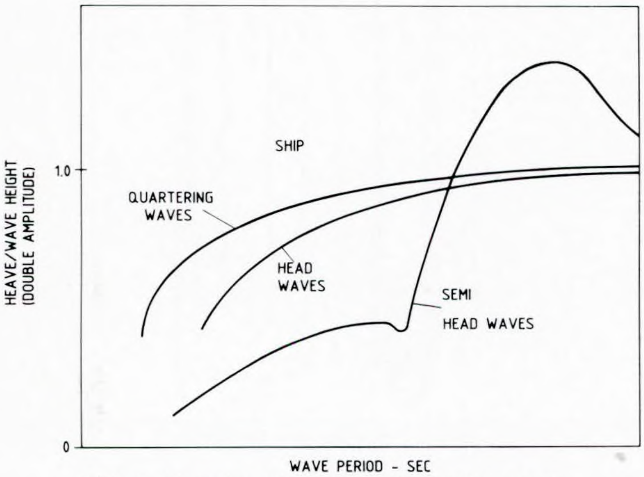


FIG. 4 Vessel motion: heave in regular waves

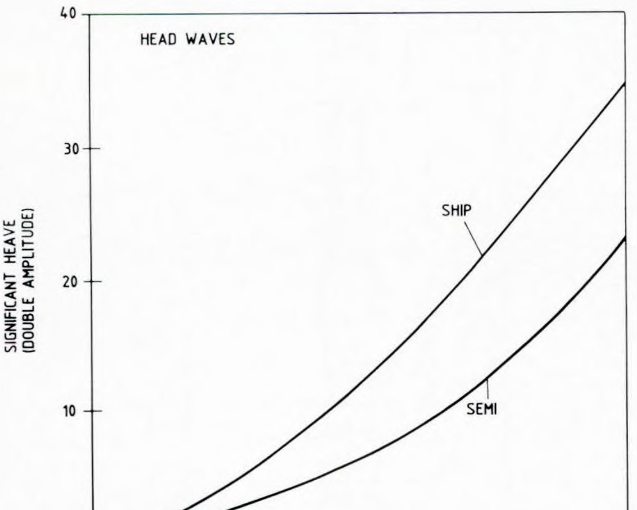


FIG. 5 Vessel motion: heave in irregular waves

Table I: Limits on motion (wave height and heave) during drilling operations[a]

OPERATION	WAVE HEIGHT (ft)	HEAVE (ft)
Drilling ahead	30	10
Running and setting casing	22	6
Landing blow-out preventer (BOP) and riser	15	3
Transferring equipment	15	—

[a]Data from *Ocean Industry* Magazine.

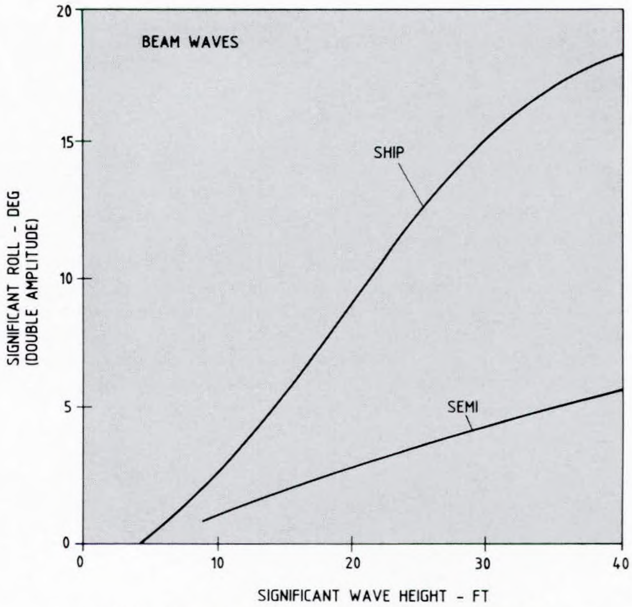


FIG. 6 Vessel motion: roll in irregular waves

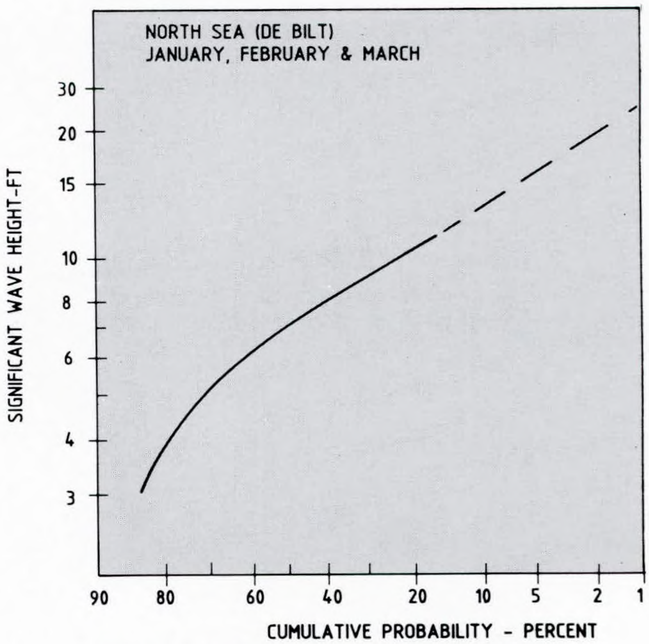


FIG. 7 Cumulative wave height distribution

Table II: Typical percentages of time spent on various operations

OPERATION	TIME (%)
Drilling and tripping	52
Logging and cementing	15
Running casing	9
Running riser and BOPs	9
Other operations	15

The relative importance of each operation can be better appreciated when the percentage of total time usually required for each operation during drilling a 'typical' well is considered (Table II). The percentage of time spent on each operation will vary between different areas.

TENSIONERS

In floating drilling it is necessary to keep the riser in tension and to allow relative movement between vessel and riser.

In the early 1960s, weights called counterbalances were used for tensioning both the riser and guidelines but, as drilling moved to deeper water and into areas of severe environmental conditions, counterbalance weights for riser tensioning became so large that they were unpractical to handle on a vessel.

Figure 8 shows a typical riser and guideline tensioner system. The advantages of such systems are that they increase the tensioning capacity relative to space requirements and facilitate changes in tension, yet for the same tension the deck loading is decreased. Applying the same principle, a constant weight is kept on a drill bit by holding a constant tension on the drill string using a drill string compensator.

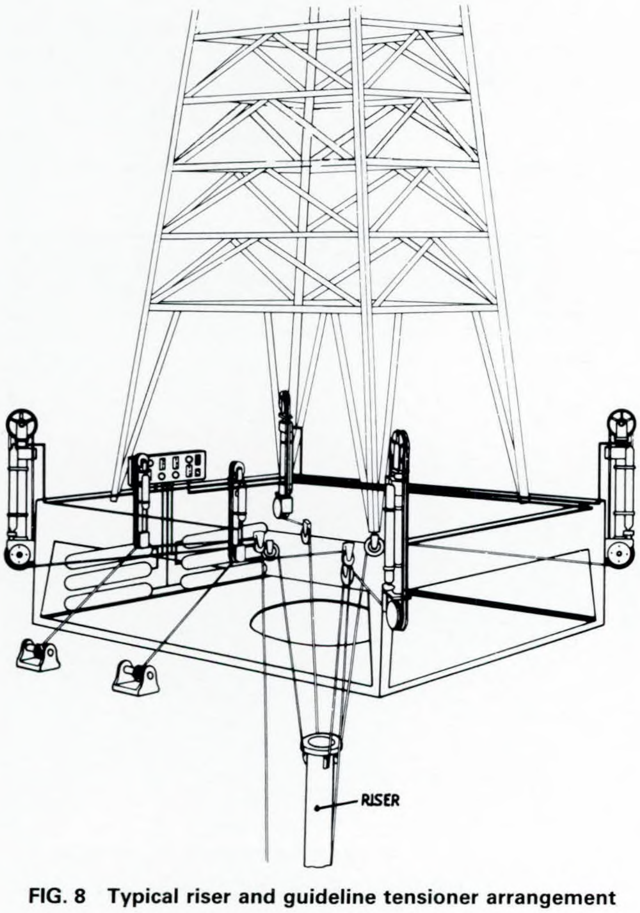


FIG. 8 Typical riser and guideline tensioner arrangement

A typical tensioner unit is shown in Fig. 9. The lower sheaves are attached to the cylinder and the upper sheaves are attached to the rod; the piston rod applies a force which tends to separate the upper and lower sets of sheaves.

This separating force determines the tension in the line reeved on the sheaves. Tension is maintained by pressure transmitted to the piston rod by oil that is pressurized by the air reservoir. To increase the tension, air is added to the reservoir through a line from a compressor; to decrease the tension, air is vented from the reservoir.

Oil is used on both sides of the piston for lubrication and inhibition of corrosion. The system includes a hydraulic safety device to slow the rod if the line breaks.

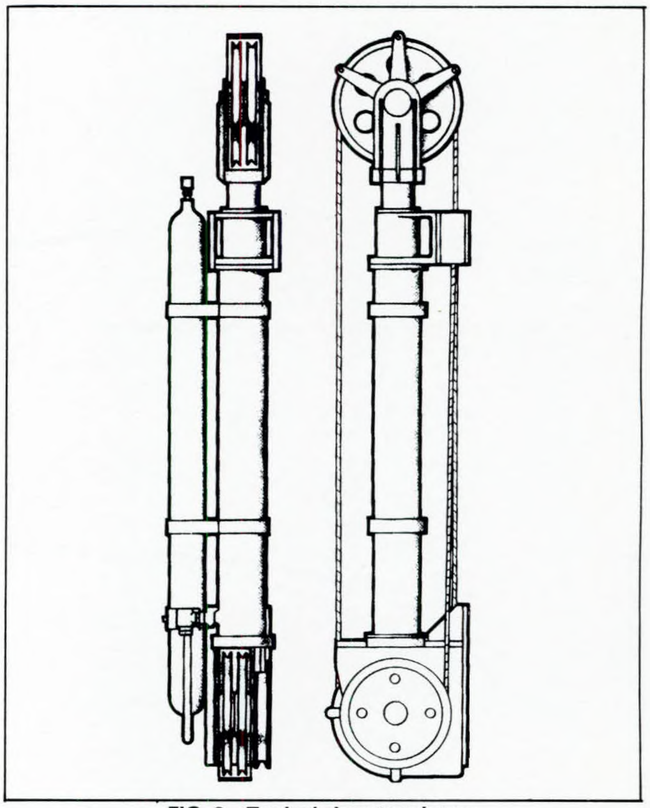
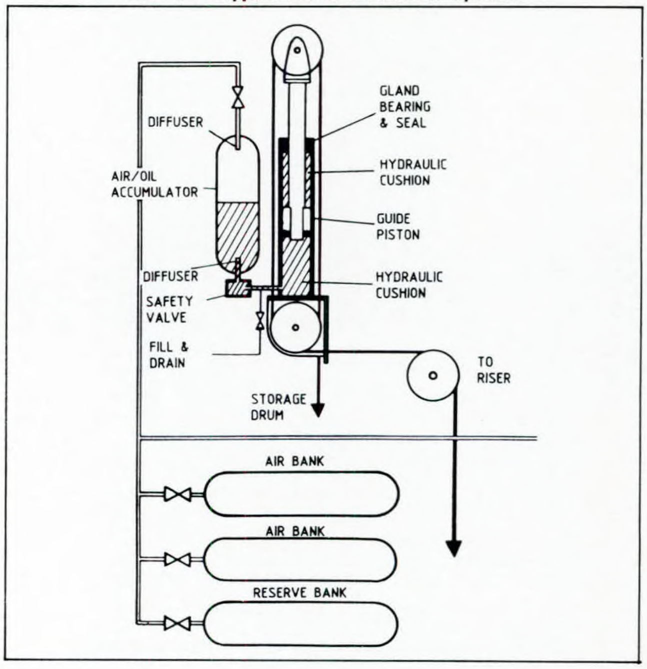


FIG. 9 Typical riser tensioner

FIG. 10 Typical riser tensioner system



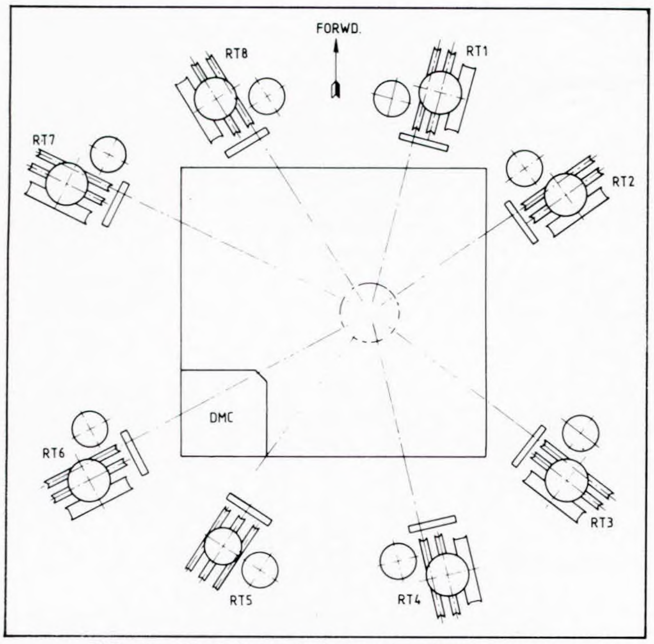


FIG. 11 Deck layout of riser tensioner system

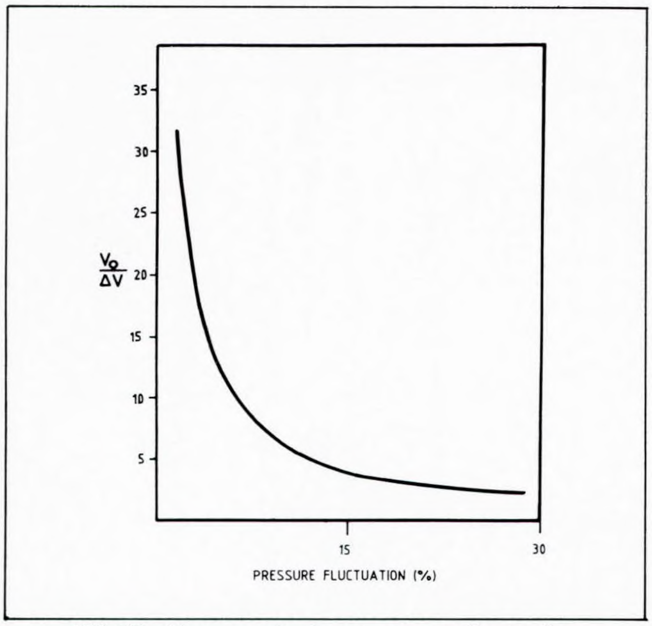


FIG. 12 Air reservoir size vs pressure fluctuation

The piston serves as a guide for the piston rod and also prevents the rod being forced out of the cylinder in the event of a failure.

The piston rod, or ram, is machined from special stainless steel which is corrosion-resistant and hard-wearing. Stainless steel, unlike chrome, does not have a catastrophic failure mode and it can be repaired *in situ*.

Reeving, which varies from four to eight turns, decreases the stroke length relative to the heave. Tensioners that are diametrically opposed on the riser should be connected to the same reservoir. This decreases the probability of cocking the slip joint or tensioning ring while tension is being adjusted or in the event of a reservoir failure.

The maximum working pressure of the type of system shown in Fig. 10 is about 2700 lbf/in^{2}g.

In field operations, four to twelve tensioners may be used for riser tensioning. Smaller tensioners, varying in capacity from 16 000 to 18 000 lb, are used for tensioning guidelines.

Figure 11 shows a typical deck layout arrangement for a riser tensioner system. Idler sheaves are used to guide the wire to the required

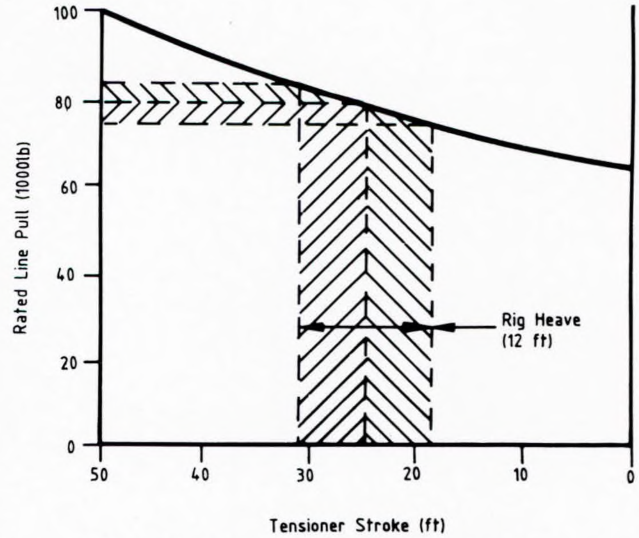


FIG. 13 Typical load variation characteristics

position. Riser tensioner systems are normally designed for a total pull of 1 to 1.6 × 10^6 lb.

Performance

The tensioner's pull will vary with the stroke, the extent of this variation being mainly dependent on the relative size of the air reservoir.

$$\Delta P = P_0[V_0/(V_0 + \Delta V)]^n \quad \text{where } n = 1.41$$

The change in volume, ΔV, of the air reservoir is directly proportional to stroke. The variation of pressure ΔP from the mean pressure P_0 is therefore dependent on the ratio between mean reservoir volume V_0 and ΔV. Riser tensioner systems are normally required to have a load variation of up to 15%. This would require a mean volume of air reservoir of greater than five times the stroke displacement (Fig. 12).

Friction losses are not critical in a tensioner application.

Figure 13 shows typical load-variation characteristics of an 80 000 lb riser tensioner unit.

DRILL STRING COMPENSATORS

A drill string compensator applies a constant force to a dynamic load and allows relative motion of the vessel to be isolated from the load. The motion compensator enables drilling operations to continue in relatively high sea states by decreasing downtime, bit wear and equipment damage, while increasing the safety of personnel. The need for heave compensation was recognized at the advent of floating drilling vessels but operational equipment was not available, even for testing, until the 1960s.

More stringent tolerances are required for drill string compensators than for tensioners, because one of the former's principal uses is to keep a nearly constant weight on the bit even in rough weather. Suppose that 40 000 lb bit weight is needed on a bit when the total load on the drill string is 200 000 lb. Information by Galle and Wood indicates that drilling efficiency does not deteriorate rapidly until load fluctuations exceed about 30% of the weight on the bit. Using this criterion in the example, fluctuations would be less than 12 000 lb, which is 6% of the total drill string weight. A performance requirement of 4–5% load fluctuation is a severe test of the principle of passive motion compensation and can only be achieved through careful design of the system.

Travelling block compensators

In these systems the operating cylinder(s) is located on the travelling block, between the block and the hook. Large air reservoirs are located on or below the deck and pressurized fluid is piped to the cylinder(s) by flexible hoses. Dual or single pistons may be used to support the load.

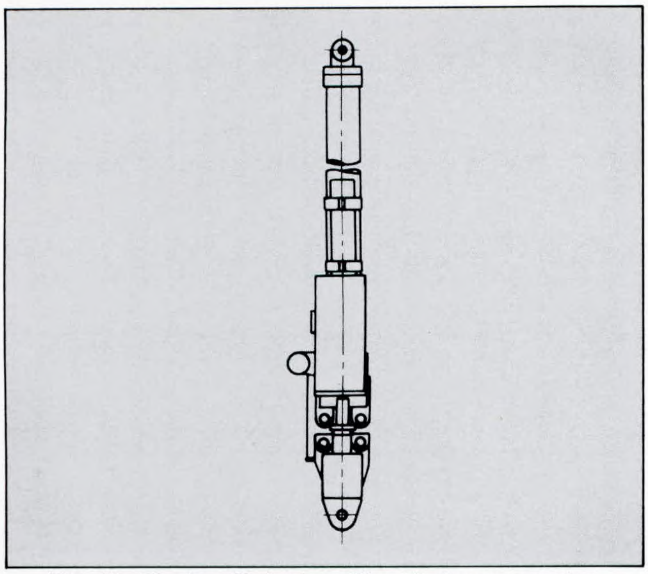


FIG. 14 Single cylinder travelling block compensator

A single cylinder (simplex) compensator (Fig. 14) would normally require more derrick height than a dual cylinder (duplex) unit unless it utilizes a split arrangement of crown and travelling blocks (Fig. 15). The crown block and travelling block must be replaced if the derrick was not initially designed for the single piston compensator. This, together with other reeving complications, have tended to make the duplex compensator (Fig. 16) the preferred equipment.

Crown block compensators

Crown block motion compensators are also pneumatic springs but the mechanical compensation is accomplished by compensating the crown block at the top of the derrick. This is done by moving sheaves attached to the piston rods and lower yoke, positioned as shown in Fig. 17.

The cylinders are located on the water table and remain essentially motionless relative to the vessel, therefore rigid piping can be used. This decreases the probability of hydraulic line failure in this type of system relative to travelling block compensators, in which flexible lines are required.

System description

Figure 18 is a simplified diagram depicting this air spring support system as applied to the drill string motion compensator.

Air pressure is translated to oil hydraulic pressure through an accumulator. The oil acts against the compensator pistons and actually supports the hook load.

The motion compensator is simple in theory but becomes complicated in practice because of the many valves, control devices, indicators and support systems required for safe and efficient operation.

The compensator based on the above principle is available as a twin cylinder unit. It has a stroke capacity of from 20 to 25 feet and maximum operating load capabilities of 400 000 to 600 000 lb.

Equipment description

The system consists of a compensator unit connected to an accumulator using rotary hoses and air pressure vessels. There are also air compressors and dryers, a control console and a hydraulic power unit to provide a topping-up facility and to activate the mechanical locking feature (Fig. 19).

All operations are performed from the control console. These operations include adding or reducing air pressure (hook load support); locking and unlocking of the compensator, both hydraulic and mechanical; control of air compressors and dryers; manual control of safety valves; hydraulic locking, and visual readout of such items as compensator stroke, valve positions and system pressures.

Compensator unit

The duplex drill string compensator unit (Fig. 20) consists of two cylinders with pistons and piston rods which carry the load in tension.

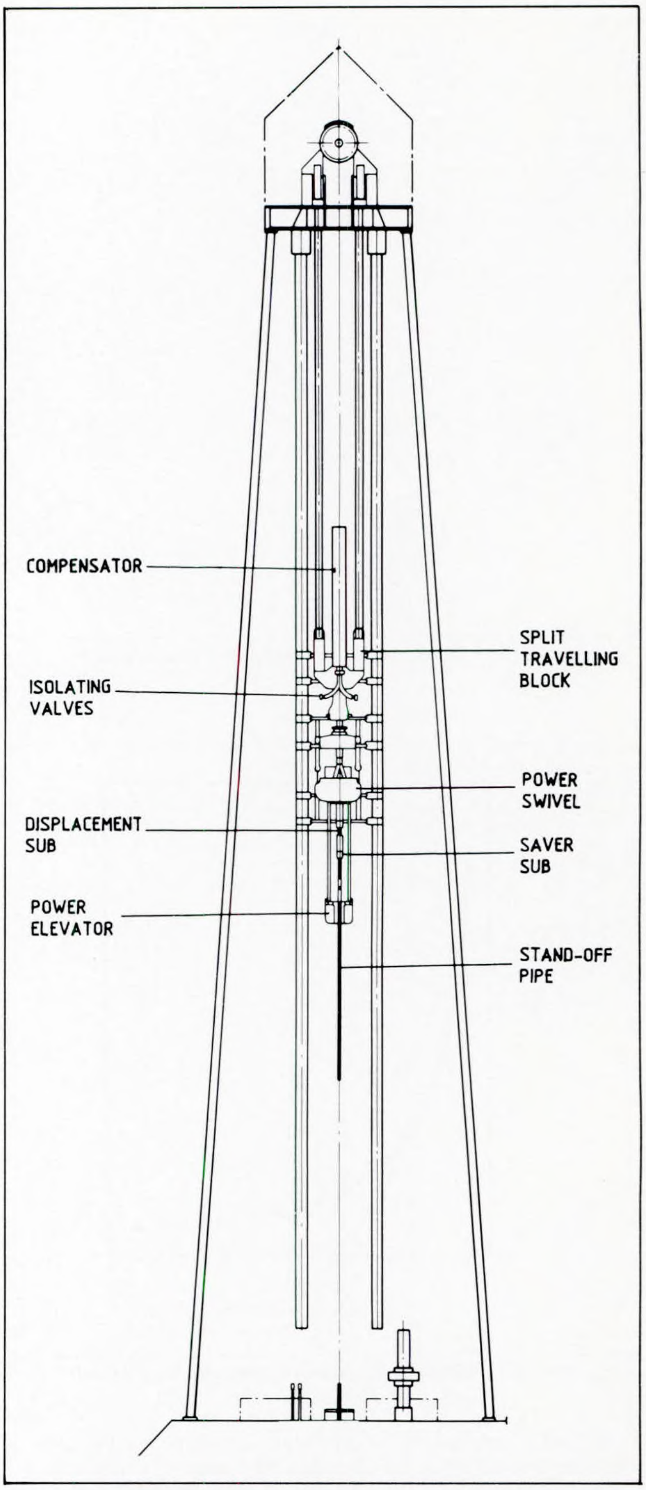


FIG. 15 Single cylinder split travelling block compensator

The two cylinders and rods are held together by an upper yoke, to which the travelling block is fitted, and a lower yoke to which the hook is fitted. Both travelling block and hook are rigidly fixed to the compensator to avoid undesirable loading on the guide-rails. The piston rods, which are made from a special stainless steel, have a large diameter in order to provide low bearing pressures in the gland bearing due to lateral accelerations.

The seals used are pressurized low-friction polytetrafluorethylene (PTFE) seals. In the gland there are two seals (Fig. 21): a primary seal which takes the system pressure and which is running wet; and a secondary seal with a rubber lip which scrapes the rod dry, with any

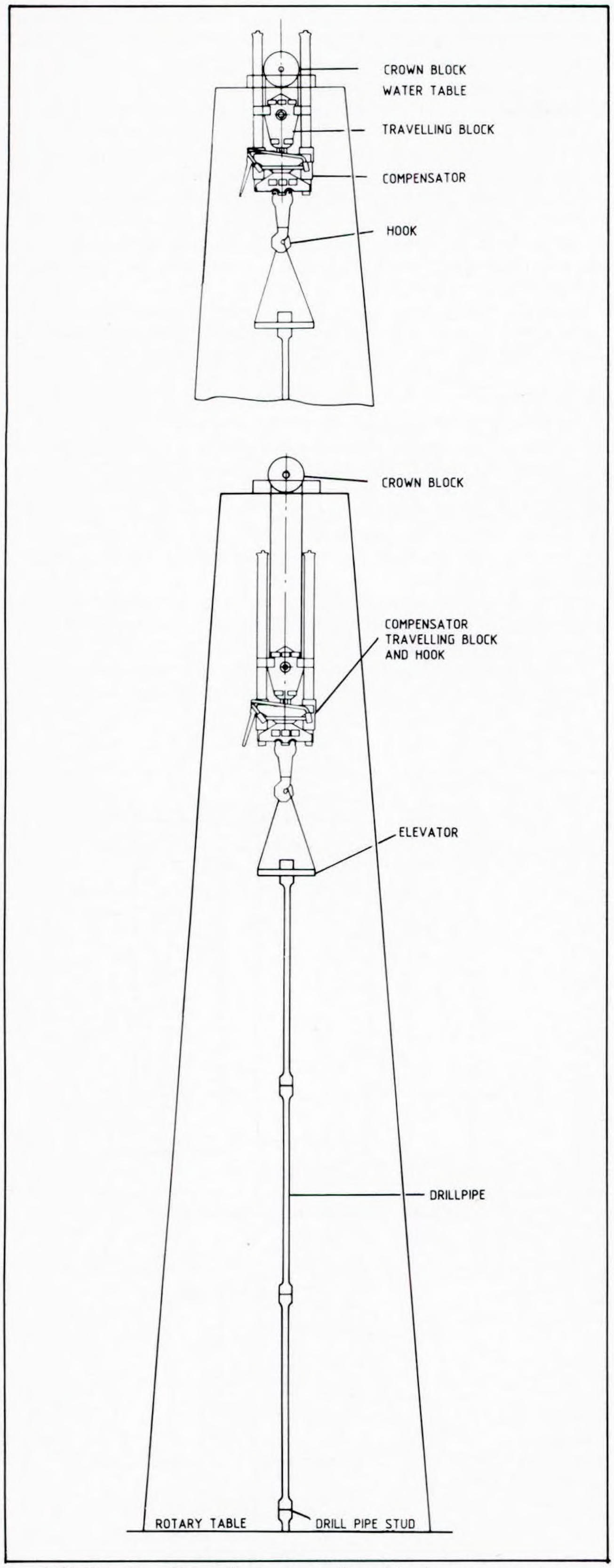


FIG. 16 Double cylinder travelling block compensator

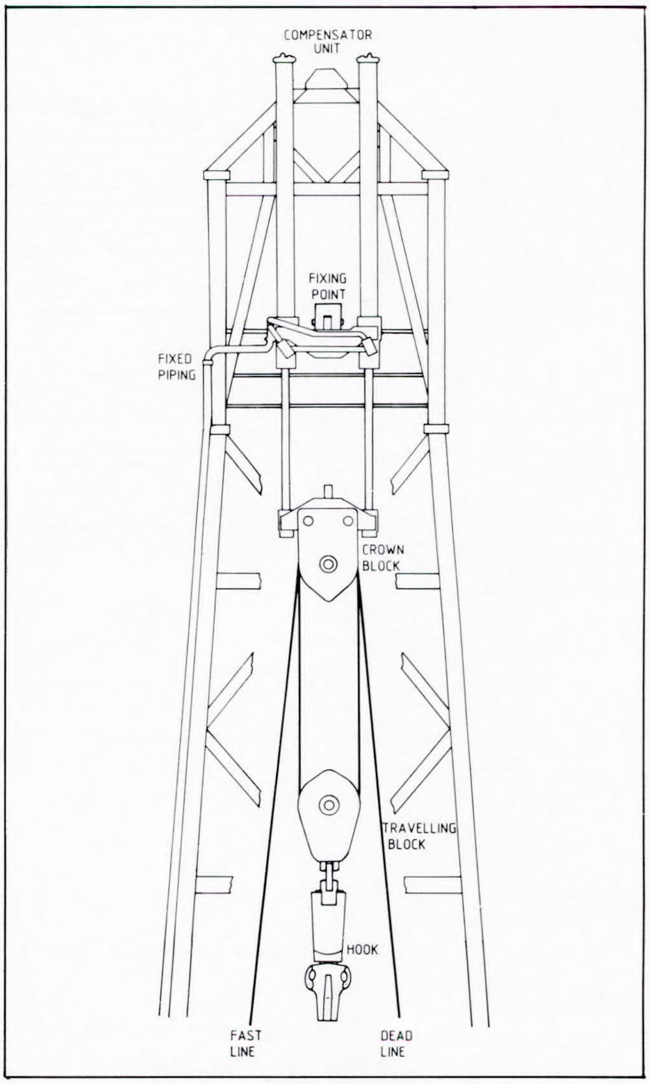
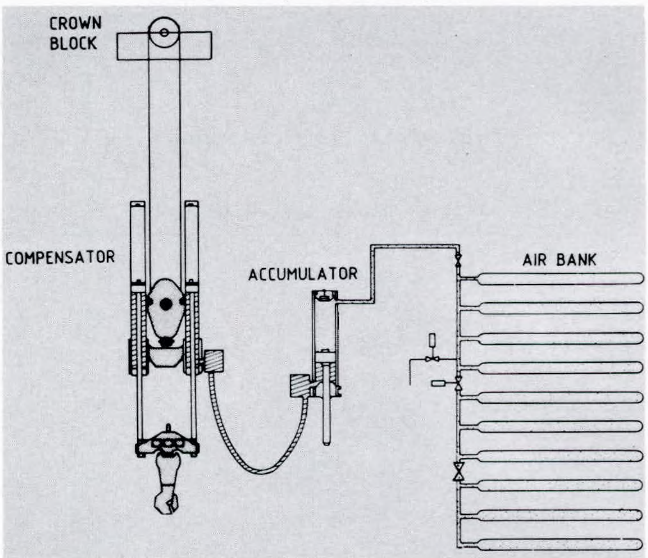


FIG. 17 Crown block compensator

FIG. 18 Air spring support system as applied to drill string motion compensator (simplified)



excess fluid draining back to the power unit tank. In the event of failure of the primary seal, excess fluid drains back to the tank. This can be easily observed and a valve can be closed. The operation can be continued with the pressure being taken by the secondary seal until a suitable time for a changeout.

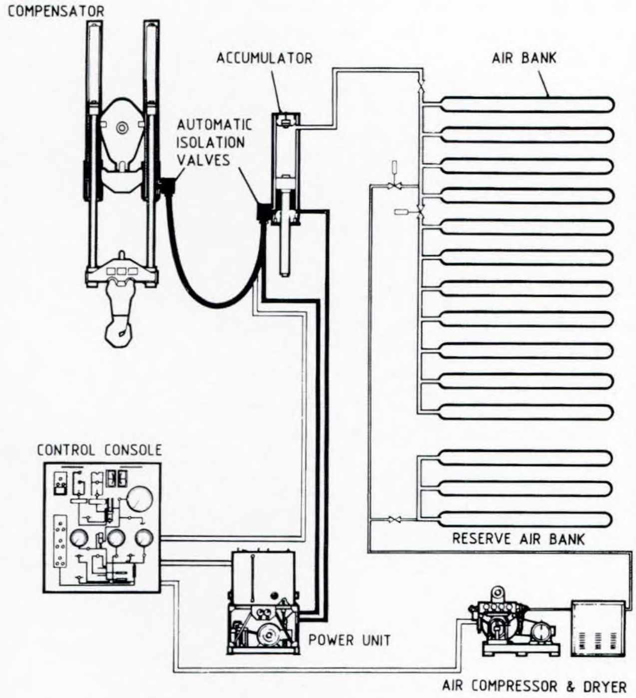


FIG. 19 Compensator system configuration

Piston sealing (Fig. 22) is accomplished by a series of piston rings backed up by a cartridge seal unit containing an energized seal, made of PTFE rubber, with a low-pressure wiping lip.

The mechanical locking mechanism (Fig. 23) consists of a single hydraulically actuated pin which, when locking, effectively removes the compensation effect from the hoisting system and makes a rigid link between the hoisting system and the load. The hoisting system then resumes its full load in a compensating mode.

The most direct method of supporting the hook's load is to put the rods in tension. This arrangement minimizes mechanical contact and potential wear. To avoid potential lateral loading effects, the piston rods are of sufficient diameter to make the bearing areas, and therefore bearing pressures, acceptable.

In a duplex compensator unit, it is necessary to ensure equal loading between the pistons. The pistons are therefore connected via piston rods through the lower yoke to a rigidly fixed hook.

Accumulator

A completely pneumatic system would soon fail because of lack of lubrication of the sealing elements and because a failure in either the air system or in the drill string would result in a disastrous bottoming-out (of the piston against the packing glands) or an equally disastrous sling-shot reaction (when the hook load was removed from the compensator). An accepted method of correcting these deficiencies is achieved through the use of hydraulic fluid and a hydro-pneumatic accumulator (Fig. 24).

The compensator pistons and rods are well lubricated by the hydraulic fluid, which is fire-retardant and maintains a specified viscosity over a wide temperature range.

The addition of safety shut-off valves at both the compensator unit and the accumulator safeguards against potential failure modes, i.e.

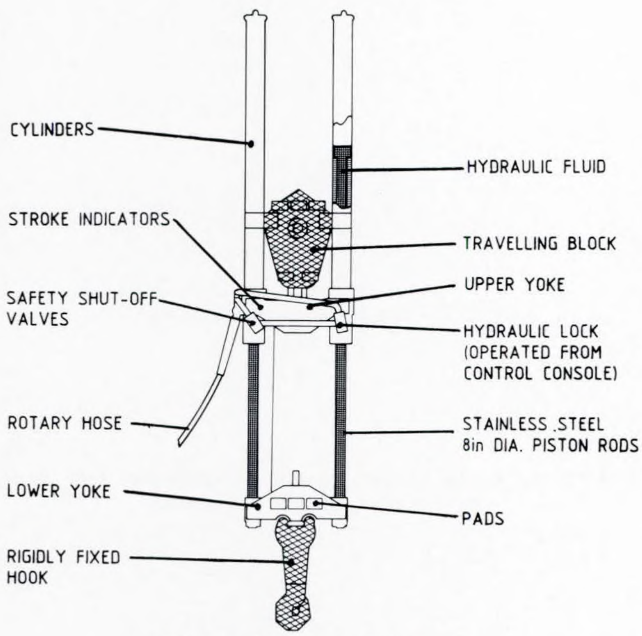
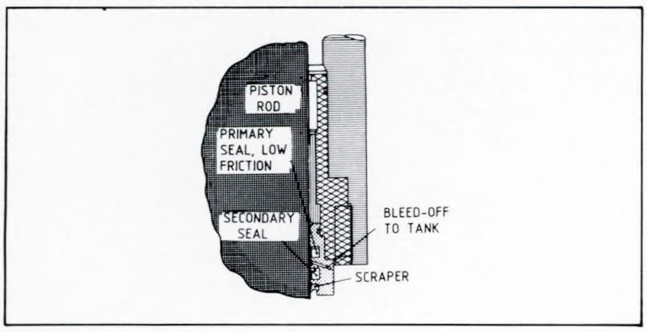


FIG. 20 Compensator features

FIG. 21 Gland seal arrangement



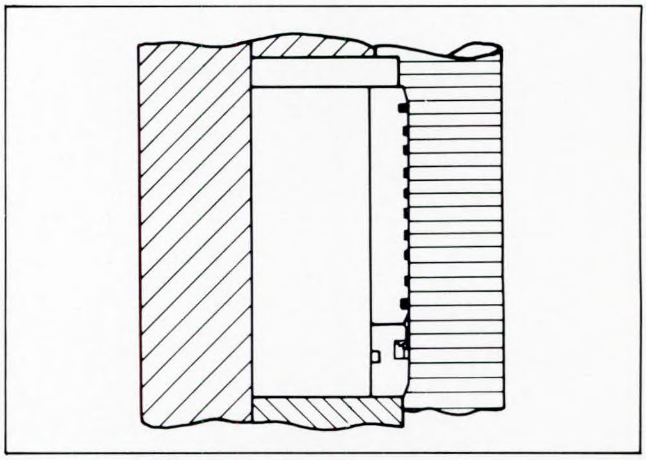


FIG. 22 Piston seal arrangement

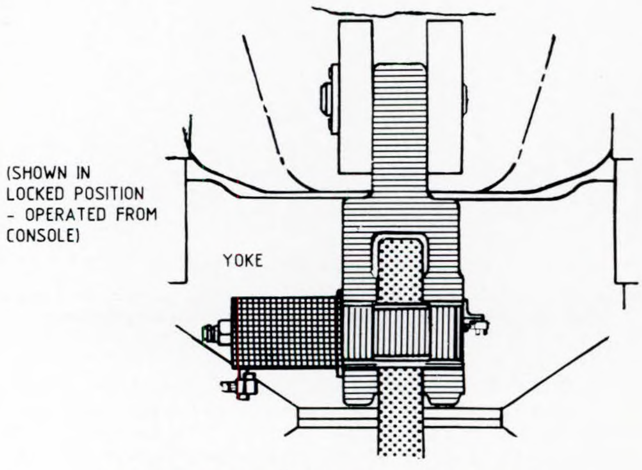


FIG. 23 Mechanical locking arrangement

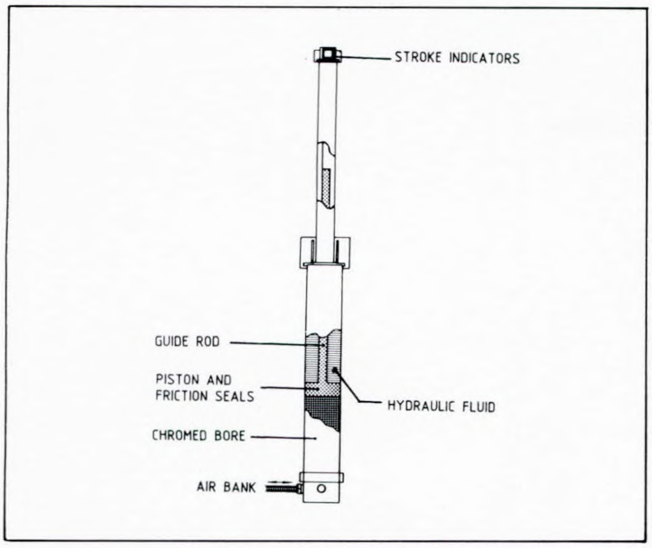
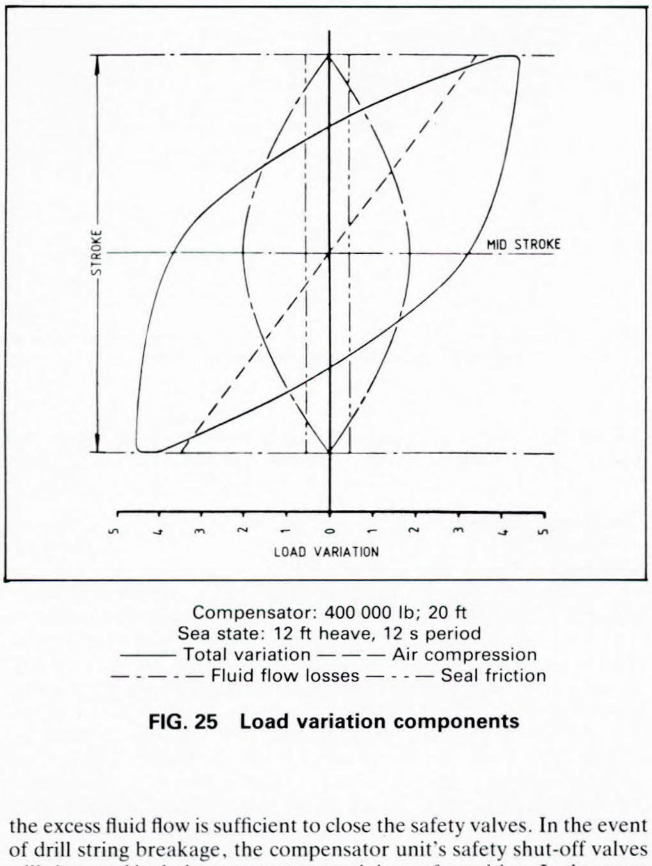


FIG. 24 Hydropneumatic accumulator



Compensator: 400 000 lb; 20 ft
Sea state: 12 ft heave, 12 s period
—— Total variation — — — Air compression
— · — · — Fluid flow losses — · · · — Seal friction

FIG. 25 Load variation components

the excess fluid flow is sufficient to close the safety valves. In the event of drill string breakage, the compensator unit's safety shut-off valves will close and lock the compensator unit in a safe position. In the event of a rotary hose rupturing, both the compensator unit and accumulator shut-off valves will close, so that the compensator unit is made safe and spillages are minimized.

A potential drawback in a hydraulic compensator is loss of hydraulic fluid pressure due to high-velocity flow. However, this drawback is easily overcome by careful selection of pipe diameters and by minimizing both pipe lengths and the quantity of pipe bends in the system.

Hydraulic fluid

Both pneumatic and hydraulic compensator systems use hydraulic fluid which is fire retardant and has a defined viscosity over a wide temperature range. Fluids currently in use are Pydraul ELT 29 and Erifon 500.

Air bank

The motion compensator's air-compressor and air-dryer equipment is used in conjunction with the riser and guideline tensioner systems. The quantity and size of the air compressors and dryers depend on the volume of back-up and reserve air required.

The volume of back-up air is determined by the percentage of load variation that is acceptable over the specified range of heave, i.e. the smaller the amplitude of heave, the smaller the compensator piston motion affecting the total volume of air in the system and, therefore, the lesser the quantity of back-up air vessels required. Total available volume of back-up air is that which will allow a 4 to 5% load variation with the compensator unit at full working stroke.

The reserve air vessels are maintained at the maximum rated pressure of the system so that, as drilling progresses and the load on the drill bit increases, air bank pressure can be increased or replenished by using air from the reserve air vessels so as to support the weight of an added drill pipe. The volume of reserve air vessels is determined as a compromise between the air compressors' capacity and operational air requirements.

Control panel

The compensator's control panel contains hand-operated valves for control of system pressure. This is accomplished by venting system air to atmosphere in order to decrease the system pressure; or valving the reserve air to the system in order to increase load support pressure.

The console contains working systems, reserve and back-up air pressure gauges. Both the compensator and accumulator have stroke indicators on the console so that the position of either unit can be clearly indicated. The console also contains controls for hydraulically and mechanically locking the compensator, together with the necessary indication.

Performance

The passive drill string compensator is essentially a hydro-pneumatic tension spring incorporated in the drill string suspension gear between the travelling block and swivel. Pretension is governed by the mean pressure in the air bank and the fluctuation about this mean is governed by:

1. Relative changes in volume (i.e. piston movement).
2. Hydraulic fluid and air friction in rotary hoses and pipes: these losses depend on pipe lengths, diameters and quantity of bends and may determine the phasing of peak load relative to heave (Fig. 25).
3. Seal friction.

Assuming a sinusoidal heave motion of amplitude Q_0 and radian frequency of ω, and a rigid drill string, the components of tension in the drill string may be expressed mathematically as follows:

Mean tension:

$$T_m = A_a K_1 P_0 \tag{1}$$

where A_a is the effective accumulator piston area; K_1 is the accumulator amplification factor, and P_0 is the mean pressure in the air bank.

Load variation:

$$\delta T_a = P_0/(1 - C_1 Q_0 \sin \omega t) \cdot C_1 Q_0 \sin \omega t \tag{2}$$

where $C_1 = (nA_a)/V_0($ constant$)$, n being a polytropic exponent in gas law and V_0 the mean air volume.

$$\delta T_b = C_2 \omega^2 Q_0^2 \cos \omega t \cdot \cos \omega t \tag{3}$$

where C_2 is a constant relating piston velocity to fluid friction force.

$$\delta T_c = F_c \text{ (sign of } \sin \omega t) \tag{4}$$

where F_c is the coulombic friction force.

It is seen that δT_a is dependent on heave amplitude and mean air volume; δT_b on the square of heave velocity and, while δT_c is constant in magnitude, its sign is dependent on direction.

The resultant tension in the drill string at any instant is:

$$T = T_m + \delta T_a + \delta T_b + \delta T_c$$

The components δT_a, δT_b and δT_c may be represented by vectors.

Optimum load variation is dependent upon the phase relationship between fluid friction losses and air bank variation. Air bank volume is optimized with respect to minimum static and fluid friction losses (Fig. 25).

A duplex passive drill string motion compensator can in practice achieve a load variation of 4–5% for maximum loads rated at normal operating loads (Fig. 26).

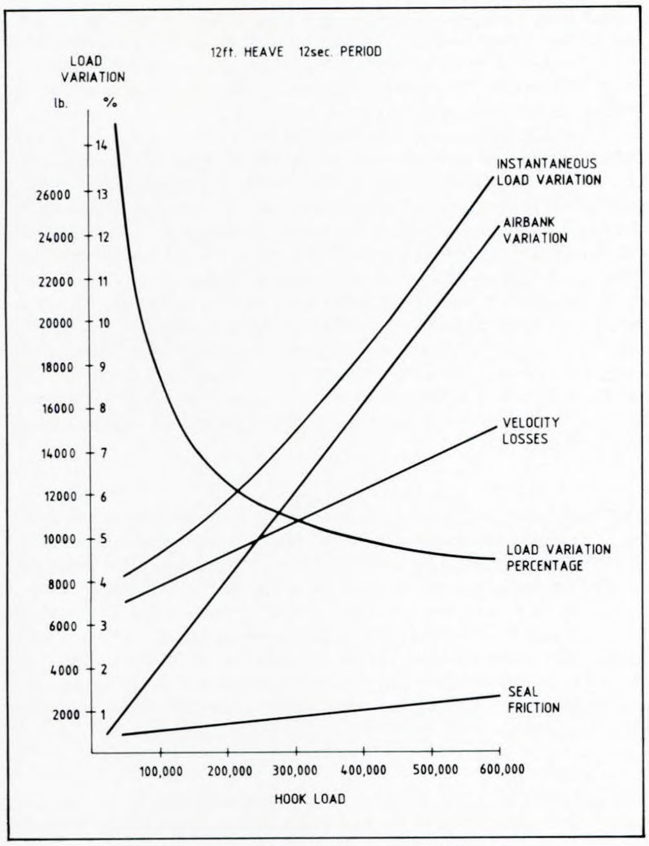


FIG. 26 Load variation vs hook load

ACTIVE COMPENSATOR

For a load variation requirement of 1%, a semi-active drill string compensator can be used.

The first semi-active compensator was developed by Brown Brothers in conjunction with Global Marine and makes use of the accumulator and hydraulic system to compensate for:

(a) Air bank variation.
(b) Viscous losses.
(c) Seal friction in the accumulator.

A servo-controlled actuator is fitted to the accumulator and the sensing and controlling elements are confined to the air bank and accumulator unit. Due to the transport delay of 35–70 ms between the accumulator and compensator, the compensator does not form part of the control loop, as this would require greatly reduced gain in order to obtain the necessary stability (Fig. 27).

As the air bank in a semi-active compensator system provides the main tension, the power requirement for the actuator is reduced by 80 to 90% of that required for a fully active compensator system.

TYPICAL OPERATIONS

Drilling

The improvements to the drilling operations may be summed up as giving the driller a better 'feel' for bit action.

Probably the biggest benefit of a heave compensator is the ability to change bit loading by simply varying the back-up air pressure. This is relevant when drilling varying formations with rock bits and is particularly advantageous when coring or turbo-drilling.

An analysis of 12.25-in and 8.5-in bits used in operations in the North Sea indicates increased service life and bit penetration rate.

In a fast drilling situation, where it is often difficult to match the fast line paired to the drilling ring, bottoming-out of a bumper-sub may occur, passing high shock-loads on to the bit and often breaking a piece. Even more serious is the damage to a diamond core head and bits and the possibility of stalling a turbine.

Spudding operations are less troublesome when using a motion compensator, particularly those entailing large diameter drilling or hole-opening and where under-reeming is unavoidable, since the shock loads on the bottom-hole assembly are minimized.

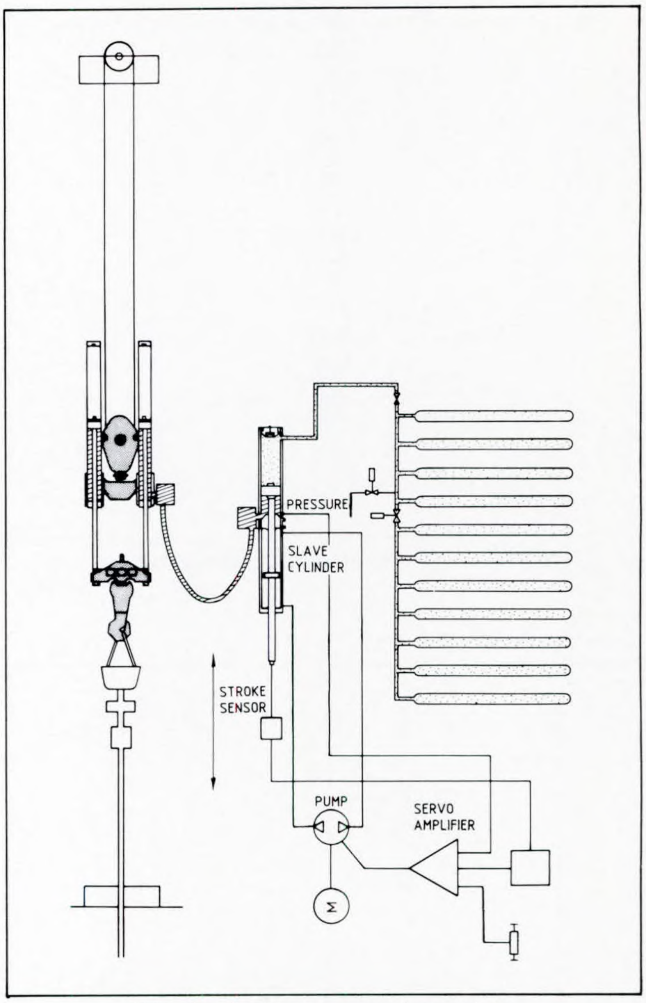


FIG. 27 Active compensator system

Landing equipment

BOP stacks

When using the bumper-subs of the motion riser's telescopic joint to set the BOP stack on the well-head, rig heave conditions should not exceed 4 ft. At the instant of landing, in order to reduce the impact as much as possible, almost 100% of the weight of the marine riser and the stack is taken by the riser tensioning system. With the rig heaving, however, and the stack connector following this motion exactly, heavy contact between the well-head and the stack connector cannot be avoided. Use of a motion compensator, however, can virtually eliminate the possibility of damage to the well-head during this operation.

The compensator will pick up 20–40% of the stack weight on the riser tensioning system, with the remainder being used to stroke the compensator, resulting in the stack connector remaining nearly motionless with respect to the well-head under most sea conditions. Once landed and locked, the compensator will continue to stroke whilst the tension is transferred totally to the riser tensioning system.

Casing strings

Landing of casing string in the well-head is altogether a safer operation if the casing's housing passes locking dogs that latch to the previous well-head; use of a motion compensator eliminates the possibility of shearing these dogs when the two housings are locked.

Cementing head installation

The installation of the cementing head is simplified using a motion compensator, particularly during high heave conditions. As the compensator strokes under the weight of the surface string, a constant distance is maintained between the hook and the casing; without compensation, the surface string would be unsupported after the casing hanger is landed. Therefore, whilst the vessel heaves, it is possible that the hook may interfere with the cementing head installation.

Hanging-off

Hanging-off because of bad weather or in the case of a kick is accomplished faster and safer with a motion compensator than with a bumper-sub. In the event of a kick, the BOPs may be closed and immediate killing of the well commenced without removal of the kelly. Once hung off, the surface string is instantly supported by the compensator and does not require the use of a small guideline tensioner or constant-tension winch.

Without the use of a compensator, the bag-type preventer must first be closed prior to installation of a circulating head and the string stripped in to hang off. Thus, the motion compensator provides a valuable improvement in the safety procedures whilst controlling the well.

Fishing

All aspects of fishing are greatly simplified with the use of a motion compensator. Increased control is possible and potential damage to spears and grapplers reduced.

Adjustment of air pressure permits controlled connection between fish and string, thus reducing the risk of jamming up the fishing tool and hence losing the ability to circulate. Milling, too, is accomplished in a more efficient manner and can be undertaken in worse weather than when using bumper-subs.

Production testing

With the string hung off in the well-head, a motion compensator can be activated to support the surface riser string and the Christmas tree. Whilst reacting to heave, the compensator will still retain the option of permitting the whole string to be lifted above the packer for instant killing of the well.

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Discussion

T. A. P. HAMILTON (Department of Energy, Petroleum Engineering Division): As deeper and deeper prospects are drilled offshore, there appears a need to increase the compensatable loads while running casing. At the other end of the scale, we need efficient compensators for light loads, e.g. during wire line operations. Are there any practical upper and lower bounds to effective compensation?

R. G. BODDIE FIMarE: The author referred several times in his presentation to the importance of weight reduction and yet the weight of the 12 in number HP air bottles must be very heavy compared with the rest of the system. Can anything be done to reduce the weight?

Could Mr Hatleskog comment on the 50-year wave (mentioned in a recent TV programme as being a fact of life) and say if it is allowed for in the design?

The author attaches great importance to fire-retardant hydraulic fluids and mentioned the risk of oil/air explosions in the pneumatic systems. Could he please explain why the fire risk is so important, as the risk of an HP fluid leak igniting in the open air would seem remote?

What problems have the author's company had with seals and for how long is a seal likely to last?

Author's Reply

In reply to Mr Hamilton, the drill string compensator's maximum working load is normally 600 000 lb, recently increased from 400 000 lb. The ability to compensate, expressed as load variation for a 600 000 lb unit as shown by Fig. 26, is $\pm 5\%$ at full load and $\pm 14\%$ at 50 000 lb. Very low loads are usually handled by tying the compensator to the top of the riser.

It is expected that the compensator will be increased to 1 000 000 lb maximum working load as the water depth increases and the compensator tasks are expanded. The dynamic range, currently at 10:1, may be increased through careful design and possibly using semi-active techniques as mentioned in the paper.

In reply to Mr Boddie, the weight of equipment on a drill rig or ship is of varying importance depending on where it is situated. The air pressure vessels (APVs) can be located in a suitable place, usually below the main deck.

A drilling vessel will discontinue drilling when the weather makes drilling impossible or in anticipation of bad weather. The disconnection process is relatively quick with a range of stages and options.

Careful consideration must be given to hydraulic fluid properties, with reference particularly to safety with regard to fire or explosions: firstly because they are situated in fire/explosion hazardous areas and, secondly, because the equipment could potentially generate an explosion 'dieseling'. The design of the equipment and selection of fluid must attempt to analyse, identify and remove all potentially hazardous conditions during normal and indeed abnormal operating conditions, assuming minimum maintenance. Indeed, it is important to consider the possibility of lubrication grease getting into the system and the lubrication oil from the compressors getting into the APVs.

The seals are particularly important in the compensator as they have direct impact on load variation, particularly at low loads, and maintenance requirements. The company initially used conventional packing but found this quite unacceptable, due in particular to high 'break-out' friction, resulting in 'jerkiness' at low loads.

A considerable design and testing programme was embarked upon, resulting in considerable improvements: 'break-out' friction reduced by a factor of 4 and 'running friction' reduced by a factor of 2.5.

The life of the seals has been found to be quite good and they can be expected to last 2 to 4 years, depending on operating conditions.

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