CSCanada
Advances in Petroleum Exploration and Development
Vol. 3, No. 2, 2012, pp. 31-41
DOI:10.3968/j.aped.1925543820120302.306

ISSN 1925-542X [Print]
ISSN 1925-5438 [Online]
www.cscanada.net
www.cscanada.org

Possible Alternatives for Deep-Water Gas Charged Accumulators

Mehdi Mir Rajabi[a]; Mahmood Amani[b],*

[a]Texas A&M University, USA.
[b]Texas A&M University at Qatar, Qatar.
*Corresponding author.

Received 8 May 2012; accepted 14 June 2012

Abstract

Gas Charged Accumulators are widely used in Drilling operations; however, the current Accumulator design methods are inadequate for Deepwater Drilling. Gas-Charged Accumulators are used in subsea drilling as well as production operations. One important application of accumulators is in blowout preventers (BOPs). BOP's are designed to shut in a well under pressure so that the well control procedures could be implemented. Control Systems for the BOPs should be highly efficient hydraulic systems and operate in as short a time as possible. Supplying enough volume of pressured hydraulic fluid to operate those emergency functions is essential. To have the necessary quantity of control fluid under pressure requires storing this fluid in accumulators.

Gas Charged Accumulators are the most commonly used accumulators in Drilling operations. These accumulators are not efficient at all in Deep waters, and there are not many alternatives for them. This paper looks into possible alternatives for Gas Charged Accumulators in Deep Waters.

Supplying enough volume of pressurized hydraulic fluid to operate the BOPs for emergency situations is essential for Deep Water Drilling. This requires storing the pressurized hydraulic fluid in accumulators. A problem may arise when the wellhead is at water depth of more than 3500 ft. In deep water drilling, the accumulators should be placed on the subsea BOP stack to reduce hydraulic response times and provide a hydraulic power supply in case of interruption of surface communication. Hydraulic fluid capacity of an accumulator may drop to 15% of its capacity on the surface and even less, depending on the water depth. The reason for this is that the nitrogen gas does not behave like an ideal gas as we go to very deep water, due to high hydrostatic pressure at that water depth.

We have to look for alternatives to Gas Charged Accumulators. It has to be something that is able to store energy, but unlike the nitrogen, its functionality should not be affected by the increasing hydrostatic pressure of water. The possibility of the use of springs and heavy weights as possible replacements for nitrogen in structure of accumulators will be discussed in this paper. High hydrostatic pressure of deepwater should not affect the functionality of these mechanical accumulators.

Transferring bank of accumulators to the surface and connecting them to the BOP with properly sized and rigid pipes can decrease response time to an acceptable extent to satisfy regulations and standards. This idea can be considered as an alternative solution too.

We have to include the hydrostatic pressure of water in the usable fluid calculation. A low pressure tank located on the sea-floor can dismiss the negative effect of high hydrostatic pressure of seawater. This alternative idea is also discussed.

Efficient deep water accumulators would reduce the number of accumulators required in deepwater and cut the cost of the project. With the advent of such efficient accumulators, we can hope that one of the numerous problems of deepwater drilling has been solved and we can think of drilling in even deeper waters.

Key words: Drilling operations; Deep-water gas charged accumulators; BOPs

Mir Rajabi, M., & Amani, M. (2012). Possible Alternatives for Deep-Water Gas Charged Accumulators. *Advances in Petroleum Exploration and Development*, *3*(2), 31-41. Available from: URL: http://www.cscanada.net/index.php/aped/article/view/j.aped.1925543820120302.306
DOI: http://dx.doi.org/10.3968/j.aped.1925543820120302.306

NOMENCLATURE

A	Cross sectional area of piston in piston accumulators,[L]^2, in.^{2}
API	American Petroleum Institute
BOP	Blowout Preventer
c	Spring Index
d	Wire Diameter, [L], in.
D	Mean diameter of cylindrical helical spring, [L], in.
D_p	Diameter of Piston in spring charged accumulator, [L], in.
D_w	Water depth, [L], ft
f	Deflection of spring, [L], in.
F_S	Force of the spring, [M][L][T]$^{-2}$, lbf
G	Shear modulus of elasticity, [M][L]$^{-1}$[T]$^{-2}$, psi
g	Gravity Acceleration, [L][T]$^{-2}$, ft/s^2
h	Vertical displacement, [L], ft
k	Curvature correction factor
K&C	Kill and Choke
L	Length of shaft, [L], ft
m	Mass, [M], lbm
MMS	Minerals Management Service
n	Number of active coils in a spring
NPD	Norwegian Petroleum Directorate
P	Load on the spring, [M][L][T]$^{-2}$, lbf
P_{max}	Maximum working pressure at any depth of water, [M][L]$^{-1}$[T]$^{-2}$, psia
P_{maxs}	Maximum working pressure at the surface, [M][L]$^{-1}$[T]$^{-2}$, psi
P_{min}	Minimum working pressure at any depth of water, [M][L]$^{-1}$[T]$^{-2}$, psia
P_{mins}	Minimum working pressure at the surface, [M][L]$^{-1}$[T]$^{-2}$, psi
P_n	Precharge pressure at any depth of water, [M][L]$^{-1}$[T]$^{-2}$, psia
P_{ns}	Precharge pressure at the surface, [M][L]$^{-1}$[T]$^{-2}$, psi
S	Shearing Stress, [M][L]$^{-1}$[T]$^{-2}$, psi
S_v	Safe shearing stress, [M][L]$^{-1}$[T]$^{-2}$, psi
S_v'	Corrected shear stress, [M][L]$^{-1}$[T]$^{-2}$, psi
V_{ac}	Actual volume of accumulator, [L]^3, gal.
V_U	Usable Fluid, [L]^3, gal.
W	Energy, [M][L]^2[T]$^{-2}$, Joule
W	Weight, [M][L][T]$^{-2}$, lbf
Z	Gas compressibility factor
Z_{max}	Gas compressibility factor at P_{max}
Z_{min}	Gas compressibility factor at P_{min}
Z_n	Gas compressibility factor at P_n

INTRODUCTION

Since the 1960's gas charged accumulators have been placed on subsea blowout preventers to reduce hydraulic response times and provide a local hydraulic power supply in case of interruption of surface communication. Accumulators are also used in subsea production control systems to provide local storage that allows smaller line sizes in control umbilicals[1]. Usable Fluid, which is declared as the amount of pressurized liquid that an accumulator can hold, noticeably decreases as drilling and subsea production moves to ever-deeper waters so that a large number of accumulator bottles is needed to store liquid required to do close and open functions in that depth of water. This issue of gas charged accumulators introduces itself as one of numerous obstacles to ultra-deepwater drilling technology. This behavior of accumulators is in part because of non-ideal behavior of compressed gas, usually nitrogen, in high ambient pressure at the sea floor where accumulators are located. Even if, nitrogen behaves like an ideal gas, the volume of usable fluid decreases, since the hydraulic fluid exhausts to the sea-water to reduce the length of umbilicals and pressure drop. So, the calculation of usable fluid should compensate for the hydrostatic pressure of water depth where hydraulic fluid is supposed to exhaust. Usable fluid volume of an accumulator decreases as the depth of operation of that accumulator increases. Figure 1 shows how the volume of usable fluid decreases as water depth increases. This graph is plotted for a 15-gallon bladder accumulator (V_{ac} = 13.7 gal.) with a maximum working pressure of 5,000 psi, minimum working pressure of 2,000 psi, and a precharged pressure of 1,800 psi.

Calculations giving the usable fluid of accumulators are based on the gas equation of state. Nitrogen, a common gas used in accumulators, behaves like an ideal gas as long as ambient pressure is not too high, so there is no problem on the surface or even in water depth of less than 3500 ft. In these circumstances, Boyle's Law, $P_1V_1 = P_2V_2$, along with a safety factor of 1.5 is used to calculate an accumulator's usable fluid. Dividing the volume of hydraulic fluid to close and open one annular-type preventer and all ram-type preventers from a full-open position against atmospheric wellbore pressure by the volume of usable fluid of each accumulator, the number of accumulators can be determined. But there is another story when accumulators are to be located in deeper waters. The subsea accumulator capacity calculations should compensate for the hydrostatic pressure gradient at the rate of 0.445 psi/ft of water depth[2]. In these circumstances, nitrogen used in accumulators does not behave like an ideal gas. It behaves like a non-ideal gas and we have to incorporate Z-factor in our calculations and also consider the process of expansion of nitrogen as an adiabatic process. This model that is the most accurate description of the expansion of the gas[3] calculates the pressure that the accumulator initially contains after a discharge. This method even gives a smaller usable fluid volume than what you see in Figure 1 for deeper waters. In ultra-deepwater drilling, the usable fluid of accumulators is much less than that of the same accumulators on the surface or in relatively shallow waters. So, we have to provide our hydraulic control system with a large number of accumulators that costs too much, and even worse, it would be very difficult to use this kind of accumulator in water depth of more than about 12,000 ft where today's investigators in the petroleum industry are trying to reach.

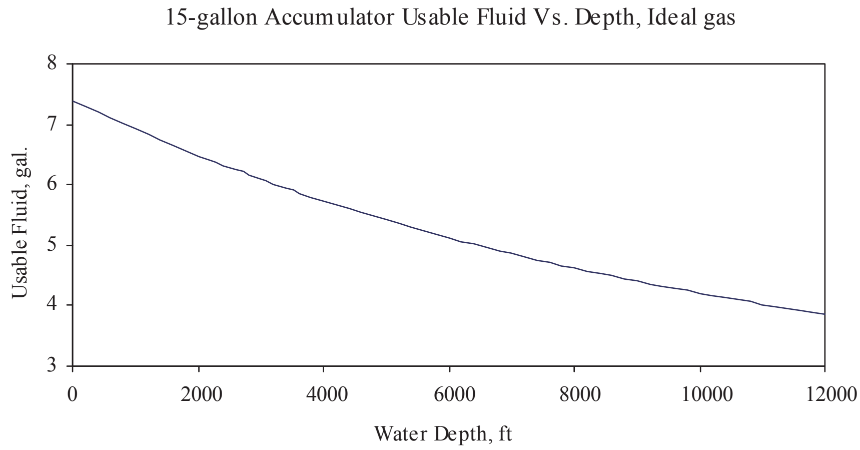

Figure 1
Usable Fluid Decreases as Water Depth Increases

Since the volumetric efficiency of gas charged accumulators is so low in ultra-deep water and no replacement for gas has been found yet, some investigators are trying to find a way to transfer all the BOP equipment to the surface but some others are looking for other alternatives.

Replacing conventional accumulators by another kind of accumulator whose functionality is not affected by the hydrostatic pressure, may provide a solution.

Spring-loaded accumulators and weighted accumulators are discussed in this paper as two possible replacements for gas charged accumulators. Lifted weight and deflected springs can store energy like compressed gas. The work done on a deflected spring is stored in it and can be released at the time required. This energy can be used to run the piston inside a cylinder like the compressed gas in the piston type of gas charged accumulator. There is the same story about lifted weight; the work of the weight of the body in vertical displacement is stored as potential energy and can be recovered to run the piston against the hydraulic power in a cylinder.

In most cases, a properly sized rigid conduit to conduct power fluid from accumulators mounted on the surface to the BOP stack is capable of providing sufficient flow rates to operate BOP functions within API mandated limits, even in extreme water depths[4].

Exhausting hydraulic fluid into a low pressure tank may dismiss the negative effect of troublesome pressure of seawater on the BOP control system.

1. ACCUMULATOR CAPACITY REQUIREMENTS

The BOP control system shall have a minimum stored hydraulic fluid volume, with pumps inoperative, to satisfy the greater of the two following requirements:

Close from a full open position at wellbore pressure, all of the BOP's in the BOP stack, plus 50% reserves[5].

The pressure of the remaining stored accumulator volume after closing all of the BOP's shall exceed the minimum calculated (using the BOP closing ratio) operating pressure required to close any ram BOP (excluding the shear rams) at the maximum rated wellbore pressure of the stack[5].

The hydraulic fluid volume recoverable from the accumulator between the maximum operating pressure and the minimum operating pressure is defined as its usable fluid. The usable fluid for a hydro-pneumatic accumulator is not constant and changes depending the maximum and minimum operating pressure and precharge pressure.

2. MODELS OF USABLE FLUID CALCULATION

2.1 Boyle's Law Method

In this method we assume that gas inside accumulators behaves like an ideal gas (Eq. 1). The parenthesis in Eq. 1 is called usable fluid fraction. Traditionally subsea accumulator capacities have been calculated based on this method: Boyle's Law model[1]. This method does not take into account temperature changes that occur temporarily in the accumulator while charging and discharging. This method has given adequate accuracy in surface and relatively shallow subsea systems, partially due to the generous safety margins that are built into mandated capacities[1]. For operation in 3,000 ft of water on 3,000 psi control systems, using Boyle's Law to calculate usable subsea accumulator volumes results in overstating capacity slightly more than 15% [3]. This has not been a problem because of design factor of 1.5[4]. This method is accepted by API 16D.

$$V_U = \left(\frac{P_{ns}}{P_{\min s}} - \frac{P_{ns}}{P_{\max s}} \right) V_{ac} \tag{1}$$

Accumulator system pressure is higher at any water depth than at the surface in order to compensate for hydrostatic head. The subsea accumulator bottle capacity calculations should compensate for hydrostatic pressure gradient at the rate of 0.445 psi/ft of water depth[2]. There is the same story about precharge pressure; the precharge pressure should also compensate for the water depth. So, we can rewrite Eq. 1 as Eq. 2 to calculate accumulator usable fluid installed in relatively shallow water, less than 3500 ft.

$$V_U = \left(\frac{P_{ns} + 0.445 D_w}{P_{\min s} + 0.445 D_w} - \frac{P_{ns} + 0.445 D_w}{P_{\max s} + 0.445 D_w} \right) V_{ac} \tag{2}$$

The best applications of Eq. 2, which has been developed based on Boyle's Low, are those of around atmospheric conditions. The greater the deviation from atmospheric conditions, the greater the discrepancy between Boyle's Law and actual gas properties.

2.2 Gas Compressibility Model

If we install the accumulator in deeper water, we can not use Eq. 2 any longer under these conditions. To get more accurate results, we have to switch to another method. We must include the gas compressibility factor into our calculations. This model gives us less usable fluid than the Boyle's Law model but is more accurate.

At pressures of 5,000 psi and above, compressibility becomes a notable factor.[4] How much to reduce the pressure in order to determine a more accurate volume depends on the temperature and pressure.

To more accurately compute the usable volume in a theoretical accumulator, the ideal precharge pressure as well as the minimum and maximum pressures will have to be adjusted by the appropriate Z-factor. The effect of gas compressibility is incorporated into Eq. 3. The Z-factor can be determined by existing tables, graphs, or methods.

$$V_U = \left(\frac{P_n / Z_n}{P_{\min} / Z_{\min}} - \frac{P_n / Z_n}{P_{\max} / Z_{\max}} \right) \times V_{ac} \tag{3}$$

Where $P_n = 0.445 \times D_w + 14.7 + P_{ns}$. The change in temperature during charging and discharging time is neglected.

A typical deepwater drilling environment is used for the accumulator in the example, and the accumulator is installed at the depth of 7,000 ft. For this example pressures and corresponding compressibility factors are as follows[4]:

$$P_n = 0.445 \times 7{,}000 + 14.7 + 1{,}800 = 4{,}930 \text{ psia}$$
$$P_{\max} = 0.445 \times 7{,}000 + 14.7 + 5{,}000 = 8{,}130 \text{ psia}$$
$$P_{\min} = 0.445 \times 7{,}000 + 14.7 + 2{,}000 = 5{,}130 \text{ psia}$$
$$Z_n = 1.19, Z_{\max} = 1.45, Z_{\min} = 1.2$$

We substituted these values in Eq. 3 and we got 3.2 gallons for the usable fluid volume of the accumulator whose usable fluid at the depth of 3,500 ft was 5.9 gallons. The value of 3.2 is not the exact usable fluid volume, but it is more accurate than the previous method (Boyle's Law method) at that depth, and yet is proving that the usable fluid volume of gas charged accumulators dramatically decreases in deeper waters.

2.3 Adiabatic Non-Ideal Gas Model

In deeper waters, higher pressure and full accumulator depletion produces rapid discharges, which suggest adiabatic gas expansion rather than the more common isothermal expansion process. Isothermal gas expansion assumes that changes in volume or pressure take place at constant temperature[4]. Functioning the BOP stack releases hydraulic fluid from the subsea accumulators, which allows the precharged gas to expand. When gas expands, its temperature decreases. So, in order for the gas temperature to be held constant and satisfy an isothermal expansion process, the gas would have to expand slowly. A slower expansion rate permits the gas to gain heat from the surrounding environment and, thus, maintain a constant gas temperature.

In contrast, an adiabatic process assumes no heat transfer. Practically any process can be made adiabatic if it occurs fast enough and higher gas precharge pressures tend to produce higher discharge rates[4]. With BOP functions occurring in the 20 second range, there is minimal time for significant heat transfer[4]. Therefore, it is logical to assume that the relatively quick discharge of gas in subsea accumulators at higher pressures tend toward an adiabatic process.

We calculate usable fluid for the accumulator mentioned above to help us figure out the significant difference in results between adiabatic and Boyle's Law model. Considering the same parameters but assuming an adiabatic expansion process, the amount of usable fluid in a 15-gallon accumulator is determined to be only 2.2 gallons. Eq. 4 is used to calculate the usable fluid in this model[6]. In this method, gas is considered as a non-ideal gas, and so we have adjusted pressures with compressibility factor (*P/Z*) before using them in Eq. 4[4].

$$V_U = \left(\left(\frac{P_n}{P_{min}} \right)^{0.71} - \left(\frac{P_n}{P_{max}} \right)^{0.71} \right) \times V_{ac} \tag{4}$$

2.4 Cameron Model

Cameron investigators believe that typical discharge times allow enough heat to flow between the gas and the accumulator to cause serious deviations from both Boyle's Law and adiabatic models. So, they used their own model to calculate usable fluid of accumulators located at ever-deeper waters. The model includes both gas compressibility and effects of heat transfer between the gas and accumulator bottle[1]. They claim that the results are particularly helpful in design of any accumulator system operating at pressures of 5,000 psi and higher. As can be seen from Figure 2, there is not a significant difference between the adiabatic expansion calculation and results of Cameron's computer program at high pressures.

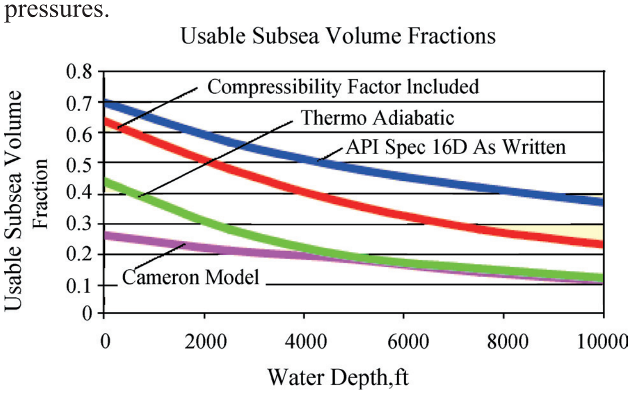

Figure 2
Comparison of Different Models of Usable Fluid Calculation[3]

The best, and thus recommended, method of calculating gas behavior is to assume constant entropy, or adiabatic expansion. Entropy is the measure of order in a system; this correction assumes that in changing temperature and pressure while the gas expands, the order remains constant. This is a textbook thermodynamic assumption that fits the physical situation well. From the starting entropy (at a given temperature and pressure), the gas is expanded adiabatically to the target pressure, P_{min}. The density of gas at this condition is then read. For precharge pressure, one minus the ratio of beginning to ending densities at the two conditions results in the usable fluid fraction[3].

However, as can be seen from Figure 2, the usable fluid fraction dramatically decreases when going to the deeper water. To compensate for this, a large number of accumulators should be mounted in the hydraulic unit at the seafloor. Figure 2 shows that usable volume fraction at the depth of 10,000 ft may be 0.12. It means that an accumulator's usable fluid at that depth should be 0.12 of its actual fluid capacity.

Usable fluid fraction of accumulators in use is used to calculate the number of accumulators required at the depth of interest. To calculate the number of accumulators, we must divide the volume of hydraulic fluid required to manipulate the stack and meet the regulatory agency requirement by the calculated usable fluid fraction. As an example assume that we have calculated this volume of fluid to be 265.5 gallon. Since the usable fluid fraction decreases as water depth increases, clearly the number of accumulators increases (Figure 3). As Figure 3 demonstrates, based on adiabatic method calculation which is the most accurate method of calculation of deepwater accumulator usable fluid, we need more than 150 accumulators at water depth of 10,000 ft.

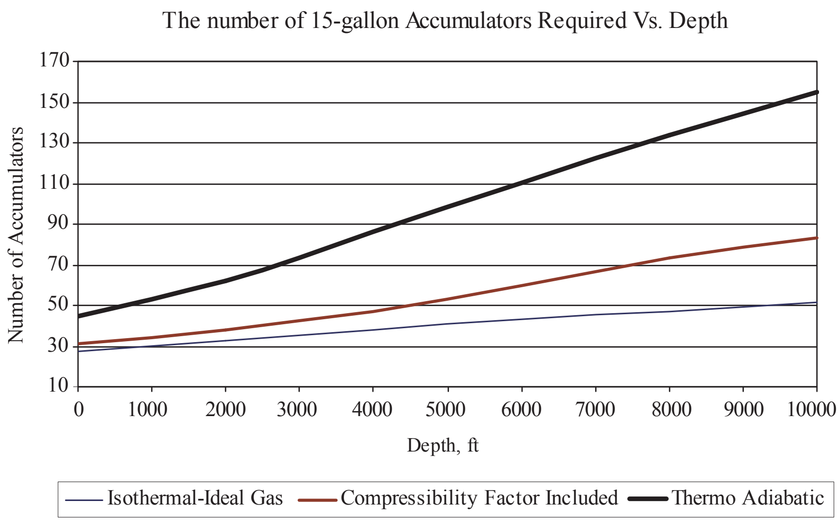

Figure 3
The Number of Accumulators Increases in Deeper Waters

3. POSSIBLE REPLACEMENTS FOR GAS CHARGED ACCUMULATORS

3.1 Spring Loaded Accumulators

The idea of using a spring instead of gas in accumulators comes from the fact that a deflected spring can store energy in it and this energy can be recovered at the time we need. And unlike the gas, the energy stored in a spring is not a function of its ambient pressure. A spring stores the same amount of energy at any water depth as it would do on the surface.

3.1.1 Structure and Operation

A spring loaded accumulator may look like what you can see in Figure 4. When the power fluid is pumped into this kind of accumulator, the spring inside the accumulator is deflected by the piston which separates the fluid side of the accumulator from its spring side. The piston must be completely sealed against the fluid leakage to the spring side of the accumulator. This problem has already been solved in the industry.

Like conventional accumulators, spring accumulators should keep 200 psi or more above the precharge pressure in the system after it has been discharged, as API 16D mandates. It means that the spring should not extend to its free original length. Figure 5 shows operation of spring loaded accumulators.

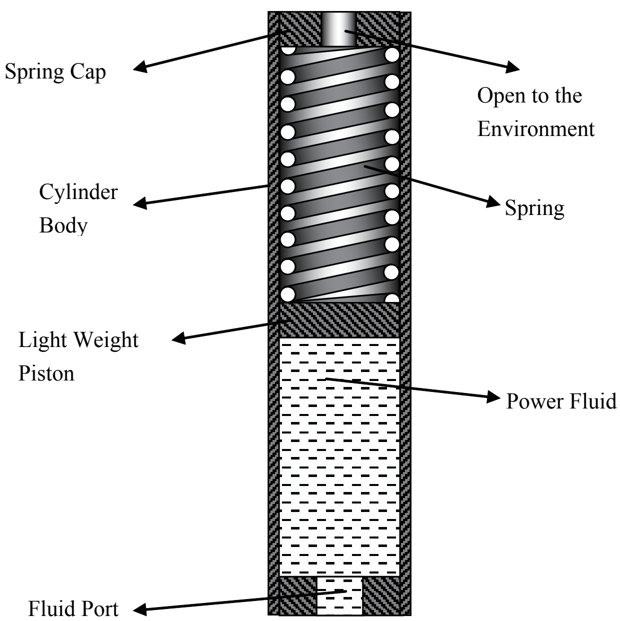

Figure 4
Subsea Spring-Loaded Accumulator

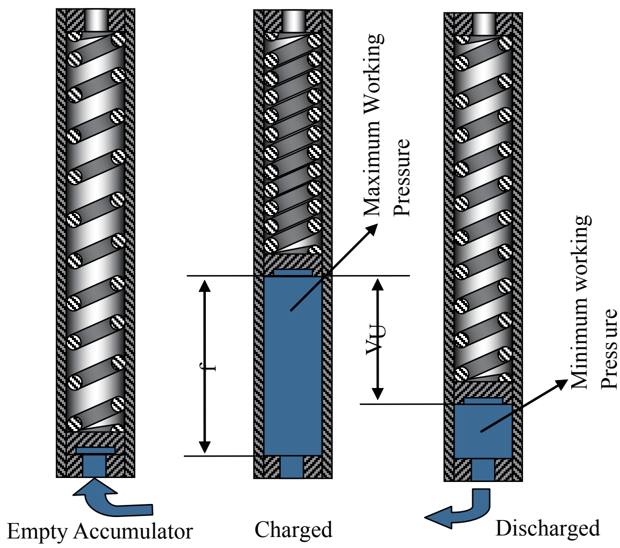

Figure 5
Operation of Spring Charged Accumulators

3.1.2 Spring Design

Our primary objective in a spring design is generally to obtain a spring which is able to be loaded high enough to provide maximum working pressure. This spring should fit into a reasonable sized cylindrical body of spring loaded accumulator. This accumulator must be able to store the highest possible volume of power fluid, so that we can reduce the number of accumulators as much as possible. Since the spring is to be exposed to the corrosive sea water, we have to consider this factor in spring material selection.

The first step in the design is to determine the load and deflection for a given maximum working pressure and the quantity of usable fluid required.

The following formulas are used to design helical compression springs of round wire[7,8].

$$S_v = \frac{8\,P\,D}{\pi\,d^3} \tag{5}$$

$$S_v' = \frac{8\,P\,D}{\pi\,d^3}k \tag{6}$$

$$k = \frac{4c-1}{4c-4} + \frac{0.615}{c} \tag{7}$$

$$c = \frac{D}{d} \tag{8}$$

$$f = \frac{8\,n\,D^3 P}{d^4 G} \tag{9}$$

where P is load on spring, D is mean diameter of spring, d = wire diameter, f = deflection, S_v = maximum allowable shear stress in wire, S_v' = corrected shear stress, k = Curvature correction factor, and n is the number of active coils in the spring.

3.1.3 Load on the Spring, P (F_s)

Figure 6 shows a free body diagram of a piston inside a subsea spring charged accumulator.

As can be seen from Figure 6, equilibrium equation of forces is as follows:

$$F_s + P_{hydrostatic} A = \left(P_{\max} + P_{hydrostatic}\right) A \tag{10}$$

We solve this equation for F_s:

$$F_s = P_{\max} A \tag{11}$$

F_s is the load on the spring that is directly proportional to the maximum working pressure and square of piston diameter:

$$F_s = \frac{\pi}{4} P_{\max} D_p^{\ 2} \tag{12}$$

3.1.4 Deflection of Spring, f

Deflection required in the spring can be determined based on the volume of stored power fluid needed in one accumulator unit. With a constant piston cross sectional area, the deflection of the spring is directly proportional to the usable fluid:

$$V_p = A \times f \tag{13}$$

where A is the cross sectional area of the piston and V_p is the volume of fluid pumped into the accumulator to get the maximum working pressure at deflection f.

It can be concluded from Eq. 13 and Eq. 11 that A, V_p, f, and the load on the spring P, are dependent on each other and none of them can be determined separately. It means that any changes in one of them would change the other parameters. Any changes in the diameter of the piston changes the load on the spring and f.

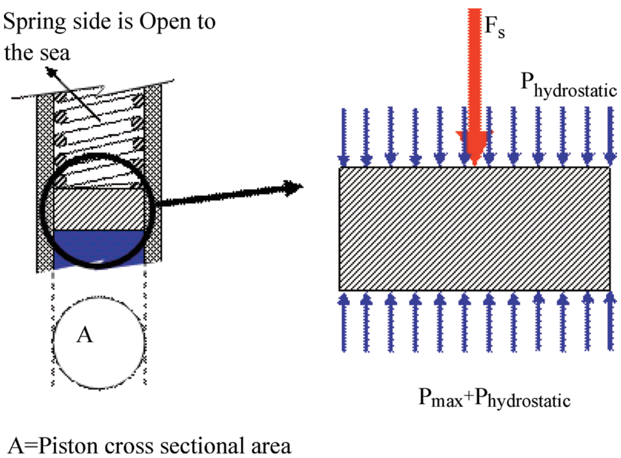

Figure 6
Free Diagram of Piston in a Subsea Spring Charged Accumulator

3.1.5 Material

Referring to Eq. 11 and substituting some common values for P_{max} and A, we can find that the spring we are going to design needs to withstand a really huge load; may be a load higher than the load which helical springs in available heavy equipment currently bear. Our primary objective of material selection is to choose a spring material which is used in manufacture of heavy duty springs.

The alloy spring steels have an important place in the field of spring materials, particularly for conditions involving high stress and where shock or impact loadings occur. Alloy spring steels can also withstand both high and low temperatures. All these alloys are used in springs for equipment in wire sizes frequently under ¼ in. diameter and up to 1/2 in. Since we need to design a spring to meet a high load, we should select a material whose larger bar sizes are available too. The alloy steels are generally recommended for the larger sizes and higher stresses.[7] Annealed bars of alloy springs are available from 3/8 to 2 in. or larger.[9] Hot-rolled Nickel-Chromium-Molybdenum alloy steel bars, ASTM A 331, are available up to 2.5 in. in diameter.[9]

Helical compression springs having bar diameters larger than about 5/8 in. are commonly coiled hot and then heat-treated, since it is not practical to wind such springs cold.[7] So, the tentative spring we intend to design would be kind of hot-wound compression spring. Currently, hot-wound springs are used in automotive, railroad, armament, and heavy equipment.

3.1.6 Shear Modulus of Elasticity, G

In calculating deflections and stresses, a modulus of rigidity of 10.5×10^6 psi is frequently used by spring manufacturers for alloy steel bars in hot coiled springs[7]. Carlson suggests 10.75×10^6 for G in his book[9].

Some processes may reduce the modulus G. Overstraining of the spring such as occurs in presetting tends to reduce this modulus. However, it would be accurate enough to use these values in order for our primary analysis of deflection, and load of spring, and consequent shear stress in spring wire.

3.1.7 Working Stress, S_v

Carlson in his book[9], presented the elastic limit of materials as a percentage of their ultimate tensile strength. For alloy steel bars where they are in torsion, the maximum shear stress without permanent set should be 60 to 70 percent of ultimate tensile strength. Ultimate tensile strength for alloy steel bars varies from 180,000 to 200,000 psi.

There are some graphs in some topic related books and manuals, which suggest working stresses with respect to bar diameters[7,8]. Maximum allowable working stresses varies with bar diameter; allowable working stresses are higher for smaller bar diameters[7,8]. However, we use the values suggested by Carlson and Wahl for our primary analysis.

Now, let's get into design of a helical compression spring for our tentative accumulator. Table 1 is based on the formulas given above in this chapter. d is diameter of wire, D = Pitch diameter (center to center of wire), P = safe working load for the maximum allowable stress, and f is deflection of one coil for the safe working load P.

Table 1
Safe Working Load *P* and Deflection *f* for Hot-Wound Helical Compression Spring

Maximum Allowable Shear Stress = 108,000 psi, G = 10,750,000 psi														
d, in.	D, in.	3	4	5	6	7	8	9	10	11	12	13	14	15
1.0	P	8,948	7,555	6,473	5,644	4,995	4,477	4,055	3,705	3,409	3,157	2,940	2,750	2,583
	f	0.184	0.368	0.616	0.929	1.305	1.747	2.252	2.823	3.457	4.157	4.921	5.749	6.642
1.5	P		21,292	19,010	16,995	15,302	13,887	12,698	11,689	10,823	10,074	9,421	8,845	8,335
	f		0.205	0.358	0.552	0.790	1.070	1.393	1.759	2.168	2.620	3.115	3.653	4.243
2.0	P			38,865	35,790	32,845	30,213	27,906	25,890	24,126	22,574	21,202	19,982	18,891
	f			0.231	0.368	0.536	0.737	0.969	1.233	1.529	1.858	2.218	2.611	3.036
2.5	P				61,634	57,855	54,031	50,457	47,208	44,285	41,662	39,307	37,187	35,272
	f				0.260	0.387	0.540	0.717	0.921	1.150	1.404	1.684	1.990	2.322

High pressure in spring charged accumulator can be obtained by exerting a high force on the piston which is separator of spring and liquid. Table 1 shows that, the maximum possible force obtained from a helical compression spring would be 61,634 lbf. It is not exactly the maximum force a deflected compression spring may provide, but it is accurate enough not to mislead us. Even a safe load of 100,000 lbf would not solve our problem.

The table above shows that the pitch diameter of this spring is 6 in. So outside diameter of this spring would be 8.5 in. (6 in. + 2.5 in. = 8.5 in.). Consider 8.5 in. (Outside diameter of spring) as diameter of piston in our accumulator. So, the pressure which this spring can provide us would be:

$$P_{\max} = \frac{P}{A} = \frac{61{,}634 lbf}{56.75 in^2} = 1{,}086 \text{ psi}$$

This pressure is not good for hydraulic BOP control purposes. Common maximum working pressure for subsea BOP control systems would be 5,000 psi. Of course we can put the spring in an order with other parts to get that much pressure. But it is not based on sound engineering principles and practices. The force of 61,634 lbf should be exerted on an area of 12.327 in^2 to provide a pressure of 5,000 psi (Figure 7). In this case, to store just 5 gallons of hydraulic fluid the stroke of the piston inside the cylinder should be 94 in. It means that the maximum deflection of spring should be 94 in. Since the deflection of one coil of spring mentioned above is 0.260 in. (Table 1), we need 361 (94 / 0.260) coils of spring. Regarding that the bar diameter is 2.5 in., solid length of such spring would be 902 in. (2.5 × 361). Since the length of this spring is 996 in. (902 + 94), the height of this accumulator would be 1090 in. (996 + 94) that is really too long and is not feasible.

We added 94 to 996, because it is the length of the shaft that connects the larger piston to the small piston (L = 94 in.).

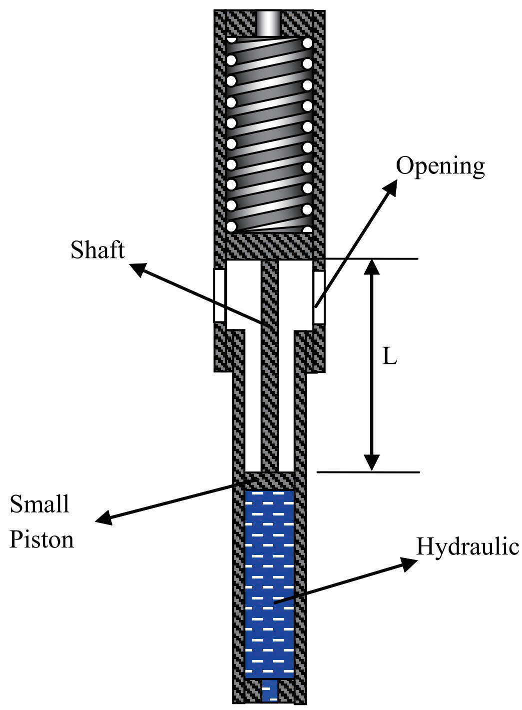

Figure 7
Smaller Piston to Get Higher Pressure

Based on Eq. 12 the load which we need the spring to exert on the piston to provide us with the pressure of 5,000 psi is 283,725 lbf which is not really a load which a compression spring can bear. Indeed, there is no helical compression spring that can tolerate this load[10].

3.2 Weighted Accumulator

When we are lifting an object, we are doing work on it. This work done on the lifted body is stored as potential energy in it and can be recovered as kinetic energy. This energy can be calculated by Eq. 14.

$$W = wh = mgh \tag{14}$$

where w is the weight of object, m is mass, g is gravity acceleration, and h is vertical displacement of object.

Figure 8
Heavy Weight Required in Weighted Accumulator

This mass should exert its weight as vertical force on a piston with a cross sectional area small enough to provide us the pressure required in subsea BOP control system, for example 5,000 psi. A piston with a cross sectional area of 50 in^2 (8 in. diameter) should be loaded by 250,000 lbf (249,912 lbm), which is 386 ft^3 of lead. Maybe a machine like what you see in Figure 8 can do that for us, which is not a good replacement for gas charged accumulators. May be these weighted accumulators had been the first shape of accumulators which were replaced by gas charged accumulators in early days when accumulators were needed[10].

3.3 Moving Bank of Accumulators to the Surface

What if we use the energy of hydraulic fluid column to compensate for the hydrostatic pressure of seawater? For that, we need to transfer the source of pressurized fluid to the surface so that the hydrostatic pressure of hydraulic fluid inside the conduits lined up from BOPs on the seafloor to the bank of accumulators on the surface is used in the aid of pressure of fluids inside the accumulators; In other words, it is suggested that all accumulators stay on the surface (Figure 9).

As we have said earlier, since hydrostatic pressure of seawater is applied to the vent point of power fluid on the pilot valve while operating a function on the subsea BOP, subsea accumulator capacity calculations should compensate for the hydrostatic pressure gradient at the rate of 0.445 psi/ft of that water depth. Now that we are installing accumulators on the surface, the effect of hydrostatic pressure of hydraulic fluid inside the conduits can dismiss the effect of hydrostatic pressure of seawater to a great extent. So, for a hydraulic fluid of 7 lb/gal (0.364 psi/ft), surface accumulator capacity calculations should compensate for the pressure gradient at the rate of 0.081 psi/ft which is the difference pressure gradients of seawater and hydraulic fluid.

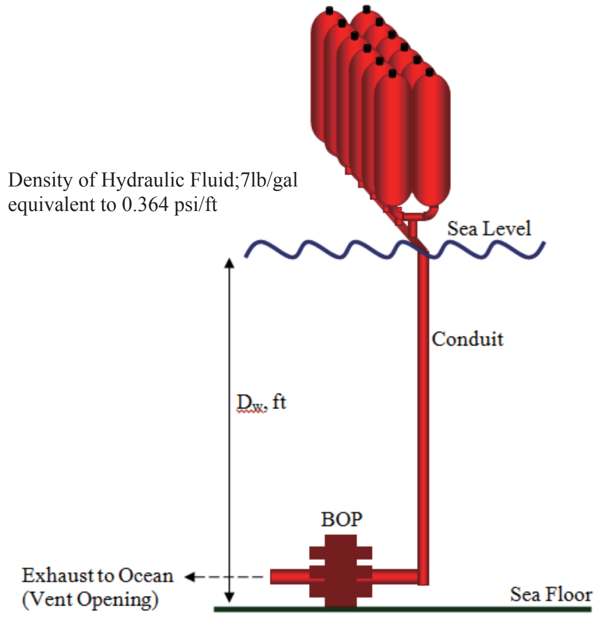

Figure 9
Hydrostatic Pressure of Hydraulic Fluid in Compensate for Hydrostatic Pressure of Seawater

0.445 psi/ft – 0.364 psi/ft = 0.081 psi/ft

So we can rewrite Eq. 1 for this case as follows:

$$V_U = \left(\frac{P_{ns} + 0.081 D_w}{P_{\min s} + 0.081 D_w} - \frac{P_{ns} + 0.081 D_w}{P_{\max s} + 0.081 D_w} \right) V_{ac} \qquad (15)$$

For example for the water depth of 10,000 ft, usable fluid calculations should compensate for just 810 psi which is equivalent to 1,820 ft of water depth; as if we are drilling at the water depth of 1,820 ft which is not considered as deepwater.

For drilling purposes, properly sized and rigid umbilicals to conduct power fluid from accumulators to the BOP stack is capable of providing sufficient flow rates to operate BOP functions to satisfy API mandated limits, even in extreme water depths.

However, certain BOP control systems must have dedicated sources of stored hydraulic fluid located on the BOP stack. In case of interruption of surface communication, it would be really a disaster not to be able to operate BOPs.

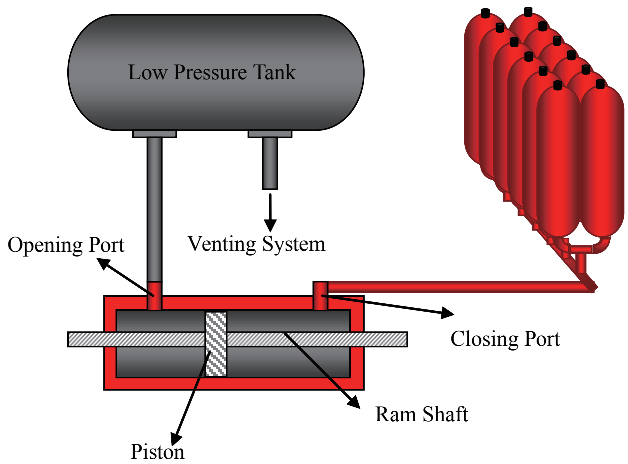

Figure 10
Low Pressure Tank System

Another alternative with the same concept as the previous system might be interesting to researchers. In this system a properly determined column of high density fluid, like mud or high density completion fluid, which is separated from hydraulic fluid, can provide pressure required for control pods to operate BOP functions. The high density fluid should be heavy enough to provide pressure required for the new BOP hydraulic control system. For example for a water depth of 4,000 ft, the minimum working pressure required right to the subsea control pods would be:

4,000 psi × 0.445 psi/ft + 2,000 psi = 3,780 psi

3,438 ft of 20 lb/gal mud along with 562 ft of 7 lb./gal hydraulic fluid can provide this pressure. Of course, in actual case, we have to include friction pressure drop. An example of this system is shown by a simple sketch in Figure 11. In such a system, as the hydraulic fluid is being depleted, the height of the mud column grows, thereby, increasing the pressure available at the BOP. Re-pressurization is simple, and is similar to current technology.

The seals may be problem in the idea presented in this accumulator system. The settling of mud into the hydraulic fluid could be a serious problem, even with multiple seals. A u-tube addition with a drain at the bottom may be a solution to that potential problem (Figure 12).

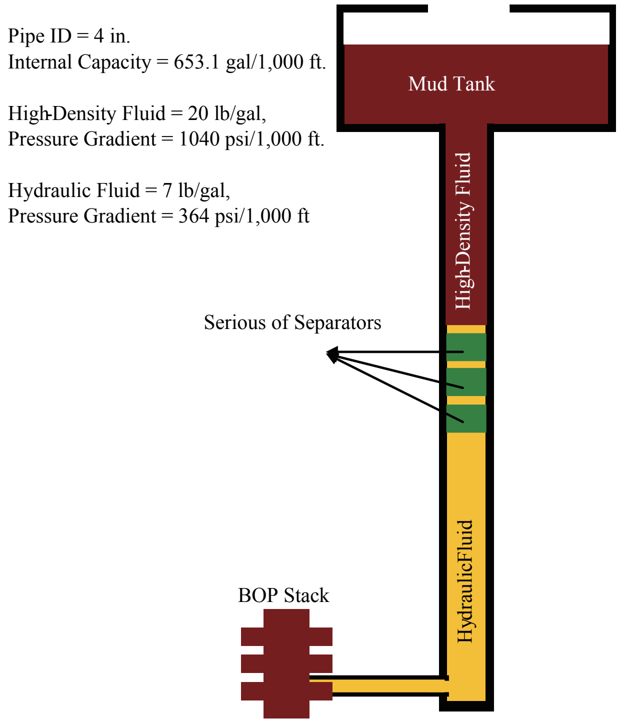

Figure 11
Schematic of High Density Liquid Accumulator System

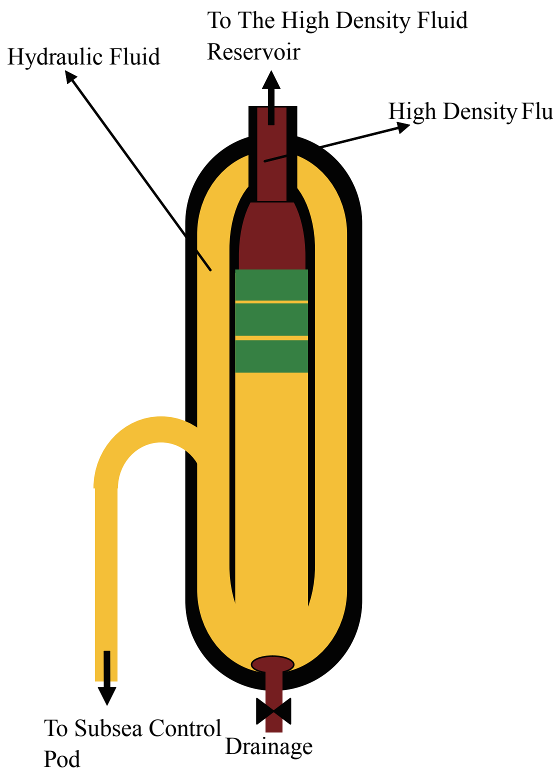

Figure 12
U-Tube Solution

3.4 Low Pressure Tanks

In conventional BOP control systems, the hydraulic fluid is supposed to exhaust to the seawater each time we perform a function on the BOP stack. That is the reason why we have to include the hydrostatic pressure of water in the usable fluid calculation. As a matter of a recommendation, a low pressure tank located on the seafloor can dismiss the negative effect of high hydrostatic pressure of seawater. If we connect pilot valves, which are installed on the control pods, to a tank with a low pressure (for example atmospheric pressure), accumulator capacity calculations will not need to compensate for hydrostatic pressure of seawater as mandated by relating regulations and standards. Implementation of this system requires a new design for pilot valve and control pod so that a properly sized conduit connects vent opening of the pilot valve to the low pressure tank. This low pressure tank should be provided with a port to a venting system so that it can be evacuated after a number of functions operated on the BOPs (Figure 10).

Work is now in progress within the deepwater drilling industry to evaluate the feasibility of all BOP equipment (BOP stack, bank of accumulators, and all other parts of hydraulic control system) to the surface as an alternative to current system.

CONCLUSION

Since the volumetric efficiency of gas charged accumulator drops in deep waters, we need a large number of this kind accumulator to do functions on the BOP stack at the sea floor. In this paper we evaluated the feasibility of some alternatives. Here are the conclusions from this study;

It is possible to build spring loaded accumulator, but this kind of accumulator is not practical for BOP control purposes where a large amount of energy and high pressure fluid is required.

Replacement of gas charged accumulators by weighted accumulators is not feasible; tons of the heaviest and yet cheapest metal is required to be lifted very high to store the energy needed to perform the functions on the BOP stack.

Using properly sized and rigid conduits from bank of accumulators on the surface to the subsea BOPs will satisfy API mandated time limits, but there is still a big concern about interruption of surface communication.

It is worth to conduct a further research on the idea of using low pressure tank installed on the seafloor to exhaust hydraulic fluid into it at any time a function is operated on BOPs.

ACKNOWLEDGEMENTS

The authors would like to express their appreciation for the valuable guidance provided by Dr. Hans C. Juvkam-Wold and Dr. Jerome Schubert.

REFERENCES

[1] Good, A. C., & McAdams, P. J. (2001). Mathematical Prediction and Experimental Verification of Deep Water Accumulator Capacity. *PAPer OTC 13234 presented at the 2001 Offshore Technology Conference, 30 April - 3 May, Houston, Texas*.

[2] RP53 (1997). *Recommended Practices for Blowout Preventers Equipment Systems for Drilling Wells* (3rd ed.). Washington, DC: API.

[3] Sattler, P. S. (2002). BOP Subsea Hydraulic Accumulator Energy Availability, How To Ensure You Have What You Need. *SPE 74469 presented at the 2002 IADC/SPE Drilling Conference, 26-28 February, Dallas, Texas.*

[4] Curtiss, J. P., & Buckley M. (2003). Subsea Accumulators – Are they a False Reliance? *SPE 79881 presented at the 2003 IADC/SPE Drilling Conference, 19-21 February, Amsterdam, The Netherlands*.

[5] Spec. 16D (1993). Specification for Control Systems for Drilling Well Control Equipment. Washington, DC: API.

[6] Hydac Corp. Accumulator Division (2002). *Hydac International Accumulator, Bladder, Piston*, *Diaphragm*. Bethlehem, Pennsylvania.

[7] Wahl, A. H. (1963). *Mechanical Spring* (2nd ed.). New York City: McGraw-Hill Book Co., Inc.

[8] Baumeister, T., Avallone, E. A, & Baumeister III, T. (1978). *Marks' Standard Handbook for Mechanical Engineers* (8th ed.). New York City: McGraw-Hill Book Co., Inc.

[9] Carlson, H. (1978). *Spring Designer's Handbook*. New York City: Marcel Dekker, Inc.

[10] Amani, M., Mir Rajabi, M., Juvkam-Wold, H. C., & Schubert, J. J. (2006). Current Accumulator Design Methods are Inadequate for Deepwater Drilling. *OTC Paper 17990, Presented at the 2006 Offshore Technology Conference, 1-4 May, 2006, Houston, TX, USA*.

[11] Amani, M., Mir Rajabi, M., Juvkam-Wold, H. C., & Schubert, J. J. (2006). Possible Alternatives for Gas-Charged Accumulators in Deep Waters. *SPE Paper 100305, presented at the 2006 International Oil & Gas Conference and Exhibition in China (IOGCEC), 5-7 December, 2006, Beijing, China*.