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The Effect of Friction in Passive and Active Heave Compensation of Crown Block Mounted Compensators

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Abstract: This paper studies the effects of friction model during passive and active heave compensation of offshore drilling equipment. The main purpose of heave compensation while drilling from vessels or semi-submersible platforms is to maintain the drilling operation unaffected by the wave induced motion. The investigated system is of an existing crown mounted compensator. A model of the system is developed which includes mechanics, hydraulics and pneumatics. The passive heave compensation scheme is described including force equalising hydraulic cylinders. In this paper the detrimental effect of friction on the heave compensation performance in both passive and active heave compensation is investigated and discussed.

Keywords: Friction, Active Compensation, Passive Compensation, Electro-Hydraulics Systems.

1. INTRODUCTION

In general, heave compensation is introduced in order to facilitate offshore operations so that they may be performed in a safe, effective and controlled way as rough weather as possible. Today this type of heave compensation is utilising large hydraulic-pneumatic cylinder-piston arrangements that effectively suspend the drill-string or the riser in a weak gas spring. The weakness of the spring is designed so that the spring force is approximately constant independent of the heave motion induced by the surrounding conditions. Active heave compensation is introduced by adding actively controlled hydraulic cylinders that counteract non-linearities of the passive system and based on position measurements attempt to maintain the position or the tension in the upper part of the drill-string at a constant level.

A major obstacle in advances within model based research and development of heave compensation equipment, in general, is the difficulties and costs associated with experimental work and the more or less total absence of prototypes. Despite this, several interesting research work has been carried out in the later years. On passive heave compensation Skaare and Egeland (2006) as well as Hatleskog and Dunnigan (2007) has investigated model based approach. Recently, Ottestad et al. (2010) has shown that the friction in the main cylinder often has a strong detrimental effect on the performance of passive heave compensation systems. It has also been shown, Hansen and Ottestad and Hansen (2011), that the ability to predict the Coulomb friction is the most important parameter when employing direct force compensation. It has, however, not been investigated whether the cylinder friction has the the same importance on an active heave compensation system using a classical motion control scheme.

The approach that is taken to investigate this is by addressing a worst case heave compensation scenario as seen from a passive heave compensation point of view. This corresponds to a situation with low weight-on-bit (WOB), short drillstring, and stiff formation. A model of such a system is modeled with a typical active heave compensation control and the best possible performance regarding variations in WOB is determined for different combinations of friction and control valve bandwidth.

2. COMPENSATOR MODEL

The compensator model consists of several parts such as active hydraulic, passive hydro-pneumatic and mechanical subsystems. In Fig. 1 and 2 a Drill String Compensator is shown which is located on the top of the derrick and consists of the passive Crown Mounted Compensator (CMC) and the Active Heave Compensator (AHC). The main compensator function is to compensate the platform heave motion while drilling or landing equipment on the seabed.

CMC consists of one plunger cylinder and two small force equaliser cylinders, see Fig. 3. The passive compensator works as a hydro-pneumatic spring and the force in it depends on its extension. The compensator uses two small equaliser cylinders to reduce the effect of a force variation due to the compressed air. AHC allows the rig to carry out operations in an extended weather window and consists of two double acting cylinders positioned besides the plunger cylinder on the CMC, see Fig. 4. AHC makes it possible to keep the crown block position with respect to the seabed constant typically within $0.01 - 0.05$ metres with a heave motion amplitude up to $4 - 5$ meters, Haaø and Vangen (2011). In purpose of the heave motion compensation the crown block is moving while the drawworks is fixed on the drilling deck. The wire line length variation is taken

Fig. 1. The Drill String Compensator. The total height of the compensator equals 20 metres.

up by the rocker arm system. This design prevents the compensator from adding any wear to the wire, Haaø and Vangen (2011).

The compensator model used in this paper was considered in Ottestad et al. (2010). The loads used in the model are summarised in table 1. A pressure source used for the active hydraulic system actuation is 345 bar. The AHC consists of two electro-hydraulically actuated 4/3-way directional control valves in parallel, see Fig. 4. For simplicity they are modeled as a single large valve. Each valve has a rated flow of $Q_r = 460$ L/min at a rated pressure drop of $\Delta p = 35$ bar per metering edge. The active heave cylinders are symmetrical actuators with a piston diameter of $d_p = 200$ mm and a rod diameter of $d_r = 170$ mm. Like the valves they are modelled as a single larger cylinder.

The two active heave cylinders have a stroke length of 7650 mm. The passive system consists of one oil filled plunger cylinder connected to a piston accumulator which intermediates the pneumatic and the hydraulic system, see Fig.3. The hydro-pneumatic system is modeled as a spring to keep WOB around $3g$ kN. The plunger cylinder is modeled as a gas spring with stiffness 42 kN/m and damping 52 kNs/m. These numbers are adopted from Ottestad and Hansen (2011), where a detailed model of the entire passive system all major flow resistances and capacitances in both the pneumatic and hydraulic systems were included. Two force equaliser cylinders (Fig. 3) are used to compensate for the force variations caused by the gas spring. The cylinders pull the Crown Block downwards when the plunger cylinder is under mid-stroke and upwards when the plunger cylinder is above mid-stroke.

Table 1. The loads supported by the compensator.

Cylinder	10[t]	Crown Block	20[t]	Drill bit	3[t]
Travel. Block	20[t]	Drill String	80[t]	DDM	30[t]

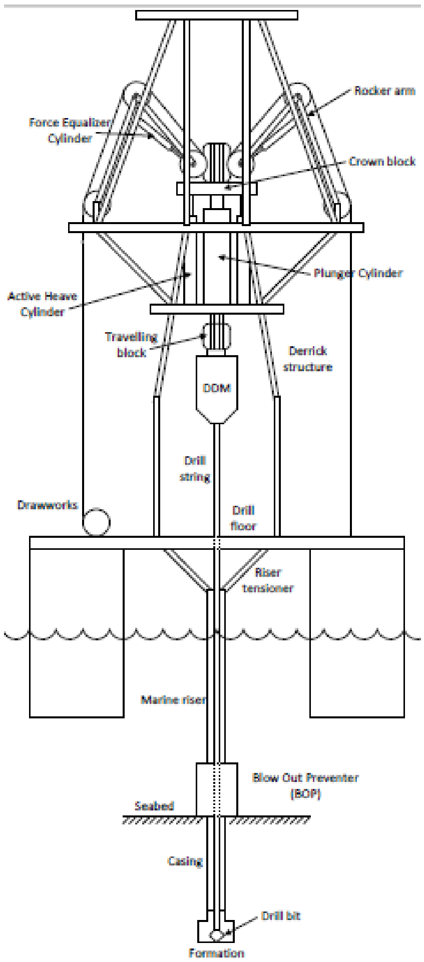

Fig. 2. Main arrangement of CMC drilling equipment.

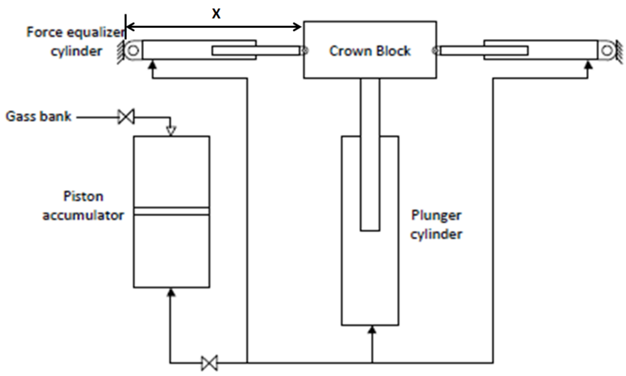

Fig. 3. The hydro-pneumatic system with a plunger cylinder and two force equaliser cylinders.

In this paper the platform motion is modeled as a single sine wave, see (1). The amplitude of the wave is 0.5 metres and the time period is 12 seconds. Clearly, this is a simplification as compared to actual wave patterns. However, the two main parameters when designing heave compensation equipment are the maximum travel and maximum velocity,

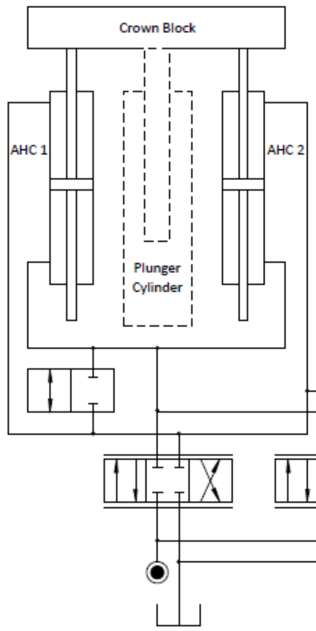

Fig. 4. The hydraulic system with active heave cylinders, two proportional valves and the bypass valve

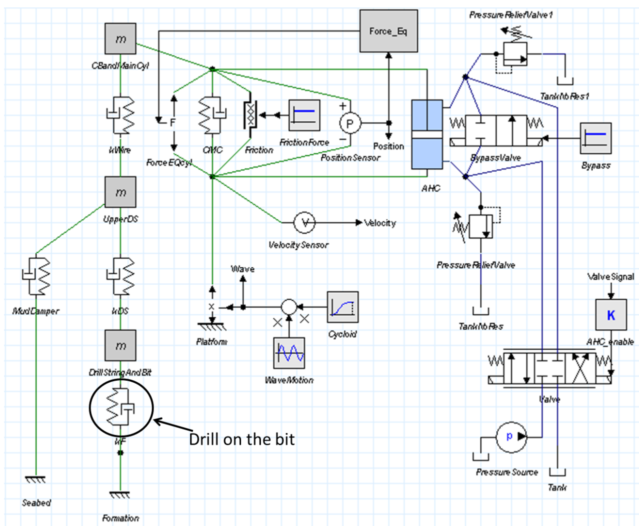

Fig. 5. 20-Sim based model of the drill string compensator without a control loop.

hence the simple sine wave is considered as adequate to represent these parameters satisfactory.

$$Z_h(t) = \Delta Z_h \sin\left(\frac{2\pi t}{T}\right) \tag{1}$$

where ΔZ_h is the amplitude and T is a time period of the wave. The force from the force equalisers is based on the geometry, see Fig. 3, and the pressure level in the main cylinder. The active and passive compensators adjust the distance between the platform and the crown block. The crown block holds payloads of different parts of the structure, as shown in Fig. 2. The parameters of the mechanical part such as the mass and the stiffness are summarised in Ottestad et al. (2010) and some are given below. The total wire stiffness is $k_w = 4000$ kN/m. Both the Drill String and wire stiffness vary with respect of their lengths but are considered constant in this paper. The Drillstring spring stiffness is $k_{DS} = 298$ kN/m. The contact stiffness with the formation is $k_F = 1000$ kN/m.

The complete model is implemented in $20-\texttt{Sim}$ software and is shown in Fig. 5. The model has four outputs, wave position, wave velocity, WOB and cylinder position. The valve signal is an input which controls the valve opening. The active heave compensator is modeled by the AHC cylinder and the hydraulic system on the right-hand side of Fig. 5, while the crown block, Drillstring and the spring dynamics are on the left-hand side. The platform is modeled as a position actuator with a sine wave input, representing the platform lateral motion. The passive compensator, CMC, along with the friction force and force equalising cylinder is placed between the crown block and platform in parallel with the AHC. The on/off bypass valve determines wether the heave compensation is passive or active.

3. FRICTION MODEL

The friction in the hydraulic cylinders is caused by the presence of elastomeric seals between piston rod and cylinder bore interfaces required to prevent leaks. The pressure loss in the hydraulic lines and in the piston accumulator of the hydro-pneumatic system is caused by friction forces. In general, there is a wide range of different resistance models used to compute the effective damping of the passive system. They include nozzles, leakage paths, pipes, hoses and orifices. As an example, the pressure drop in a hydraulic line is calculated as follows:

$$\Delta p = \lambda \frac{L}{D_l} \rho \frac{V_l^2}{2} \tag{2}$$

where λ is a dimensionless friction factor that depends on the flow regime (laminar or turbulent), L is the length of the hydraulic line, D_l and V_l are the line diameter and velocity respectively and ρ is the oil density. When considering the friction in the main cylinder it is often considered as a sum of a constant force such as Coulomb friction, a force proportional to the pressure level, a force proportional to the relative motion velocity (viscous friction) and Stribeck friction at low velocity. Steady-state models provide an expression of the friction force for a constant relative velocity as shown below.

$$F_{fric} = F_c + (F_s - F_c)e^{-\left(\frac{|v|}{v_s}\right)^{n_s}} + \sigma_2 v + k_p p \tag{3}$$

where F_c is Coulomb friction, F_s is the maximum static friction (stiction) force, v_s is Stribeck velocity, v is a velocity between the contact surfaces, $n_s = 2$ is an appropriate exponent, k_p represents the pressure dependant friction term and σ_2 is the viscous friction coefficient. The friction model that has been implemented in $20-\texttt{Sim}$ is presented in (4). The different parameters are summarised in table 2.

$$F_{fric} = F_c + (F_s|\tanh(\xi * v)| - F_c)e^{-\left(\frac{|v|}{v_s}\right)^2}(sign(v)) + \sigma_2 v + p k_p$$

where ξ represents the steepness of the Coulomb friction curve and p is the pressure level in the main cylinder.

Table 2. Coulomb and Stribeck Friction model and hydraulics parameters

F_c	12 [kN]	σ_2	10 [kNs/m]
F_s	22.8 [kN]	v_s	100 [mm/s]
ξ	1000 [s/m]	k_p	42 [kN/m]
d_r	170 [mm]	d_p	200 [mm]
Q_r	460 [L/min]	Δp	35 [bar]

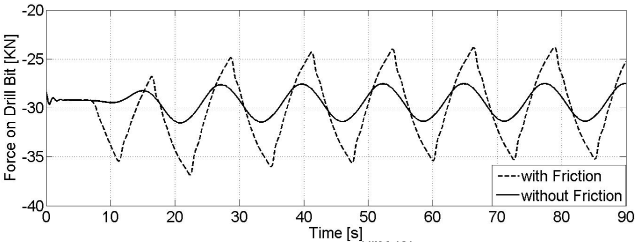

Fig. 6. Force on the Drill Bit where the passive compensation is used.

4. FRICTION EFFECT

In order to evaluate the effect of friction in the main cylinder it is necessary to distinguish between passive and active heave compensation. In the passive case previous work has indicated that friction plays a significant role in the compensation performance. This is especially pronounced when a force equalising system as the one used on the considered system is present. This is verified with the model used in this work, as maybe seen in Fig. 6 where the variation in WOB is shown with and without friction. The difference in WOB is substantial and with a nominal WOB of $3000g$ we have had to reduce the original wave amplitude of 1.75 meter to 0.5 meter to avoid slamming, i.e., the drill bit losing contact with the formation. In the following, we maintain an amplitude of 0.5 meter and employ the AHC system to increase the safety against slamming. In this work a simplified standard motion control scheme is used to investigate combinations of control valve bandwidth and friction in the main cylinder. The task of the control system is obviously to minimize the variations in WOB, however, this is often translated into maintaining the crown block position constant with respect to the seabed. The wave data is collected by a Motion Reference Unit (MRU) which measures effective heave motion of the rig at the drillcenter. The controller is presented in Fig. 7 and it uses a velocity feed-forward loop with a fixed gain K_v. It also has the position feedback loop to counteract deviation in position. The quality index ϕ is used to find two gains K_p, K_v and given below.

$$\phi = 0.5 \left(\sqrt{\frac{\int_0^T z_{cb}^2 dt}{\int_0^T z_{cb}^{*2} dt}} + \sqrt{\frac{\int_0^T \dot{z}_{cb}^2 dt}{\int_0^T \dot{z}_{cb}^{*2} dt}} \right) \tag{4}$$

where z_{cb} is the position and $\dot{z}_{cb}$ is the velocity of the crown block, the asterisk $*$ indicates the reference passive compensation. The two gains K_v and K_p are found by using a built-in parameter sweep tool in $20-\texttt{Sim}$. Firstly, K_p is set to zero and $20-\texttt{Sim}$ runs the simulations where K_v is adjusted for each iteration. The value of K_v that gives the best result as compared to the passive compensated system is saved as the optimal gain for velocity compensation. Secondly, the parameter sweep

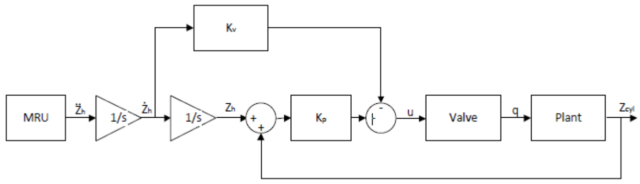

Fig. 7. Typical motion control.

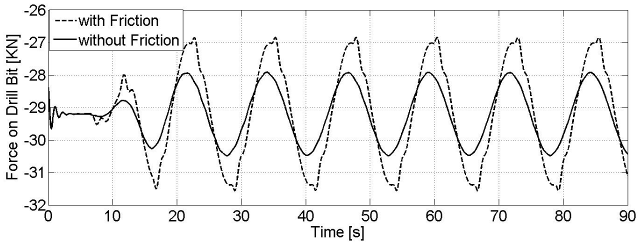

Fig. 8. Force on the Drill Bit where the active compensation is used. Both frictionless and the simulations with friction are shown.

tool is used to find K_p with multiple iterations. In this paper several different control valves with a wide variety of bandwidths have been considered and the optimal controller gains has been tuned in for each valve. For example, $K_p = 3.02$ [1/m] and $K_v = 0.55$ [s/m] when valve natural frequency equals 0.75 [Hz] and friction in the main cylinder is considered. However, the controller gains are $K_p = 6.15$ [1/m] and $K_v = 0.55$ [s/m] for the same valve but without friction.

The simulation results of the $20-\texttt{Sim}$ model have revealed when the valve bandwidth reaches a certain level, hence no more improvement is obtained in the variations in WOB. This is the case for the both systems where the friction in the main cylinder has been omitted and for systems where the full friction model as described in (4) is included. As an example, the WOB variation obtained with a valve bandwidth of 0.75 [Hz] is shown in Fig. 8. Clearly, friction increases the WOB variation, however, not dramatically. In table 3 some results have been assembled, and the trend is quite clear that whether friction is present or not the improvement in WOB variation is more or less non-existent from valves with [0.75..1] [Hz] and up. This is, in fact, in quite good accordance with on the basic rules of thumb regarding valve bandwidth which states that the valve bandwidth should be approximately 3..4 times larger than the bandwidth of the hydro-mechanical system to be manipulated. In this case, we have an eigenfrequency of approximately 0.23 [Hz] of the mass-spring system in $20-\texttt{Sim}$ which seems to correspond well with the rule of thumb and the observations in table 3, where $*$ indicates the simulations without friction in the main cylinder, ΔF_{WOB} represent the force variation in both negative and positive directions from the baseline of 30 [KN].

There is no doubt, that the valves used in heave compensation equipment as the one presented here has much higher bandwidth and there may be other good reasons for this. However, this study seems to indicate that low-response valves, width bandwidth below 4..5 [Hz] must be disqualified as AHC valves based on other load cases than those investigated here.

Table 3. Valve bandwidth and WOB variations.

$Valvefreq., [Hz]$	0.25	0.5	0.75	1.0	2.5
$\Delta F_{WOB}, [kN]$	15	6.2	2.1	2	1.9
$\Delta F^{*}_{WOB}, [kN]$	15	7.1	1.1	1	1

5. DISCUSSIONS AND CONCLUSIONS

The Drillstring compensator model in $20-\mathtt{Sim}$ consists of active and passive parts, where the active compensator is a hydraulic system connected to the controller and the passive hydro-pneumatic compensator is modeled as a spring with friction components. The controller is used to compensate the heave motion of the platform, where the crown block is fixed with a respect to the seabed. The AHC system is able to reduce the motion to 2 cm with a wave amplitude of 0.5 meter and period of 12 seconds while keeping the contact force between the drill bit and the formation within the limitations ± 1.1 [kN]. The force variation becomes higher (2.5 [kN]) than a wave amplitude is increased to 1.75 meters, however the use of the passive compensator only will increase the force variation to 10 [kN] in this case.

The model presented in this paper has been used to verify the usefulness of force equalisers during the passive compensation and to identify the impact of the friction in the main cylinder on the AHC. The performance of heave compensation equipment in practice depends on the variation in WOB. The worst scenario might be a short and stiff Drillstring with very small tolerance on WOB. The friction in the main cylinder has a significant impact on the heave compensation performance when the passive heave compensation and force equalizers are used. However, it was proved in this paper that the friction in the main cylinder seems to have much smaller impact on the heave compensation performance when active compensation is implemented.

In additional, the ratio between valve bandwidth and hydro-mechanical system bandwidth is more important than friction. Generally, the valve bandwidth should be three times higher than the mechanical system bandwidth. The threshold values for the valve bandwidths are surprisingly low as compared to what is usually used in Oil and Gas industry but they are in good accordance with rule of thumb. However the results maybe different than more/other load cases are considered. In fact, the market price for the valves with lower bandwidths is smaller and one of the further directions of this research is an investigation of the performance changes than more specific load cases are applied.

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