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Compact Hydropneumatic Heave Compensator

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Abstract: A hydropneumatic heave compensator and a semi-active control are proposed according to the requirements of offshore drilling at a depth of 6 km. This semi-active control is used to compensate mass changes in the system, to reduce the accumulator volume and improve the passive compensator performance. In this control, the system damping is modified at each instant of time through changes in the servo valve position, which is the only manipulated variable, and facilitates the low energy consumption of the compensator. The compensator with semi-active control has a satisfactory frequency response for each drill string mass and an accumulator volume comparable with the accumulator volume of the active compensators that are currently used in the offshore industry.

Keywords: Semi-active, heave compensators, control, frequency-response, hydropneumatic.

1. INTRODUCTION

The heave compensator is of utmost importance in the drilling of an offshore well, because it mitigates the vibrations transmitted by the platform to the drill string. The compensator prevents drill bit and drill string damage, collapse of the borehole wall, well kick from stopping the drilling, which causes loss of operation time and money, and the risk of major accidents.

The current commercial compensators are active or passive systems with an active subsystem to improve the performance, allowing high attenuation rate of the movement transmitted to the drill string. This attenuation is better than 95% but the energy consumption is very high. The semi-active compensator has lower energy consumption and it allows some force control. The authors have proposed semi-active hydropneumatic heave compensator, but the semi-active control has different actuators. Xiaojian and Shaojun (2010) designed a compensator with a H_∞ robust control, using a magneto-rheological damper in parallel as actuator. In the design proposed by Hao and Yancong (2011), the hydraulic oil is pumped in and out of the compensator cylinder. Cuellar and Fortaleza (2014) used a servo valve to change the system damping when the drill string mass changes (look at Fig. 1), this compensator is very robust regarding mechanical/ electrical failures, it has low consumption energy and its frequency response is satisfactory, but the accumulator is large, $170m^3$, in size.

This article proposes a semi-active hydropneumatic heave compensator for the drilling in water depths of up to 6000 m deep. This compensator is similar to the compensator proposed by Cuellar and Fortaleza (2014), but the semi-active control modifies system damping in high frequency manner to optimize the compensator performance and to reduce the accumulator volume.

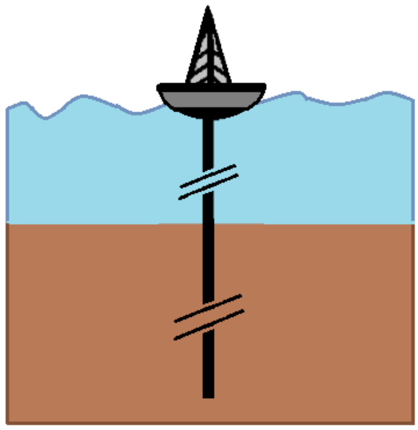

Fig. 1. Drilling process

This compensator has a reduced energy consumption, because only the servo valve requires external energy. This technology allows maintaining an acceptable performance and the reference position when the drill string mass changes. The compensator with a redundant system in parallel (two parallel servo valve), allows the drilling process to continue, in case of a servo valve failure, because the redundant system is opened, this redundant system is an easy and cheap option

2. MODELLING

The modelling is the same as shown by Cuellar and Fortaleza (2014). The compensator was designed as a traditional hydropneumatic suspension, Bauer (2011). The principal components of this compensator are accumulator, cylinder, servo valve and spring mass in Fig. 2.

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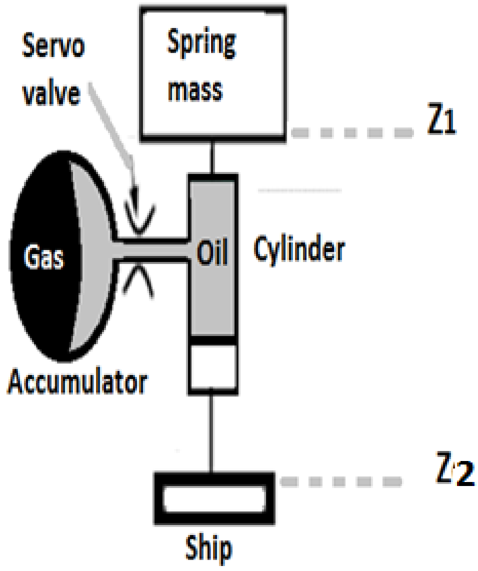

Fig. 2. Diagram of traditional hydropneumatic suspension system

The compensator operation is as follows: when the ship heaves up, the oil of the cylinder is forced into accumulator and the gas of accumulator is compressed to compensate the increased displacement and to store energy. This energy is dissipated by the valve when the oil flow passes through it. When the ship goes down, the cylinder housings descend and the gas of the accumulator is expanded. Meanwhile, the oil flows from accumulator to the cylinder. This process is described in more detail by Ni and Lui (2009).

The modelling goal is to find dynamic equation representing the movement mass function of the ship movement. Four assumptions are made to derive this transfer function: oil in the cylinder is not compressible, friction in the cylinder is negligible, no heat exchange is possible between the gas and the environment, and the system is not linear with regard to the servo valve saturation.

First, the forces exerted on the mass in the static case are the weight and cylinder force. Therefore, the static pressure in the cylinder is the weight mg divided by the cylinder area S_v:

$$P_{st} = \frac{mg}{S_v} \tag{1}$$

Applying the fundamental law of dynamics leads to the equations of motion mass:

$$mz_1''(t) = P_C(t)S_v - mg \tag{2}$$

The forces applied on the spring mass are the weight and the cylinder force. The cylinder pressure P_C results from the sum of pressure drop across the valve hydraulic resistance and the inside accumulator pressure .

$$P_C = \triangle P + P_{Ac} \tag{3}$$

The pressure drop $\triangle P$ could be linearized assuming small variations of flow $q_{Ac}(t)$. The hydraulic resistance R represents the pressure drop for some flow. The hydraulic resistance value could physically change by the position of servo valve.

$$\triangle P = Rq_{Ac}(t) \tag{4}$$

The flow $q_{Ac}(t)$ is due to the movement of the ship and the spring mass movement. The flow is calculated with the distance between the spring mass and the ship position.

$$q_{Ac} = S_v(z_2'(t) - z_1'(t)) \tag{5}$$

Finally the pressure drop $\triangle P$ is given by introducing the relation (5) in equation (4).

$$\triangle P = RS_v(z_2'(t) - z_1'(t)) \tag{6}$$

The accumulator pressure $P_{Ac}(t)$ is described by the non-linear equation (7). This equation represents an adiabatic change of state. The value of the adiabatic exponent r is 1.66 for monoatomic gases, 1.40 for biatomic gases and 1.30 for triatomic gases.

$$P_{Ac}(t)(V(t))^r = Po(Vo)^r \tag{7}$$

Assuming small variations, expression (7) is linearized around the static pressure P_{st}:

$$P_{Ac} = P_{st} - r\frac{\partial V(t)}{C} \tag{8}$$

The pneumatic capacity of accumulator C is given by the partial derivative evaluated around operating point P_{st} and V_{st}.

$$\left.\frac{dP}{dV}\right|_{P_{st},V_{st}} = -r\frac{P_{st}}{V_{st}} = -\frac{1}{C} \tag{9}$$

The gas volume variation is equal to the variation of the cylinder volume, but with opposite sign. The cylinder volume is calculated with integration of the fluid flow of equation (5).

$$dV(t) = -\int_0^t S_v(z_2'(t) - z_1'(t))dt \tag{10}$$

Substituing (10) in equation (8), the accumulator pressure is:

$$P_{Ac} = P_{st} + r\frac{S_v}{C}(z_2(t) - z_1(t)) \tag{11}$$

Finally, the expression of the cylinder pressure is:

$$P_C = RS_v(z_2'(t) - z_1'(t)) + P_{st} + r\frac{S_v}{C}(z_2(t) - z_1(t)) \tag{12}$$

Substituting the above equation in dynamic equation of the spring mass:

$$\begin{aligned} mz_1''(t) &= r\frac{S_v^2}{C}(z_2(t) - z_1(t)) \\ &+RS_v^2(z_2'(t) - z_1'(t)) + S_vP_{st} - mg \end{aligned} \tag{13}$$

The expression of viscous damping coefficient is b_2 and the stiffness k_2 of the sphere are given by:

$$b_{control} = RS_v^2 \quad k_2 = r\frac{S_v^2}{C} \tag{14}$$

In the equation (13), the weight is canceled with the force of static pressure. After which, the expression (14) is

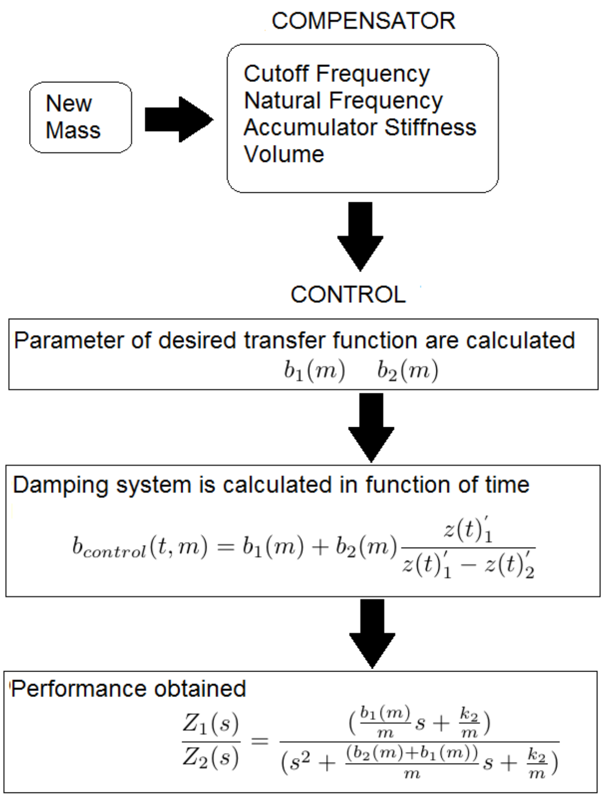

Fig. 3. Control diagram

introduced in equation (13). This equation represents de compensator performance in time domain and in function of the ship motion:

$$mz_1''(t) = k_2(z_2(t) - z_1(t)) + b_{control}(z_2'(t) - z_1'(t)) \quad (15)$$

3. CONTROL

The hydropneumatic performance is affected by the spring mass changes. Thus, the control calculates the desired transfer function for each mass, so its parameters b_1 and b_2 are a function of the supported mass, but these parameters are virtual, they have no direct relation with a physical parameter. This desired function is designed with a cutoff frequency of $0.056Hz$, because most energy of Brazilian ocean waves is distributed in frequencies higher than 0.06Hz. The desired function parameters are used to calculate the real damping system $b_{control}$ at each instant of time, obtaining a heave compensator performance with variable damping similar to the desired function performance (look at Fig. 3). Cuellar and Fortaleza (2014) showed that when the hydropneumatic compensator with semi-active control has a damping constant for each spring mass, a large accumulator is needed to achieve the desired cutoff frequency.

3.1 Desired Function

The desired function is:

$$\frac{Z_1(s)}{Z_2(s)} = \frac{(\frac{b_1(m)}{m}s + \frac{k_2}{m})}{(s^2 + \frac{(b_2(m)+b_1(m))}{m}s + \frac{k_2}{m})} \quad (16)$$

Where, b_1 and b_2 are parameters related with damping, but these parameters have no relation with any of the compensator's physical components, could be called of virtual damping. However, they have physical meaning, b_2 refers to a damping in the system output and b_1 is the generated damping due to the relative displacement between input and output system. The values of b_1 and b_2 are chosen so:

$$b_1(m) = 2zw_n m(1 - 0.85) \quad (17)$$

$$b_2(m) = 2zw_n m(0.85) \quad (18)$$

The value of 0.85 makes the zero of desired function 6 times bigger than the real part of desired function poles. Therefore, the performance is determined by the transfer function denominator.

3.2 Getting the Desired Function

The desired function is found with the expression for the valve damping $b_{control}$, this expression cancels a dynamic part of compensator performance and generates the virtual damping (b_1 and b_2). The valve damping should be calculated at each instant of time. The compensator in time domain is:

$$z(t)_1'' = -(z(t)_1' - z_2(t)')\frac{b_{control}}{m} - (z(t)_1 - z(t)_2)\frac{k_2}{m} \quad (19)$$

The desired transfer function in time domain is:

$$z(t)_1'' = -(z(t)_1' - z_2(t)')\frac{b_1(m)}{m}$$
$$-z(t)_1'\frac{b_2(m)}{m} - (z(t)_1 - z(t)_2)\frac{k_2}{m} \quad (20)$$

Start by equating the two above equations, then, clearing the valve damping $b_{control}$

$$b_{control}(t,m) = b_1(m) + b_2(m)\frac{z(t)_1'}{z(t)_1' - z(t)_2'} \quad (21)$$

In theory, this expression for the valve damping allows getting desired function at each instant of time. This control expression could require really great or negative damping values, the valve saturation does not allow the determination of these damping values; the damping just has physical sense for a positive interval with lower and upper limit.

4. COMPENSATOR DESIGNED

The discovered reservoirs made in the pre-salt have high comercial value, these reservoirs are drilled in deep water, between 5 km and 10km in which the water blade is between 1 km and 3 km deep. The compensator was designed to work in a depth of between 2km and 6km, because the drill string movement should be compensated when the drill bit touches the ocean floor in 2km, and when the drilling process finishes in 6km.

The drill string mass is calculated with the data from Cayres (2013); as a result, the drill string mass is 150 in 2km and 280 in 6km. Furthermore, according to Haao and Vangen (2012), the mass supported by the compensator without the drill string are the mass of crown block 20t, the Derrick Drilling Machine (DDM) 30t, travel block 20t and cylinder 10t. Thus, the compensator spring mass is between 230t and 360t.

The maximum mass is 360t; consequently, the maximum pressure is 22.8 MN/m^2. This value is calculated with (1), using the cylinder area of $0.16m^2$. The compensator pressure maximum has a value similar to the value used by Ni and Lui (2009) in passive compensator of 570t, and commercial compensators used currently, the pressure used is $20.3MN/m^2$ and $26.6MN/m^2$ respectively.

The project mass is the mass used to design the compensator. This mass is chosen with the equation (21). The project mass was chosen 43% of the possible mass interval; as a result, the project mass value was 286t. The accumulator volume in the project mass is $10.4m^3$. The stiffness is $60.2KN/m$, it is calculated with (14). Table 1 summarizes the values of compensator parameters in the project mass

Table 1. The compensator parameter values

Parameter	Unit	Value
Maximum pressure	MN/m^2	22.8
Cylinder area	m^2	0.16
Project mass	t	286
Project Volume	m^3	10.4
Diameter of opened valve	m	0.069
Diameter of closed valve	m	0.016
Maximum damping of valve saturation	MNs/m	2
Minimum damping of valve saturation	MNs/m	0.05

The principal components were described in the project mass except the servo valve; its main characteristic is the saturation (see control section). The valve has a diameter of $0.016m$ and $0.069m$ in opened and closed state respectively. Then, the damping coefficient value is between $2MNs/m$ and $0.05MNs/m$.

5. RESULTS

The compensator has a servo valve, meaning it is a nonlinear system due to the valve saturation, so it is not possible to get the frequency response with the transfer function. The compensator with control, valve saturation and a sinusoidal input is simulated in Simulink. The input has an amplitude of $1m$ and a frequency value between 0.01Hz and 1Hz and the frequency value is constant during each simulation. The simulation is repeated with a different input frequency, this frequency and the output amplitude are registered to plot the frequency response.

The Fig. 4 shows the compensator frequency response with the three masses. The responses are similar to frequency response of a Butterworth low pass filter, because it has a flat frequency response in the passband, and an acceptable attenuation in the transitions band. The compensator cutoff frequency is 0.56Hz; hence, the compensator filters the frequencies of ocean waves.

The semi-active compensators designed by Cuellar and Fortaleza (2014) and Ni and Lui (2009) always have a frequency zone, with maximum gains bigger than 2.3db and 12db respectively. The compensator designed in this article has a better frequency response, because its flat passband filter.

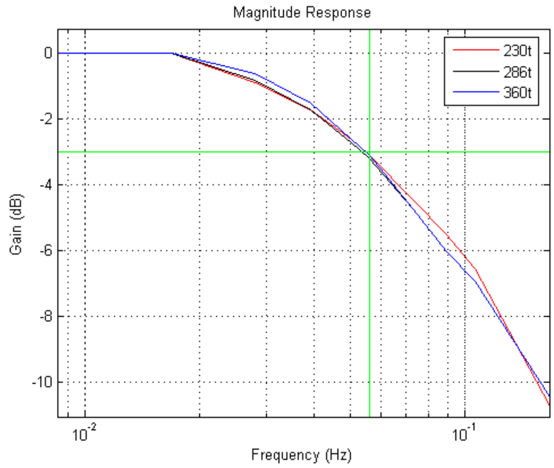

Fig. 4. Frequency response

Table 2 summarizes the values when there are changes in the spring mass. The maximum and minimum volume are 12.2 m^3 and 8.9 m^3, respectively, the accumulator is 7% of the accumulator volume value designed by Cuellar and Fortaleza (2014), this volume is comparable with the volume used in commercial compensators. The damping coefficient without saturation is used in the desirable function, using this damping, the performance with control is the same as the desired function performance, but when the saturation is introduced in the control, the compensator performance is different. The solution is increase the damping value, the damping with saturation should be bigger due to the effect of valve saturation.

Table 2. The compensator parameter values

Massa (t)	Damping with saturation z	Damping without saturation z	Volume (m^3)
230	1.20	0.85	12.2
286	1.45	0.9	10.4
360	1.70	0.95	8.9

The frequency response with the least attenuation was the response of 360t. The compensator performance is simulated when the platform is moved by the ocean wave. Fig. 5 shows the platform movement (signal taken from Ni and Lui (2009)) and the performance compensator. The response is acceptable, because the compensator mitigates the input motion; so that the maximum output motion is 25% of the value of maximum input motion. This relation is 17% in passive compensator designed by Ni and Lui (2009), but the volume in this case is the $20m^3$ and it does not adapt to mass changes. The active compensator is used in industry currently with attenuation better than 95% for any wave, but its energy consumption is high.

Fig. 6 shows the step responses for three masses with semi-active control and valve saturation, these responses are overdamped, with the damping z between 1.2 and 1.7. The semi-active control allows keeping the step response for any spring mass. The best step response is in the minimum mass, because it is the fastest. Furthermore, The maximum settling time is 25 s in the maximum mass, this is the slowest response.

The frequency responses of compensator with dynamic and static damping are shown in Fig. 7. The compensator

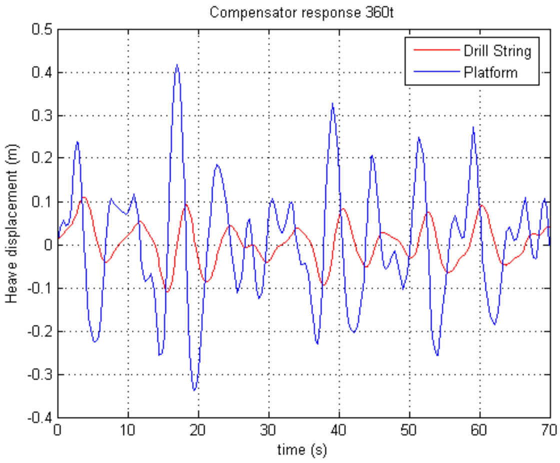

Fig. 5. Compensator response under the random wave influence

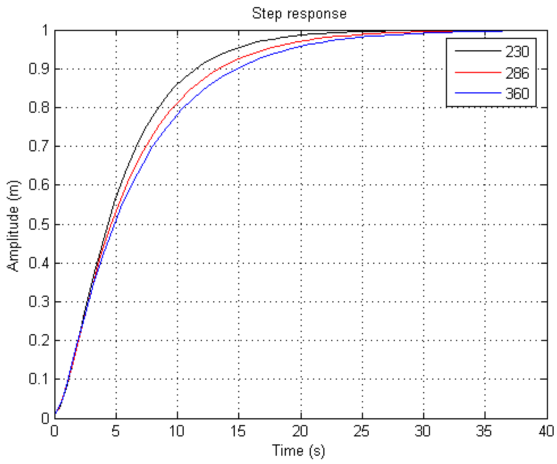

Fig. 6. Step response for three masses

with static damping z of 1.4 has a cutoff frequency of 1.2Hz and its passband is not flat. When the desired function damping z is 1.4, the compensator with dynamic damping has the desirable cutoff frequency (0.056 Hz)and a passband flat, this response is better than response with static damping.

6. CONCLUSION

The compact heave compensator reaches its objectives to be used by the industry. With an accumulator volume of just $12.2m^3$, it is able to attenuate heave displacements (all the wave with period small than the $17.8s$ are filtered). The control system only actuates on a servo valve, thus it presents low energy consumption and robustness. This robustness is obtained using a spare actuator in parallel, which is used if there is a fail of the main actuator.

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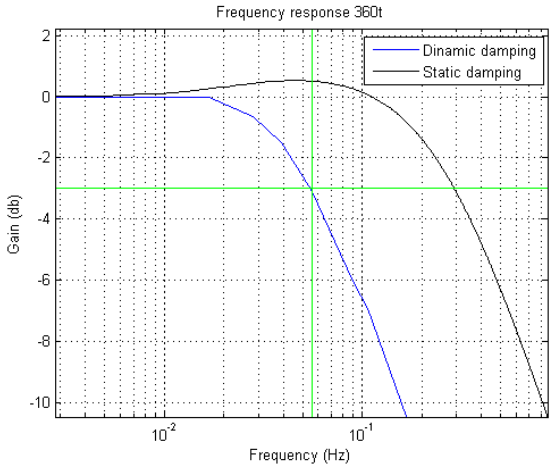

Fig. 7. Compensator frequency response with dynamic and static damping.

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