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NUMERICAL SIMULATION OF A PASSIVE HEAVE COMPENSATOR FOR SCIENTIFIC RESEARCH SHIPS

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Abstract:

Research vessels are commonly used on a daily basis for ocean exploration and payload handling. However, due to unpredictable wave motion on the ship and the flexibility of the cable, the heave of the ship is unavoidable and causes danger during operations, loss of payload, possible damages to expensive equipment and prolong period of downtime. A compensator system is an essential part of operations to mitigate this effect and to ensure safety, reduce down-time of operation and increase efficiency while providing longer and better duration of operation even in harsh conditions. In this article, a passive heave compensator system with cylinder, accumulator and depth compensator connected in series by pressured pipes develeped for a scientific research ship with length of 68m and breath of 16m is analyzed along the coast of Guyana, South America. The payload used in this analysis is 200ton. The working principle of the heave compensation system is described, the parameters affecting the performance of the system are simulated and analyzed using MatLab. A 3D model of the system is built using SolidWorks and schematic drawings are produced from AutoCAD. The compensation rate of the system is higher than 77% under the influence of the input wave and the system has a response of an average setting time of 18s. The point of maximum load exerted is at the splash zone. For a typical most probable extreme significant wave height, Hs= 2.3m, period T= 6s and direction $\mu = 45^0$ in the operational area, the reduction in heave motion when the vessel is equipped with the heave compensator is approximately 77% compared to 47% reduction when the vessel is without a compensator.

Keywords: Passive heave compensator, accumulator, depth compensator, towed bodies, numerical simulation, payload handling

NOMENCLATURE

AHC	Active heave compensation
A_{zh}	Movement of ship in the vertical direction
A_{zp}	Movement of PHC in the vertical Direction
C_v	Damping
D	Cylinder diameter
d	Piston diameter
HHC	Hybrid heave compensation
L	Length
η	Compensation rate
m	Meters
PHC	Passive heave compensation
SAHC	Semi-active heave compensation
s	Seconds

1. Introduction

Offshore operations are very challenging but have great value waiting to be uncovered and as stated in many articles the ocean has enough resources to supply the entire human species (energy, food, commercial commodities, recreational, etc.) Due to the depletion of land resources for the past decade, the ocean is targeted as one of the most important source for sustainability. Research, exploration, shipping, payload handling are just a few operations that are conducted on a daily bases, however these operations are usually affected by environmental factors such as; tides, currents, weather, wind and waves which poses treats to vessels, loss of payload and human life. For safe operational performance, weather condition should be monitored since this phenomenon causes operations to be delayed.

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Ship travels along the surface of the ocean, and is therefore subjected to surface wave excitation which varies in height, frequency and is unpredictable in nature. This motion will cause an effect of heave, roll and pitch on the vessel, which will cause the stability of the ship to vary with time. Roll refers to the rotation about the longitudinal axis, pitch is the rotation about the transversal axis, while heave refers to movement in the vertical direction. These three motions are important when considering the motion of the ship, but when considering the motion affecting payload or towed operations, heave motion is considered to be the most important since the cable that connects the ship and the payload can be consider as a spring and is elastic which can reduce the roll and pitch effect on the payload. Vessels handling payload operations are continuously working all year round and are very expensive to operate. During rough sea state, equipment handling becomes a difficult task to manage, additionally activities such as research, exploration and other offshore operations will be hindered and possible damages to expensive equipment, resulting; long down time period which leads to high economical cost.

Ship motion resulting from waves causes disturbances to the crane which is connected to the towing cable, the cable then transfer this motion underwater to the towed body creating a disturbance, hence; the need for a compensator to effectively attenuate the unwanted towed body motion resulting from wave motion at the surface. Additionally, when lifting, lowering or handling a load at sea, heave compensation issued to remove vessel heave motion from the load, resulting in the decoupling of load motion (Woodacre et al., 2015).

As stated by Feng et al., (2011) and Yan et al., (2005), heave compensators has an increasingly important role in ocean resources exploration such as; deep-water oil and gas, offshore engineering installation and maintenance, maritime search and rescue, oceanographic research and other field.

Generally heave compensation can be divided into two main categories: passive heave compensation (PHC) and active heave compensation (AHC) which depends on the method of compensation employed. Additionally, hybrid heave compensator (HHC) systems and Semi-Active Heave compensator (SAHC) system are also implemented. Researchers for decades have been constantly developing ideas and implementing system designs to reduce heave effect and are still in the process of trying to completely eliminate this effect. A time line for heave compensation development is as shown in Fig 1.

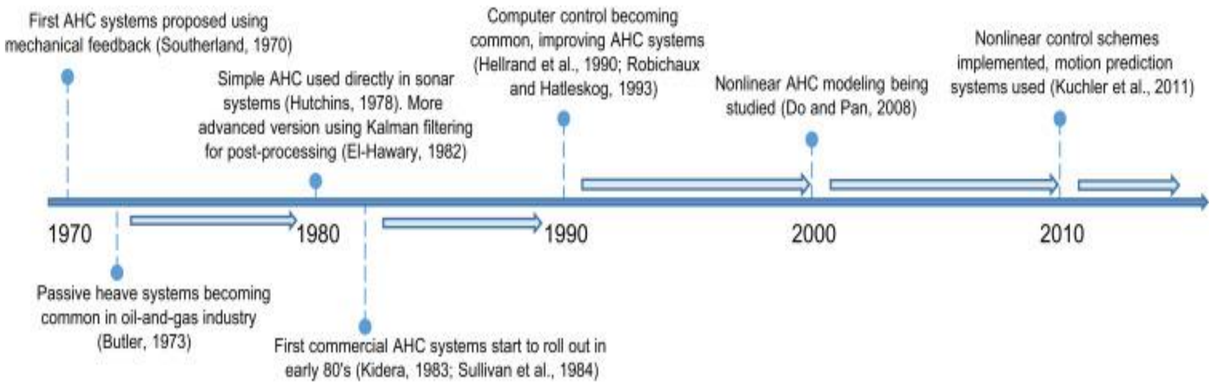


Fig. 1: An approximate timeline of heave compensation development (Hellrand et al., 1990; Sullivant et al.,1984,)

In this paper, a PHC system initially presented by Ormond (2011) is modified and studied using numerical simulation analysis. The parameters affecting the performance of the system are simulated and analyzed by MatLab 2014. The simulation result aims at determining if the PHCs presented is effective and feasible, and also aims to provide a theoretical base for future development of heave compensation system. A schematic of a research vessel equipped with a heave compensator is shown in Fig. 2.

The design of a towed underwater system is often constrained by the dynamic tensions and deflections induced by motions of the surface vessel in harsh weather. A towed system has a nonlinear response because of large curvatures and quadratic fluid drag, so either lengthy simulation or an approximate analysis is required (Hover et al., 1994).

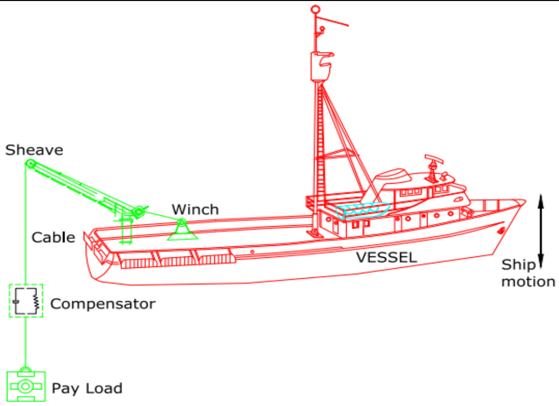


Fig. 2: Schematic of a research vessel equipped with a heave compensator

1.1 Heave compensation system

A heave compensation system acts as a spring device which is used to reduce the vertical motions and forces during heavy lift operation. Wave height and weight are key factors when considering a suitable compensation system, as it tends to be very large at certain periods during the year resulting in larger heave motion than regular. Regardless of the compensator type, the main function is to decouple load motion from ship. Compensation is generally based on two kinds of equipment, cylinder or winch (Jakobsen, 2008). A cable configuration diagram is as shown in Fig.3.

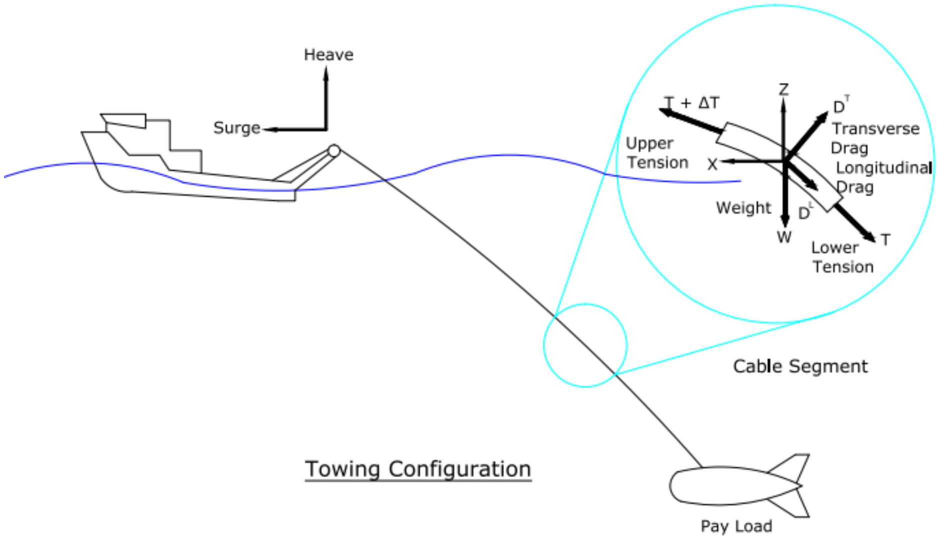


Fig. 3: Schematic drawing of a vessel hauling a load and the cable configuration

1.1.1 Passive heave compensation (PHC)

Passive heave compensation is based on a collective system of cylinder and accumulator working simultaneously which contains gas (high and low pressured) and hydraulic fluid (high and low pressured) between the compensating cylinder and accumulator, this result in a spring-damper effect with predefined, relatively low stiffness. The passive cylinder usually reliefs the system of static forces which can be adjusted automatically with the pressure in the accumulator; the volume of gas determines how much pressure increases during compensation. The system can be applied on a broader range of operating conditions as the pressure inside the accumulator can be set to match the given load case. Additionally; it requires no external power source for its operation and utilizes the motion of system to develop control forces. However, it is stated that while PHC systems are generally less expensive and less complex, they are also less effective at attenuating heave motion effect on the payload (Woodacre et al., 2015).

1.1.2 Active heave compensation (AHC)

Compared with PHC system, AHC system can provide higher compensation efficiency, at a rate above 95%. However; this type of system required high energy consumption, which means higher economical cost. However; the forces are adjustable. Generally; this type of system comprises of various types of sensors that continuously monitor the system and signals which are used to compute the control force (Rexroth, 2016). Usually; this system is controlled by signals from MRU (motion reference unit) that measures the heave of the ship and send feed-back to the monitoring base, the system can also be based on electrical winches.

Active heave compensation systems are normally located on the deck of the vessel, while PHCs can be found on deck and on the cable connecting the crane and payload. AHC system supports both static load and compensate for motion in one single hydraulic system. One major disadvantage of this system is that high flow in the system is difficult to control and the system can easily become unstable, additionally, the maintenance cost of this type of system is relatively high. The performance of this type of system is highly dependent on the sensors and actuators, also the required power to operate is large and it is more complex and expensive.

1.1.3 Hybrid heave compensation (HHC) and Semi-active heave compensation (SAHC)

HHC is obtained by a combination of an AHC and a PHC or a SAHC and a PHC system (Jakoben, 2008). This type of compensator usually consist of a passive cylinder carrying the load and an active controlled cylinder assist the none-ideal spring to overcome the friction losses. This combined arrangement gives additional movement to ensure effective compensation. SAHC system is designed with a combination of different set-ups of PHC and AHC systems. These combinations assist in reduction of the power requirement of the heave compensation system. Which system uses server hydraulic motors with variable displacement and in compensation mode several of these motors are connected to an accumulator system. The variable displacement in the engines adjusts the weight to be compensated, while the other engines that are still connected to the hydraulic circuit perform the compensation motion. . A semi-active system will be able to reduce the power requirement with up to 75% related to an equal rated active system (Huang, 2013). Semi-active heave compensation systems may also be used as a precautionary design measure in the event of an AHC system failure, thereby increasing overall system robustness (Quan et al., 2016a).

2. Methodology

The methodology adapted for this research was executed as shown in Fig. 4.

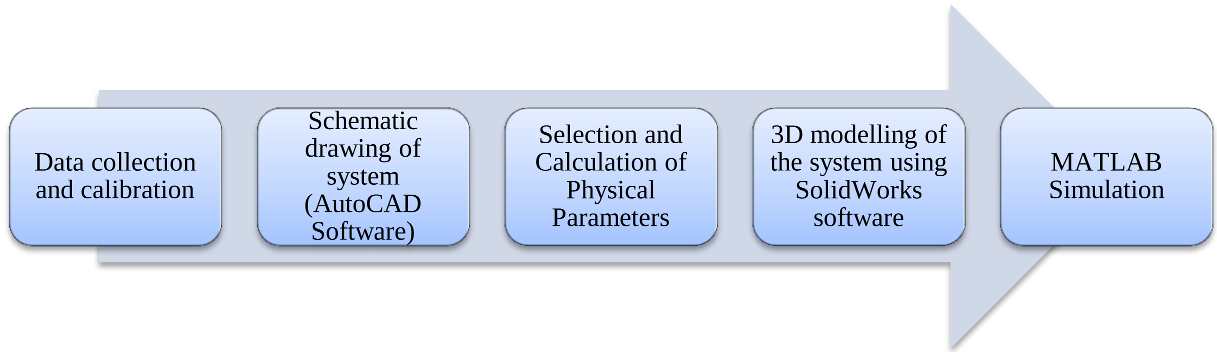


Fig. 4: Flow chart of the adapted methodology for this project

In the data collection phase, data was sourced from previous literatures which includes; journals, articles, books, etc. these data was then combined, calibrated and filtered for useful information concerning the related topic under study. This data served as guidelines throughout this research. Schematic drawings were then produced in AutoCAD 2013 of the passive heave compensator and the preliminary parameters were selected and calculated and the most suitable parameters were selected based on literature. These parameters were then calculated and compared to parameters in pervious literatures. A 3D model of the compensator was then produced using SolidWorks 2016, to have a better view and produce an image of the compensator system proposed. The model was then designed in MatLab 2014 and simulation was conducted on the model using the selected parameters. The output of the results is presented in this paper.

2.1 System design

2.1.1 Working principle

An illustration of the proposed heave compensator used mainly for research vessels or small vessels handling payload is as shown in Fig. 5. The system comprises of three major components which includes the accumulator, depth compensator and cylinder which is connected in series by pressured pipes. The compensator is attached to a cable on the top pad-eye which is attached to the crane of the vessel. The payload is attached to the clevis which is connected to the piston on the cylinder.

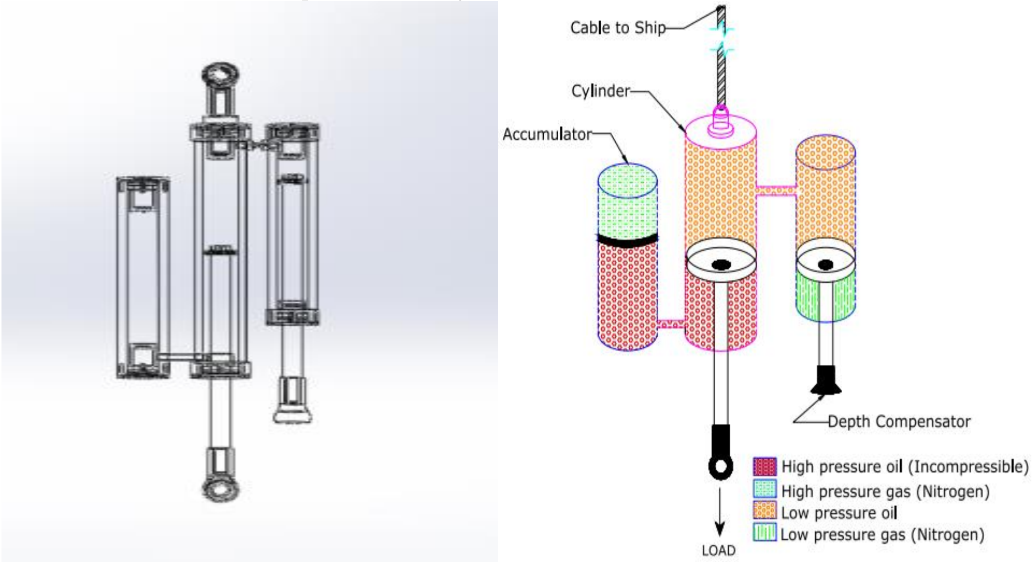


Fig. 5: Illustration of the passive heave compensator

The accumulator contains high pressured gas at the top section and high pressure oil in the lower section which is separated by a balloon at a certain depth in the accumulator (this is influenced by the desired load). The accumulator is a reservoir for the high pressure oil which is connected by high pressured pipe leading to the cylinder. The cylinder consist of high pressured oil at the lower section under the piston which is received from the accumulator as a result from the movement of the piston in each cylinder, the top section of the cylinder above the piston contains low pressured oil which is received through a low pressured pipe from the depth compensator. The piston in the depth compensator has an upwards and downwards motion, once the piston pushes upwards it compresses the low pressured oil in the depth compensator which then travels to the cylinder and compresses the low pressure oil in the cylinder and pushes downwards on the piston of the cylinder due to the motion of the vessel.

When the system is fully submerged, the external water pressure produces a net hydrostatic pressure acting on the cross sectional area of the rod (depth compensator) which generates a force on the rod. This force is countered by applying a pressure to the low pressure hydraulic fluid in cylinders and upper section of the depth compensator. The hydrostatic pressure on the rod is translated to a force on bottom of the rod, which is translated to a pressure on the fluid in the upper section of the cylinder and depth compensator. That pressure translates to a force on the piston, which counteracts the hydrostatic force generated on the rod. The net effect of hydrostatic pressure on the rod in the cylinders and the depth compensator is zero or a balance force that has negated the depth effect. This allows the accumulator chamber to be enlarged such that the stiffness of the system can be lowered. The resulting effect generates a balanced system that is not affected by hydrostatic pressure due to varying depths. This effect keeps the tension in the cable to avoid snapping.

2.1.2 Main design parameters

For effective compensation, the compensation cylinder lifting force should be equal to the gross weight in order to support the weight of the payload. The pressure of the system is influenced by the weight of the piston system. For this research, the static load of the system is considered as 200*ton*. It is important to limit the compensation cylinder size and keep the system pressure as reasonable as possible and not too high in order to maintain stability and efficiency. Taking into consideration the mass of 200*ton*, one compensation cylinder is adapted in the system design.

According to the strength theory, the diameter of piston-rod is $d \geq 222.4mm$, rounding this value and taken as $d \geq 250mm$ in accordance to national standard. Based on calculation and consideration for the volume of the system and the effective system pressure, the inner diameter of the cylinder is chosen as $D = 450mm$. The effective length of housing for the cylinder is chosen as $L = 2.5m$. This value was chosen based on wave height data collected for the project area (northern coast of Guyana) from the Maritime administration department, Georgetown, Guyana. The data shows hourly wave height changes on specific days. The plotted data is as shown in Fig. 6.

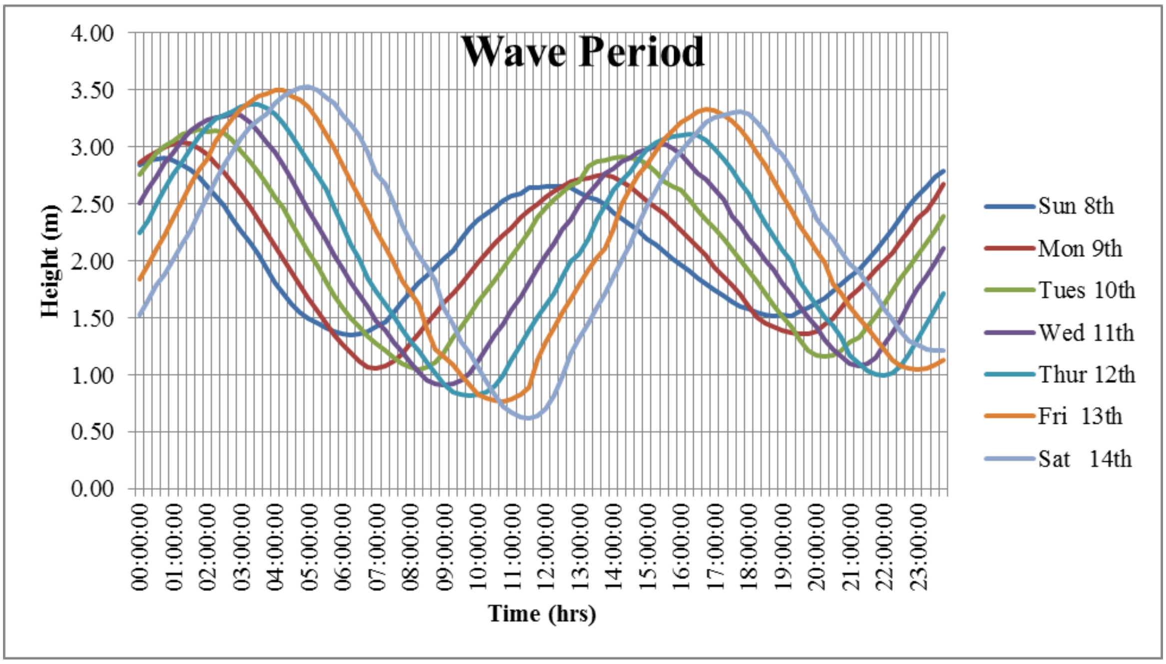


Fig. 6: Ocean waves considered in the simulation of the research vessel installed with PHC

The main parameters of the cylinder were discussed in the previous section. The parameters of the accumulator, compensator and parameters of the pipeline will be discussed in this section. The accumulator can be considered as the main component of the passive heave compensation system, its function is to store and release the energy generated by the movement of the ship (Jia et al., 2009). Assuming, A_{zh} and A_{zp} are the maximum heave displacement differences of ship and compensation platform respectively.

The compensation rate could be expressed as:

$$\eta = \left(1 - A_{zp} / A_{zh}\right) \times 100\% \tag{1}$$

Simulation has been conducted using the following data: pipe diameter of 0.08m, pipe length 0.5m and wave period of 6s. The pipe parameters were selected based on a study done by Wang (2009) which showed that the set of data with pipe length of 0.5m is the most optimal for an allowable amount of flow. However; a study on pipe sizing done by Ni et al. (2009) shows that a pipe diameter of 0.08m has a higher efficiency. The diameter and length of the pipeline which connects the accumulator and cylinder and the cylinder and depth compensator, will directly influence the velocity flow of the fluid, in this case, the fluid is oil during the compensation process and the dynamic performance of the system. From the result of the simulation, it was shown that the PHC system could not achieve 100% efficiency with the selected parameters. In order to achieve a higher percentage of efficiency, the accumulator size would have to be increased. However, if the parameters of the system were increased the efficiency would only increase in very small percentages, so the selected parameters were deemed suitable and can be useful for application. The result could also vary according to different wave period. The compensator with a sinusoidal input is simulated in MatLab. The input has amplitude of 1m and a frequency 0.17Hz. The component of the PHC system is as shown in Fig. 7 and listed in Table 1.

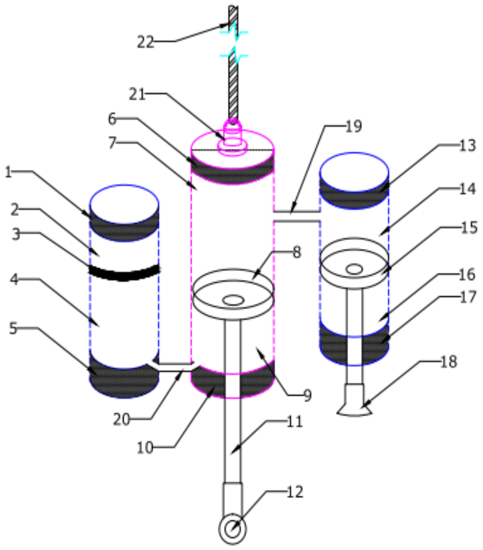


Fig. 7: Shows components of the PHC system

Table 1: Main and sub-section of the heave compensator

Parts of the Compensator		
Sub-Component	**Component Group**	**Description**
1	Accumulator	End Cap
2		High Pressure Nitrogen
3		Balloon for separation of Oil and Gas
4		High Pressure Oil Reservoir
5		End Cap w/ports
6	Cylinder	End Cap
7		Low Pressure Oil Chamber
8		Low Pressure Oil Piston
9		High Pressure Oil Chamber
10		End Cap w/seal
11		High Pressure Piston Rod
12		Bottom Padeye
13	Depth Compensator	End Cap
14		Low Pressure Oil Reservoir
15		Low Pressure Piston
16		Low Pressure Gas (atmospheric)
17		End Cap w/seal
18		Low pressure Piston Rod
19	Depth Compensator/Cylinder Connector	Low Pressure pipe
20	Cylinder/Accumulator	High Pressure pipe
21	Top Padeye	Top Padeye
22	Cable	Cable

2.1.3 Influence of environmental loads

The PHC system is located on the cable between the crane and payload, as the compensator submerged underwater below the splash zone (transition zone from air to water), it will be influenced by ocean waves and/or underwater current which will affect its stability if the force is great. When the compensator is absent, the balance position of the payload will be lifted gradually due to the action of ocean current acting on the cable. The cable will be subjected to shape deformation; the degree of deformation will be based on the actual current speed. From a study done by Quan (2016b), who developed a graphical representation to determine the deformation process showing the initial shape of the cable as being a vertical line which showed a tendency to incline under the influence of ocean current. In the same paper, he further stated, that in the case of compensation there was a smaller heave than the uncompensated one, the load is still lifted under the action of ocean current, but the cable tension is compensated effectively. A model of the vessel is simulated in waves of 6*s* is as shown in Fig. 8.

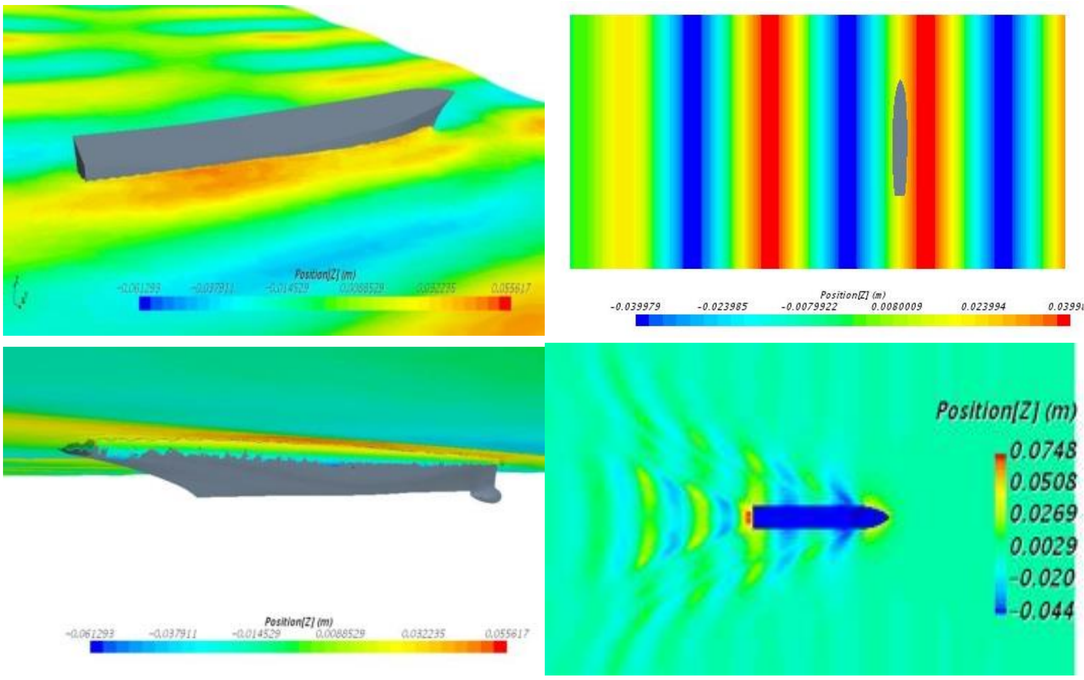


Fig. 8: Model of a research vessel hull part simulated in Ocean waves

Ocean currents mainly affect the cable shape, and the balance position of the load has to be lifted during the shift procedure of the cable, this event will reduce the compensation performance for the displacement. The passive heave compensator is used to compensate the cable tension more effectively. During a certain range of ship heave period, the compensation performance for heave or cable tension is lowered with larger period.

2.1.4 Governing equations:

A. Newton's second law can be used to describe the acceleration of the payload:

$$(m+m_A)\ddot{y} = -K_c(z+H\cos\omega t) \tag{2}$$

m – Mass of the load under the PHC system
m_A – Added mass of the load under the PHC system
$\ddot{y}$ – Acceleration of the mass of the load under the PHC system
K_c – Stiffness of the PHC system
H – Wave amplitude
z – Vertical position of the mass under the PHC system
ω – Angular wave frequency
t – Time

B. Hydrostatic pressure:

When a body is submerged in water, it experiences a uniformly distributed force on its surface area, as the depth increases, this force also increases. This is due to an increase in hydrostatic pressure, the force per unit area exerted by a liquid on an object. As ocean depth increases, the pressure acting on a body increases. For every 10.06*m* (33*ft*) pressure increases by 0.10*MPa*. The pressure increases about one atmosphere for every 10*m* of water depth. One atmosphere is equal to the weight of the earth's atmosphere at sea level, about 0.10*MPa*. The water pressure at various depths is Table 2. The hydrostatic pressure can be determined by the following equation:

$$P_{st} = \frac{mg}{A_{cylinder}} \tag{3}$$

Where:

P_{st} – Static pressure
m_g – Weight
$A_{cylinder}$ – Area of cylinder

Table 2: Water pressure acting on the depth compensator piston at various depths

Note: For every 10.06m deep, water pressure increases by 0.10MPa

Depth		Water Pressure
m	*ft*	*MPa*
10	32.808	0.101
20	65.617	0.201
30	98.425	0.301
40	131.234	0.402
50	164.042	0.502
60	196.850	0.603
70	229.659	0.703
80	262.467	0.804
90	295.276	0.904
100	328.084	1.005
125	410.105	1.256
150	492.126	1.506
175	574.147	1.758
200	656.168	2.009
300	984.252	3.014
400	1312.336	4.018
500	1640.42	5.023
610	2001.3124	6.128
1524	5000.00016	15.309
3048	10000.00032	30.619

C. Force acting on payload:

The forces acting on the payload can be determined by the following equation:

$$F_h = F_B + F_A + F_D \tag{4}$$

F_B – Force due to buoyancy
F_A – Force due to added mass

F_D – Force resulting from drag (drag force)

D. Equation of motion:

Assuming the force on the depth compensator will be cancelled with the force of the cylinder piston. For the system, applying the fundamental law of dynamics gives the vertical motion equation:

$$mz_1^{"}(t) = p_c(t)A_1 - p_{DC}(t)A_2 - mg - c(z_2 - z_1) \tag{5}$$

Where m is the load, t is time, z_1, z_2 are the vertical displacement of piston and vertical displacement of cylinder housing respectively. A_1, A_2 represents the area of the cylinder and depth compensator, p_c , p_{DC} is the pressure in the cylinder and depth compensator and C is the viscosity damping coefficient.

This can be reduced to:

$$z_1^{"} = -\frac{c}{m}z_1 + \frac{c}{m}z_2 + \frac{A_1}{m}\Delta p_c \tag{6}$$

E. Flow equation of cylinder:

For this type of system, assuming the depth compensator force and cylinder force cancel each other, the relation between the pressure and flow of cylinder is expressed as:

$$p_c = K / V_E\left[-q_R - A_1\left(z_2 - z_1\right)\right] \tag{7}$$

Where , q_R is the volumetric flow out the cylinder, K is the bulk modulus of hydraulic oil and V_c is the volume of cylinder.

F. Mathematical expression for accumulator:

When the accumulator is at equilibrium position, assuming that that (P_A, V_A) and (P_{A0}, V_{A0}) represent the gas pressure and volume. From ideal gas law:

$$m = \frac{PV}{RT} \tag{8}$$

P – Pressure
V – Volume
R – Gas constant
T – Absolute temperature

For a piston in two different state:

$$m_1 = m_2$$

$$Therefore: \frac{P_1V_1}{R_1T_1} = \frac{P_2V_2}{R_2T_2} \tag{9}$$

$$Assume: R_1 = R_2, T_1 = T_2$$

$$Therefore: \mathrm{P_1V_1} = \mathrm{P_2V_2} \tag{10}$$

For an adiabatic process, the equation can be expressed as:

$$P_1V_1^{\ k} = P_2V_2^{\ k} \tag{11}$$

$$PV^{\ n} = C \tag{12}$$

P_1 – maximum gas pressure at full charged accumulator
V_1 – volume of gas at maximum pressure
P_2 – minimum operating pressure

V_2 – Volume of gas at pressure P_2
k – factor for linearity.

This equation can further be expressed as:

$$p_A = kP_{A0}V_{A0}^{-1}q_R \tag{13}$$

G. Flow equation of pipeline:

The cylinder and accumulator and cylinder and depth compensator are connected by pipeline. These pipelines have a hydraulic conductivity Cq_R that indicates the characteristic of the pipeline to transmit oil when subjected to pressure gradient. Therefore; the relationship between the volumetric flow and pressure can be expressed by:

$$q_R = \frac{\pi d_g{}^4}{128\mu l}(p_C - p_{AC}) \tag{14}$$

Where d_g is the inside diameter of the pipeline, μ is the dynamic viscosity of hydraulic oil and l is the length of the pipeline. It can be assumed that $p_{E0} = p_{G0}$, which is the cylinder pressure, is equal to the accumulator pressure.

Therefore;

$$\begin{aligned} p_C - p_{AC} &= (p_C - p_{C0}) - (p_{AC} - p_{AC0}) \\ &= \Delta p_C - \Delta p_{AC} \end{aligned} \tag{15}$$

Hence; the hydraulic conductivity can be expressed as:

$$C_{qR} = \frac{\pi d_g^4}{128\mu l} \tag{16}$$

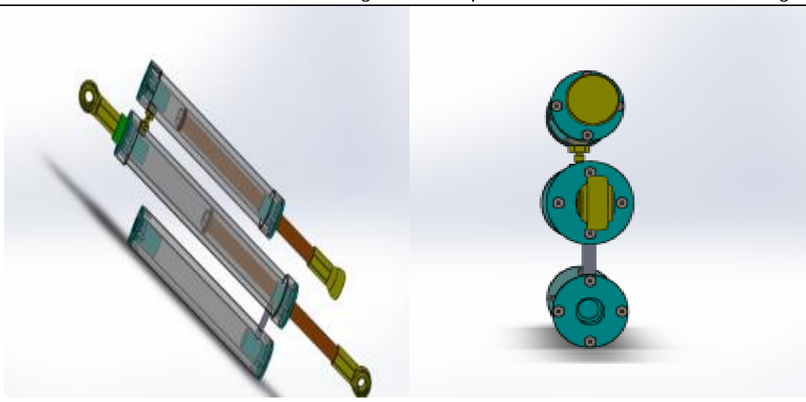
The system's damping coefficient is changed by controlling the servo valve's opening area. The damping can be calculated using the equation:

$$C_v = \frac{\rho_{oil}}{2C_f^2\alpha^2}A_0 \tag{17}$$

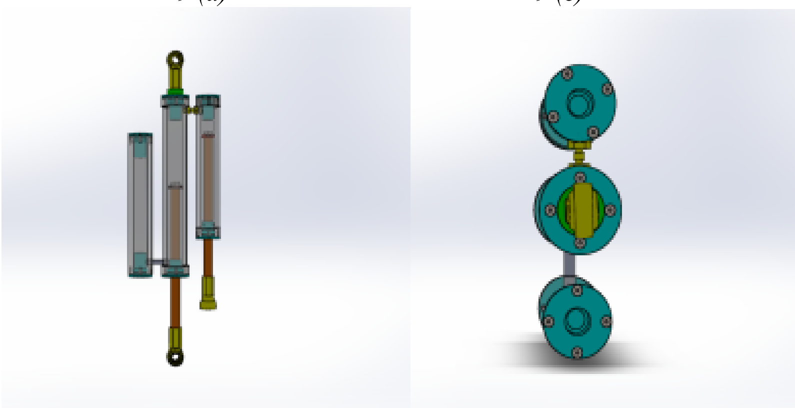
ρ_{oil} – Density of the oil
α – Ratio of the area of the valve opening
A_0 – Area of the connection of the cylinder
C_f – Damping coefficient

3. Results and discussion

The different views of the compensator used in the analysis of this project are shown in Fig. 9. The compensator is simulated with a sinusoidal and random wave input in MatLab. The input has amplitude of 1m and a frequency value of 0.17*Hz* and the frequency value is constant during the simulation. The mass of the input is 200*ton*, pipe diameter is 0.08*m* and the pipe length is 0.5*m* with other parameters stated in previous section in the paper. The response of the input was then observed and is shown in Fig. 10 and Fig. 11.



9 (a) *9 (c)*



9 (b) *9(d)*

Fig. 9: Showing different views of the PHC system [9(a) - 3D view, 9(b) - front view, 9(c) – end view, and 9(d) – top view]

In this case the PHC system with cylinder, accumulator and depth compensator connected in series with pressured pipes is investigated. The authors of this paper built a simulation model of the system using MatLab/Simulink. Parameters describing the model (cylinder cross-sectional area, connecting pipe diameter, bolts and nuts, etc.) are calculated using the same software. Under the impact of sinusoidal wave the compensation rate of the proposed passive heave system is approximately 77%. The proposed PHC system is more effective for high sea wave frequency than for low wave frequencies. The compensator can compensate all of the frequency in sea condition 4. The result of the compensator response to the input wave is as shown in Fig. 10.

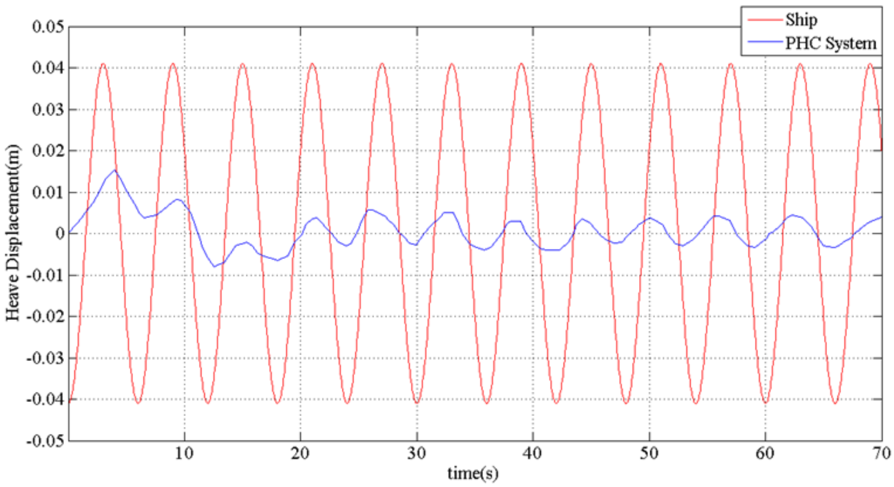


Fig. 10. Corrected response of the PHC system under the influence of sinusoidal wave

3.1 Response of random wave

The significant wave height of sea condition 4 in Guyana's deep sea area is 2.3*m*. The simulation for the random wave was done using the same parameters of the PHC system and the response is shown in Fig. 11. The distribution of the wave energy, which is relative to frequency under sea condition 4 is obtained by spectrum analysis method.

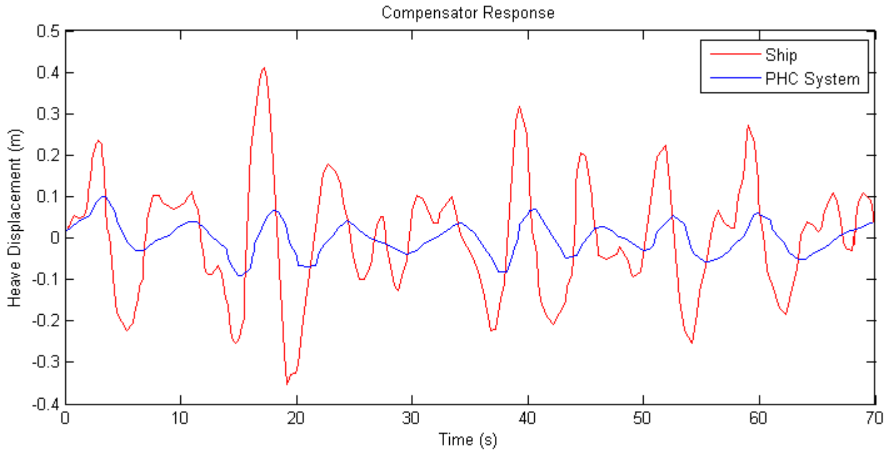


Fig. 11. Response of the PHC system under the influence of random wave

The compensator shows good response under the wave conditions and the selected parameters. The compensator shows reasonable compensation value during constant simulation. The response is more linearize and the values are small, hence; the compensator is suitable for application. Fig. 12 shows the step response for the mass of 200*ton*, this response is observed to have a damping C_v of 1.4. The depth compensator allows keeping the step response for the spring mass of 200*ton*.

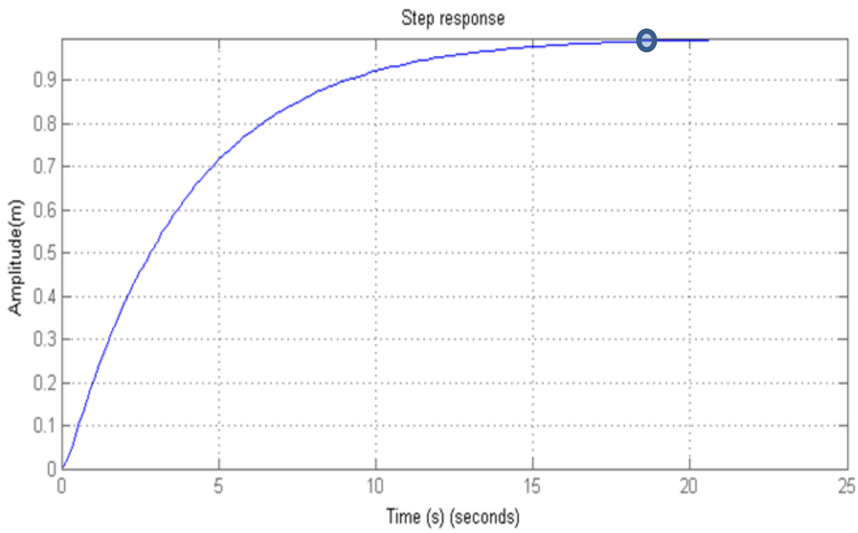


Fig. 12. Step response of the mass

The project mass of 200*ton* has a good response with an average setting time of 18*s*. This means when the compensator and payload is submerged below the splash zone it will take approximately 18*s* before the compensator becomes fully active and provide it's fully efficiency at the design amplitude of 1m. Fig. 12. shows as the compensator is submerged the system will start to respond and initiate compensation but it will take approximately 18*s* to work at its capacity of 100% and becomes stable.

4. Conclusions and Future Plan

4.1 Conclusion

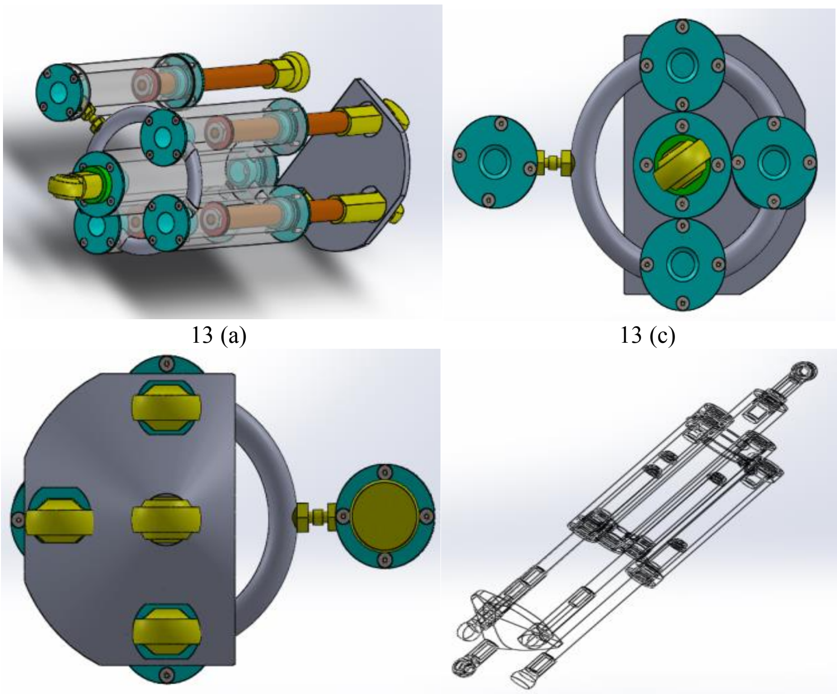
The heave compensator presented in this article has reached its objectives. The compensator with a sinusoidal and random input is simulated with MatLab. Moreover, the main and sub parameters are calculated. The input wave has a wave period of 6*s*.The input has amplitude of 1*m* and a frequency value of 0.17*Hz* and the frequency value is constant during the simulation. The compensator performance is simulated when the vessel is moved by the ocean waves. The following conclusions are drawn:

1. The response of the PHC system is acceptable. It mitigates the input motion under the selected parameters and environmental condition.
2. The compensator has an efficiency of approximately 77%.
3. The step response for the project mass of 200*ton* is approximately 18*s*.
4. The point of maximum load exerted on the system is at the splash zone, the transition area from air to water.
5. The efficiency of the PHC system can be improved by using a larger cylinder. However, the response time may be longer and this will not be feasible because of the high economic cost.
6. The system should be validated experimentally before real world application.

In the future the authors will conduct additional research on the compensator. In addition, a physical model will be constructed and checked with experimental investigation.

4.1 Future plan

The future plans of the author are to conduct experimental test of a model of the PHCs presented in this paper and determine its efficiency in real world application. Besides, the authors would like to present in this paper a new type of PHC system which can be applicable for heavy load lifting operations in offshore environment. This model was created and design using the software SolidWorks, with the aim of creating a more accurate heave compensation system, improving accuracy and providing additional features for application. The proposed system is shown in Fig. 13.



13 (a) 13 (c)

13 (b) 13(d)

Fig. 13: Showing different views [13(a) – 3D view of the model, 13 (b) bottom view, 13(c) – top view, and 13 (d) – schematic view of the model]

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References

Ablow, C M. and Schechter, S., (1983): Numerical simulation of undersea cable dynamics, Ocean Engineering, 10(6), 443-457. https://dx.doi.org/10.1016/0029-8018(83)90046-X

Bliek, A. (1984): Dynamic analysis of single span cables, Ph. D thesis, Dept. of Ocean Engineering, Massachusetts Institute of Technology, United States.

Cannell, D., Labbe, C., Odigie, E. and Riddell, S. (2016): Passive heave compensator, US Patent 20,160,002,012. https://patents.google.com/patent/US9718652B2/en

Feng, X., Li, Y. and Xu, H. (2011): The next generation of unmanned marine vehicle dedicted to the 50 anniversary of the human world record diving 10912m, Jiqiren (Robot) 33, 113-118. http://en.cnki.com.cn/Article_en/CJFDTotal-JQRR201101020.htm

Hover, F. S., Grosenbaugh, M. A. and Triantafyllou, M. S. (1994): Calculation of dynamic motions and tensions in towed underwater cables, IEEE Journal of Ocean Engineering, Vol. 19, No. 3. https://ieeexplore.ieee.org/abstract/document/312921

Frederick, R., Driscoll (2000): A comparison of Ship-Mounted and Cage-Mounted passive heave compensation systems. Department of Mechanical Engineering and School of Earth and Ocean Sciences, University of Victoria. J. Offshore Mech. Arct. Engg., Vol.122, No. 3, pp. 214-221. http://offshoremechanics.asmedigitalcollection.asme.org/article.aspx?articleid=1454023

Hellrand, A., Moen, L.K., Faanes, T. (1990): Crane control system with active heave compensation and constant tension modes onboard the vessel stena wel/ servicer. http://dx.doi.org/10.4043/6259-MS.

Huang, L., Zhang,Y., Zhang, L., and Liu, M. (2013): Semi-active drilling drawworks heave compensation system, Petroleum Exploration and Development, Vol. 40, No. 5, pp. 665-670. https://www.sciencedirect.com/science/article/pii/S1876380413600890

Jakobsen, S. (2008): Passive heave compensation of heavy modules, Master's Thesis, University of Stavanger, Norway. https://brage.bibsys.no/xmlui/handle/11250/182671

Jia, N., Shaojin, L., Mingfeng, W., Xiaozhou, H., and Yu, D. (2009): The simulation research on passive heave compensation system for deep sea mining, College of Mechanical and Electrical Engineering Central South University Changsha, Hunan Province, China.10.1109/ICMA.2009.5246165 https://ieeexplore.ieee.org/abstract/document/5246165

Korde, U. (1998): Active heave compensation on drill-ships in irregular waves, Ocean Engineering, Vol. 25, No. 7, pp. 541-561. https://doi.org/10.1016/S0029-8018(97)00028-0

Ni, J., Liu, S., Wang, M., Hu, X, Dia, Y. (2009): The simulation research on passive heave compensation system for deep sea mining. In Mechatronics and Automation. International Conference on IEEE, 2009: 5111-5116. https://www.ncbi.nlm.nih.gov/pmc/articles/PMC4505866/

Ormond, M.J. (2011): Depth Compensated subsea passive heave compensator, US Patent 7 934 651, 3 May, 2011.

Quan, W., Liu, Y., Zhang, Z., Li, X. and Liu, C. (2016a): Scale model test of a semi-active heave compensation system for deep-sea tethered ROV, Ocean Engineering, Vol. 126, pp. 353-363. http://dx.doi.org/10.1016/j.oceaneng.2016.09.024

Quan, W., Liu, Y., Zhang, A., Zhao, X. and Li, X. (2016b): The nonlinear finite element modeling and performance analysis of the passive heave compensation system for the deep-sea tether ROVs, Ocean Engineering, Vol. 127, pp 246-257. http://dx.doi.org/10.1016/j.oceaneng.2016.10.011

Sullivan, R.A., Davenport, M.R. and Clements, R.E., (1984): Multipurpose active/passive motion compensation system, 16th Annual Offshore Technology Conference, April, pp. 235–240, ISBN 978-1-61399-077-3. http://dx.doi.org/10.4043/4737-MS.

Yan, Y., Ma, P.S., Wang, D.Y. and Gao, X.G. (2005): Development of deep sea ROV and its working system, Robot 27, 82-89. http://en.cnki.com.cn/Article_en/CJFDTotal-JQRR200501016.htm

Wang, M. and Jia, N. (2009): The simulation research on passive heave compensation system for deep sea mining, Conference: Mechatronics and Automation. https://ieeexplore.ieee.org/abstract/document/5246165

Woodacre, J., Bauer, R., Irani, R. (2015): A review of vertical motion heave compensation systems, Ocean Engineering, 104,140-154. https://dx.doi.org/10.1016/j.oceaneng.2015.05.004