Dynamic simulation of the passive heave compensator for a strand jack

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Abstract. The passive heave compensator for a strand jack unit used in deep sea shipwreck salvage is studied. A dynamic model based on the Lagrange equations is developed, which considers the strand elasticity. Numerical simulations are performed for a range of compensator stiffness, which are determined by the gas volume. Numerical results show that increasing gas volume reduces the stiffness of the compensator, and the resulting displacements of compensator platform and sunken vessels decrease.

1. Introduction
Strand jack lifting is a new method to lift sunken vessel especially suitable for deep sea situations. During the lifting operation, the maximum forces and lowest weather restrictions are expected. But in rough seas, large heave motions cause high peak loads at the sunken vessel and the lift system. Therefore, the method of load reduction is needed. Passive heave compensator (PHC) is a common method to reduce peak load and increase the workability in offshore operation.

A general PHC works as a hydro-pneumatic spring absorbing high load peaks due to the ship movement. The PHCs are widely used in situations with large size and heave load lifting field as they do not require an external source of energy to operate [1]. Nam et al. [2] investigated the PHC by varying the stiffness of hydro-pneumatic spring and the period of heave movement. He used a spring to model the PHC without consideration the damper. Duc and Trong [3] pointed out the PHC is effective to reduce the vibration displacement, and keeps the wireline with a nearly constant tension independent of the heave motion of the ship. Further, it is more effective to keep cable tension constant than motion compensation [4].

This paper presents a dynamic model based on the Lagrange equations for the passive heave compensator, which mitigates the strand jack unit from the motion of barge. The dynamic model considers the strand elasticity. The paper is organized as follows. Section 2 provides details on the strand jack synchronous lifting system. In section 3, the proposed system dynamic model is presented. Section 4 presents the results of the simulation. The last section presents the conclusion.

2. the Strand jack Lifting System
The strand jack lifting system consists of a number of strand jack unit. All strand jack units are controlled to lift a sunken vessel synchronously. Fig.1(a) shows a schematic of the strand jack unit with a passive heave compensation system, which comprises two sets of compensators. Each one of the proposed compensators includes a compensator cylinder mounted on the deck of the barge, a nitrogen accumulator connected to the compensator cylinder by a pipeline. The compensator platform is connected to the
piston rode of the compensator cylinder, which supports the weight of the sunken vessel. The load is lifted by strand jack through strand.

![Diagram](image)

**Figure 1. Strand jack lifting system**

The working principle of the proposed compensator is as follows: when the salvage barge heaves up with the wave, the oil of the passive hydraulic cylinders flows into the oil chamber of the piston accumulator and the volume of the nitrogen decreases to compensate the increased displacement of the salvage barge. Conversely, when the salvage barge heaves down, the gas of the piston accumulator is expanded and the oil flows back to the passive hydraulic cylinder to compensate the decreased displacement of the salvage barge.

3. **Dynamic Model of the proposed Compensator**

Fig.1(b) shows the proposed dynamic model. In the model, the passive heave compensator is modeled as two spring-dampers characterized by constant stiffness and damping parameters, $k_1$ and $c_1$ respectively. Each strand is broken down into two linear-elastic elements with stiffness of $k_2$, which are connected inline.

When lifting operation is performed, the sunken vessel experiences damping force, since it is surrounded by a viscous fluid. The damping force may be interpreted as added mass and hydrodynamic damping $c_2$.

3.1. **Accumulator modelling**

The thermodynamic process of the nitrogen within accumulator can be modelled by adiabatic process. Assuming relatively small volume change during the compensator operations, the stiffness of the pneumatic spring can be expressed as

$$k_i = \frac{\lambda P_{d0} S_a^2}{V_{d0}}$$  \hspace{1cm} (1)

Where $P_{d0}$ and $V_{d0}$ are the gas pressure and volume within the accumulator at the equilibrium position, respectively; $\lambda$=1.4 is adiabatic coefficient for nitrogen; and $S_a$ is the area of non-rod chamber of the compensator cylinder.

3.2. **Pipe modelling**

A pipe connects the compensator cylinder and the accumulator. Using the Hagen-Poiseuille equation, pressure difference across the pipe length is

$$P_a - P_e = \frac{32\mu d u}{d^2}$$  \hspace{1cm} (2)
Where \( P_a \) and \( P_c \) are the instantaneous oil pressure within the compensator cylinder and the accumulator, respectively; \( \mu \) is the dynamic viscosity of the oil; \( d \) and \( l \) are the cylindrical pipe inner diameter and length respectively; and \( u \) is the average oil flow rate within the pipe.

The damping force \( F_c \) provided by the pipe is

\[
F_c = \frac{128 \mu d}{\pi l^4} S_a^2 \left( x_1 - y_1 \right)
\]

(3)

Where \( x_1 \) is the vertical displacement of the salvage barge, and \( y_1 \) is the vertical displacement of the compensator platform.

Hence, the pipe damping coefficient is

\[
c_1 = \frac{dF_c}{d(x_1 - y_1)} = \frac{128 \mu d}{\pi l^4} S_a^2
\]

(4)

3.3. Strand modelling

According to strain stress relationship, the strand tension can be determined by

\[
F = A_s E \frac{\delta}{L}
\]

(5)

Where \( A_s \) is cross-section area of the strand; \( E \) is Elastic Modulus of strand; \( \delta \) and \( L \) are the length change and the initial length of the strand, respectively.

the stiffness of strand can be expressed as

\[
k_2 = \frac{A_s E}{L}
\]

(6)

3.4. Fluid damping

In lifting operation, the motion of sunken vessel submerged in water generates fluid damping force, which can be described by the Morison Equation.

\[
F_d = -\frac{1}{2} \rho |v| c_d A_v
\]

(7)

Where \( \rho \) is the water density, \( v \) is the average fluid velocity, \( c_d \) is the drag coefficient, and \( A_v \) is the cross-section area of the sunken vessel.

Since \( F_d = -c_2 v \), the damping coefficient \( c_2 \) can be expressed as follows:

\[
c_2 = \frac{F_d}{v} = \frac{1}{2} \rho |v| c_d A_v
\]

(8)

3.5. System dynamic model

The kinetic energy of the lifting system is

\[
T = \frac{1}{2} M_l \dot{y}_1^2 + \frac{n}{2} (0.5 \cdot M_s \cdot \dot{y}_2^2) + \frac{1}{2} M_p \cdot \dot{y}_3^2
\]

(9)

Where \( M_l \) is the mass of strand jack unit and the compensator platform; \( M_s \) is the mass of a strand; \( M_p \) is a virtual mass, including the mass of the sunken vessel, strand and the add mass; \( n \) is the total number of strand; \( y_2 \) and \( y_3 \) are the vertical displacement of the first element of the strand and the sunken vessel respectively.

The added mass coefficient takes 0.15.\textsuperscript{6} Then \( M_p \) equals to

\[
M_p = 1.15 \cdot M_s + 0.5 \cdot n M_l
\]

(10)

The potential energy of the lifting system is

\[
V = k_1 (y_1 - x_1)^2 + \frac{n}{2} k_2 \left[ (y_1 - y_2)^2 + (y_2 - y_3)^2 \right]
\]

(11)
The dissipation energy of the lifting system is

$$D = c_1 \cdot (\dot{y}_1 - \ddot{x}_1)^2 + \frac{1}{2} c_2 \cdot \ddot{y}_2^2$$  \hspace{1cm} (12)

The Lagrangian is

$$L = \frac{1}{2} M_j \cdot \dot{y}_1^2 + \frac{n}{4} M_r \cdot \dot{y}_2^2 + \frac{1}{2} M_p \cdot \dot{y}_3^2 - k_2 (y_1 - x_1)^2 - \frac{n}{2} k_2 [(y_1 - y_2)^2 + (y_2 - y_3)^2]$$  \hspace{1cm} (13)

Then the dynamic equations of the lifting system are derived using the Euler-Lagrange equations as follows:

$$\begin{bmatrix}
M_j & 0 & 0 \\
0 & \frac{n}{2} M_r & 0 \\
0 & 0 & M_p
\end{bmatrix}
\begin{bmatrix}
\ddot{y}_1 \\
\ddot{y}_2 \\
\ddot{y}_3
\end{bmatrix}
+
\begin{bmatrix}
2k_1 + nk_2 & -nk_2 & 0 \\
-nk_2 & 2nk_2 - nk_2 & 0 \\
0 & -nk_2 & nk_2
\end{bmatrix}
\begin{bmatrix}
y_1 \\
y_2 \\
y_3
\end{bmatrix}
+
\begin{bmatrix}
2c_1 & 0 & 0 \\
0 & 0 & 0 \\
0 & 0 & c_2
\end{bmatrix}
\begin{bmatrix}
\dot{y}_1 \\
\dot{y}_2 \\
\dot{y}_3
\end{bmatrix}
= \begin{bmatrix}
2c_1 \ddot{x}_1 + 2k_2 \ddot{x}_1 \\
0 \\
0
\end{bmatrix}$$  \hspace{1cm} (14)

4. Simulation results

4.1. Simulation Setup
In this Section the simulation setup is given. The sunken vessel weights 8000t in air and has dimension of 150 × 25 × 16 m. Twenty stand jacks are used, then the load of each strand jack is 400t, namely $M_b=400t$. The parameters of strand used in this paper are listed in Table 1. The other parameters values are chosen as $n=20$, $L=300$, $M_l=1t$, $\rho=1.02 \text{g/cm}^3$, $c_d=0.95^{[6]}$.

| Diameter (mm) | Cross-sectional Area (mm$^2$) | Elastic Modulus (GPa) | Weight (kg/m) |
|---------------|-------------------------------|-----------------------|---------------|
| 15.2          | 139                           | 195                   | 1.19          |

The input signal of the salvage barge motion is of a sinusoidal signal. Assuming that the sinusoidal signal is with an amplitude of 2.1m and time period of 5.8s.

4.2. Simulation results
We analysis the displacement of compensator platform and the tension variation of strand, both with the compensator on and off. The simulation is performed with the value of $V_{sh}=4m^3$, $d=19mm$, $l=2m$, and the diameter of compensator cylinder is 240mm. The tension Variation of strand is calculated by

$$\Delta F_p = (y_1 - y_3)k$$  \hspace{1cm} (15)

Figure 2 shows the simulation results. In general, the compensator is able to reduce the motion of the compensator platform, the motion of the sunken vessel, as well as strand tension variation, effectively.
Figure 2. Numerical results with compensator on and off

Figure 3 shows the effectiveness of the compensator with different gas volumes. The results clearly show that increasing gas volume reduces the stiffness of the compensator, the resulting displacements of compensator platform and sunken vessels decrease. Meanwhile, compensator with more gas volume reduces the tension variation of strand.
5. Conclusion
The performance of a passive heave compensator for a strand jack unit is investigated under different gas volume using a dynamic model based on the Lagrange equations. The compensator is able to reduce the motion of the compensator platform and the sunken vessel, as well as the tension variation in strand. The effectiveness of the compensator increases with increasing the gas volume.

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