# SLIDING MODE CONTROLLER FOR SALVAGING OF SUNKEN VESSELS

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## SUMMARY

This paper presents a mathematical model and numerical time-domain approach to simulate the dynamics of a sunken ship/vessel being raised from the seafloor by a gas inflating system. Based on the simplified four-degree-of-freedom equations of rigid-body vessel motion, a sliding mode-depth controller is designed to ensure the hydrodynamic stability within the assumed diving vertical plane. Numerical simulations are carried out by solving the standard State Dependent Ricatti Equation in a body-fixed coordinate reference frame. Preliminary results in terms of coordinate positions or trajectories, velocities and angular velocity components of the raising body are evaluated based on an experimental pontoon model. The efficacy of the sliding mode controller is investigated for the exemplified marine salvage operation.

## NOMENCLATURE

| | |
|---|---|
| $A_S$ | Vessel surface area (m$^2$) |
| $B$ | Buoyancy force (N) |
| $b$ | Vessel breadth (m) |
| $C_D$ | Drag coefficient |
| $C_g$ | Centre of gravity |
| $F$ | Total breakout force (N) |
| $G$ | Vessel submerged weight (N) |
| $f$ | Gas flow rate (m$^3$s$^{-1}$) |
| $g$ | Gravitational acceleration (ms$^{-2}$) |
| $I$ | Mass moment of inertia (kgm$^2$) |
| $I_{yy}$ | Moment of inertia about Y axis (kgm$^2$) |
| $l$ | Vessel length (m) |
| $M$ | Pitch moment (Nm) |
| $M'_{\dot{w}}, M'_{\dot{q}}$ | Hydrodynamic coefficients (moments) |
| $m$ | Vessel mass (kg) |
| $p, q, r$ | Roll, pitch and yaw rates (rad/s) |
| $u, v, w$ | Surge, sway and heave velocities (ms$^{-1}$) |
| $u_c$ | Control parameter |
| $V$ | Volume of gas (m$^3$) |
| $V_{min}$ | Minimum volume of gas required (m$^3$) |
| $W$ | Vessel dry weight (N) |
| $X, Z$ | Surge and heave forces (N) |
| $X_e, Y_e, Z_e$ | Earth- fixed reference frame |
| $X_b, Y_b, Z_b$ | Body- fixed reference frame |
| $x_s$ | State-space vector |
| $x, y, z$ | Displacement variables (m) |
| $x_G, z_G$ | Coordinates of centre of gravity |
| $x_B, z_B$ | Coordinates of centre of buoyancy |
| $Z'_{\dot{w}}, Z'_{\dot{q}}$, | Hydrodynamic coefficients (forces) |
| $\rho$ | Sea water density (kgm$^{-3}$) |
| $\rho_g$ | Gas density (kgm$^{-3}$) |
| $\nabla$ | Volumetric form displacement (m$^3$) |
| $\Phi, \theta, \psi$ | Roll, pitch and yaw angles (deg.) |
| $\mu$ | Increase in buoyancy with depth (Nm$^{-1}$) |
| $k, \sigma, \eta, \Phi_b$ | Sliding mode control parameters |

## 1. INTRODUCTION

Marine salvage is an operation of rescuing a ship/vessel, its cargo or other properties from impending peril. The salvage comprises of towing and refloating a sunken or stranded vessel with the main purposes to prevent the marine environment from the pollution and to clear a channel for the navigation. Ships sink or capsize because they lose their buoyancy or stability due to the collision, battle or weather damage, flooding and other means. The rescue of a damaged vessel is a very difficult task when compared to an intact ship in the same location [1-3]. Salvaging of sunken ships requires both the recovery of sufficient buoyancy to bring the ship afloat and the suitable buoyancy distribution to regain the satisfactory condition of stability, trim and strength.

The concept of using a buoyancy system (e.g. the gas inflated bags) for salvaging sunken vessels from the deep ocean has been around for centuries. This operation is based on the well-known 'Archimedes' principle for which the force on the object can be determined by subtracting the dry weight of the object from the weight of the fluid displaced by that object [4]. In general, the bottoms of inflatable bags (e.g. balloons) are attached to the payload to be lifted and inflated using pipes from the gas generating system. The main drawback of using the inflating bags for marine salvage operation is due to the difficulty in controlling the vertical speed as the ship ascends. A large buoyancy force may be initially required to separate the ship from the seabed, resulting in an excessive vertical speed. During the ascent, any trapped air inside the hull may also expand and further increase the buoyancy; thus the balloons themselves may slightly expand as the hull ascends. Excessive speed will result in a potentially-hazardous working environment to divers and salvaging crews and this may cause the lifting bag to breach the surface of the water so fast that the air escapes from the bottom. This purge cause the payload to sink back to the bottom which, in turn, results in a loss of time, damage to the hull, high operating and maintenance costs, and the risk to divers and crew members [5, 6].

Hence, an automatic controller is required to regulate the volume flow rate of filling gas inside the balloon (i.e. the pump flow rate) to ensure a safe and steady ascent [6]. In addition, it is necessary to deal with the excessive buoyancy force once the ship is free from the sea bottom. This circumstance may require a controlled but rapid gas release. In the present study, a sliding mode controller is designed to overcome the above-mentioned issue.

This paper is structured as follows. In Section 2, the dynamic equations of rigid-body motion describing a raising sunken vessel are formulated and presented in a state-space form. In Section 3, a sliding mode-depth controller is overviewed. Numerical simulation results based on a pontoon experimental model are discussed in Section 4. The scope for future works is also outlined. The paper ends with the conclusions in Section 5.

## 2. PROBLEM FORMULATION

Ship calculations for the salvaging operation are typically less detailed than those in the preliminary design. The dynamic analysis of raising sunken vessels depends on the available information pertaining to a particular ship scenario. A number of assumptions are usually required to simplify the problem. The loads acting on a sunken vessel consist of the breakout forces, hydrostatics and hydrodynamics. The hydrodynamic forces with velocity and acceleration components depend on the empirical coefficients derived from a physical model test or theory. In the following, the dynamic equations of motion are presented based on the assumed body-fixed variables in the diving plane. These variables are then transformed to the earth–fixed coordinates using the kinematic relations.

### 2.1 MODEL OF A RAISING VESSEL

To describe the motion of a raising vessel, two reference frames are considered as in Figure 1, including the earth- and body-fixed frames. The origin of the body-fixed frame coincides with the centre of gravity ($C_g$) of the vessel being in the principle plane of symmetry [8, 10]. The positions and orientations of the vessel (kinematic variables) are expressed with respect to the earth-fixed coordinates whereas the linear and angular velocities of the vessel (dynamic variables) are expressed in the body-fixed coordinates. The relation of the two coordinate systems is based on the Euler angles ($\Phi$, $\theta$, $\psi$).

To describe the dynamics of the sunken vessel ascending from the seafloor, it is preliminarily assumed that:

- the vessel behaves as a rigid body,
- the acceleration of a point on the surface of the earth is neglected,
- the external loads comprise of the breakout, hydrostatic and hydrodynamic drag forces, and
- the seabed is flat creating a breakout force of 1.3 times the ship wet weight (see Section 2.2 c).

### 2.2 EQUATIONS OF VESSEL MOTION

As the problem is concerned with the dynamics of raising sunken vessels (i.e. the control surfaces are inactive), it is further assumed to consider only the diving vertical-plane (surge, heave and pitch) motions in the stability analysis. Thus, the system model variables include the surge velocity ($u$), heave velocity ($w$), pitch angle ($\theta$), pitch rate ($q$) and global depth position ($z$).

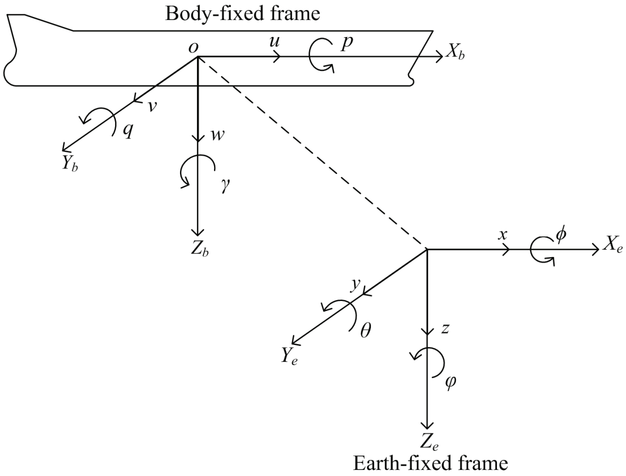


**Figure 1**: Vessel model and reference coordinates

Accordingly, the first-order equations of motion read [8]

$$m\left(\dot{u} + wq - x_G q^2 + z_G \dot{q}\right) = X \quad (1)$$

$$m\left(\dot{w} - uq - x_G \dot{q} - z_G q^2\right) = Z \quad (2)$$

$$I_{yy}\dot{q} + m\left[z_G(\dot{u} + wq) - x_G(\dot{w} - uq)\right] = M \quad (3)$$

Not that the surge motion couples with the heave and pitch motions through the meta-centric height. This dynamic coupling could be eliminated by redefining hydrodynamic coefficients with respect to the ship's $C_g$ [11-14]. For a sunken vessel, it is also known that the forward speed is zero. However, due to the nonlinear coupling and external forces, the surge motion may exist. In this study, we implement the sliding mode control strategy to the linear equations of motion with the neglected surge motion.

#### 2.2 (a) Hydrostatic Force and Moment

Hydrostatic force and moment are due to the ship weight $W$ and buoyancy $B$. The buoyancy of the sunken vessel may be changed due to the sea density variation and to the compressibility of the hull. Herein, the former case is accounted for through a linear volume change with depth,

$$B = \rho \nabla g + \mu z \ (\rho / \rho_o) \quad (4)$$

In the body-fixed coordinate system, the hydrostatic components of force and moment for heave/pitch motions are [8, 10]:

$$Z_{HS} = (W - B)\cos\theta \quad (5)$$

$$M_{HS} = -(z_G W - z_B B)\sin\theta - (x_G W - x_B B)\cos\theta \quad (6)$$

#### 2.2 (b) Hydrodynamic Force and Moment

The hydrodynamic components of force and moment for heave/pitch motions are [8, 11]:

$$Z_{hydro} = \frac{1}{2}\rho\ell^3 Z_{\dot{w}}'\dot{w} + \frac{1}{2}\rho\ell^4 Z_q'\dot{q} - \frac{1}{2}\rho C_D A_s w^2, \quad (7)$$
$$M_{hydro} = \frac{1}{2}\rho\ell^4 M_{\dot{w}}'\dot{w} + \frac{1}{2}\rho\ell^5 M_{\dot{q}}'\dot{q}\ , \quad (8)$$

Let,

$$Z_{\dot{w}} = \frac{1}{2}\rho l^3 (Z_{\dot{w}}'), \quad (9)$$

$$\dot{Z}_{\dot{q}} = \frac{1}{2}\rho l^4 (Z_{\dot{q}}'). \quad (10)$$

Consequently, Eq. (7) may be simplified as

$$Z_{hydro} = Z_{\dot{w}}\ \dot{w} + Z_q\ \dot{q} - \frac{1}{2}\rho C_D A_s w^2. \quad (11)$$

Similarly, by letting

$$M_{\dot{w}} = \frac{1}{2}\rho l^4 (M_{\dot{w}}'),$$

$$M_{\dot{q}} = \frac{1}{2}\rho l^5 (M_{\dot{q}}') \quad (12)$$

Therefore, Eq. (8) may be simplified as

$$M_{hydro} = M_{\dot{w}}\,\dot{w} + M_{\dot{q}}\,\dot{q} \quad (13)$$

2.2 (c) Breakout Force

The breakout or suction force accounts for the difference between the total lift force required and the object's wet weight. It is theoretically and empirically difficult to estimate this breakout force due to the involvement of several variables and unknowns [1]. In general, the amount of breakout force depends on the seafloor soil characteristics (i.e. the compressibility of soil skeleton and pore water, permeability etc.), the embedment depth and time, the object shape parameters and the loading conditions. The lift force (*F*) required for the complete extraction of the object from the sea bottom should be greater than their submerged weight (*G)* due to the ground reaction (*R*) exerted by the soil (Figure 2).

Sawicki and Mierczynski [15] proposed a simple formula for the estimation of lift force as $F = (1 + k_p)G$ where $k_p$ is an empirical coefficient depending on the subsoil.

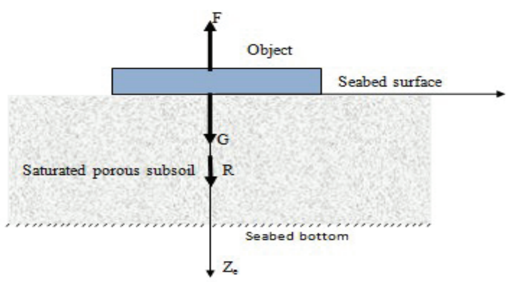


**Figure 2**: Lift force model to extract an object from the seabed [15]

Vaudrey [16] investigated the efficacy of 3 analytical methods (i.e. Muga, Liu and Lee methods) for the prediction of breakout forces with different object shapes such as a cylinder, sphere and block, with and without breakout force reduction techniques. From the analysis, it was observed that the use of breakout reduction methods such as the mud suction tubes, water flooding and air jetting would reduce the total lift force by approximately 15% and eliminate the snap loading condition. The selection of breakout reduction methods depends on the particular salvage operation, bottom soil conditions and the availability of equipments. From the above literature, the total lift force is assumed to be 1.3 times the wet weight of the vessel whereas the breakout time is assumed to be the same as the inflating time.

2.2 (d) Additional Buoyancy by the Inflating System

For the sunken vessel resting on the seafloor, the vessel weight is balanced by both the buoyancy and the ground reaction [1]. Additional force required to lift the vessel should overcome both the in-water object weight and the ground reaction. This force, described in terms of the buoyancy, could be provided by the volume of gas inside the balloons. The gas-generating system (solid, liquid or cryogenic pressurised system) is used such that the produced gas is pumped into the balloons at a desired flow rate using pipes. During the ascent, a lift bag may experience a pressure decrease due to the depth variation. This decreased pressure makes the volume of gas inside the balloon increase or makes the lift bag expand, following the Boyle's law,

$$P_1 V_1 = P_2 V_2 \text{ or } V_2 = \frac{P_1 V_1}{P_2} \quad (14)$$

where $P_1$ is the initial absolute pressure, $V_1$ the initial volume inside the lift bag, $P_2$ the final absolute pressure and $V_2$ the final volume inside the lift bag.

Nevertheless, as the pressure relief valves are provided either mechanically (spring loaded) or electrically at the top of the lift bag to purge out the excess gas inside the balloon, a constant buoyancy could be maintained [6]. The variation of volume with respect to the time during inflating process can be accounted for by considering the volume flow rate of filling gas inside the balloon (i.e. the control parameter) as follows:

$$\frac{dV}{dt} = \dot{V} = f \quad (15)$$

Consequently, the additional buoyancy provided by the inflating system can be written as

$$B_a = (\rho - \rho_g) g V \quad (16)$$

## 2.3 KINEMATIC RELATIONS

The kinematic relations are used to transform the motion variables from the local to global coordinate system [8, 10]. The simplified kinematic relations for heave and pitch motions are

$$\dot{z}_e = w \tag{17}$$

$$\dot{\theta}_e = q \tag{18}$$

## 2.4 STATE-SPACE MATRIX MODEL

For small values of pitch angle, it is assumed that $sin\theta = \theta$ and $cos\theta=1$. Imposing the linearization and neglecting the products of small motions, the equations of motion can be written in the state-space matrix form as

$$\left[M_0\right]\left\{\dot{x}_s\right\}=\left[A_0\right]\left\{x_s\right\}+\left[B_0\right]\left\{u_c\right\} \tag{19}$$

where

$$\left[M_0\right]=\begin{bmatrix} m-Z_{\dot{w}} & -mx_G-Z_{\dot{q}} & 0 & 0 & 0 \\ -mx_G-M_{\dot{w}} & I_{yy}-M_{\dot{q}} & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 1 \end{bmatrix} \tag{20}$$

$$\left[A_0\right]=\begin{bmatrix} 0 & mz_G & 0 & 0 & -(\rho_{water}-\rho_{gas})g \\ 0 & 0 & -(z_GW-z_BB) & 0 & (\rho_{water}-\rho_{gas})gx_B \\ 1 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \end{bmatrix} \tag{21}$$

$$\left\{B_0\right\}=\begin{bmatrix}0\\0\\0\\0\\1\end{bmatrix},\quad \left\{x_s\right\}=\begin{bmatrix}w\\q\\z\\\theta\\V\end{bmatrix},\quad u_c=f \tag{22-24}$$

Eq. (19) can be reduced in the form

$$\left\{\dot{x}_S\right\}=\left[A\right]\left\{x_S\right\}+\left[B\right]\left\{u_C\right\} \tag{25}$$

which is the State Dependant Riccati Equation (SDRE) where $[A]$ and $[B]$ are the input matrices given by

$$\left[A\right]=\left[M_0\right]^{-1}\left[A_0\right],\quad \left[B\right]=\left[M_0\right]^{-1}\left[B_0\right] \tag{26}$$

## 3. SLIDING MODE CONTROLLER

A sliding mode controller (SMC) is selected for regulating the volume flow rate of filling gas inside the balloons in order to maintain the stability of the raising vessel within the diving plane. This selection was made due to the following reasons [9-13]:

- SMC compensates for nonlinear behaviours
- SMC provides robustness to uncertainty
- SMC is straightforward to implement

Based on the derived SDRE (Eq. 25), the control law ($u_c$) becomes [9-13]:

$$u_C=-\left[s^TB\right]^{-1}s^TAx_S-\left[s^TB\right]^{-1}\eta\tanh\left(\frac{s^Tx_S}{\Phi_b}\right) \tag{27}$$

where $s^Tx_S = \sigma(x_S)$ is the weighted sum of errors in the state $x_s$, $s$ is the right eigen-vector of the desired closed loop system matrix , $\sigma$ is the sliding surface and $\Phi_b$ is the boundary layer thickness. $\eta$ is an arbitrary positive quantity provided to satisfy the Lypunov stability condition [14]. The values of A and B can be obtained from Eq. (26).

Let $k=[s^TB]^{-1}s^TA$, then the above equation becomes:

$$u_c=-kx_s-\left[s^TB\right]^{-1}\eta\tanh\left(\frac{s^Tx_S}{\Phi_b}\right) \tag{28}$$

The gain vector $k$ can be calculated in Matlab using the pole placement method.

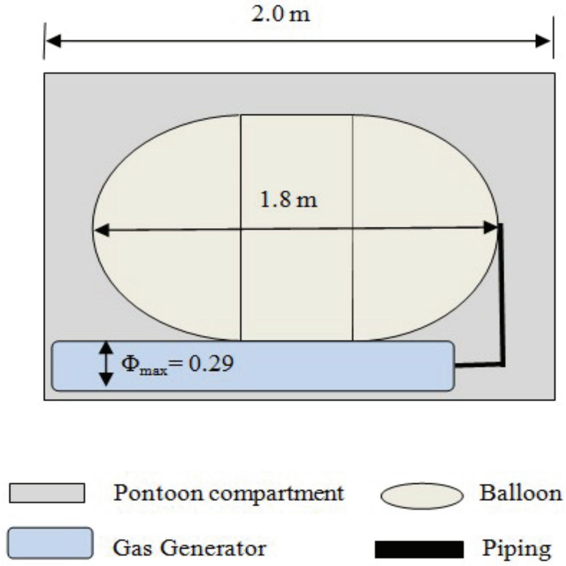


**Figure 3**: Pontoon compartment with inflating system [3]

## 4. RESULTS AND DISCUSSION

Numerical simulations based on a small-scale pontoon model have been carried out using Matlab. This model with the gas-inflated balloons will be tested at sea in the mid-2012. The pontoon is a rectangular-shaped structure with watertight compartments for internal deployment of balloons and gas generating system. The balloon length is 1.8 m and the maximum space in a single compartment for the installation of a gas generator including piping is 0.29 m in diameter and 2 m in length [3]. The minimum volume of gas required for lifting the given pontoon to a target depth may be calculated based on the hydrostatic principles. For instance, for a given water depth of 30 m, the gas volume required is about 15 $m^3$. As the volume of one single inflated balloon is 1.8 $m^3$, a total of 8 balloons are required for the inflation. Figure 3 exemplifies the installation of inflating system inside a single pontoon compartment.

Table 1: Input parameters in the simulation program

| Parameters | Values & Units |
|---|---|
| $W$ | 9320 kg |
| $l$ | 6.0 m |
| $b$ | 3.0 m |
| $\rho$ | 1025 kgm$^{-3}$ |
| $\rho_g$ | 1.017 kgm$^{-3}$ |
| $I_{yy}$ | 1481.31 kgm$^2$ |
| $Z_{\dot{w}}'$ | - 15.7x 10$^{-3}$ |
| $Z_{\dot{q}}'$ | - 0.41x10$^{-3}$ |
| $M_{\dot{w}}'$ | - 0.53x10$^{-3}$ |
| $M_{\dot{q}}'$ | - 0.79x10$^{-3}$ |

Some input physical and empirical parameters are given in Table 1 for the pontoon model. The inflation time of filling gas inside the balloons depends on the initial flow rate whereas the breakout time of the pontoon from the seafloor is assumed to be 100 s. The latter would be changed if the appropriate suction force model was considered. In the following, two cases of numerical simulations are considered for different target depths being equal to 30, 40 and 50 m. In the first case, the initial flow rate is variable for different depths, whereas in the second case the initial flow rate is fixed. The latter would result in different numerical simulation time depending on the sliding-mode control.

4.1 CASE 1: VARYING INITIAL FLOW RATE

With 3 target (30, 40 and 50 m) depths, 3 different initial flow rates (0.15, 0.1875 and 0.25 m$^3$s$^{-1}$, respectively) are considered such that the payload reaches the desired depths at the same time (about 1000 s). The inflation time of filling gas inside the balloons is suitably taken as 100, 80 & 60 s respectively. The obtained vertical dynamic responses (vertical trajectory, ascent velocity, pitch angle and pitch rate) and the variation of the control parameter (i.e. the flow rate) are presented in Figures 4-7 and 8, respectively.

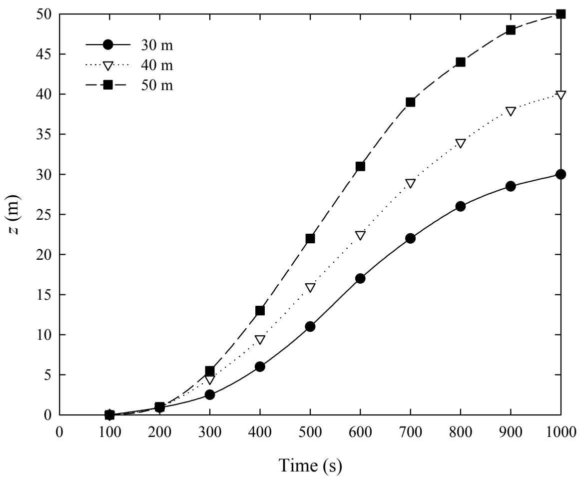


**Figure 4**: Case 1 - Variation of ship vertical position

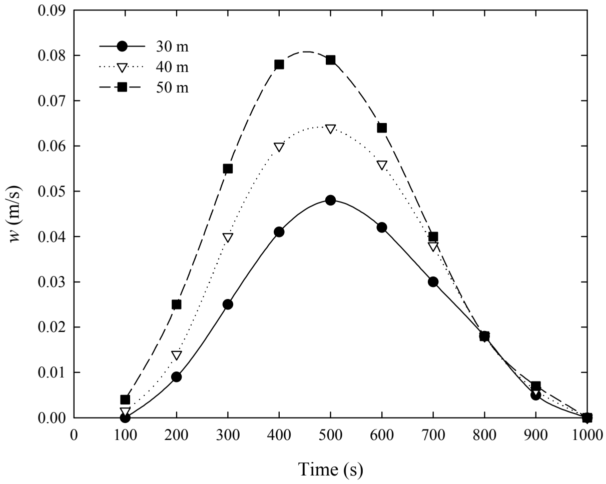


**Figure 5**: Case 1 - Variation of ship ascent velocity

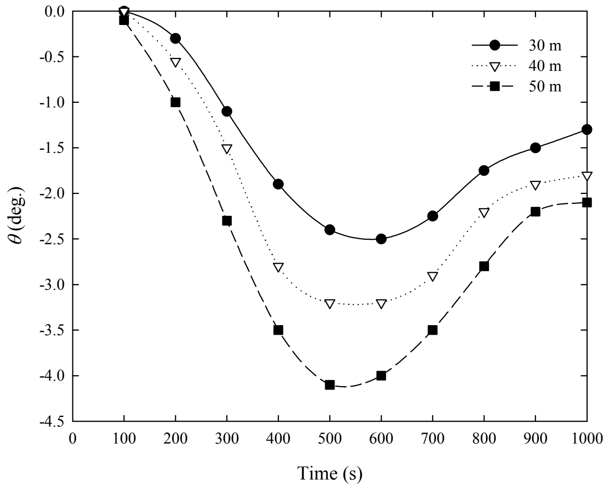


**Figure 6**: Case 1 - Variation of ship pitch angle

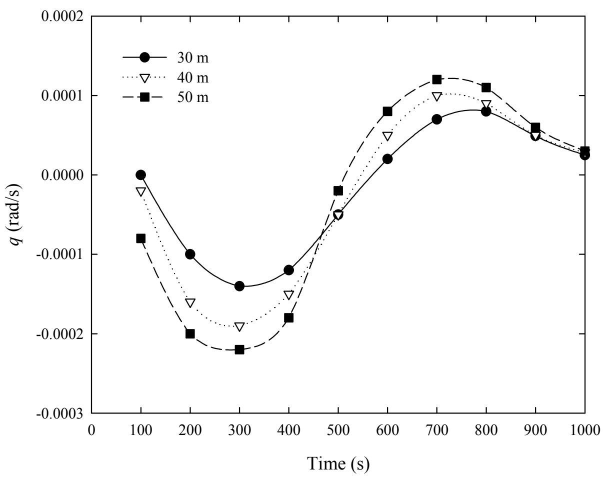


**Figure 7**: Case 1 - Variation of ship pitch rate

It is worth noting that the ascent velocity of a raising vessel is vital to the successful salvage operation. This entails a comparatively slow process (about 17 min in the present case). It is seen from Figure 5 that the maximum heave velocity (ascent velocity) increases with the target depth as does with the initial flow rate. The maximum value of the ascent velocity is found to be 0.08 m/s - being within the required range (< 0.6 m/s [5]). This implies how the pontoon motion is stable. When the vessel reaches the commanded depth, the controller reduces the ascent velocity to the nearly-zero value. The depth rate and ascent velocity responses reveal similar trend to the results of Nicholls-Lee et al. [2].

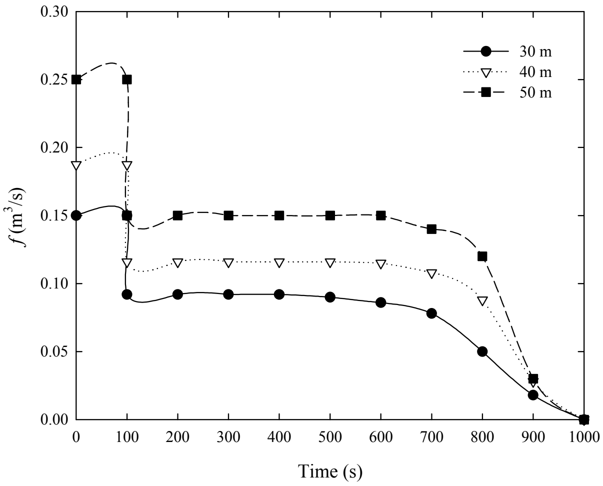


**Figure 8**: Case 1 - Variation of gas flow rate

From Figure 6, the pitch angle is found to increase with time, reaching a maximum value during a half of period and thereafter decreasing. The maximum value of the pitch angle is found to be about 4.2 deg. which is within the required limit (< 15 deg. [5]). Similar to the ascent velocity in Figure 5, the pitch angle in Figure 6 increases with the initial input flow rate. Nevertheless, the pitch angle decreases when the payload reaches the commanded depth due to the fact that the controller generates a pitch angle command as per the depth error. At the beginning, the depth error is large thereby the controller produces a high value of pitch angle to eliminate the error. Figure 7 shows how the pitch rates of all analysis depths become nearly equal to zero when the pontoon reaches the required positions.

In all three depth (30, 40 and 50 m) cases, it is seen from Figure 8 that soon after the breaking out period (at $t$ = 100 s) the sliding-mode controller instantly reduces the initial input flow rate value from 0.15, 0.1875 and 0.25 to 0.092, 0.116 and 0.15 m$^3$s$^{-1}$, respectively, in order to compensate the presence of excessive buoyancy induced by a sudden release of the pay load from the sea bottom. Thereafter, the flow rate of gas filling maintains a constant value until the vessel nearly reaches the target depth. Once the target depth is fulfilled, the controller further reduces the flow rate to almost the zero value. The variation in the buoyancy force with respect to the varying depth is compensated by the pressure relief valves. Thus, by the combined use of the sliding mode controller for regulating the flow rate of filling and pressure relief valves, a constant and stable ascent rate can be reasonably maintained.

4.2 CASE 2: FIXED INITIAL FLOW RATE

Now, the analysis is performed in the case of fixed initial flow rate (0.25 m$^3$s$^{-1}$) for different target depths. The simulation times required for the depths of 30, 40 and 50 m are 700, 800 and 1000 s, respectively. The obtained dynamic responses are displayed in Figures 9-13.

It can be seen from Figures 9-12 that the obtained responses are stable (based on the limiting values of ascent velocity and pitch angle). They are also quite similar to those illustrated in Figures 4-7. The maximum ascent velocity, pitch angle and pitch rate values seem to depend mainly on the initial input flow rate, irrespective of the simulation time. After the pontoon reaching the commanded target depth, even though the simulation time is variable, it has a slight effect on the response values because of the sliding-mode controller action. As a result, Figure 13 displays 3 different patterns – i.e. reduction, maintenance and further reduction to zero value – of the flow rate, as in Figure 8.

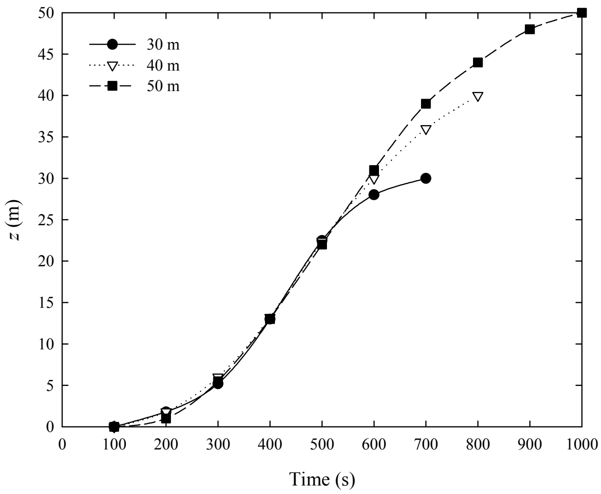


**Figure 9**: Case 2 - Variation of ship vertical position

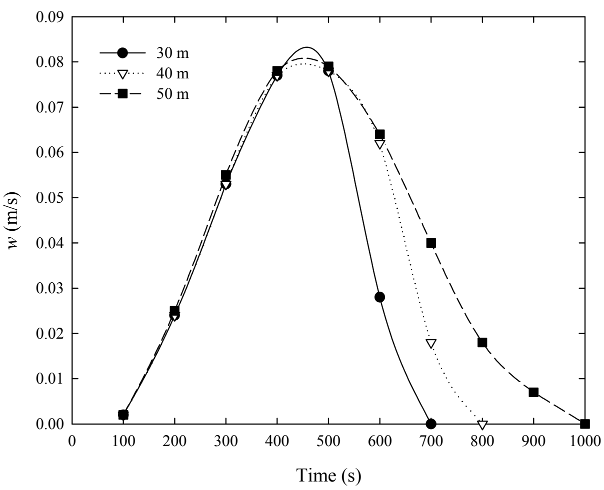


**Figure 10**: Case 2 - Variation of ship ascent velocity

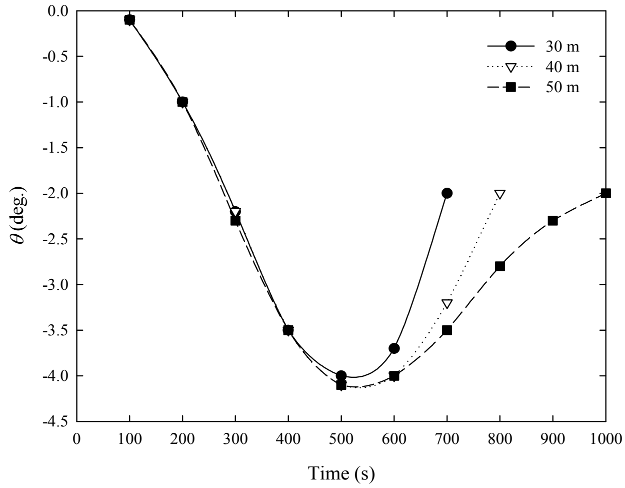


**Figure 11**: Case 2 - Variation of ship pitch angle

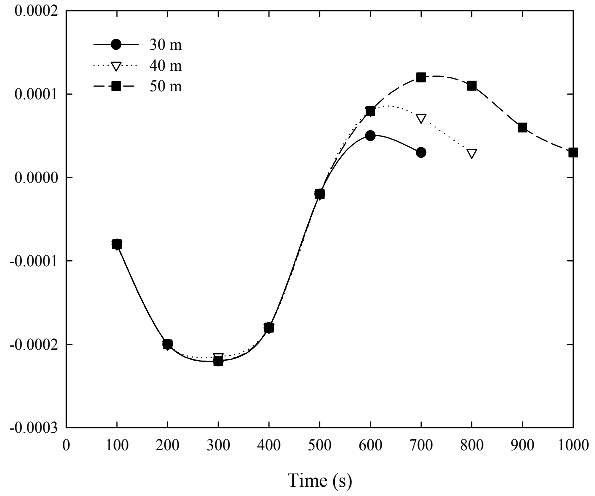


**Figure 12**: Case 2 - Variation of ship pitch rate

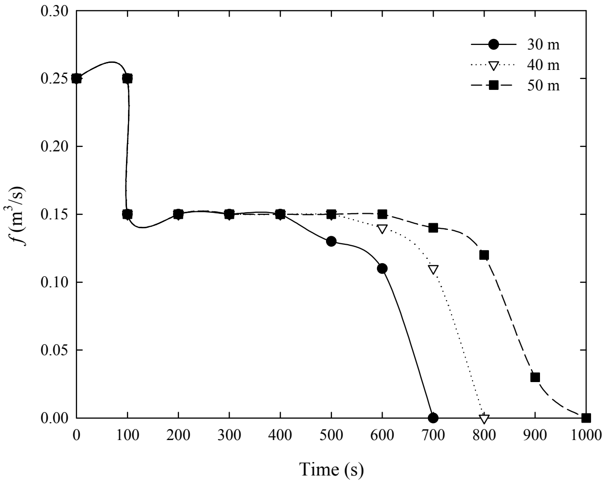


**Figure 13**: Case 2 - Variation of gas flow rate

It should be noted that the present preliminary model and numerical results are based on the rigid-body ship behaviour and several assumptions. This could be further improved by considering, for instance,

- the appropriate breakout force model in order to evaluate the suitable breakout time,
- the effect of surge degree of freedom: this would however lead to some nonlinear coupling terms which require a nonlinear control strategy such as a Fuzzy or Adaptive sliding-mode controller,
- the elastic beam model with free–free boundary conditions for the ship to account for the spatial variation of hydrodynamic forces, moments and pressures as well as ship displacements and velocities,
- the evaluation of shear forces and bending moments and corresponding fatigue damage thresholds.

## 5. CONCLUSIONS

A mathematical modelling and numerical time-domain approach to simulate the dynamics of a sunken vessel being raised from the seafloor by a gas inflating system have been presented. Some preliminary parametric studies have been carried out and the obtained numerical results highlight the following findings.

- The flow rate of filling gas inside the balloon is the key parameter determining the ship stability during the raising dynamics.
- For a safe and viable salvage operation, the ascent velocity and pitch angle should be controlled by the combined use of sliding mode controller and pressure relief valve.
- The sliding mode controller may be utilized to effectively maintain the hydrodynamic stability of the raising vessel in the diving plane during the salvage operation.

## 6. ACKNOWLEDGEMENTS

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Presentation

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