Flexible Body Analysis of Lifting a Sunken Chemical Tanker using Multiple Controlled Gas Inflating Bags

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Abstract: In this paper, a mathematical model and numerical time domain approach to simulate the dynamics of a sunken chemical tanker being raised from sea floor by multiple controlled gas inflating bags is presented based on the principles of flexible body modeling & control. In this regard, the vessel or payload is modeled as an Euler-Bernoulli beam with free – free boundary conditions. Free vibration analysis or eigen value analysis of the vessel is carried out in MATLAB using finite element method to obtain the natural frequencies (eigen values) and mode shapes (eigen vectors). From the mode shape plots, the buoyant systems or lift bags are located suitably on the "nodes of a mode" of the beam. Then the eigenvectors are normalized with respect to mass, and the equation of motion is developed in principal coordinates after defining the nodal forces and moments. Then the modal contributions of individual modes are analyzed according to their dc gain/peak gain value to define, which ones have greatest contribution and later several modal reduction techniques such as 'modred-mdc' and 'modred-del' are used to obtain the smallest state space model that represents the pertinent system dynamics. The full and reduced order modal responses are compared in both frequency and time domains. It is observed that 'unsorted modred-mdc' is the preferred choice for modal order reduction compared to the other reduction methods. Longitudinal distribution of shear force and bending moments across the tanker are also evaluated. Finally, a supervisory fuzzy logic controller is integrated with the flexible state space model of each lift bags to obtain the controlled stable responses.

Keywords: Marine Salvage, Buoyant Systems, Flexible Body Modeling & Control, Supervisory Fuzzy Logic Controller.

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1 Introduction

The concept of using buoyancy systems (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 weight of the object in air from the weight of the fluid displaced by that object (Farrell, 2008; Rawson & Tupper, 2001). In general, the bottoms of inflatable bags are attached to the payload to be lifted and inflated using pipes from the gas generating system. In salvage industry, there are mainly two types of lift bags are generally available for recovering sunken objects; one is parachute type and other is cylindrical type. Parachute type (external) bags are generally preferred for lifting purpose, whereas cylindrical type (internal) lift bags are used for providing stability (SMIT, 2010; Subsalve, 2010).

The main drawback of using the inflating bags for marine salvage operation is due to the difficulty in controlling the vertical speed and pitch motion as the ship ascends. Due to the suction break out force, a large buoyancy force may be initially required to separate the ship from the seabed, resulting in an excessive vertical speed and pitch angle after break out. During the ascent, any trapped air inside the hull may also expand and further increase the buoyancy. Also due to the pressure difference between gas inside the lift bag and surrounding sea water pressure in accordance with the decrease in water depth during the ascent, the gas inside the lift bag expands. All these factors lead to an increase in buoyancy force and hence result in an excessive vertical speed as well as pitch angle during the ascend. Excessive vertical speed results in a potentially-hazardous working environment to divers and salvaging crews and this may cause the lift bag to breach the surface of the water so fast that the air escapes from the bottom. High values of pitch angle cause the lift slings to break loose from payload and hence results to a further buoyancy loss. These all factors make 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 (Farrell, 2008; JW Automarine, 2010; SuSy, 2011).

Hence, in order to ensure hydrodynamic and structural stability during the ascent, it is required to design the following control systems; a primary controller for regulating the flow rate of filling gas inside the lift bags according to the buoyancy requirement with respect to hydrostatic force due to weight, buoyancy and suction breakout, hydrodynamic force and uncertainty arises due to external disturbances. A secondary controller to regulate the opening of a purge valve fitted on the lift bags in accordance with the excess buoyancy available after suction breakout and to the variation in pressure difference b/w gas inside the lift bags and surrounding seawater. Then a supervisory controller needs to be designed to monitor or switch between the primary and secondary controller according to the depth error and ascent rate.

In the rigid body modeling & control approach, the state space model is created by considering the total additional buoyancy provided by all lift bags together and the responses are available for the whole motion of the payload (Velayudhan et al., 2011 & Velayudhan et al., 2012). But in actual practice, lift bags are located

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at different locations on the vessel and their location significantly affects the hydrodynamic and control responses. Whilst rigid body modeling can be extended to include the response of individual lift bags and to control them separately, i.e. to use multiple controlled lift bags to ensure both hydrodynamic and structural stability, it cannot deal with more complicated lifts, such as a flexible pipeline. Although it continues to consider the case of ship salvage, this paper extends the theory to allow for this possibility. For meeting these objectives, the rigid body modeling & control approach is extended to a detailed flexible body modeling & control.

2 Problem Formulation

In the flexible body modeling & control approach, the vessel or payload is modeled as an Euler-Bernoulli beam with free – free boundary conditions as shown in Figure 1.

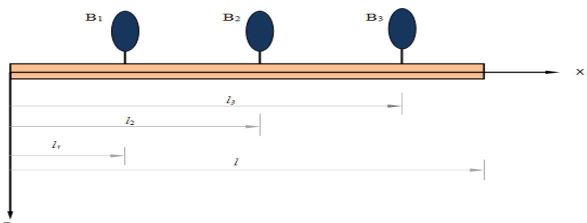

Fig.1: Concept of a beam model with lift bags for marine salvage

The Euler-Bernoulli equation for the transverse vibration of a beam within the *X-Z* plane is given by (Clough & Penzien, 1975),

$$\frac{\partial^2}{\partial x^2}\left[EI\frac{\partial^2 z}{\partial x^2}\right]+\rho A\left[\frac{\partial^2 z}{\partial t^2}\right]=F(x,t) \quad (1)$$

In which, $z\ (x,\ t)$ is the deflection at location x along the beam at time t, E –Young's modulus or modulus of elasticity of the beam material, I – Second moment of area of the beam cross section about the neutral axis, ρ is the density of the material and A is the cross sectional area of the beam.

$F(x,\ t)$ is the force excitation on the beam, which is a function of both space & time, can be represented as the summation of all the point forces (Sun et al., 2007),

$$F(x,t)=\sum_{i=1}^{r} f_i(x,t) \quad (2)$$

In which r is the total number of input forces. $fi(x,\ t)$ is the set of point forces $fi(t)$ located at $x=l_i$, which can be expressed as a distributed force per unit length according to:

$$f_i(x,t)=F_i(t)\delta(x-l_i) \quad (3)$$

Where $\delta(x\text{-}l_i)$ is the Dirac Delta function for i = 1, 2….r and $Fi(t)$ is the universally distributed force per unit length.

For the beam with free-free boundary conditions, the shear force and bending moment at the two ends are zero (Clough & Penzien, 1975),

i.e.

$$\left.\frac{\partial^2 z(x,t)}{\partial x^2}\right|_{x=0}=0,\qquad \left.\frac{\partial^2 z(x,t)}{\partial x^2}\right|_{x=l}=0, \quad (4)$$

$$\left.\frac{\partial^3 z(x,t)}{\partial x^3}\right|_{x=0}=0,\qquad \left.\frac{\partial^3 z(x,t)}{\partial x^3}\right|_{x=l}=0, \quad (5)$$

In Euler-Bernoulli beam theory, the assumption is plane cross section of the beam remains plane and normal to the neutral axis before and after the bending. i.e. Euler-Bernoulli beam theory neglects rotational inertia and deformation due to shear forces.

2.1 Finite Element Solution

For a beam with free-free boundary conditions, the displacement $z(x,\ t)$ can be represented as (Huo et al, 2004),

$$z=z_0 e^{-i\omega t} \quad (6)$$

Where z_0 is the amplitude and ω is the natural frequency of vibration.

Therefore,

$$\dot{z}=z_0 e^{-i\omega t}\times -i\omega \quad (7)$$

$$\ddot{z}=z_0 e^{-i\omega t}\times -i\omega\times -i\omega=-\omega^2 z \quad (8)$$

Substituting Eq. (8) in Eq. (1), the eigenvalue problem is obtained as,

$$\frac{\partial^2}{\partial x^2}\left[EI\frac{\partial^2 z}{\partial x^2}\right]-\rho A\omega^2 z=0 \quad (9)$$

The above eigenvalue problem in differential form can be converted to finite element formulation as (Bathe, 1996; Huo et al, 2004; Manning et al, 2000):

$$-\omega_i^2[M]\{z_i\}+[K]\{z_i\}=0 \quad (10)$$

Where,

ω_i is the eigenvalue or natural frequencies of the beam and z_i is the eigenvector or mode shape of the beam, which can be obtained from eigenvalue analysis in MATLAB. $[M]$ - Mass matrix of the system, $[K]$ - Stiffness matrix of the system, which are calculated based on finite element principles as (Chhabra et al., 2011; Huo et al, 2004).

$$[K]_e=\frac{EI_e}{l_e^3}\begin{bmatrix}12 & 6l_e & -12 & 6l_e\\ 6l_e & 4l_e^2 & -6l_e & 2l_e^2\\ -12 & -6l_e & 12 & -6l_e\\ 6l_e & 2l_e^2 & -6l_e & 4l_e^2\end{bmatrix} \quad (11)$$

$$[M]_e = \frac{\rho A_e l_e}{420}\begin{bmatrix} 156 & 22l_e & 54 & -13l_e \\ 22l_e & 4l_e^2 & 13l_e & -3l_e^2 \\ 54 & 13l_e & 156 & -22l_e \\ -13l_e & -3l_e^2 & -22l_e & 4l_e^2 \end{bmatrix} \quad (12)$$

2.2 Development of State Space Model

Within the controller it is not possible to integrate the coupled motions equation of motion, so it is required to convert from physical to principal coordinates. For that, the eigenvectors obtained using FEM are normalized with respect to mass and the equation of motion is developed in principal coordinates as (Hatch, 2001):

$$\ddot{z}_{pi}(t) + 2\zeta_i\omega_i\dot{z}_{pi}(t) + \omega_i^2 z_{pi}(t) = F_{pi}(t) \quad (13)$$

Where $z_p(t)$ is the vector of amplitudes of different vibration modes in principal coordinates, ζ is the critical damping, $F_p(t)$ is the force in principal coordinates and i denote the number of modes.

Suppose, if we are considering only the first three modes, then the equations of motion in principal coordinates can be written in a state space form as (Hatch, 2001; Huo et al, 2004):

$$\begin{bmatrix} \dot{z}_1 \\ \dot{z}_2 \\ \dot{z}_3 \\ \dot{z}_4 \\ \dot{z}_5 \\ \dot{z}_6 \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 & 0 & 0 & 0 \\ -\omega_1^2 & -2\xi_1\omega_1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & -\omega_2^2 & -2\xi_2\omega_2 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \\ 0 & 0 & 0 & 0 & -\omega_3^2 & -2\xi_3\omega_3 \end{bmatrix} \begin{bmatrix} z_1 \\ z_2 \\ z_3 \\ z_4 \\ z_5 \\ z_6 \end{bmatrix} + \begin{bmatrix} 0 \\ F_{p1} \\ 0 \\ F_{p2} \\ 0 \\ F_{p3} \end{bmatrix} u_c \quad (14)$$

This is the form $\dot{x} = Ax + Bu$, where the system matrix A is made up of each eigenvalue and damping for each mode and input matrix B is made up of applied force at the nodes.

2.3 Model Reduction Techniques

The vessel structure or beam can vibrate with many modes. In the development of flexible state space model, n uncoupled second order differential equations are converted into $2n$ first order differential equations. Therefore the size of system matrix is twice the number of modes to be included in the analysis. Hence if large number of modes are included in the analysis, the process become cumbersome and leads to high computational time. However all the modes do not contribute significantly to the overall responses. Hence it is sensible to use the important modes, that cause the maximum disturbances, and to develop a state space model that includes the most significant modes in the analysis at the same time it might account for the effect of eliminated modes in the remaining system. The objective of the modal reduction technique is to provide the smallest state space model that accurately represents the flexible system dynamics (Hatch, 2001; Huo et al, 2004; Khot et al., 2011; Khot & Yelve, 2011).

2.3.1 Sorting of Modes by dc gain approach

The common method for reducing the modal size is to simply truncate the higher frequency modes. This kind of elimination is not valid for all cases and leads to less desired accuracy. Therefore, accurate reduced models can be obtained by sorting the modes based on their individual contribution to the overall response and keeping only important modes. This can be carried out in terms of 'dc gain', which can be defined for i^{th} mode in the form of transfer function as (Hatch, 2001; Huo et al, 2004; Karagulle et al., 2004):

$$\frac{z_{ji}}{F_{ki}} = \frac{z_{nji} z_{nki}}{\omega_i^2} \quad (15)$$

Where, $z_{nji} z_{nki}$ are the product of the j^{th} (output) row and k^{th} (input or force applied) row terms of i^{th} eigenvector.

2.3.2 Modred method

Though reduced models can obtained effectively by sorting of modes based on their dc gain value, still there is an error introduced due to neglecting the contribution of eliminated modes in the overall dc gain. The MATLAB function "modred" (MODel order REDuction) is introduced to eliminate this error, which is based on the assumptions that some modes being more important than other (Hatch, 2001; Huo et al, 2004). This allows reducing size of the problem to that of the 'important modes'. The Modred function has two options or sub functions; the 'mdc' (Matched DC gain) option reduces defined states by setting the derivatives of the state to be eliminated to zero and then solving for the remained states, which is analogous to Guyan reduction in that the low frequency effects of the eliminated states are included in the remaining states. The other option 'del' simply eliminates the defined states, typically associated with the higher frequency modes (Hatch, 2001).

3 Modal Analysis Responses

For carrying out the flexible body analysis and control, a Chemical Tanker is suitably taken as shown in Figure 2. Geometric particulars of the chemical tanker are given in Table 1. Initially modal analysis of the chemical tanker is performed, without the controller, to obtain the free vibration analysis and forced vibration analysis responses and supervisory fuzzy logic controller is integrated later to obtain the controlled stable responses.

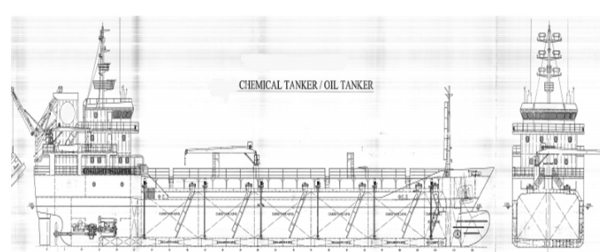

Fig.2: Chemical Tanker model (SuSy, 2010)

Table 1 Geometric Particulars of the chemical tanker

Main Dimensions	
LOA	78.02 m
LBP	72.40 m
Breadth	12 m
Depth	5.4 m
Displacement	3200

3.1 Free Vibration Analysis

A two dimensional Euler - Bernoulli beam model with free - free boundary conditions is developed in MATLAB. Eigenvalue analysis is carried out by finite element method using different number of elements, a convergence check is also carried out and it is found that an 11 element beam model is the preferred one.

The mode shapes of the chemical tanker obtained from finite element free vibration analysis are shown in Figures 3-9, in which the first two modes are rigid body modes (heave and pitch) corresponding to zero frequency and rest of them are flexible modes. Note that, the eigenvectors are normalized with respect to unity for plotting. Figure 10 shows the resonant frequency corresponding to each mode.

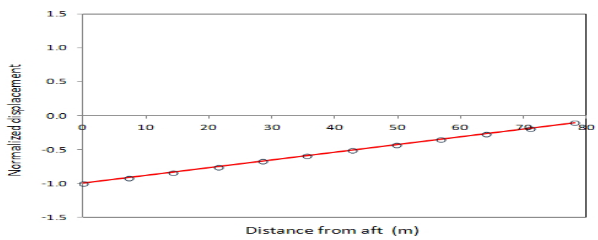

Fig.3: First rigid body (heave) response, 0 Hz

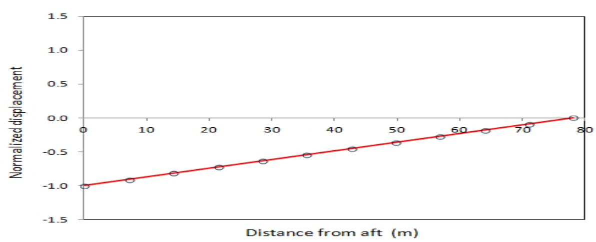

Fig.4: Second rigid body (pitch) response, 0 Hz

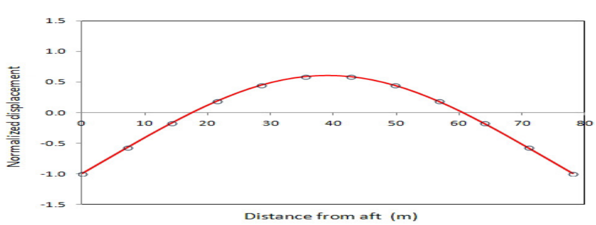

Fig.5: First flexible body response, 5 Hz

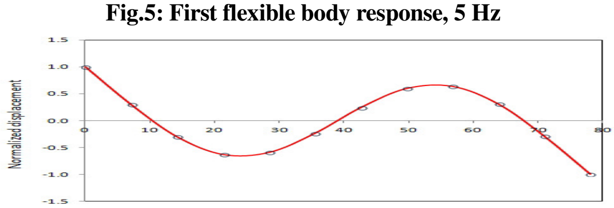

Fig.6: Second flexible body response, 13 Hz

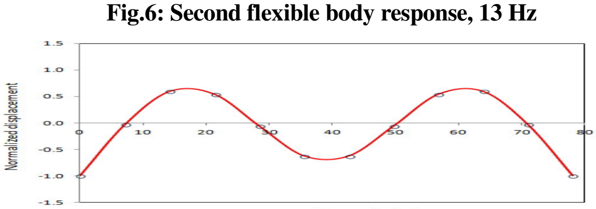

Fig.7: Third flexible body response, 25 Hz

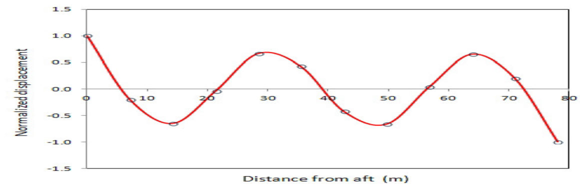

Fig.8: Fourth flexible body response, 42 Hz

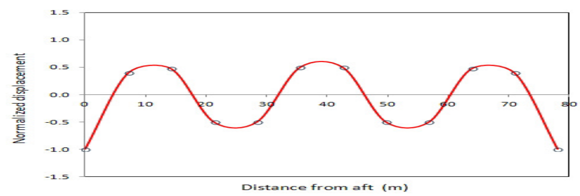

Fig.9: Fifth flexible body response, 63 Hz

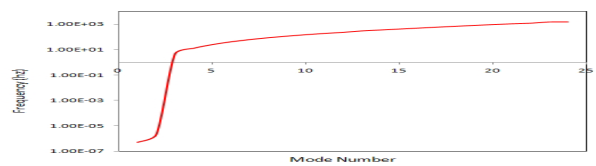

Fig.10: Resonant frequency versus mode number

3.2 Forced Vibration Analysis

In order to carry out the forced vibration analysis of raising sunken chemical tanker, primarily it is required to carry out the force modeling, i.e. to define the nodal force and moments. For the chemical tanker salvage using buoyant systems, the major forces to be considered are hydrostatic force due to weight and buoyancy, suction break out force and the additional buoyancy provided by the inflating system as explained in Velayudhan et al. (2011) & Velayudhan et al. (2012). Total lift force required to extract the sunken tanker from sea bottom is estimated to be 1.3 times the wet weight (*W-B*), which is equals to 3616.815 tonnes.

Suppose if we are using lift bags (Ever Safe, 2012) with 1.5m diameter and 18 m length that can displace 565 tonnes with 2.8 bar working pressure.

Therefore number of lift bags required
= total lift force required/ lifting capacity of a lift bag
= 3616.815/565=6.40~ 7

Considering buoyancy losses, total 7 lift bags are required for inflating. Therefore additional buoyancy provided by lift bags= 7*565=3955 tonne, which is much higher than the total lift force required. Therefore for mathematical calculations, we are redesigning lift bags having lift capacity 517 tonnes each so that in total they can offer 517*7=3619 tonne, which is equal to the lift force required.

Thus we are attaching two vertical parachute type lift bags externally on each side of the tanker and three cylindrical bags (internal) horizontally in the ballast tank to maintain hydrodynamic stability as shown in Figure11. The positions of external lift bags can be located according to the eigenvalue or free vibration analysis of the chemical tanker. The lift bags can be attached at the "node of a mode", the point where the displacement is negligible.

As we need to fix the four external bags, it is relevant to consider the third flexible mode response as plotted in Figure 7. From the response, it is seen that there are four “node of modes”, which are at nodes 2, 5, 8 and 11. Therefore, the external lift bags can be fixed at node 2, 5, 8 & 11 respectively. Internal lift bags can be placed suitably on the ballast tanks of the tanker.

From the chemical tanker GA (Figure 2), it is found that ballast tanks are situated at frame 27 to frame 113 and one longitudinal frame spacing is 0.57 m.

i.e. 27th frame= 27*0.57=15.39 m from aft.
113th frame = 113*0.57=64.41 m from aft.

Therefore, 3 internal lift bags can be placed horizontally between the length 15.39 m and 64.41 m from the aft end of the vessel as shown in Figure 12. In order to incorporate this in to the beam model, suitably internal lift bags are placed between node 3 and node 10 for a length of 49.645m.

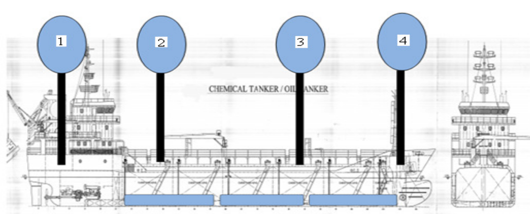

Fig.11: Arrangement of lift bags for the chemical tanker salvage

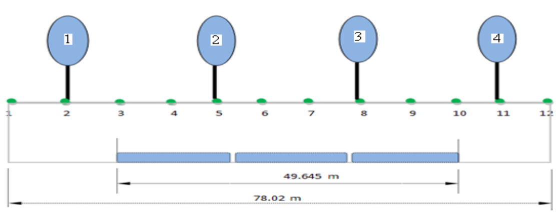

Fig.12: Arrangement of lift bags on beam nodes

According to Velayudhan et al. (2012), hydrostatic force acting downwards due to weight, buoyancy and suction breakout is 1.3*(W-B)*, which is assumed to be uniformly distributed along the vessel length. Therefore, 1.3*(W-B)/l* is the net hydrostatic force acting downwards at a particular beam node. Hydrodynamic force can be computed from Eqs. (6) & (7) of Velayudhan et al. (2012), which are also considered as uniformly distributed along the vessel length. Additional buoyancy provided by external or parachute type lift bags are taken as point loads where as buoyancy provided by internal or cylindrical type lift bags are treated as universally distributed load (UDL).

Nodal force and moments across the vessel length are computed according to classical strength of material’s principles (Rawson & Tupper, 2001) and input in the MATLAB program for obtaining the forced vibration analysis results. Shear force and bending moment at any point on the vessel length are also estimated and its longitudinal distribution across the beam length is plotted in Figures 13 & 14 respectively. The maximum value of shear force is 3.51MN, which is at node 8 (i.e. at lift bag 3) and the bending moment is 44.16 MN.m at node 5 (i.e. at lift bag 2).

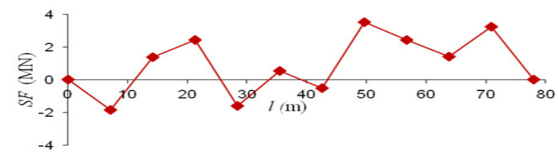

Fig.13: Longitudinal distribution of shear force

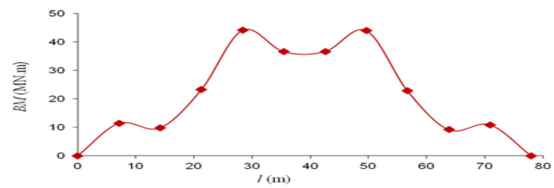

Fig.14: Longitudinal distribution of bending moment

3.2.1 Forced Vibration Analysis Responses

The first step in any modal analysis is to understand the resonant frequencies of the model. From Figure 15, it is found that the resonant frequencies of the tanker are within the order 10^0 to 10^4. As we are using 11 element beam, total 22 modes are considered for the analysis, out of which the first two modes are rigid body modes corresponding to 0 Hz frequency and rest are flexible modes. Now the sorting of modes are carried out by their ‘dc gain’ value to measure the individual modal contribution to the overall response. Figure 16 shows the low frequency gain for the first two rigid body modes (mode 1 & mode 2) and dc gains for all other modes versus mode number. The modes are sorted according to their dc gain in the order as:

1	2	3	4	6	8	7	9	11
12	10	14	18	13	20	19	22	17
16	21	15	5					

As expected, it is observed that, for this problem, the first two rigid body modes are more important than the flexible modes.

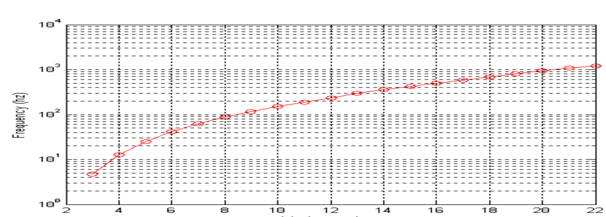

Fig.15: Resonant frequency versus mode number

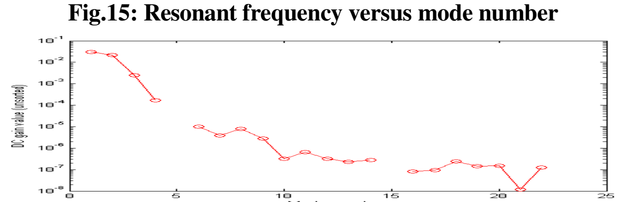

Fig.16: Unsorted dc gain value of each mode versus mode number

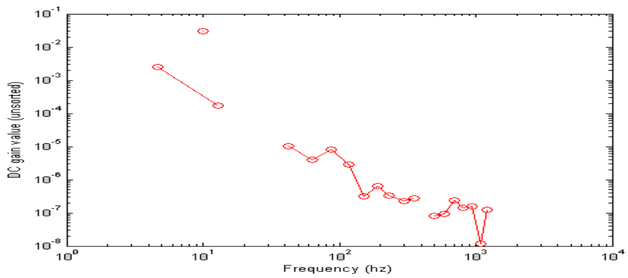

Fig.17: Unsorted dc gain value of each mode versus resonant frequency

Unsorted dc gain value of each mode versus its resonant frequency is shown in Figure 17. It is noted that as the frequency increases, the general trend is to have modes with lower dc gain values. Highest dc gain is obviously for the first two rigid modes.

Now modes are sorted according to their dc gain value and allocate index numbers from higher dc gain modes to lower dc gain modes as shown in Figure 18.

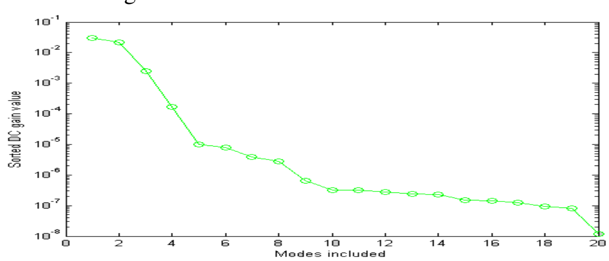

Fig.18: sorted dc value of each mode vs. number of modes

It should be noted that for hull bending effects the gain associated with hull curvature will be important and this should be taken into account when deciding frequency cut-off values.

3.2.1.1 Full and Reduced Model Responses

As controller needs to integrate with the smallest state space model, which accurately represents the flexible body dynamics, the responses obtained from various model reduction techniques using either sorted or unsorted modes with full model responses are compared in both frequency and time domain. Finally state space models are developed using the optimum modal reduction technique.

Table 2 shows the comparative performance of various reduction methods used for modal order reduction.

Table 2: Reduction methods summary

Reduction method	Dc gain error %	Peak error %
Unsorted	+0.0796	-0.0023
Sorted	-0.3429	0.0109
Unsorted Modred-del	+0.0796	-0.0023
Unsorted Modred-mdc	-0.00087794	0.000072472
Sorted Modred-del	+0.0624	0.0109
Sorted Modred-mdc	-0.0064	0.00085478

From Table 2, it is found that unsorted reduced models show better performance compared to sorted reduced models. This is due to the reason that low frequency modes are more significant compared to high frequency modes for the marine salvage problem. The overall transient response of the system is matched well by the 'mdc' option while the 'del' option has slight error, which might be due to the reason that 'modred-mdc' option minimizes the low frequency errors by including the contribution of the unused modes while 'modred-del' option does not account for the dc gains of the eliminated modes in the reduced system. Thus it is observed that 'modred-mdc' is the preferred method for model order reduction in which, '*unsorted modred-mdc*' option is the optimum choice for marine salvage.

4 Integrating the Controller

For maintaining hydrodynamic stability in a salvage operation using buoyant systems, a SIFSMC is the optimum choice as the primary controller for regulating the flow rate of filling gas inside the lift bags as explained in Velayudhan et al. (2012) and a PID controller is chosen as the secondary controller to regulate the purging of gas through the valves fitted on the lift bags as described in Farrell (2008). Now for a safe and stable salvage operation, it is required to monitor or switch between these two sub controllers by a supervisory controller as per the depth error and depth rate. In such situations, the only possibility is to choose an intelligent controller such as fuzzy logic controller (FLC) as the supervisory controller for monitoring the primary and secondary controllers as shown in Figure 19.

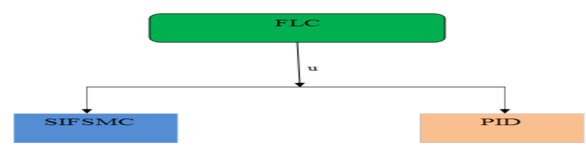

Fig.19: Design of a Fuzzy logic controller for monitoring two control subsystems

Based on the experience learned while conducting numerical simulations on primary and secondary controllers, a supervisory fuzzy logic controller is designed by utilizing MATLAB Fuzzy Logic toolbox and integrated in SIMULINK as shown in Figure 20. Here inputs to the FLC are the depth error (z_e) and depth rate (w). Depth error is defined as the commanded depth minus the measured depth. The output or control variable is 'u' which regulates the buoyancy with respect to the depth error and depth rate. After carrying out the stability check using different kinds of membership functions, Gaussian membership functions are finally used for representing the input and output variables as shown in Figures 21-22. Using a trial and error approach, the best inference mechanism to use in this case seems to be the prod-probor method. Because of simplicity and availability of the graphical user interface (GUI) in MATLAB, the Mamdani inference engine (Palm et al., 1997) is employed for designing the FLC that uses the minimum operator for a fuzzy implication and max-min operator for composition. The defuzzification technique used is found using a trial and error and centroid method is the one which provides least integral square error. Table 3 shows fuzzy rule base consists of 49 rules for computing the output variable, which are formulated based on the author's experience in performing numerical simulation using depth error and depth rate as the system states and change in volume of gas inside the lift bag as the output. The definition of fuzzy control actions are defined in Table 4.

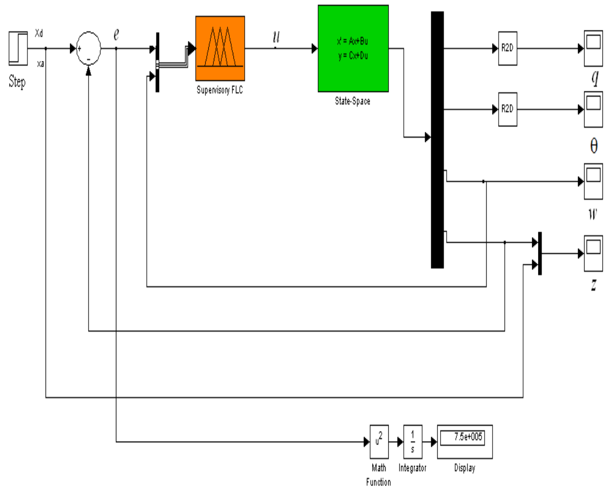

Fig.20: SIMULINK block diagram of a supervisory FLC for marine salvage

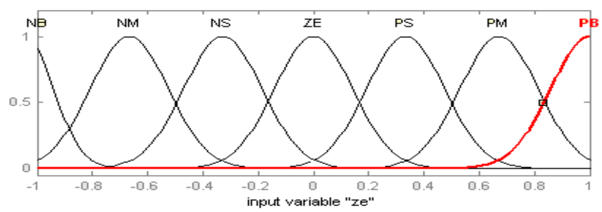

Fig.21 (a): Membership functions for the input variable 'z_e'

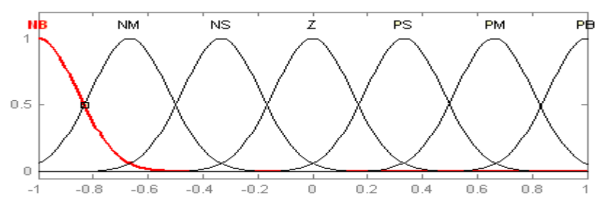

Fig.21 (b): Membership functions for the input variable w

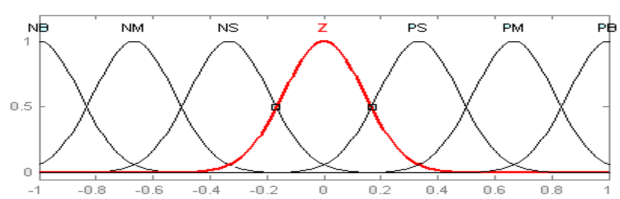

Fig.22: Membership functions for the output variable 'u'

Table 3: Two dimensional fuzzy rules to compute u

z_e	w						
	NB	**NM**	**NS**	**ZE**	**PS**	**PM**	**PB**
PB	**Z**	**PS**	**PM**	**PB**	**PB**	**PB**	**PB**
PM	**NS**	**Z**	**PS**	**PM**	**PB**	**PB**	**PB**
PS	**NM**	**NS**	**Z**	**PS**	**PM**	**PB**	**PB**
ZE	**NB**	**NM**	**NS**	**Z**	**PS**	**PM**	**PB**
NS	**NB**	**NB**	**NM**	**NS**	**Z**	**PS**	**PM**
NM	**NB**	**NB**	**NB**	**NM**	**NS**	**Z**	**PS**
NB	**NB**	**NB**	**NB**	**NB**	**NM**	**NS**	**Z**

Table 4: Definition of fuzzy output control action

Output 'u'	Meaning	Control Action
Z	Zero	Both Primary and Secondary controllers are off
PS	Positive Small	Small rate of filling gas in to the lift bag : operating primary controller
PM	Positive Medium	Medium rate of filling gas in to the lift bag : operating primary controller
PB	Positive Big	Large rate of filling gas in to the lift bag : operating primary controller
NS	Negative Small	Small purging of gas from lift bag: operating secondary controller
NM	Negative Medium	Medium purging of gas from lift bag : operating secondary controller
NB	Negative Big	Large rate of purging gas from lift bag: operating secondary controller.

The variation of control action 'u' with respect to the depth error (z_e) and depth rate (w) is shown in Figure 23. Positive value of u implies filling gas inside the lift bags, where as negative value implies taking gas or purging gas out from the bags. Thus by the combined action of filling gas in to the lift bag and by regulating the purging of gas through the valves in accordance with the depth error and its derivative, a stable ascent can be ensured.

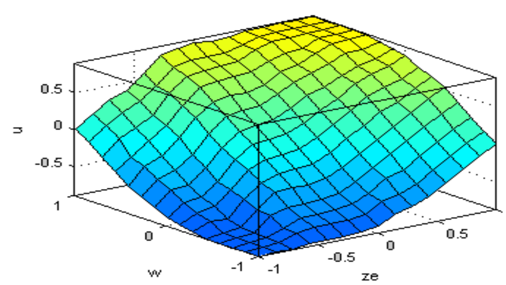

Fig.23: Variation of control action with depth error and depth rate

Now the controlled response of the chemical tanker is trying to obtain using the supervisory fuzzy logic controller. By flexible body modeling approach the state space model is available for individual nodes on the beam. Thus controlled response of individual lift bags can be simulated. This is the advantage of flexible body modeling & control over rigid body modeling & control. The reduced state space model (4*4) obtained using the optimum model order reduction technique '*unsorted modred-mdc*' is used for carrying out the simulation in SIMULINK. Supervisory fuzzy logic controller is integrated to state space models corresponding to each lift bags separately to get the individual controlled responses.

The total lift force required for breakout is 1.3 times wet weight and obtained as 3616.815 tonne (Velayudhan et al., 2012) and the break out time for the estimated force is found to be 400 s (Foda, 1982; Mei et al., 1985). The controlled responses and variation of control parameter obtained for a target depth of 300 m from sea bottom is plotted for the four external lift bags separately as shown in Figures 24-43.

Lift bag 1

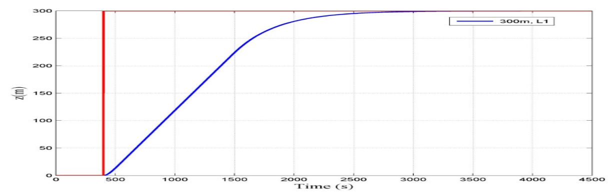

Fig.24: Variation of vertical position of lift bag 1 from sea bottom

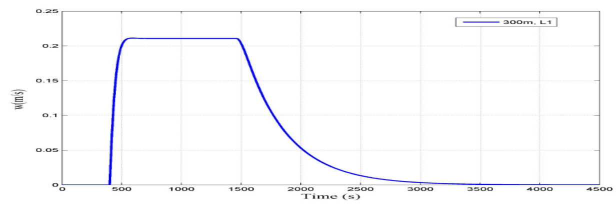

Fig.25: Variation of ascent velocity of lift bag 1

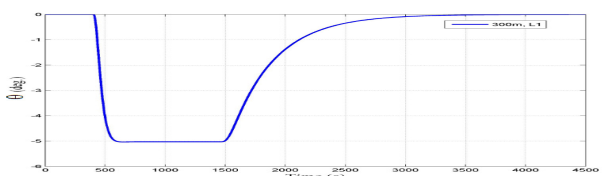

Fig.26: Variation of pitch angle of lift bag 1

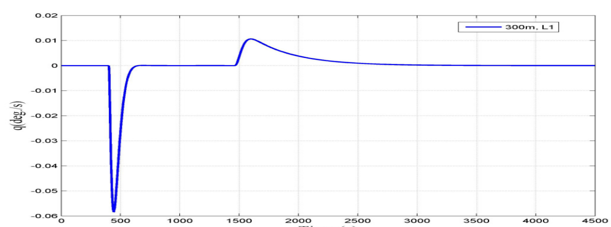

Fig.27: Variation of pitch rate of lift bag 1

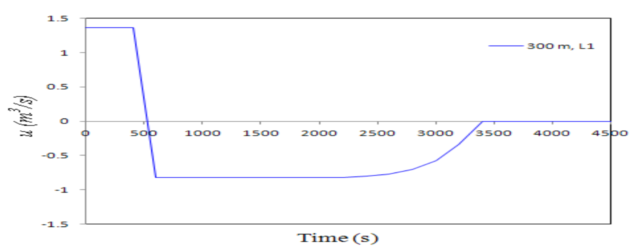

Fig.28: Net flow rate (at local pressure) in and out of lift bag 1

Lift bag 2

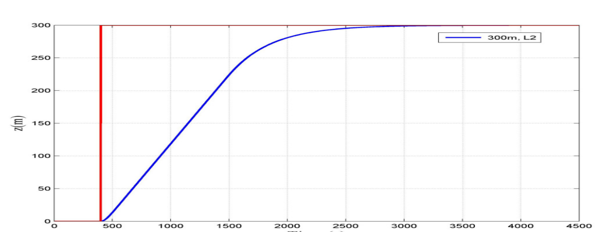

Fig.29: Variation of vertical position of lift bag 2 from sea bottom

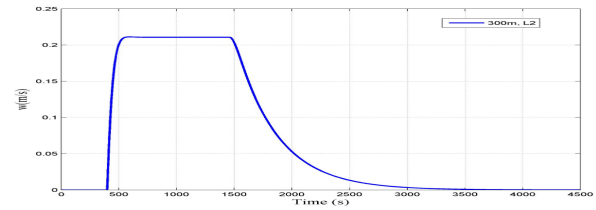

Fig.30: Variation of ascent velocity of lift bag 2

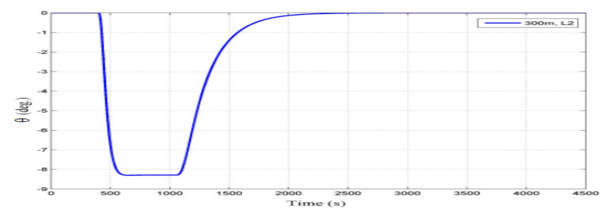

Fig.31: Variation of pitch angle of lift bag 2

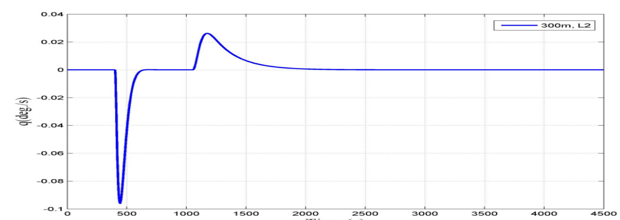

Fig.32: Variation of pitch rate of lift bag 2

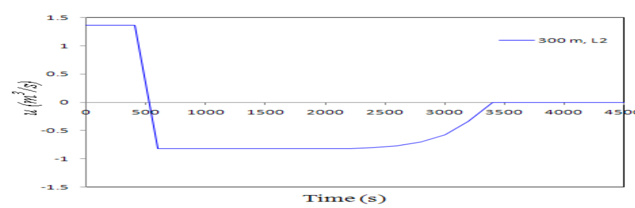

Fig.33: Net flow rate (at local pressure) in and out of lift bag 2

Lift bag 3

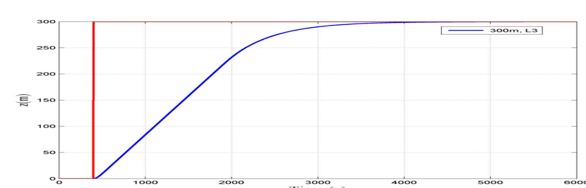

Fig.34: Variation of vertical position of lift bag 3 from sea bottom

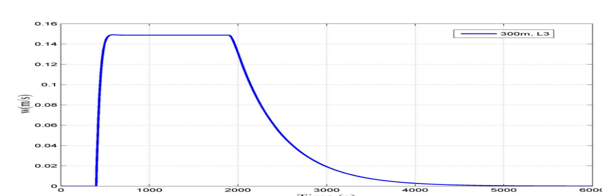

Fig.35: Variation of ascent velocity of lift bag 3

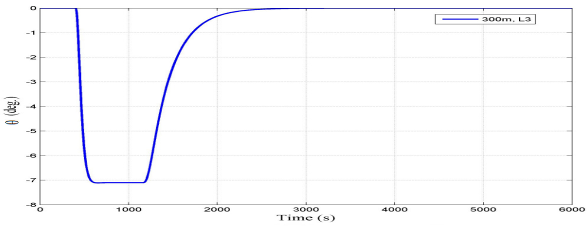

Fig.36: Variation of pitch angle of lift bag 3

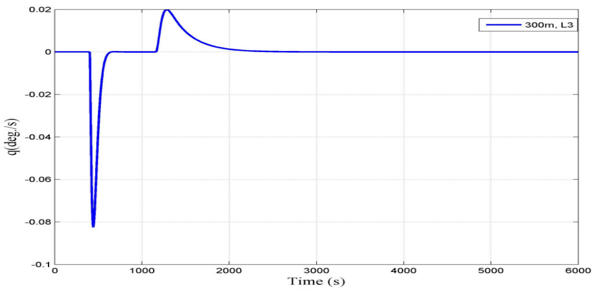

Fig.37: Variation of pitch rate of lift bag 3

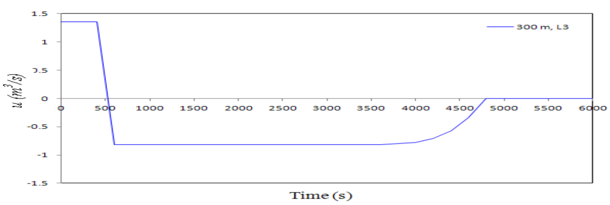

Fig.38: Net flow rate (at local pressure) in and out of lift bag 3

<u>Lift bag 4</u>

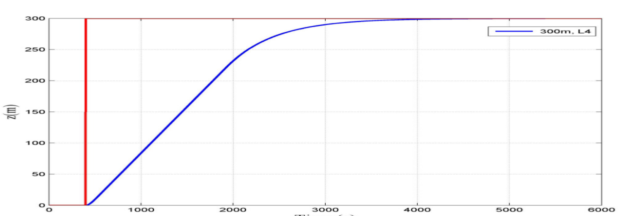

Fig.39: Variation of vertical position of lift bag 4 from sea bottom

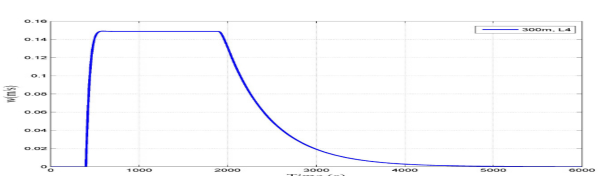

Fig.40: Variation of ascent velocity of lift bag 4

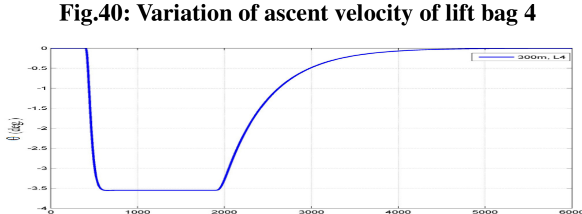

Fig.41: Variation of pitch angle of lift bag 4

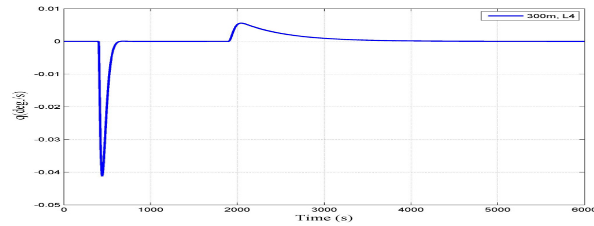

Fig.42: Variation of pitch rate of lift bag 4

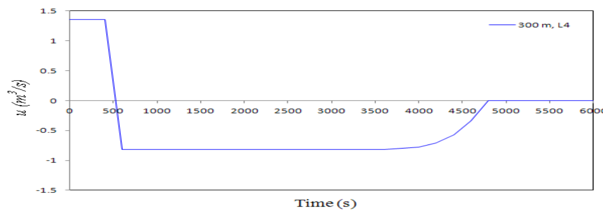

Fig.43: Net flow rate (at local pressure) in and out of lift bag 4

From the response plots (Figures 24-43), it is seen that all the four external lift bags have steady state behavior by integrating the controller. I.e. the heave velocity of lift bags initially increases after breakout and reaches a higher stable value and then decreases to zero when the tanker reaches the target depth. Pitch response of the tanker follows the same trend as the ascent velocity curve. It is also noted that by integrating the controller, fluctuating pitch motions of lift bags avoided, hence stability ensured. From Figure 24, 29, 34 & 39, lift bag 1 and lift bag 2 reaches the target depth in 2800s, where as lift bag 3 and lift bag 4 takes 4000 s to achieve the target depth. Therefore the aft part of the tanker rises faster than bow part. This is mainly because controller is integrated to each lift bag separately and there is no communication between the lift bags. This is one of the drawbacks of the present model. The maximum value of ascent velocity among the four lift bags is found to be 0.21 m/s < 0.6 m/s, which implies that the ascent is stable. The maximum value of pitch angle for lift bags (see Figure 31) is found to be about 8.2 degrees, which is within the required limit (<15 degrees). Pitch rates of the lift bags become nearly equal to zero when the bags reach the target depth. Figures 28, 33, 38 & 43 shows that the control action for lift bag 1 & lift bag 2 are identical, while lift bag 3 and lift bag 4 are having the same control output. This is due to the reason that state space models of lift bag 1 and lift bag 2 are almost same, whereas lift bag 3 and lift bag 4 are having the same state space model. It is also due to the reason that there is no communication between the lift bags.

Even though the system is stable by using multiple controlled lift bags, for more uniform lifting of the vessel, it is required to build an integrated network of control system, in which there is a master controller which gives commands to all the lift bags in order to attain a uniform ascent. For that sensors need to provide at individual lift bags and there should be proper communication between a master controller and the various subsidiary lift bags by a network. Purge valves in the individual lift bags should also be controlled by another network integrated with pressure sensors connected to each lift bag. Such a control system can be integrated with our flexible beam modeling & control approach. This could be extended to model and control the salvage, or installation of long flexible structures such as pipes.

5 Conclusions

This paper presented a mathematical model and numerical time domain approach to simulate the dynamics of a sunken chemical tanker being raised from sea floor by multiple controlled lift bags based on the principles of flexible body modeling & control. Initially modal analysis of the chemical tanker is performed without controller to obtain the free vibration and forced vibration analysis responses and supervisory fuzzy logic controller is integrated later with the state space model of individual lift bags to obtain the controlled stable responses. The longitudinal distribution of shear force and bending moment across the vessel length is estimated and the maximum value of shear force is found to be 3.51MN, which is at node 8 (i.e. at lift bag 3) and the bending moment is 44.16 MN.m at node 5 (i.e. at lift bag 2). The modal contributions of individual modes are analyzed according to their dc gain value and highest dc gain is obtained for the first two rigid modes, which implies that rigid body modes are more significant compared to flexible modes for marine salvage. Finally the effectiveness of various modal reduction techniques are investigated in both frequency and time domain to obtain the smallest state space model (4*4) that accurately represents the pertinent flexible body dynamics and 'unsorted modred-mdc' is found to be the optimum choice for modal order reduction as it minimizes the low frequency errors by including the contribution of the unused modes in the reduced model. From the modal analysis results, the heave response of the tanker is found to be increasing with time, whereas pitch motion is seen to be fluctuating with time. Therefore, in order to maintain hydrodynamic stability, it is necessary to integrate a control system with the model. Using the flexible body modeling approach the state space model is available for individual nodes on the beam. Thus the controlled response of individual lift bags can be simulated. This is the advantage of flexible body modeling & control over rigid body modeling & control. Hence supervisory fuzzy logic controller is integrated with the 4*4 flexible state space model obtained using '*unsorted modred-mdc*' method to obtain the controlled stable responses of each external lift bags. From the response plots, it is seen that all the four external lift bags have steady state behavior. Lift bag 1 and lift bag 2 reach the target depth in 2800s, whereas lift bag 3 and lift bag 4 take 4000 s to achieve the target depth. Therefore the aft part of the tanker rises faster than bow part. This is mainly because controller is integrated to each lift bag separately and there is no communication between the lift bags. This is one of the drawbacks of the present control system. Even though the system is stable in the application of using multiple controlled lift bags, it is required to build an integrated network of controllers, in which there is a master controller which gives commands to subsidiary lift bags for attaining a more uniform ascent.

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References

Bathe KJ (1996). *Finite Element Procedures*, Prentice Hall, Upper saddle River, New Jersey. (textbook)

Chhabra D, Chandna P & Bhushan G (2011). Design and Analysis of Smart Structures for Active Vibration Control using Piezo-Crystals, *International Journal of Engineering and Technology*, Volume 1, No.3. (journal)

Clough RW, Penzien J (1975). *Dynamics of Structures*, McGraw-Hill Kogakusha Limited, New Delhi. (textbook)

Ever Safe (2012). Salvage Catalog, Retrieved on 10th July 2011, www.eversafe-marine.com. (catalog)

Farrell JE (2008). *Automated Purge Valve*, M.S.Thesis, Florida Institute of Technology, Florida. (degree thesis)

Foda MA (1982). On the Extrication of large objects from the ocean bottom (the breakout phenomenon), *Journal of Fluid Mechanics*, Vol.117, 211-231. (journal)

Hatch MR (2001). *Vibration Simulation Using MATLAB and ANSYS*. Chapman & Hall/CRC, Boca Raton, Florida, ISBN 1-58488-205-0. (textbook)

Huo S, Short SR, and Xue X (2004). FEM Based Modal Analysis of a Damaged Free-Free Beam. *SEM X International Congress & Exposition on Experimental Applied Mechanics*, Costa Mesa, California, USA. (conference)

J W Automarine (2010). The theory of lifting bags. Retrieved on 4th October 2010, www.jwautomarine.co.uk. (catalog)

Karagulle H, Malgaca L and Oktem HF (2004). Analysis of active vibration control in smart structures by ANSYS. *Smart Materials and Structures 13*, 661–667. (journal)

Khot et al. (2011). Active vibration control of cantilever beam by using PID based output feedback controller. *Journal of vibration and control -18(3)*, 366–372. (journal)

Khot, SM, Yelve NP (2011). Modeling and response analysis of dynamic systems by using ANSYS and MATLAB. *Journal of Vibration and Control - 17 (6)*, 953-958. (journal)

Manning, WJ, Plummer AR and Levesley MC (2000). Vibration control of a flexible beam with integrated actuators and sensors. *Smart Materials and Structures 9*, 932–939. (journal)

Mei, CC, Yeung RW & Liu KF (1985). Lifting of a Large Object from a Porous Seabed, *Journal of Fluid Mechanics*, Vol.152, 203-215. (journal)

Palm R, Driankov D and Hellendoorn H (1997). *Model Based Fuzzy Control*, Springer-Verlag, Germany. (textbook)

Rawson KJ, Tupper EC (2001). *Basic Ship Theory - Vol. 1*, Butterworth-Heinemann publishing, Oxford. (textbook)

SMIT (2010). Salvage Brochure. Retrieved on 5th October 2010, www.smit.com. (brochure)

Subsalve (2010). Salvage Catalog. Retrieved on 5th October 2010, www.subsalve.com. (catalog)

Sun et al. (2007). Design of Vibration Controllers for Flexible Beams Using the Mechatronic Design Quotient (MDQ) Approach, *Journal of Vibration and Control*, 13 (1), 65-94. (journal)

SuSy (2010).Requirement of the Vessel Design Selected in the Typical Scenarios. (project report)

SuSy (2011). Project Mid-term Report. (report)

Velayudhan, AKD, Srinil N, Barltrop N (2011). Sliding Mode Controller for Salvaging of Sunken Vessels, *Proceedings of 15th International Conference on Computer Applications in Ship Building*, Trieste, Italy. (conference)

Velayudhan, AKD, Srinil, N, Barltrop N (2012). A New Ship Recovery Concept and Design Using Adaptively Controlled Buoyancy Systems, *Proceedings of 11th International Marine Design Conference*, Glasgow. (conference)