# Simulator and Control System Design for a Free Floating Surface Effect Ship at Zero Vessel Speed

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**Abstract:** This paper covers vertical motion damping of a free floating Surface Effect Ship (SES) at zero vessel speed. Vertical motion damping is requested at the bow deck of the SES during offshore wind turbine docking operation. Vertical motion damping provides a safer transfer of personnel and goods from vessel to the wind mill in moderate/rough sea states. This will provide an increased operational weather window access for maintenance and repair.
The paper has two main contributions. 1) A simulator model for a SES. This is referred to as the cushion process plant model and provides a valid simulation tool for the control system. 2) A simple controller for actuating air cushion pressure. The presented controller actively controls the air flow actuators to the air cushion in order to minimize vertical motions at the vessel bow. A numerical stability investigation for the controller is included.

Keywords: Ship control, Feedback control

## 1. INTRODUCTION

Surface Effect Ships offer reduced hydrodynamic forces acting on the side hull compared to conventional catamarans. The SES is known to offer high speed as well as great sea-keeping capabilities in rough seas.

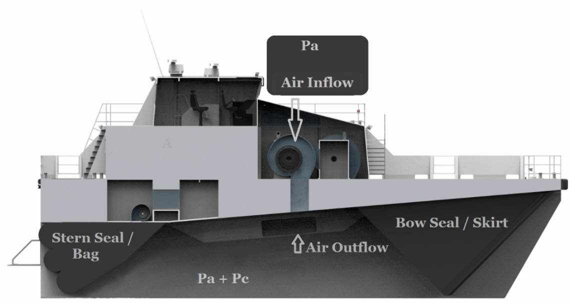



Fig. 1. The SES Concept (Umoe Mandal (UM) Proprietary)

The SES rides on an air cushion enclosed by catamaran type side hulls and flexible rubber seals in stern and bow end. During transit, the air cushion approximately lifts 80 % of the vessel mass, leaving only a minor part exposed to hydrodynamic disturbances. One or more lift fans provide cushion air inflow. The fans are assumed to run at constant rotational speed. The air cushion actuator is composed of adjustable ventilation valves for controlling the air outflow. By closing the ventilation valve the vessel will increase its vertical position (upwards). The opposite effect will appear by opening the valve. This paper presents a SES simulator model as well as a controller that actively adjusts the ventilation valve in order to compensate for encountered wave propagation.


Fig. 2. This paper deals with UM's Wave Craft (UM Proprietary)

The SES simulator was developed for design and numerical testing of the proposed control system. The simulator calculates enclosed air cushion volume, air flows and a set of equations in order to compute cushion pressure at each time instance. The cushion pressure acts on the wet deck area which induce forces acting on the craft.

The control problem of this paper is different from traditional air flow control on a SES which is known as a Ride Control System (RCS) where one is striving to damp out pressure variations which corresponds to high vertical accelerations during transit, see [Sørensen and Egeland, 1995]) and [Kaplan and Davies, 1978]. These accelerations result in a passenger comfort problem.

In contrast, the controller presented here encourage large pressure variations in order to minimize vertical motions at the bow tip excited by medium/ large wave disturbances. This is done by changing the rotation point in pitch from center of gravity to the vessel bow tip.

A similar task was introduced by Basturk et al. [2011], where disturbance cancellation was performed on another Umoe Mandal developed vessel, the T-Craft (Hybrid Air Cushion - Hovercraft ). This work involves active control of the airflow in order to minimize wave induced motion between two ramp-connected ships. Another work by Basturk and Krstic [2012] presents an observer for first order wave propagations that uses measurement of state derivatives. This utilizes the fact that one can implement the system using only a heave accelerometer.

## 2. SES SIMULATOR - PROCESS PLANT MODEL

The SES simulator is implemented using the integrated ship design tool *ShipX* [Marintek, 2012]. *ShipX* consist of several necessary plug-ins in order to simulate a SES: VeRes calculates offline hydrodynamic vessel response using strip theory according to various predetermined parameters such as vessel dimension, mass distribution, radii of gyration, vessel velocity and a desired set of wave propagation's and headings.

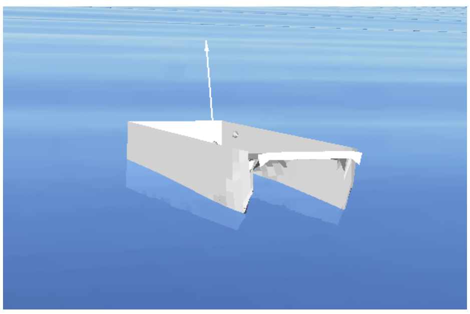


Fig. 3. SES Simulation and visualization using ShipX plug-ins

Online simulation of the vessel and environmental disturbances acting on the vessel is performed by the Vessel Simulator (VeSim). All hydrodynamic forces acting on the craft are calculated by VeSim. The air cushion forces and moments acting on the vessel are not build into ShipX. This is solved by externally sending these forces and moments to VeSim, see figure 4. The calculation is based on Faltinsen [2005] with some minor adjustments. The cushion process plant was first implemented by Espeland [2008] and further developed in [Auestad, 2012]. The model also includes dynamics from the flexible rubber bow and stern seals [Wu, 2011]. The seal dynamics includes seal leakage and seal forces acting on the vessel. The seal dynamics are not further discussed in this article.

Figure 4 illustrate the overall SES Simulator implemented in VeSim. It is the green subsystem on the upper left corner that is implemented and discussed in this paper, the rest is handled by ShipX and VeSim.

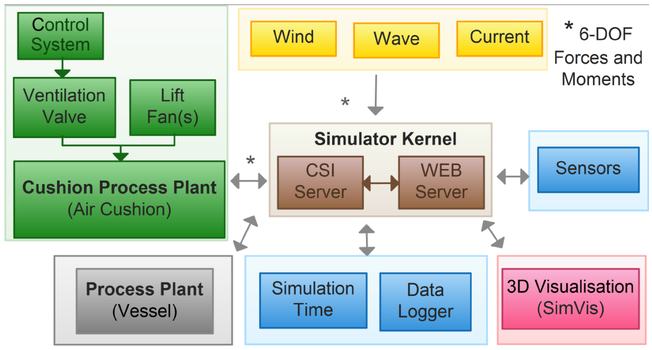



Fig. 4. Architecture of the SES Simulator. CSI: Common Simulation Interface. The WEB server enables the simulator to receive parameters and input via an internet browser.

### *2.1 Cushion Process Plant Model*

In this section we derive an expression for the cushion pressure variations which enable us to calculate the air cushion forces acting on the vessel. Only the most vital aspects of this process are discussed. We define a moving coordinate system, $B$, whose origin is located at the mean water plane below the center of gravity (CG). The **x**,**y** and **z** axes are defined positive forward, to the port and upwards respectively, as illustrated in figure 5.

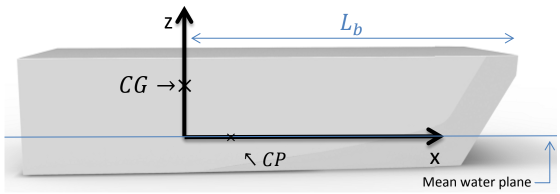



Fig. 5. Body fixed coordinate system $B$. CG and CP are set to illustrate possible longitudinal positions but not actual design parameters.

$CP$, as illustrated in figure 5, denotes the longitudinal center of pressure which is the attack point of the cushion pressure, $\eta_{1,2,3}$ represent surge, sway and heave which respectively denotes the translation along the **x**, **y** and **z** axes. $\eta_{4,5,6}$ respectively represent roll, pitch and yaw which is rotation around the given axes according to the right hand rule. The equation of motion are formulated in the $B$-frame.

Using a 3D model of the craft, the enclosed air cushion volume $\Omega$ is numerically calculated as a function of time according to:

$$\Omega(t) = \iint_{A_c} h_c + \eta_3(t) + y\,\eta_4(t) - x\,\eta_5(t) - T - \zeta(x, y, t)\, dA, \tag{1}$$

where $h_c = h_c(x, y)$ is the spatial varying height from the baseline to the wet deck (cushion roof), $A_c$ is the cushion area, $T$ and $\zeta$ are, respectively, draught and wave elevation.

Total cushion pressure is $P_c(t) + P_a$, where $P_a$ is the atmospheric pressure, and $P_c(t)$ is cushion excess pressure. The cushion dynamics are linearized around an equilibrium air cushion excess pressure $P_0$. $P_0$ corresponds to the cushion excess pressure in the case of constant lift fan rotational frequency, constant valve leakage area ($A_0^{ctrl}$) and no waves. The non-dimensional uniform pressure variation $\mu_u(t)$ is defined according to:

$$\mu_u(t) = \frac{P_c(t) - P_0}{P_0}, \tag{2}$$

The air cushion air inflow ($Q_{in}$) is given by a designed lift fan characteristic as a function of $P_c(t)$. The cushion air outflow is given by:

$$Q_{out}(t) = c_n A_L(t) \sqrt{\frac{2P_c(t)}{\rho_a}} \tag{3}$$

where $\rho_a$ is the atmospheric density of air and $0 < c_n < 1$ is a correction term for leakage orifice. This term is assumed constant. $A_L(t)$ is the total cushion leakage area.

The cushion process plant is linearized. Using Taylor Expansion on (3) with respect to $\mu_u(t)$ around the equilibrium $\mu_u = 0$ yields:

$$\begin{aligned} Q_{out}(t) &\approx c_n A_L(t) \sqrt{\frac{2P_0}{\rho_a}} + \frac{1}{2} c_n A_L(t) \sqrt{\frac{2P_0}{\rho_a}} \mu_u(t) \\ &= c_n A_L(t) \sqrt{\frac{2P_0}{\rho_a}} \left(1 + \frac{\mu_u(t)}{2}\right), \end{aligned} \tag{4}$$

Note that the total cushion leakage $A_L(t)$ include leakage contribution from under the side hulls, seals and the active controllable ventilation valve:

$$A_L(t) = A^{SEALS}(t) + A^{HULL}(t) + A^{CTRL}(t) \tag{5}$$

Two more equations must be satisfied in order to calculate the cushion pressure $P_c(t)$.

(1) A continuity equation for the air mass inside the cushion using the chain rule yields

$$\begin{aligned} \dot{m} &= \dot{m}_{in} - \dot{m}_{out} \\ &= \frac{d}{dt}(\rho_c \Omega) \\ &= \dot{\rho}_c(t)\,\Omega(t) + \rho_c(t)\,\dot{\Omega}(t), \end{aligned} \tag{6}$$

where $\rho_c$ is the density in the air cushion. Faltinsen [2005] and Kaplan and Davies [1974] shows that the mass rate can be expressed as:

$$\rho_{c0}\left[Q_{in}(t) - Q_{out}(t)\right] = \dot{\rho}_c(t)\,\Omega(t) + \rho_c(t)\,\dot{\Omega}(t), \tag{7}$$

where $\rho_{c0}$ is the air mass density at equilibrium excess pressure $P_0$. A normal simplification is setting $\rho_{c0} = \rho_a$.

(2) An adiabatic equation relating pressure and mass density, assuming ideal gas:

$$\frac{P_c(t) + P_a}{P_0 + P_a} = \left(\frac{\rho_c(t)}{\rho_a}\right)^{\gamma}, \tag{8}$$

where $\gamma = 1.4$ is the ratio of heat capacities for air. Rewriting (8) using (2) and linearizing $\rho_c(t)$ around $P_0$ using Taylor expansion results in:

$$\rho_c(t) \approx \rho_a \left(1 + \frac{\mu_u(t) P_0}{\gamma(P_0 + P_a)}\right). \tag{9}$$

Note that equation (6) requires $\dot{\rho}_c$. Differentiating (9) with respect to time yields:

$$\dot{\rho}_c = \frac{\rho_a P_0}{\gamma(P_0 + P_a)}\,\dot{\mu}_u. \tag{10}$$

Using (4), (9) and (10) and inserting this into (7) yields:

$$\begin{aligned} &\rho_a \left[Q_{in}(t) - c_n A_L(t) \sqrt{\frac{2P_0}{\rho_a}} \left(1 + \frac{\mu_u(t)}{2}\right)\right] = \\ &\frac{\rho_a\,P_0}{\gamma(P_0 + P_a)}\,\dot{\mu}_u(t)\,\Omega(t) + \rho_a \left(1 + \frac{\mu_u(t) P_0}{\gamma(P_0 + P_a)}\right) \dot{\Omega}(t) \end{aligned} \tag{11}$$

which can be rewritten:

$$\dot{\mu}_u + A^* \mu_u(t) = B^* \tag{12}$$

Where:

$$\begin{aligned} A^* &:= \left[\frac{\gamma(P_0 + P_a) c_n A_L(t)}{2P_0\,\Omega(t)}\right] \sqrt{\frac{2P_0}{\rho_a}} + \frac{\dot{\Omega}(t)}{\Omega(t)} \\ B^* &:= \frac{\gamma(P_0 + P_a)}{P_0 \Omega(t)} \left[Q_{in}(t) - c_n A_L(t) \sqrt{\frac{2P_0}{\rho_a}}\right] \end{aligned} \tag{13}$$

Finally, in the simulator, (12) is solved for each time instance $i$ using the integration factor:

$$\begin{aligned} g(t) &:= \int_{t_{i-1}}^{t_i} A^*\,dt \\ \mu_u(t) &= e^{-g(t)} \int_{t_{i-1}}^{t_i} B^*\,e^{g(t)}\,dt \end{aligned} \tag{14}$$

The cushion pressure induce forces and pitch moments acting on the vessel according to:

$$\mathbf{F}_{\mathbf{cushion}}(t) := -\iint_S P_0(1 + \mu_u)\mathbf{n}\,dS \tag{15}$$

$$\mathbf{M}_{\mathbf{cushion}}(t) := -\iint_S P_0(1 + \mu_u)(\mathbf{r} \times \mathbf{n})\,dS, \tag{16}$$

where $\mathbf{r}$ is a vector from the origin to the point where the moments are calculated. $\mathbf{F}_{\mathbf{cushion}}$ and $\mathbf{M}_{\mathbf{cushion}}$ are sent to VeSim as illustrated in figure 4, where they are merged with the environmental and hydrodynamic forces acting on the hull. The location of the surface $S$ and normal vector $n$ is shown in figure 6.

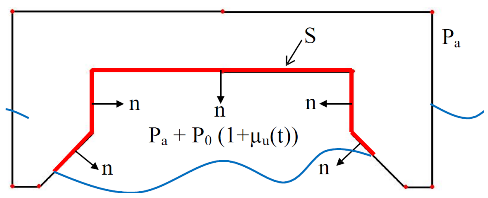



Fig. 6. Calculating Air Cushion Forces and Moments for the Simulator

## 3. CONTROL SYSTEM - CONTROL PLANT MODEL

The Control Plant Model differs from the Process Plant Model since it is a simplified mathematical description of the process plant that is relevant to the control problem.

While the equations in section 2.1 are based on Faltinsen [2005] the following are based on Sørensen [1993]. Sørensen presents both coupled and decoupled equations for heave, pitch and uniform pressure variations. The equations used in this analysis are quasi-coupled. With this, we understand that there is no coupling between heave and pitch with respect to hydrodynamic added mass, water wave radiation damping and hydrostatic coefficients, as this his is assumed negligible in the $B$-frame (figure 5). However, there is coupling between uniform pressure variations and pitch velocity. This coupling is necessary to include since the Wave Craft has very narrow side hulls at the fore-end compared to the stern. To obtain a reasonable trim angle when lift fans are turned off, the longitudinal center of gravity is forced relatively far to the stern. Compared to other SES, this leads to a non-negligible gap between the center of pressure and longitudinal CG. This is also accounted for in the Process Plant Model. There is also a coupling between heave and pitch in the controller when transforming the motion from the control point to the chosen coordinate system. Therefore, we assume that translation and velocity for both heave and pitch are available for measurement.

*Remark 1.* Sørensen and Egeland [1995] also presents the solution to damp spatial varying pressure variations around the resonance frequencies of the vessel, this is not discussed here or in the cushion process plant model since such a short SES, at zero speed, does not experience acoustic vibrations. This result in certain simplifications.

### 3.1 Craft Dynamics

The following control plant model include contributions from hydrodynamic buoyancy and air cushion dynamics.

The active controlled leakage area of the ventilation valve can be expressed as:

$$A^{CTRL} = A_0^{CTRL} + \Delta A^{CTRL}, \tag{17}$$

where $A_0^{CTRL}$ is the reference leakage area allowing two sided control as discussed in Kaplan and Davies [1974].

The dynamics in heave according to Sørensen [1993] is given according to:

$$(m+A_{33})\,\ddot{\eta}_3(t)+B_{33}\,\dot{\eta}_3(t)+C_{33}\,\eta_3(t)-A_c\,P_0\,\mu_u(t) = F_3^e(t), \tag{18}$$

and the dynamics in pitch can be written:

$$(I_{55}+A_{55})\ddot{\eta}_5(t)+B_{55}\dot{\eta}_5(t)+C_{55}\eta_5(t)+A_c p_0 x_{cp}\mu_u(t) = F_5^e(t), \tag{19}$$

where $m$ and $I_{55}$ are vessel mass and the moment of inertia around the body fixed y-axis. Let j = 3,5 respectively denote heave and pitch. $A_{jj}$ is hydrodynamic added-mass coefficient, $B_{jj}$ is the water wave radiation damping coefficient and $C_{jj}$ is found by integrating over the water plane area of the side hulls. $F_j^e$ is the hydrodynamic excitation force acting on the side-hulls acting in $j$ direction. $A_c$ is equilibrium air cushion area. $x_{cp}$ is the longitudinal position of the center of pressure. The hydrodynamic excitation force in heave can be expressed as:

$$F_3^e(t) = 2\zeta_a e^{-k\,d}\frac{sin\frac{k\,L}{2}}{\frac{k\,L}{2}}\left(C_{33}-\omega_0^2 A_{33}\right)sin\,\omega_0 t, \tag{20}$$

where $k = 2\pi/\lambda$. $\zeta_a$, $\lambda$ and $\omega_0$ are respectively sea wave elevation amplitude, length and frequency, $d$ is draft of side hulls. The hydrodynamic excitation force in pitch is given by:

$$F_5^e(t) = 2\zeta_a e^{-k\,d}\Big[\left(\frac{1}{k}cos\frac{k\,L}{2}-\frac{2}{k^2\,L}sin\frac{k\,L}{2}\right)\left(C_{33}-\omega_0^2 A_{33}\right)\Big]cos\,\omega_0 t \tag{21}$$

The uniform cushion pressure equation is given by:

$$K_1\,\dot{\mu}_u(t)+K_3\,\mu_u(t)+\rho_{c0}\,A_c\,\dot{\eta}_3(t)-\rho_{c0}A_c x_{cp}\dot{\eta}_5 = K_2\,\Delta A^{CTRL}(t)+\rho_{c0}\,\dot{V}_0(t), \tag{22}$$

where:

$$\begin{aligned} K_1 &= \frac{\rho_{c0}\,h_0\,A_c}{\gamma\left(1+\frac{P_a}{P_0}\right)},\\ K_2 &= \rho_{c0}\,c_n\sqrt{\frac{2P_0}{\rho_a}},\\ K_3 &= \frac{\rho_{c0}}{2}\left(Q_0-2\,P_0\,q\frac{\partial Q_{in}}{\partial P}|_0\right), \end{aligned} \tag{23}$$

$\Delta A^{CTRL}$ is the controlled air flow leakage out of the air cushion, $h_0$ is the height from waterline to wetdeck at equilibrium pressure $P_0$, $Q_0$ is the equilibrium air flow, $\frac{\partial Q_{in}}{\partial P}|_0$ is the lift fan characteristic slope at equilibrium point, $\dot{V}_0(t)$ is the rate of wave volume pumping of dynamic pressure, $q$ is the total number of lift fans (if more than one, assume they run at similar rotational speed and share the same fan characteristic).

The rate of wave volume pumping is expressed as:

$$\dot{V}_0(t) = A_c\,\zeta_a\,\omega_0\,\frac{sin\frac{k\,L}{2}}{\frac{k\,L}{2}}\,cos(\omega_0 t) \tag{24}$$

### 3.2 State Space Model

Consider the following control plant model of the linear time-invariant (LTI) system of the form

$$\begin{aligned} \dot{\mathbf{x}} &= \mathbf{A}\,\mathbf{x}+\mathbf{B}\,u+\mathbf{E}\,\mathbf{v}\\ \mathbf{y} &= \mathbf{C}\,\mathbf{x}, \end{aligned} \tag{25}$$

where

$$\mathbf{x} = \left[\,\eta_3\ \eta_5\ \dot{\eta}_3\ \dot{\eta}_5\ \mu_u\,\right]^T \quad ,\mathbf{v} = \left[\,F_3^e\ F_5^e\ \dot{V}_0\,\right]^T, \tag{26}$$

where $\mathbf{x}(t)$ is the 5-dimensional state vector, $u(t) = \Delta A^{CTRL}$ is the scalar control input, $\mathbf{y}(t)$ is the 2-dimensional measurement vector which will be discussed in the next section. See appendix A for the time invariant matrices $\mathbf{A}$, $\mathbf{B}$ and $\mathbf{E}$.

### 3.3 Control System

It is desired to minimize the bow motion acting in the defined $\mathbf{z}$ axis located at the vessel bow. Defined this as coordinate system $B_2$ which has the same axes direction of the coordinate system $B$, but the origin is positioned at the vessel bow. This corresponds to placing an accelerometer at the bow, and integrating the signal twice, respectively to $\dot{\eta}_{3\_bow}$ and $\eta_{3\_bow}$. The coordinate system as defined in section 2.1 has its origin defined below CG, hence one must transform the motion from the bow to the origin in the same coordinate frame. In the absent of pitch ($\eta_5 = 0$), $\eta_{3,bow} = h+\eta_3$, where $h$ is the height from the mean water plane to the bow deck. It becomes clear that $\eta_{3\_bow}$ will increase its value when the bow/nose is pointing upwards (negative pitch). It can easily be shown that:

$$\begin{aligned} \eta_{3\_bow} &= h + \eta_3 + L_b \, tan(-\eta_5) \\ \eta_{3\_bow} &= h + \eta_3 - L_b \, tan(\eta_5), \end{aligned} \tag{27}$$

By linearizing (27) around $\eta_5 = 0$, (27) can be rewritten:

$$\eta_{3\_bow} = h + \eta_3 - L_b \eta_5 \tag{28}$$

Differentiating (28) with respect to time yields:

$$\dot{\eta}_{3\_bow} = \dot{\eta}_3 - L_b \dot{\eta}_5, \tag{29}$$

using (28) and (29), $\mathbf{y_0}$ and $\mathbf{y}$ can be expressed as:

$$\begin{aligned} (\mathbf{y_0} - \mathbf{y}) &= - \begin{bmatrix} \eta_{3\_bow} \\ \dot{\eta}_{3\_bow} \end{bmatrix}, \\ \mathbf{y_0} = \begin{bmatrix} -h \\ 0 \end{bmatrix}, \quad \mathbf{y} = \mathbf{C}\,\mathbf{x} &= \begin{bmatrix} 1 & -L_b & 0 & 0 & 0 \\ 0 & 0 & 1 & -L_b & 0 \end{bmatrix} \mathbf{x} \end{aligned} \tag{30}$$

Therefore, the following feedback controller is proposed for minimizing motion along the $\mathbf{z}$ axis of the $B_2$ frame:

$$u = \mathbf{K}\,(\mathbf{y_0} - \mathbf{y}) \tag{31}$$

where $\mathbf{K}$ is a time invariant feedback gain matrix given by:

$$\mathbf{K} = [K_p \;\; K_d] \tag{32}$$

Consequently, the controller tries to change the rotation point in pitch from CG to the bow tip.

### *3.4 Discussion*

The control system is virtually moving the center of rotation in order to minimize heave position and velocity at the vessel bow. In order to try to understand this concept mathematically the closed loop pressure equation can be written:

$$\dot{\mu}_u = -\frac{K_3}{K_2}\mu_u - \frac{A_c \rho_{c0}}{K_1}\dot{\eta}_{3\_cp} - \frac{K_p}{K_2}\eta_{3\_bow} - \frac{K_d}{K_2}\dot{\eta}_{3\_bow}, \tag{33}$$

where $\dot{\eta}_{3\_cp} = \dot{\eta}_{3\_cp}(\eta_3, \eta_5)$ is heave velocity acting at the center of pressure (se fig 5). Lets first discuss the behavior when the control system is inactive. We set $K_p = K_d = 0$. Since the constants are defined positive, the first two terms in (33) will stabilize the pressure and heave velocity acting on the wetdeck.

By gradually increasing the control gains, the two last terms in (33) will dominate the two first. The cushion pressure will try to minimize $(\mathbf{y_0} - \mathbf{y})$ by altering the cushion pressure in order to change the longitudinal rotation point in pitch.

### *3.5 Stability properties of the control system*

Due to limited space, this paper does not consider parametric uncertainties, classification of disturbance, optimal or robust control properties. Therefore a very brief numerical stability investigation will be given. It can be shown that the the pair $(\mathbf{A}, \mathbf{B})$ is controllable and $(\mathbf{A}, \mathbf{C})$ is observable. Note that we will be investigating stability of the origin of the $B$-system. The closed loop perturbed system in (25) can be expressed in the frequency domain as:

$$\begin{aligned} \mathbf{y} &= \mathbf{C}\,\mathbf{x} \\ &= \mathbf{C}\,(s\,\mathbf{I_{5x5}} - \mathbf{A} + \mathbf{B}\,\mathbf{K}\,\mathbf{C})^{-\mathbf{1}}\,(\mathbf{B}\,\mathbf{K}\,\mathbf{y_0} + \mathbf{E}\mathbf{v}) \\ &:= \mathbf{H_1}(s)\mathbf{y_0} + \mathbf{H_2}(s)\mathbf{v}, \end{aligned} \tag{34}$$

where $\mathbf{H_1}$ and $\mathbf{H_2}$ share the same poles. $\mathbf{H_2}$ is the $2\,x\,3$ transfer function matrix from disturbance vector $\mathbf{v}$ to measurement vector $\mathbf{y}$. Figure 7 shows the bode plot for each element in $\mathbf{H_2}$:

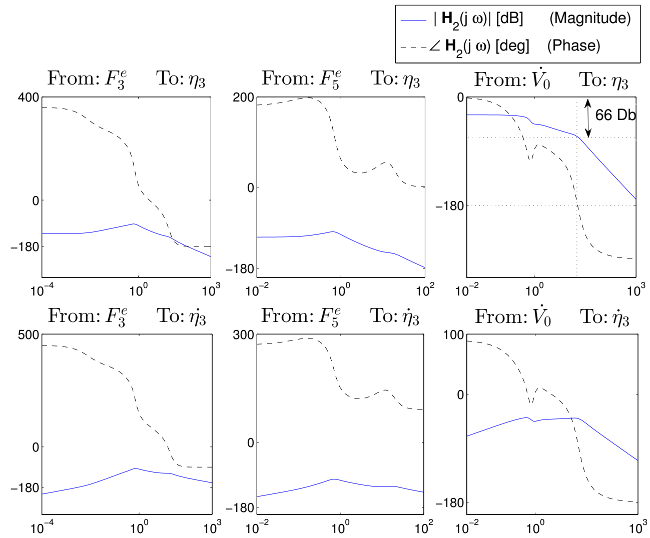


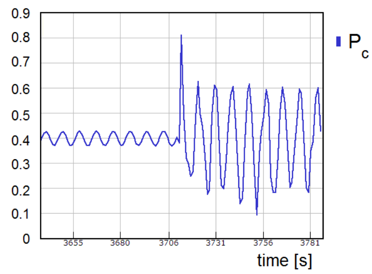

Fig. 7. Bode plot for $\mathbf{H_2}$. The x-axis shows frequenzy [Hz]

The Nyquist Stability Criterion, using Bode plot information, states that $\mathbf{x} = \mathbf{0}$ is stable if the zero decibel magnitude crossing occurs at a lower frequency than $-180$ degrees. Since all of the signals have negative magnitude, for all frequencies, the system is stable. The upper right figure has the smallest proportional gain margin of 66.2 decibel. Hence, if $K_p > 66.2\,[dB]$, the system will become unstable. If we want to vary the derivative gain, it can be shown that stability is achieved for the former assumption in addition to: $K_p < 21.6 K_d + 36$, where the numbers are given in decibel. Due to lack of space, this will not be shown here.

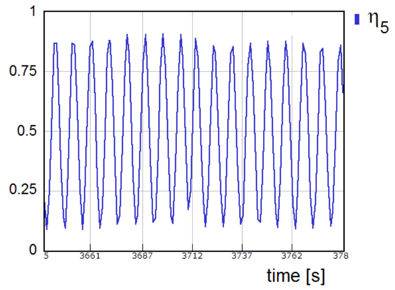
## 4. RESULTS

The objective of this paper is to develop a functional and valid SES simulator and to investigate the concept of damping wave induced motion at the vessel bow. Figure 8 shows the time series for $P_c$, $\eta_5$ and $\eta_3$ vs $\eta_{3\_bow}$ for a simulation run in regular wave head sea. The control system is initially inactive and turned on at $t \approx 3710[s]$. The figure illustrates the concept of vertical motion damping at the bow deck by changing the rotation point in pitch using the controller given in (31). In this sea state, $\eta_3$ and $\eta_5$ independently remain relatively unchanged, regardless if the control system is active or not.

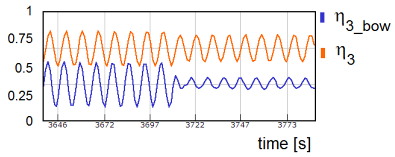
Figure 9 illustrates that the comprehensive computational simulator (Process Plant Modell) in VeSim is modeling the actual real world. The heave position at bow deck is shown for the simulator and an actual model test of the Wave Craft. The model test has a scale factor 1/8 compared to full-scale craft and possess realistic, scaled actuators, sensors and seals. Although this paper is not meant to discuss model test results, the comparison figure is given to validate the simulator. It must be noted that the model test provided some uncertain simulator parameters. The control system is initially active, and then turned off.




(a) Cushion Pressure - $P_c$




(b) Pitch - $\eta_5$




(c) Heave position at $\eta_{3\_bow}$ and $\eta_3$

Fig. 8. Simulation run, regular head sea, Wave height and period are 0.5m and 8s. All variables have been normalized.

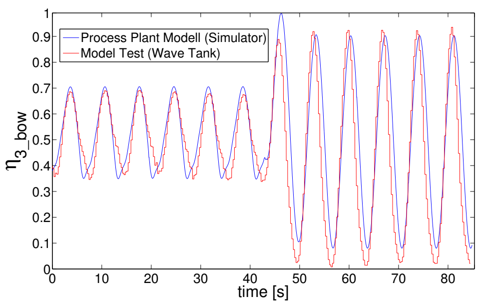



Fig. 9. Simulator validation and vertical bow motion damping, showing the control system respectively toggled on and off. The y-axis is normalized.

## 5. CONCLUSIONS AND FURTHER WORK

This paper contains a study in modeling and control of a Surface Effect Ship at zero vessel speed. The main focus is to create an overall credible simulation toolbox for a SES. Optimal performance has not been a topic, although the results clearly illustrates the concept of motion damping.

A process plant for modelling a SES has been succesfully implemented. A simplified control plant model is presented for the preliminary stability investigation of the control system.

Further work involves three important studies. Due to parametric uncertainties a global stability analysis is necessary. Obtaining a heave position signal which is a control input is possible but often an expensive task. It is therefore desired to implement a heave observer. For optimal control, control gains need to vary as functions of wave height and period. Some sort of adaptive control system is therefore desired.

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## Appendix A. SYMBOLIC MODEL MATRICES

$$\mathbf{A} = \begin{bmatrix} 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 \\ \frac{-C_{33}}{m+A_{33}} & 0 & \frac{-B_{33}}{m+A_{33}} & 0 & \frac{A_c P_0}{m+A_{33}} \\ 0 & \frac{-C_{55}}{I_{55}+A_{55}} & 0 & \frac{-B_{55}}{I_{55}+A_{55}} & 0 \\ 0 & 0 & \frac{-\rho_{c0} A_c}{K1} & 0 & \frac{-K_3}{K_1} \end{bmatrix}$$

$$\mathbf{B} = \begin{bmatrix} 0 & 0 & 0 & 0 & \frac{K_2}{K_1} \end{bmatrix}^{\mathsf{T}}$$

$$\mathbf{E} = \begin{bmatrix} 0 & 0 & \frac{1}{m+A_{33}} & 0 & 0 \\ 0 & 0 & 0 & \frac{1}{I_{55}+A_{55}} & 0 \\ 0 & 0 & 0 & 0 & \frac{1}{K_1} \end{bmatrix}^{\mathsf{T}}$$