Surface Effect Ship with Four Air Cushions Part I: Dynamic Modeling and Simulation ⋆

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Abstract: This paper deals with dynamic modelling and numerical simulations of a Surface Effect Ship (SES) with split cushion. Traditionally, a SES is a catamaran with front and aft seals equipped with lift fans that could fill an air cushion with pressurized air to lift up to 90 % weight of the vehicle. A new SES concept is designed by UMOE Mandal AS that consists of four air chambers, each one equipped with variable vent valves, through which cushion pressures can be controlled by adjusting the air out-flow from the cushion. The new design makes it possible to actively regulate the motion of the SES in all degrees of freedom but surge. In this paper, we develop a dynamic model of the four cushion SES. Furthermore, we present a high fidelity numerical simulator that can effectively simulates the dynamics of the vessel. A companion paper (Haukeland, Hassani, and Auestad 2019) studies the performance of the control system through numerical simulation using the presented high fidelity model of SES subject to waves.

Keywords: Surface Effect Ship, Split Cushion, Motion control, Process Plant Model

1. INTRODUCTION

1.1 Motivation

The rough seas surrounding offshore structures such as oil platforms, oil-rigs and wind turbines provide a challenge for both crew transport and offshore structure inspections. The use of surface effect ships has emerged as a competitive alternative to helicopter transport, proving high levels of safety, comfort, fuel efficiency and overall reduced cost of offshore logistics, Mandal 2018; Hassani, Fjellvang, and Auestad 2019. The main challenge for sea transport is safety at crew transfer and ride comfort at high transit speeds. Increased motion control and motion damping provides the solution to these challenges, and contributes to further expand the window of operations for the surface effect vessels at harsh weather conditions.

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1.2 Surface effect ship

The surface effect ship can be described as a hybrid between a catamaran and hovercraft. The hull of a SES is formed like the hull of a catamaran except the bow and stern is sealed off. Figure 1 gives an illustration of a SES cross-section as seen from the side. The purpose of this special shape, is to create a volume in which the air will be trapped and the flow of air can be controlled. This volume is referred to as the cushion chamber, where as the air inside the volume is referred to as the air cushion. This cushion allows the vessel to glide on the layer of air, rather than floating due to the displacement of the hull like conventional vessels. The design brings a few benefits such as reduction in drag, improved ride comfort and added flexibility with respect to heave. The new SES design proposed in this paper is to further divide the cushion into four air chambers. Doing so, one can add the pitch and roll control to the list of benefits. Due to the low drag and improved ride comfort of the SES, most SES-vessels are high-speed vessels. The drawback with the cushion solution is that the seals in the front and aft of the vessel suffer a lot of wear and tear. Depending on use and

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the transit velocities of the vessel, these seals might need replacement more than once per year. This project is done in collaboration with Umoe Mandal. At their request, and to avoid the publication of sensitive data, the results for the simulations and experiments will be normalized with respect to angles, pressure or volume flow.

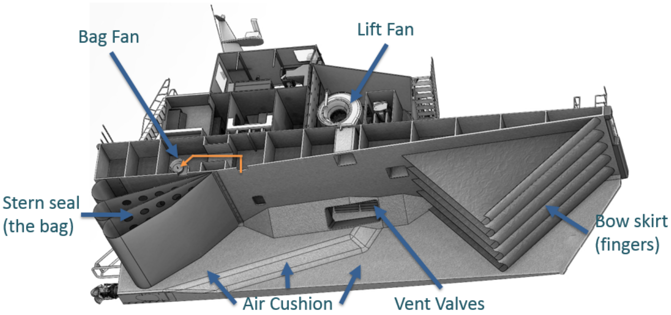

Fig. 1. Cross section of a SES. Illustration by Umoe Mandal.

1.3 SES cushion-control

Other systems for controlling a SES-vessel by manipulating the cushion pressure are already developed and in use, among them is the ride control system and the boarding control system. The ride control system was first featured in Kaplan and Davis 1978, and has since been further developed by Sørensen and Egeland 1995. The purpose of the system is to create a more smooth ride at high transit velocities. The system provides active damping of vertical motions by manipulating and reducing the cushion pressure fluctuations caused by rough sea. The boarding control system has been developed by Ø. F. Auestad 2015. The main use of the boarding control system is to reduce the movements of the vessel's bow, so that it is possible to secure safer transfer from the ship to offshore structures, specifically, offshore wind-turbines. The boarding control system relies on manipulating the pressure of the single cushions to induce heave motions, counteracting the wave-induced motions. These are examples of manipulating the cushion pressure as a way of controlling the vessel motions, that have been of great practical importance, and that provide a good basis and motivation for further development within the field.

1.4 Cushion division

In this paper, we consider a SES design in which the single cushion is divided into four sections by the use of solid walls or inflatable separators. The division and subsequent cushion numbering can be seen in figure 2. The implementation of the four chambers solution using inflatable bags allows for usage of a the traditional one-cushion solution when the four cushion division is not needed.

2. MATHEMATICAL MODELING

The system is modeled around a body-fixed coordinate system, centered at the center of gravity in the x and y-direction, and at the center of buoyancy in the z-direction. The heave center of buoyancy will vary for a

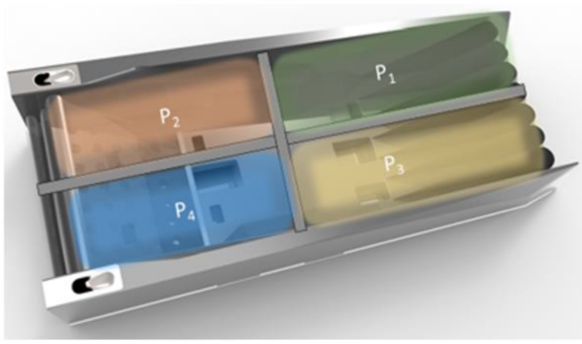

Fig. 2. The cushion separation and numbering as seen from below.

SES, depending on the current cushion pressure. With respect to the coordinate system, it is defined at the initial cushion pressure p_0, also known as the equilibrium pressure. The coordinate system is illustrated in figure 3.

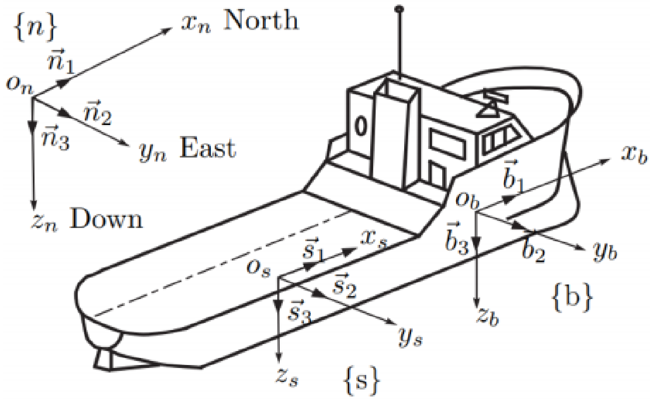

Fig. 3. Coordinate system used for the mathematical model, {b}.

2.1 Air-cushion volume and wave volume pumping

The volume of air inside each of the cushions can be described as

$$V_i = -V_{0_i} + A_{c_i}(-\eta_3 - y_{cp_i}\eta_4 + x_{cp_i}\eta_5) \qquad (1)$$

Here V_{0_i} is the volume occupied by waves entering or exiting each of the cushion chambers. A_{c_i} is the area of the water surface inside each cushion, while y_{cp_i} and x_{cp_i} describe the cushion center point, which is also referred to as the center of pressure. Lastly, η_3, η_4 and η_5 represent the heave, roll and pitch of the vessel, respectively. The cushion volume rate of change, $\dot{V}_{c_i}$, can be calculated by differentiating the formula for the cushion volume given in equation (1) with respect to time. The equation then becomes

$$\dot{V}_{c_i} = A_{c_i}(-\nu_3 - y_{cp_i}\nu_4 + x_{cp_i}\nu_5) - \dot{V}_{0_i} \qquad (2)$$

where $\dot{V}_{0_i}$ is called the wave volume pumping and ν_3, ν_4 and ν_5 represent the heave speed, roll rate and pitch rate of the vessel, respectively. Wave volume pumping is the rate of volume change due to waves passing through the cushion chambers. The wave volume pumping is calculated according to equation (3). The volume that the wave occupies inside the cushion can be described as in equation

(4). These equations are inspired by Auestad 2015, as well as Sørensen and Egeland 1995.

$$\dot{V}_{0_i}(t) = \int_{y_{1_i}}^{y_{2_i}} \int_{L_{1_i}}^{L_{2_i}} \dot{\zeta}(x, y, t) dA \quad (3)$$

$$V_{0_i}(t) = \int_{y_{1_i}}^{y_{2_i}} \int_{L_{1_i}}^{L_{2_i}} \zeta(x, y, t) dA \quad (4)$$

In these equations y_{1_i} and y_{2_i} constitute the width of cushion i, while L_{1_i} to L_{2_i} constitutes the length of cushion i measured at the current water-level. In equation (3), the term $\zeta(x, y, t)$ is the wave elevation function expressing the wave elevation at position x and y, at time t. According to Perez 2005, this wave elevation can be described as

$$\zeta(x, y, t) = \bar{\zeta} \sin\Big(\omega t + \epsilon - k(x\cos(\chi) + y\sin(\chi)\Big) \quad (5)$$

$$\dot{\zeta}(x, y, t) = \bar{\zeta}\omega \cos\Big(\omega t + \epsilon - k(x\cos(\chi) + y\sin(\chi)\Big) \quad (6)$$

In the equations above, $\bar{\zeta}$ is the wave amplitude, ω is the circular wave frequency, and k is the wave number. The wave number can be expressed as $k = \frac{2\pi}{\lambda}$, where λ is the wave length. Furthermore, ϵ is the phase of the wave, and the term $x\cos(\chi) + y\sin(\chi)$ expresses the direction of the wave propagation relative to the vessel body-frame.

Differentiating equation (5), provides equation (6), which is used to calculate the wave volume pumping in equation (3). The wave elevation and corresponding wave-forces acting on the vessel, outside of the cushion pressure, will be modeled using the *Marine Systems Simulator* created by Fossen and Perez 2004, and will not be further elaborated on in this section.

2.2 Non-linear cushion pressure dynamics

The non-linear pressure dynamics equations are in inspired by Sørensen and Egeland 1995. Some assumptions regarding linearization of the air flow, and spatial variations in pressure vary from Sørensen and Egeland 1995, leading to a few changes to these equations. Further, the equations are adapted to the current model, such that it includes the four cushion solution. The basis for the non-linear pressure dynamics is the equation for continuity of mass flow and the relation between pressure and density, shown in equations (7) and (13), respectively. The cushion pressure is considered to be uniform within each of the cushion chambers.

$$\dot{m}_{in_i} - \dot{m}_{out_i} = \frac{d}{dt}(\rho_{c_i}(t)V_{c_i}(t)) \quad (7)$$

In equation (7), the left part of the equation can also be stated as

$$\dot{m}_{in_i} - \dot{m}_{out_i} = \rho_a(Q_{in_j}(t) - Q_{out_j}(t)). \quad (8)$$

By performing the differentiation of the right side expression in equation (7), and substituting the left side with (8), we get equation (9).

$$\rho_a(Q_{in_j}(t) - Q_{out_j}(t)) = \dot{\rho}_{c_i}(t)V_{c_i}(t) + \rho_{c_i}(t)\dot{V}_{c_i}(t) \quad (9)$$

Here $V_{c_i}(t)$ and $\dot{V}_{c_i}(t)$ is the chamber air volume and its rate of change, which can be expressed as derived in section 2.1. Q_j is the air flow caused by fan j, while ρ_a and ρ_{c_i} are the respective air-densities of the atmosphere and the cushion. Assuming an adiabatic process, the first law of thermodynamics, $\delta U + \delta W = \delta Q^{heat} = 0$, can be used to derive the adiabatic pressure-volume relationship

$$\frac{p}{p_0} = \left(\frac{V_0}{V}\right)^{\gamma}. \quad (10)$$

where γ is the ratio of specific heat for air. Since there is no loss of mass in the air flow traveling from the volumes V_0 to V, the volume-density relationship between the two spaces can be written as

$$\frac{V_0}{V} = \frac{\rho}{\rho_0}. \quad (11)$$

The pressure-density relationship then becomes

$$\rho = \rho_0\left(\frac{p}{p_0}\right)^{\frac{1}{\gamma}}. \quad (12)$$

As we are interested in the pressure differential from the atmosphere to the cushion, the values for the cushion and atmospheric pressure and density are substituted into equation (12).

$$\rho_{c_i}(t) = \rho_a\left(\frac{p_a + p_{u_i}(t)}{p_a}\right)^{\frac{1}{\gamma}} \quad (13)$$

Differentiating equation (13) with respect to time leads to

$$\dot{\rho}_{c_i}(t) = \frac{\rho_a}{p_a^{\frac{1}{\gamma}}\gamma}(p_a + p_{u_i}(t))^{\frac{1-\gamma}{\gamma}}\dot{p}_{u_i}(t). \quad (14)$$

By combining equations (9), (13) and (14), the final equation representing the non-linear cushion pressure can be expressed as shown in equation (15).

$$\dot{p}_{u_i}(t) = \frac{\gamma(p_a + p_{u_i}(t))}{V_{c_i}(t)} \times \left(\frac{p_a}{p_a + p_{u_i}(t)}^{\frac{1}{\gamma}}(Q_{in_j}(t) - Q_{out_j}(t)) - \dot{V}_{c_i}(t)\right) \quad (15)$$

2.3 Cushion air flow

The flow of air into the cushions are caused by fans. The volumetric flow rate produced by each of these fans, can be described as

$$Q_{in_i} = \frac{s_i}{s_{max}}Q_i^*. \quad (16)$$

Here Q_i^* is the flow of the fan for cushion i which is set by the fan characteristics. s_i and s_{max} is the current and maximum fan rotation speed. The pressure generated by the lift fan is given as $p = (\frac{s_i}{s_{max}})^2 p_i^*$, where the specific fan pressure p_i^* can be substituted with p_{u_i}.

Airflow out of a cushion is called air leakage. There are two types of leakages to consider, passive leakage and controlled leakage. Controlled air leakage is the airflow that exits out of the controlled ventilation valves at the sides of the cushion chambers. The passive leakage is the uncontrolled leakage that occurs when air is forced out from under the sides of the cushion chambers, or leaves the chamber in any way that is not controlled or intentional. The amount of air leakage is dependent on the leakage area and the cushion pressure. The total air leakage of a chamber can be described as:

$$A_{L_i}(t) = A_{p_i}(t) + A_i^{ctrl}(t). \quad (17)$$

$A_{L_i}(t)$ represents the total leakage area, A_{p_i} is the passive leakage area, and A_i^{ctrl} is the controlled leakage area.

Controlled leakage The controlled leakage area can be described as

$$A_i^{ctrl}(t) = A_{i_{max}}^{ctrl} \frac{A_{i_{min\%}}^{ctrl} + (1 - A_{i_{min\%}}^{ctrl})u_i(t)}{100} \tag{18}$$

Here $A_{i_{max}}^{ctrl}$ is the leakage area at maximum vent valve opening. $A_{i_{min\%}}^{ctrl}$ is the lowest possible area of the vent valve opening, given as a percentage of the maximum vent valve opening. Lastly $u(t)_i$ is the control input signal, for the opening of the vent valves.

We wish to express the flow out of the cushion. The volumetric flow rate is defined as

$$Q = vA \tag{19}$$

where v is the velocity of the fluid, while A is the cross-sectional area. Due to the geometry of the leakage area some flow reduction will occur, such that $A = A_L c_n$, where A_L is the leakage area, and c_n is a orifice coefficient for the leakage area. Starting with the Euler equation,

$$\frac{dp}{\rho} + vdv + gdz = 0, \tag{20}$$

and assuming an adiabatic pressure-density relationship, such that

$$\frac{p}{\rho^\gamma} = C, \tag{21}$$

the Bernoulli equation for an adiabatic, compressible flow can be derived. Assuming none, or negligible elevation change, the simplified Bernoulli equation for adiabatic compressible flow can be written as

$$\left(\frac{\gamma}{\gamma - 1}\right)\frac{p_{c_i}(t)}{\rho_{c_i}(t)} + \frac{1}{2}v_{c_i}^2 = \left(\frac{\gamma}{\gamma - 1}\right)\frac{p_a}{\rho_a} + \frac{1}{2}v_a^2. \tag{22}$$

From this, an expression for the air velocity can be derived. We assume no initial air-velocity in the chambers, such that $v_{c_i} = 0$. Further, we assume that all excess pressure in the ventilated air from a cushion is converted to air-velocity upon release. Thus equation (22) can be rearranged to express the velocity, as shown in equation (23).

$$v_a(t) = \sqrt{2\left(\frac{\gamma}{\gamma - 1}\right)\left(\frac{p_{c_i}(t)}{\rho_{c_i}(t)} - \frac{p_a}{\rho_a}\right)} \tag{23}$$

The pressure inside the cushion is described as;

$$p_{c_i}(t) = p_a + p_{u_i}(t) \tag{24}$$

where p_{c_i} is the total cushion pressure, which can be split into the atmospheric pressure p_a and the uniform cushion pressure p_{u_i}. By inserting this expression for $p_{c_i}(t)$ into (23) and combining with equation (19), we get a final expression describing the volumetric flow of the controlled leakage as shown in (25).

$$Q_i(t) = A_i^{ctrl}(t)c_n(i)\sqrt{\frac{2\gamma}{\gamma - 1}\left(\frac{p_{u_i}(t) + p_a}{\rho_{c_i}(t)} - \frac{p_a}{\rho_a}\right)} \tag{25}$$

Here $\rho_c(t)$ is found from the expression derived in (13).

Passive leakage The passive leakage area is calculated as the sum of any area that occurs under the hull, fingers or bag of the vessel that is no longer below water-level, as seen from inside the cushion. The passive leakage area can be further divided into inter-cushion leakage, and atmospheric leakage. To calculate the passive atmospheric leakage area for a chamber, the bottom edge of every wall surrounding a chamber is represented as a line. Each of these four lines enclosing a chamber are divided into ten points from the start of the wall to the the end of the wall. At each of these points, the height from the water-surface to the draft of the given point is calculated. This height is calculated following equation (26). If the height is negative, leakage occurs.

$$\begin{aligned} H_{L_{ijk}}(t) = &- T - \eta_3(t) - y_{ijk}\eta_4(t) \\ &+ x_{ijk}\eta_5(t) + h_{p_i}(t) - \zeta_{ijk}(t) \end{aligned} \tag{26}$$

In the equation above, $H_{L_{ijk}}$ is the height of point k on line j for cushion i, relative to the water plane. T is the draught of the vessel at the initial heave position, $\eta_3 = 0$. Furthermore y_{ijk} and x_{ijk} is the distance from the center of gravity to point ijk. The term h_{p_i} is added to equation (26) as the cushion pressure leakage will be dependent on the height of the water column inside the cushion. The over-pressure inside the cushion will displace water from inside the cushion, such that the water-level inside the cushion will be lower than that on the outside. The term h_{p_i} can be expressed as shown in equation (27), and can simply be explained as a pressure to water-column height conversion.

$$h_{p_i}(t) = \frac{p_{u_i}(t)}{g\rho_w} \tag{27}$$

In equation (26), ζ_{ijk} represents the sea level elevation at the relative position of point ijk. This elevation can be expressed as

$$\begin{aligned} \zeta_{ijk}(t) = \sum_n \Big(&\zeta_{a_n} \sin(\omega_n t - \kappa_n(x_{ijk}\cos(\chi_n) \\ &+ y_{ijk}\sin(\chi_n)) + \epsilon_n)\Big) \end{aligned} \tag{28}$$

Most terms in equation (28) are previously addressed and explained in section 2.1. Note that in the equation above, the letter k is used as a subscript, therefore the letter κ is used to denote the wave number. For an irregular seastate, the specific wave elevation at a point will be the sum of many different regular waves. The elevation for each of these regular waves are expressed as previously explained in equation (5). Therefore, the total wave elevation at point ijk, becomes the sum of the elevation from every regular wave at the given time and the given position of the point. The position of point ijk is expressed by x_{ijk} and y_{ijk}. Lastly, ϵ_n represents the phase of regular wave number n. The total passive cushion leakage area can be expressed as

$$A_{p_i}(t) = A_{p_i}^a(t) + A_{p_i}^c(t) \tag{29}$$

where the superscripts a and c denotes the cushion to atmosphere leakage area, and the cushion to cushion leakage area, respectively. The passive leakage area from cushion to atmosphere can be expressed as

$$A_{p_i}^a(t) = \sum_{j_a=1}^{2}\left(H_{L_{ij_a}}(t)\sum_{k=1}^{10}(\Delta x_{ij_a k} + \Delta y_{ij_a k})\right), \tag{30}$$

where subscript j_a signifies the lines j which represent a wall separating the cushions from the atmosphere. Similarly, the passive leakage area from cushion to cushion, can be expressed as

$$A_{p_i}^c(t) = \sum_{j_c=1}^{2}\left(H_{L_{ij_c}}(t)\sum_{k=1}^{10}(\Delta x_{ij_c k} + \Delta y_{ij_c k})\right), \tag{31}$$

where subscript j_c signifies the lines j separating cushions. The volumetric flow for passive cushion to atmosphere leakage can be expressed in the same way as the controlled leakage in equation (25), by simply substituting the leakage area $A_i^{ctrl}(t)c_n(i)$ with $A_{p_i}^a$.

For the volumetric flow for the inter-cushion leakage, we assume that the pressure in cushion i is higher than the pressure in cushion j. Like before, we also assume no initial air-velocity in i, and that all pressure differential between cushions i and j is converted to air-velocity when the air enters cushion chamber j. The expression formed for v_j in (23) can then be applied. The volumetric airflow between cushions can be expressed by substituting the areas, pressures and densities in equation (25) with the relevant values for the two cushions affected by the leak. The cushion air density ρ_{ci} can be calculated from the expression derived in equation (13).

$$Q_i(t) = A_{p_i}^c(t)c_n\sqrt{\frac{2\gamma}{\gamma-1}\left(\frac{p_{c_i}(t)}{\rho_{c_i}(t)} - \frac{p_{c_j}(t)}{\rho_{c_j}(t)}\right)} \tag{32}$$

Equation (32) expresses the air flow out from cushion i, due to leakage into cushion j. There is no loss of mass in the exchange, and so the corresponding air-flow out from cushion j is the same as in (32), with a negative sign. Note that if the cushion pressure is reversed, that is cushion j holds a higher pressure than cushion i, the i and j subscripts in equation (32) are switched. Also note that the cushion separating walls are considered solid, such that no air can pass though them. However in practice, as previously explained, they are inflatable bags, such that some unknown leakage might occur. The sealing properties of these dividers will not be known until a model is created. To account for this, the model includes the option of adding a percentage, fixed or varying leakage area for each of the cushion dividers.

2.4 Cushion forces

The model is created for stationary or very low vessel speeds. Therefore only the forces in heave, roll and pitch are considered. The forces generated by the pressure in the cushions is calculated following the pressure force relation, $F = pA$. In similar fashion, the moments generated by the cushion pressures is calculated as $M = d_{cp_i}pA$, where d_{cp_i} represents the distance from the vessel center of gravity to the center of pressure in cushion i. Adding the forces and moments induced by each of the cushions, provides the total forces and moments acting on the vessel caused by the cushion pressures. The resulting expression for these forces and moments are given below in equations (33) to (35).

$$F_{3_c}(t) = \sum_{i=1}^{4} -p_0\mu_{u_i}(t)A_{xy_i} \tag{33}$$

$$M_{4_c}(t) = \sum_{i=1}^{4} -y_{cp_i}p_0\mu_{u_i}(t)A_{xy_i} \tag{34}$$

$$M_{5_c}(t) = \sum_{i=1}^{4} x_{cp_i}p_0\mu_{u_i}(t)A_{xy_i} \tag{35}$$

In the equations above, A_{xy_i} is the area of the water surface, mapped onto the x-y-plane. Note that because the positive z-direction is defined as down, and a positive pressure will cause a force working upwards, the heave force, $F_{3_c}(t)$ and the roll moment, $M_{4_c}(t)$ have a negative sign.

The last force caused on the vessel from the cushions is caused by the ventilation of cushion air through the ventilation valves. Following Newton's third law of motion, as the pressurized air escapes out from the ventilation valves in the cushions, an equal and opposite force caused by the velocity and mass of the escaping air, will act on the vessel. This force can be calculated as

$$F = \rho_a A^{ctrl} v_a^2 \tag{36}$$

where A^{ctrl} is the controlled leakage area of the chamber, and v is the velocity of the air. Note that v can also be written as $\frac{Q}{A}$, as these are the units that have mainly been used through the modeling. This generates a force in sway as well as a moment in roll and yaw. As previously mentioned, only the generated moment in roll, which can be described by equation (37), is regarded in this model.

$$\begin{aligned} M_{4_Q}(t) = &-\sum_{i=2}^{3} z_{ca_i}\rho_{c_i}(t)\frac{Q_{out_i}^2(t)}{A_i^{ctrl}(t)} \\ &+ \sum_{i=\{1,4\}} z_{ca_i}\rho_{c_i}(t)\frac{Q_{out_i}^2(t)}{A_i^{ctrl}(t)} \end{aligned} \tag{37}$$

In the equation above, Q_{out_i} is the volumetric flow out of the cushion, as expressed in equation (25). x_{ca_i} and z_{ca_i} is the distance from the vessel center of gravity to the center of area for the cushion ventilation valve, also known as the controlled leakage area, A^{ctrl}. The combined resulting forces acting on the vessel from the cushions and corresponding air-flow can be written as

$$\boldsymbol{\tau_{combined}} = \boldsymbol{\tau_c} + \boldsymbol{\tau_Q}. \tag{38}$$

Here $\boldsymbol{\tau}$ is a vector represented as
$\boldsymbol{\tau} = [F_1, F_2, F_3, M_4, M_5, M_6]$, with the corresponding subscripts of c and Q.

3. SES-SIM

The mathematical model derived in the section above is implemented in MATLAB®/Simulink®. The equations governing the SES-cushion dynamics and the corresponding forces are integrated into a larger system referred to as SES-sim. An illustration of the structure of this system is given in figure 4.

The orange blocks are taken from the Marine System Simulator (MSS), created by Fossen and Perez 2004, with a few additions and changes by Umoe Mandal. The blue square contains a simplified overview of the different equations derived above gathered into modules. The MSS toolbox takes forces acting on the vessel as input, and returns the 6 degrees of freedom movement for the vessel. It is also used to generate the waves and other environmental forces acting on the vessel.

3.1 Simulated pitch movement

Figure 5 shows the results from a pitch induced motion caused by opening and closing the aft and bow ventilation by using two sinusoidal functions in counter-phase.

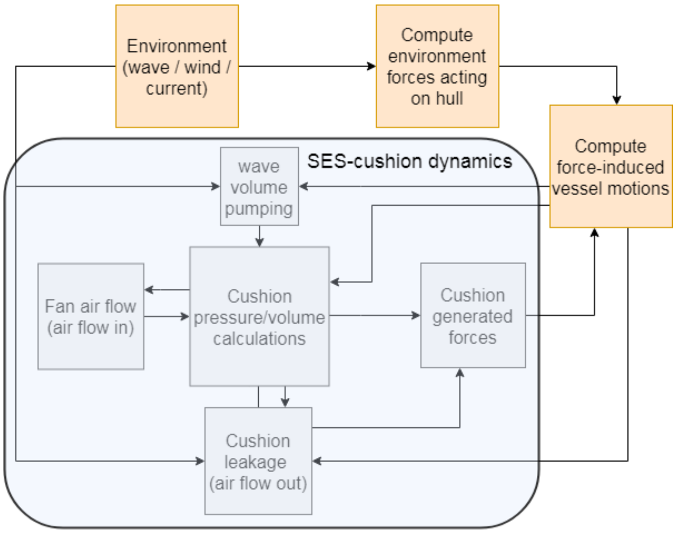

Fig. 4. Diagram of the SES-sim model structure

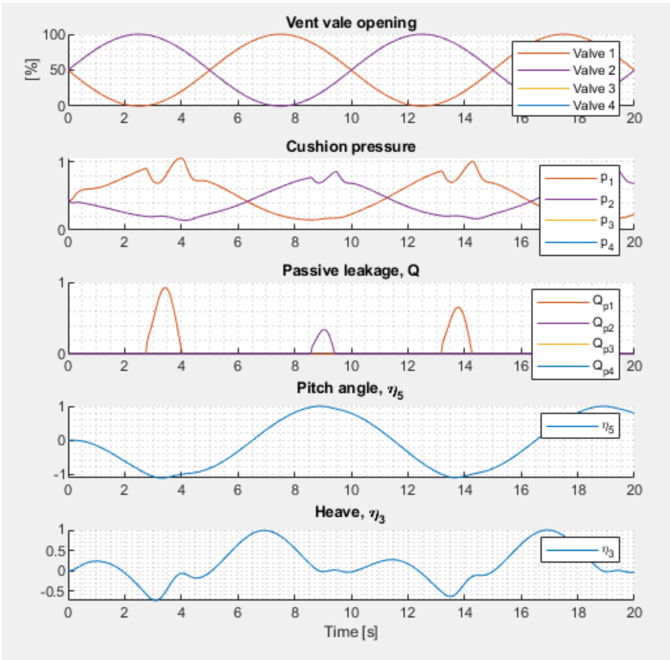

Fig. 5. Vessel dynamics from two sinusoidal vent valve signals

The purpose of the test is to confirm that the mathematical model developed above and its implementation into Simulink® produces a reasonable vessel response. The valves and cushions are numbered as illustrated by figure 2. Opening the aft valves causes a reduced pressure in the aft cushions, which leads to an induced pitch angle. When the pitch angle becomes too large, the front of the vessel is lifted out of the water, and passive air leakage occurs from the front chambers. As expected, this causes a pressure drop in the front cushions. From the graph displaying pitch angle, it's possible to recognize that the pitch velocity changes at this point. Furthermore, as the pressure drops in the front cushions, the vessel heave increases, which is expected as the positive heave direction is defined as down. The following pressure spike can be explained by the vessel bow accelerating downwards due to the pressure loss, and subsequently the cushion being isolated again as the leakage stops.

4. CONCLUSION

The mathematical model seems to provide a good representation of the physical response for a surface effect ship. The exactness of the model seems to be quite dependent on factors regarding the leakage of air from the cushion chambers. Unfortunately these factors are hard to know before any physical testing has been done. Over all, the tests show that a four cushion solution provides a very potent solution to vessel pitch control. Due to the limited length of this paper, and limited experimental data, the focus for model verification has been on pitch. Similar results are expected for roll, with further experimental data planned for near future.

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