Oscillations of Air Cavity Under Hull of Accelerating Boat

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Substantial reduction of ship drag is achievable with air cavities arranged on the hull bottom. However, the air cavities can become unstable, particularly in unsteady environments such as sea waves. Oscillations of cavities can reach high amplitudes, especially near-resonance conditions, and this may lead to the cavity collapse and degradation of drag reduction. In the present study, a lumped-element model is applied for the analysis of dynamics of the air cavity under a boat hull moving in the presence of sea waves. When the hull accelerates from a low initial speed to a high operational speed, the forcing frequency can pass through the cavity resonance condition. Numerical modeling demonstrates that cavities on slowly accelerating hulls may exhibit dangerously high amplitudes. On the other hand, when the hull acceleration is sufficiently fast, then a cavity will not have enough time to develop large-amplitude oscillations. It is found that higher damping and faster acceleration can limit magnitudes of the cavity oscillations, preventing the air cavity from disintegration and enabling the hull to attain high speeds by keeping the drag-reducing air cavity intact. [DOI: 10.1115/1.4054044]

Keywords: air-cavity boats, drag reduction, oscillations, resonance, dynamics and control, modeling, vehicle dynamics and control

1 Introduction

One of the promising methods to reduce ship drag is to use thin air-ventilated cavities on the hull bottom. A configuration suitable for fast planing boats is shown in Fig. 1. The feasible drag reduction on successful air-cavity boats can reach 20–30%, whereas it takes about 2–3% of the propulsive power to supply air into the cavity [1,2]. In addition, underwater radiation of ship-borne noise can decrease due to the presence of air layer between a hull and water [3].

While this concept seems rather straightforward, its broad implementation has not happened yet. One of the problems is possible air cavity instability in realistic sea environments with waves, as opposed to ideal calm water. On unsuccessful air-cavity boats, air cavities tend to disintegrate in some conditions. One of the causes is related to oscillations of the cavity that inevitably occur in the presence of water waves. If oscillation amplitudes become large, the cavity surface would reach the recessed ceiling (hull bottom) and/or extend well below the surrounding hull parts. Both scenarios may lead to excessive loss of air and the cavity collapse. Once the cavity is lost, the ship drag will increase abruptly; and it usually takes a much higher air supply rate to recreate the cavity in comparison with air supply needed to maintain a large stable cavity [4]. Therefore, the understanding of these instabilities and developing means for their prevention are very important steps toward the practical adaption of this energy-saving technology. Somewhat related to the current subject are oscillations of hovercraft and surface-effect ships [5,6], although the collapse of their air cushions is usually not an issue due to much larger volumes of cushions and much higher air supply rates.

The current study focuses on a simplified modeling of oscillations of an air cavity on a boat accelerating from a low speed toward an intended operational speed. A typical dependence of resistance of planing hulls on speeds, as well as a thrust curve, is shown in Fig. 2. The air-cavity hull should be able to attain a higher speed u_a than a speed u_h achievable by a similar hull without air in the cavity. However, if the hull's forward acceleration occurs in wave conditions (Fig. 3), then the air cavity will be affected by oscillating water pressure on the hull bottom. This will make the air cavity volume oscillate as well. If the forcing frequency coincides with the cavity natural frequency, a resonance will occur. In the case of sufficiently large amplitudes of the cavity surface, a significant fraction of air may be lost and the air cavity will collapse. This process is schematically illustrated in the hull resistance graph (Fig. 2) by arrows: first, the drag increase follows the air-cavity-hull curve (arrow 1). After achieving a speed u_r, corresponding to the cavity resonance, the cavity disintegration may take place, which leads to drag increase (arrow 2). Since drag is now above the thrust curve, the boat speed will drop down to the operational state of a hull without air-filled cavity at speed u_h (arrow 3). Thus, an efficient high-speed regime will not be achieved.

The transient nature of the boat forward motion (forward acceleration) also affects the occurrence of the cavity breakdown. If this acceleration is sufficiently high, then large cavity oscillations may not have enough time to develop large amplitudes, since the forcing frequency will move away from the cavity natural frequency once the boat speed increases beyond u_r. In the rest of the paper, a simplified dynamic model for the air-cavity oscillations is presented, and some realistic scenarios are explored.

2 Model

A lumped-element approach, which is commonly used for simplified modeling of acoustic oscillations in enclosures [7], is adopted in the present analysis. A problem schematic is depicted in Fig. 4. The cavity pressure is assumed to be uniform, and the external pressure is also treated as a function of time only. The

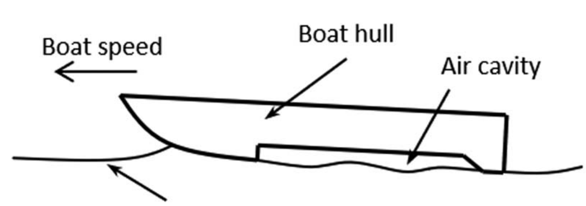

Fig. 1 Schematic of air-cavity boat with bottom recess

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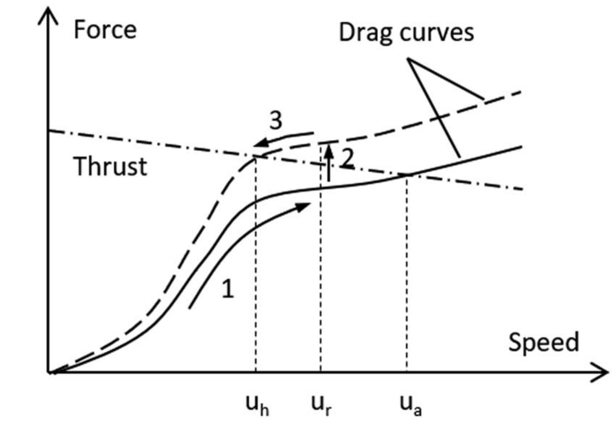

Fig. 2 Typical resistance curves of a hull with air-filled cavity (solid line) and without large air pocket (dashed line). Available thrust is shown by the dash-dotted line. Arrows indicate a possible scenario of losing air cavity due to resonance oscillations.

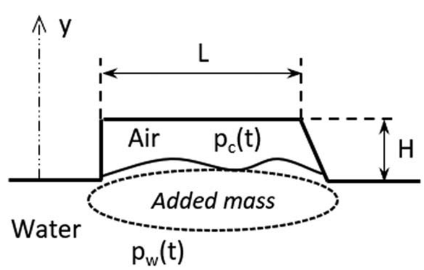

Fig. 3 Lumped-element acoustic model for air cavity

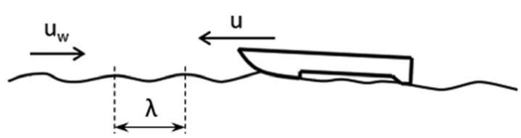

Fig. 4 Boat moving in head waves

dynamics equation for the cavity surface elevation y can be written as follows:

$$M\ddot{y} = p_w(t)A - p_c(t)A - 2\beta M\dot{y} \tag{1}$$

where M is the effective mass, p_w and p_c are the external water pressure in the cavity vicinity and the cavity pressure, respectively, A is the cavity horizontal area, and β is the effective damping coefficient. The effective mass is taken as the added water mass under the cavity with length L and width B [8]

$$M = \frac{\pi}{8}\rho_w L^2 B \tag{2}$$

where ρ_w is the water density. The water pressure is assumed to have a harmonic time-dependent component

$$p_w(t) = p_0 + \rho_w g h_1 \cos(\omega t) \tag{3}$$

where $p_0 = p_{atm} + \rho_w g h$ is the equilibrium pressure, p_{atm} is the atmospheric pressure, g is the gravitational constant, h is the hull draft, h_1 is the pressure amplitude head, and ω is the angular frequency.

The air in the cavity is assumed to oscillate isentropically. Hence, the cavity pressure relates to the volume change as follows:

$$\frac{p_c(t)}{p_0} = \left(\frac{V_0}{V_c(t)}\right)^{\gamma} \tag{4}$$

where V_0 is the initial cavity volume equal to LBH (with H being the equilibrium cavity height), and $V_c(t) = V_0 - yA$ is the cavity volume in the unsteady process. The last term in Eq. (1) is responsible for damping which can include several mechanisms, such as acoustic radiation into the water and the generation of surface water waves.

In the case of low-amplitude cavity oscillations, Eqs. (1) and (4) can be linearized, and the resulting dynamics equation can be presented in a common form

$$\ddot{y} + 2\beta\dot{y} + \omega_0^2 y = F\cos(\omega t) \tag{5}$$

where $\omega_0 = \sqrt{\gamma p_0 LB/(HM)}$ is the natural frequency and $F = \rho_w g h_1 A/M$ is the corresponding amplitude of external forcing. Solutions of Eq. (5) with constant ω are well known [9]. However, sufficiently close to resonance, one would need to use the complete nonlinear equations. The main specific of the considered problem is the variable frequency ω of the external force. For situations considered below, it is assumed that the boat speed u increases in time starting from u_0 up to u_a with a constant acceleration a

$$u = \min(u_0 + at, u_a) \tag{6}$$

In the considered case of head waves, the angular frequency of encounter (or forcing frequency) can be expressed as follows:

$$\omega = 2\pi(u_w + u)/\lambda \tag{7}$$

where λ is the wavelength of surface water waves and $u_w = \sqrt{g\lambda/(2\pi)}$ is the propagation speed of surface waves in deep water. Using the forcing frequency evaluated with Eq. (7), one can numerically integrate nonlinear equation (1) to predict the cavity dynamics. (A linear equation (5) can be used far from resonance conditions at very slow forward accelerations.) If the cavity amplitude reaches a significant fraction of the cavity height, the cavity disintegration and significant air loss will happen, accompanied by a sudden drag increase. It should be noted that the cavity surface is not completely flat even in calm-water sailing. Usually, a system of surface waves (stationary with respect to the hull) is present inside the cavity [10]. Hence, the critical oscillation amplitude Y, at which the cavity collapses, will be smaller than the cavity height H.

3 Results

To illustrate the model outlined aforementioned, a specific cavity configuration and operational conditions are selected with fixed parameters that are listed in Table 1. The resonance frequency of this setup is close to 2 Hz. The damping phenomena that attenuate cavity oscillations may include various mechanisms, such as underwater sound radiation, generation of waves on the water surface, and possibly others. For the selected parameters, the sound radiation losses estimated following [7] can be considered negligible.

Table 1 Parameters of the air-cavity system

Cavity length, L	15 m
Cavity width, B	5 m
Cavity height, H	0.5 m
Hull draft, h	1 m
Initial hull speed, u_0	14 m/s
Final hull speed, u_a	18 m/s
Wavelength of water waves, λ	10 m
Pressure amplitude head, h_1	0.2 m
Critical amplitude, Y	0.25 m

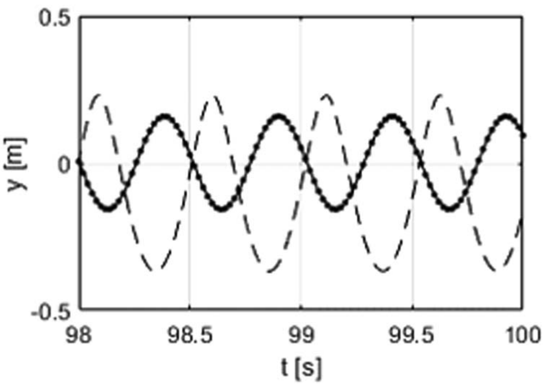

Fig. 5 Limit-cycle oscillations of the cavity surface at $\omega/\omega_0 = 0.98$ with a damping coefficient of 0.1. Solid line, analytical linear solution; dots, numerical linear solution; dashed line, numerical nonlinear solution.

Damping caused by surface wave generation can be roughly evaluated using empirical correlations obtained for air-cushion vehicles [11]. At the given conditions, the damping coefficient β appears to be around 0.1 s^{-1}. Since the considered air cavity is surrounded by substantial hull elements and is submerged deeper into the water, this coefficient may be lower. For illustrative calculations, two values of β are chosen: 0 (no damping at all) and 0.1 s^{-1}. Another variable parameter in this study is the forward acceleration of the hull. Two values are considered, 0.05 and 0.2 m/s^2, corresponding to slow and fast accelerations.

Prior to carrying out simulations with an accelerating hull, both linear and nonlinear dynamics equations (1) and (5) were numerically integrated at fixed forcing frequencies to verify computational solutions. It was found that a time-step of 10^{-4} s produces sufficiently accurate results with the first-order explicit numerical time-stepping. One illustration of the resulting cavity surface oscillations in time is shown in Fig. 5 for a steady-state regime near the resonance frequency. The numerical approach is therefore verified by demonstrating good agreement between the analytical and the numerical solutions. The nonlinear system response in this specific near-resonance regime deviates from the linear solution (Fig. 5).

The steady-state amplitudes are also shown at various frequencies in Fig. 6. For the selected setup, differences between linear and nonlinear models, as well as between systems with small damping and no damping, are noticeable only very close to the resonance. Also plotted in Fig. 6 is the chosen critical amplitude level

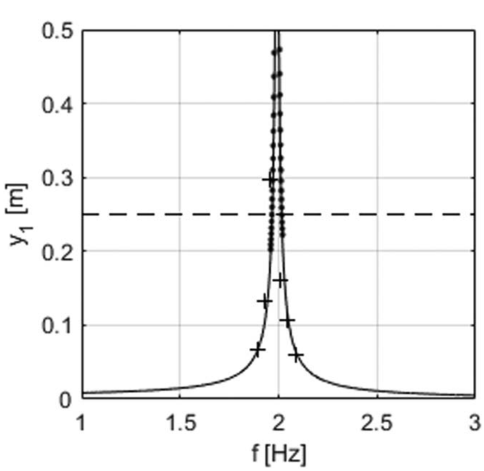

Fig. 6 Limit-cycle amplitudes of the cavity surface elevation. Solid line, linear model with damping; dots, linear model without damping near resonance; crosses, nonlinear model. Horizontal dashed line indicates the selected critical amplitude.

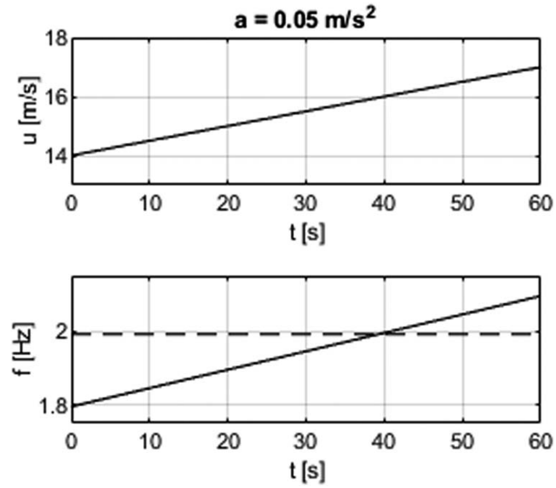

Fig. 7 Time variations of the boat speed and the exciting frequency at low forward acceleration. Dashed line indicates the resonance frequency.

$Y = H/2$, at which the cavity is assumed to disintegrate. This value is lower than the cavity height, because a significant air loss may occur if the cavity extends well below the surrounding hull surfaces or when portions of the cavity surface (which is wavy in reality) will hit the cavity ceiling. Thus, in steady states, only a very narrow range of forcing frequencies would destabilize the cavity.

Next, time integration of the dynamics equation was carried out for cases with accelerating hull. A constant lower acceleration value a = 0.05 m/s^2 was applied first. The time variations of the hull speed and the frequency of the hull encounter with sea waves are shown in Fig. 7. This frequency crosses the resonance frequency at the physical time of about 40 s from the initial state when the boat speed reaches about 16 m/s. Results for the cavity surface oscillations obtained numerically with two damping coefficients

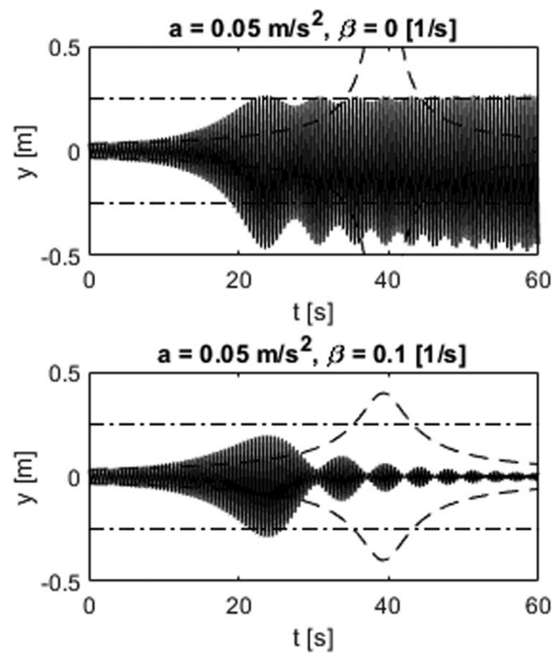

Fig. 8 Solid lines indicate time variations of the cavity surface elevation obtained with the unsteady nonlinear model. Dashed lines indicate amplitudes predicted by the quasi-steady linear model. Dash-dotted lines indicate boundaries of acceptable oscillation magnitudes.

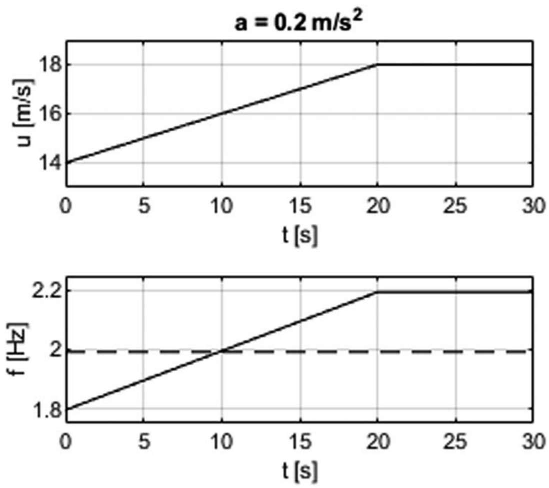

Fig. 9 Time variations of the boat speed and the exciting frequency at high forward acceleration. Dashed line indicates the resonance frequency.

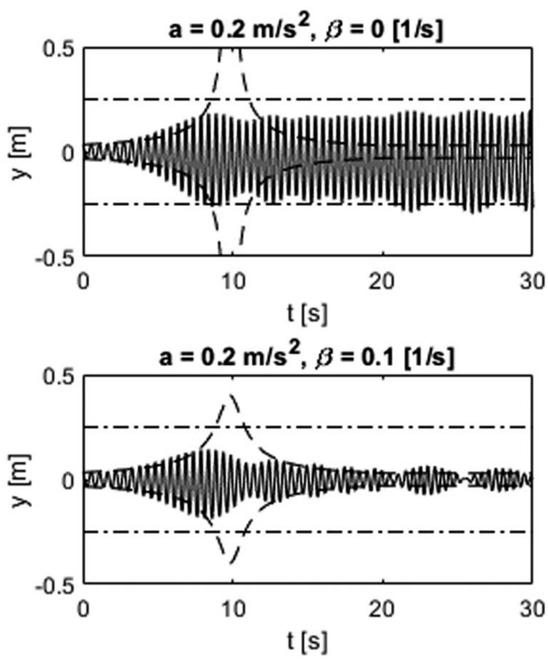

Fig. 10 Solid lines indicate time variations of the cavity surface elevation obtained with the unsteady nonlinear model. Dashed lines indicate amplitudes predicted by the quasi-steady linear model. Dash-dotted lines indicate boundaries of acceptable oscillation magnitudes.

(0 and 0.1 s^{-1}) are given in Fig. 8, where they are also compared with quasi-steady amplitudes of the linear system.

Initially, at a low hull speed far from the resonance, the nonlinear model produces results close to the linear system. However, as the boat accelerates, magnitudes of the nonlinear system grow faster and exceed the critical values even before approaching the resonance frequency. Beating behavior is noticeable in the nonlinear responses. For the case with no damping, the magnitude of oscillations increases even after passing the resonance condition, whereas in the damped case, oscillations attenuate with a further increase of the hull speed. The oscillation amplitudes are symmetric for the linear system (with respect to zero cavity surface position), while significant asymmetry is present in the nonlinear case. Thus, it is expected that a setup with very low damping will certainly result in the cavity disintegration. A strong damping may help avoid the cavity collapse, although in the considered case the cavity is expected to lose a significant amount of air at a time of about 24 s (Fig. 8).

A possible way to prevent occurrences of large amplitudes of the cavity surface oscillations is to sail faster through a resonance zone, so that growing oscillations will not have enough time to reach critical values. (This would of course put more demand on the propulsion system.) To demonstrate such a situation, the hull acceleration of 0.2 m/s^2 was analyzed. The hull speed and frequency of encounter are shown in Fig. 9, which indicate that the resonance condition will happen at about 10 s. Simulation results with the same damping coefficients as before are shown in Fig. 10.

In the case of zero damping, the oscillation amplitudes still exceed the critical values (Fig. 10), but they are noticeably smaller than in a situation with low hull acceleration (Fig. 8). In the presence of damping and with a high acceleration, the oscillations grow relatively little, and their magnitudes do not reach the critical levels (Fig. 10). Thus, this scenario will help avoid the cavity collapse, and the hull will be able to achieve a high operational speed while keeping the air cavity on the bottom.

4 Conclusion

A simple mathematical model has been developed for acoustic oscillations of drag-reducing air cavities under accelerating ship hulls. During the transition from a low initial speed to a high operational speed in the presence of sea waves, the cavity may experience a resonance condition, when external forcing may excite large-amplitude cavity oscillations. This may result in the cavity collapse and the inability of the boat to reach the intended speed. It is found that faster acceleration and higher damping can limit magnitudes of the cavity oscillations. The present model can be extended further by accounting for the nonuniform properties in the cavity, hull motions, and more complex operational scenarios. The model can be used for developing methods for suppressing cavity oscillations, such as passive or active control systems, including acoustic dampers, variable air supply, and morphing hull elements. Experiments with oscillating air cavities below hull surfaces will be of great value for both validation and extension of the presented model.

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Conflict of Interest

There are no conflicts of interest.

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