The Effect of Waves and Ventilation on the Dynamic Response of a Surface-Piercing Hydrofoil

Yin Lu Young, Hyunse Yoon, Tristan Wright, Casey Harwood

1Department of Naval Architecture and Marine Engineering, The University of Michigan, Ann Arbor, MI (USA)
2IIHR - Hydroscience and Engineering, The University of Iowa, Iowa City, IA (USA)

ABSTRACT

Hydrodynamic lifting bodies such as propellers, rudders, and ride control fins of surface vessels are regularly subject to random wave excitations, which induce dynamic load fluctuations and possibly ventilation. Ventilation is defined as entrainment of gas into the low-pressure region around the submerged portion of the body, which could lead to sudden and large changes in hydrodynamic loads, as well as flow-induced vibrations. Moreover, changes in immersion and ventilation will affect the system stability and controllability. Hence, the objective of this work is to study the effects of waves and ventilation on the natural frequencies of the foil through experimental studies. The results show that, in general, the natural frequencies of the foil decrease with increased submergence. Compared to the response in calm water, the frequency responses of the hydrodynamic loads show peaks corresponding to the encountered wave frequency and its harmonics, the foil natural frequencies, and the vortex shedding frequencies. Dynamic load amplification is observed when the harmonics of the wave excitation frequency are near the first natural frequency of the foil. In fully wetted conditions at a submerged ratio of two, the second and third resonant modes practically coalesced, which lead to significant dynamic load amplifications in both calm water and wave conditions. Natural or spontaneous ventilation develops when the yaw angle exceeds 12 degrees. The natural frequencies are higher in fully ventilated flow compared to fully wetted flow due to reduction in added mass. The relative increase in natural frequency in fully ventilated flow compared to fully wetted flow is mode dependent because of the directional-dependency of the added mass. Hence, while the second and third resonance frequencies practically coalesced in fully wetted flow, they separated in fully ventilated flow, which resulted in significant reduction in the amplitude of the dynamic load fluctuations and vibrations compared to the fully wetted case.

INTRODUCTION

Many hydrodynamic lifting bodies (propellers, turbines, rudders, control fins, energy saving devices, and energy harvesting devices) operating at or near the free surface are subject to dynamic load fluctuations caused by waves, alternating partial emersion or immersion, and ventilation. In particular, when a surface vessel is maneuvering in waves, parts of the propeller, rudder, and/or ride control fins may emerge, leading to large reductions in hydrodynamic lift/thrust, which can in turn affect the vessel maneuverability. Thus far, the effects of waves on the propeller performance and hull-propeller-rudder interactions are not yet fully understood (Nakamura & Naito, 1977; Taskar et al., 2016; Reed & Beck, 2016).

Natural ventilation is defined as the entrainment of atmospheric gas into liquid flows, and has profound implications in the design and operation of hydrodynamic lifting bodies. As noted in Breslin & Skalak (1959), Rothblum et al. (1969), Swales et al. (1974), Harwood et al. (2016a,b,c; 2017; 2019a,b), and Young et al. (2017), the flow around a partially or shallowly submerged body can transition between fully wetted (FW), partially ventilated (PV), partially cavitating (PC), and fully ventilated (FV) flow regimes, which can lead to dynamic load fluctuations and flow-induced vibrations. In particular, sudden and unexpected transition from FW to PC flow to FV flow can lead to large drops in lift/thrust, which may require voluntary speed reduction to avoid propeller racing and engine damage. Previous experiments have shown that free surface waves can promote ventilation of partially submerge bodies (McGregor et al., 1973). Experimental studies of highly loaded and shallowly submerged propellers by Koushan et al. (2009) demonstrated large drops in propeller thrust and torque due to ventilation, and the standard deviation of the dynamic loads increased significantly when operating in waves.

The fluid-structure interaction response and stability of hydrodynamic lifting bodies are also affected by emergence, ventilation, and waves. The experimental results in Harwood et al. (2016a,c; 2017; 2019a,b) demonstrated large drops in propeller thrust and torque due to ventilation, and the standard deviation of the dynamic loads increased significantly when operating in waves.

*Corresponding author. E-mail: ylyoung@umich.edu, Tel.: +1-7346470249.
showed that increasing immersion of surface-piercing bodies will result in reduction of the modal frequencies due to increased fluid added mass. The magnitude of decrease is dependent on the mode shape, relative flow speed, and presence of cavitation and/or ventilation (Harwood et al., 2017, 2019a,b; Young et al., 2016, 2017). Moreover, changes in added mass can lead to mode switching and frequency coalescence, which occurs when two modes (typically one bending and one twisting) share the same natural frequency. Coalescent modes can lead to flutter instability and even catastrophic structural failure. Thus, the objective of this work is to study the effects of waves and ventilation on the dynamic response of a canonical hydrodynamic lifting body through experiments at the University of Michigan.

EXPERIMENTAL SETUP

Experiments were conducted in the Physical Model Basin (PMB) at the Marine Hydrodynamics Laboratory at the University of Michigan. The PMB has dimensions of 110 m length by 3.2 m depth by 6.7 m width (360 ft × 10.5 ft × 22 ft), and the towing carriage is capable of speeds up to 6.1 m/s (20 ft/s). Additional tests were performed with the hydrofoil mounted to a freestanding test frame, with the tip immersed in a large drum filled with water.

The foil was chosen as a canonical proxy to more-complex hydrodynamic lifting bodies such as propeller blades, rudders, and ride control fins. The foil was constructed of PVC with an aluminum strip 0.6 cm thick × 2.79 cm wide (0.25 in × 1.1 in) affixed to the blunt trailing edge (TE) to increase the bending rigidity and to shift the elastic axis towards the TE to increase the flow-induced twist deformations. A drawing of the foil cross sectional details, including the relevant dimensions, is shown in Figure 1a. Details of the foil instrumentation and coordinate system are shown in Figure 1b.

Photographs of the hydrofoil setup in the PMB, including close-up views from behind the foil when operating in FW and FV flows, are shown in Figure 2. The hydrofoil was vertically cantilevered, hanging below a steel box frame and piercing the water surface to a prescribed depth of h. The angle of attack, \( \alpha \) (identically referred to as the yaw angle), shown in Figure 3, was set by rotating the hydrofoil inside of the frame and clamping it in place. Forcing conditions on the hydrofoil were measured using a 6-DOF load transducer (ATI Omega 190 load cell, with an estimated measurement uncertainty of ±2.6%). During post-processing, the load transducer signals were filtered using a Butterworth digital low-pass filter with a cutoff frequency of 50 Hz, in order to minimize the broadband noise from the vibrations of the carriage.

![Figure 1](image.png)
Figure 2. Photographs of the mounted hydrofoil in the Physical Model Basin at the University of Michigan. The top two photos show the hydrofoil in FW conditions, while the bottom photo shows the close-up view from behind the foil when operating in FV condition.

The instantaneous carriage speed was measured via a wheel-mounted optical rotary encoder with a 0.28 mm/pulse linear resolution. The operating principle is integer pulse counting at the encoder to obtain the linear distance as a function of time, and the speed is calculated by numerical differentiation of the distance function with time. A total of six SENIX® TSPC-30S1-232 ultrasonic type wave probes (with an estimated uncertainty of ±0.5%) were used to measure the free surface elevations around the hydrofoil. Figure 3 shows the location of each wave probe relative to the hydrofoil’s midchord position.

All the data were digitized and recorded via hardware synchronized A/D converters with rates of 1000-2000 Hz and at 16-bit resolution. All the tests at the PMB were conducted in an ambient pressure environment, and the water temperature was measured to vary between 15.0 and 16.0 degrees Celsius during the test campaign. Video was recorded at a resolution and rate of 1920 x 1080 x 60Hz, both above and below the waterline, using GoPro® Hero® model cameras.

A series of experiments in FW, PV, and FV flows were previously conducted in calm water conditions, the results of which have been reported in Harwood et al. (2016b,c) and in Harwood (2016). Experiments using the same hydrofoil as in the present work were also conducted in FW, PC, and FV conditions in the free surface cavitation channel at CNR INM (formerly INSEAN) in Rome, Italy. The CNR INM results are reported in Harwood et al. (2017; 2019a,b) and in Harwood (2016). In these previous works, only calm water conditions were examined. In 2018, another series of experiments were conducted at the PMB at the University of Michigan, which were specifically designed to study the influence of waves and ventilation on the dynamic response and stability of the flexible surface-piercing hydrofoil. This latter-most experimental campaign is the topic of the present paper.

A summary of the experimental conditions used in the present work is provided in Table 1. For each immersed aspect ratio, tests were conducted at angles of attack between five and twenty degrees. The depth-based Froude number, $F_{n_h}$, was varied between zero and three, and the immersed aspect ratio, $AR_h$ was varied between 1.0 and 2.0. $F_{n_h}$ and $AR_h$ are respectively defined in Equations (1) and (2).

**Figure 3.** Wave probe (WP) locations (shown not to scale) around the hydrofoil model. WP₁, WP₂, and WP₃ are located 6.5 c upstream and WP₄, WP₅, WP₈ are 16.5 c downstream of the foil mid-chord location. Transversely, WP₃ and WP₇ are aligned with the hydrofoil centerline when $\alpha = 0^\circ$, whereas WP₁ is 1.8 c, WP₂ is 5.6 c, and WP₄ and WP₈ are 4 c apart from the foil centerline.

$$F_{n_h} = \frac{U}{\sqrt{gh}}$$

$$AR_h = \frac{h}{c}$$
\[ F_{h} = \frac{v}{\sqrt{gh}} \]  
\[ AR_{h} = \frac{h}{c} \]

Here, \( h \) denotes the immersed depth of the foil’s free tip, \( c \) is the chord-length of the foil, \( v \) is the forward speed of the foil, and \( g \) is gravitational acceleration.

**Table 1.** Summary of test matrix conducted at the Physical Model Basin (PMB) at the University of Michigan in 2018. Experiments covered a range of immersion levels represented by the submerged aspect ratio \( AR_{h} \), angles of attack represented by \( \alpha \), speeds represented by the immersion-depth based Froude number \( F_{h} \), as well as regular wave conditions represented by the wave frequency \( f_{w} \) and wave amplitude \( A_{w} \). Also shown are the wave length to submerged depth ratio, \( \lambda_{w}/h \), and wave amplitude to wave length ratio, \( A_{w}/\lambda_{w} \).

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Test conditions</th>
</tr>
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<tbody>
<tr>
<td>( AR_{h} )</td>
<td>1.0, 1.5, 1.8, 2.0</td>
</tr>
<tr>
<td>( \alpha ) [(^{\circ})]</td>
<td>5, 10, 15, 20</td>
</tr>
<tr>
<td>( F_{h} )</td>
<td>0, 0.5, 1.0, 1.5, 2.0, 2.5, 3.0</td>
</tr>
<tr>
<td>( f_{w} ) [Hz]</td>
<td>0.067, 1.33, 1.49</td>
</tr>
<tr>
<td>( A_{w} ) [cm]</td>
<td>0, 3, 5, 6.5, 10</td>
</tr>
<tr>
<td>( \lambda_{w}/h )</td>
<td>1.23 – 12.17</td>
</tr>
<tr>
<td>( A_{w}/\lambda_{w} )</td>
<td>0 – 0.14</td>
</tr>
</tbody>
</table>

To study the effects of waves on the dynamic response, waves were generated in the PMB using a computer controlled plunger-type wave maker capable of forming regular or irregular waves with periods, \( T_{w} \), between 0.5 and 2.0 seconds and with amplitudes, \( A_{w} \), up to 0.46 m. Due to limited tank time, wave cases were limited to regular waves with periods (\( \lambda_{w} \)) of 0.75 or 1.5 seconds, and commanded amplitudes between 0 and 10 cm.

To limit the scope of this article, reported results are limited to flow conditions of \( AR_{h} = 2.0 \) and \( F_{h} = 1.5 \), with commanded wave amplitudes of 0 cm (calm water), 3 cm, and 5 cm and a fixed wave period of 1.5 seconds. Both reported wave conditions corresponded to deep water, non-breaking regular waves with \( \frac{\lambda_{w}}{h} = 6.15 \) and \( \frac{\lambda_{w}}{h} = 0.053 \) or 0.088, where \( h \) is the submerged depth of the foil. The water depth is 3.2 m. The wave length to chord ratio is \( \frac{\lambda_{w}}{c} = 12.29 \).

Experimental results indicated that for \( AR_{h} = 2 \) and \( F_{h} = 1.5 \) in calm water, the boundary

for spontaneous ventilation was approximated by a critical yaw angle of \( \alpha_{c} = 12.5^{\circ} \). Hence, for \( \alpha < \alpha_{c} \), the flow was fully wetted (FW); for \( \alpha > \alpha_{c} \), the flow was fully ventilated (FV). To study the effect of waves for FW and FV cases, results are reported here for yaw angles of \( \alpha = 5^\circ \) (FW) and \( \alpha = 20^\circ \) (FV).

For each test in waves, the wave maker was started forty seconds prior to moving the carriage so that the waves could travel down the tank into the hydrofoil’s range of steady-state operation. The wave maker continued generating waves until the carriage came to a rest. For the range of conditions shown in Table 1, the number of encountered waves varied between 12 to 36 cycles depending on carriage velocity and wave frequency.

**IN SITU DEFLECTION MEASUREMENTS**

The *in situ* deflections of the hydrofoil were measured using custom-built shape-sensing (SS) spars embedded inside the hydrofoil, as shown in Figure 1. The measurement method operated as part of a kinematic model, and has been described previously (Harwood et al., 2016a,c; Ward et al., 2018; Di Napoli et al., 2018). An SS spar consists of a \( t = 3/16 \) in (0.47 cm) thick aluminum beam, to which are affixed four strain gauge half-bridges to measure pure bending strain with an approximate half-cosine spacing along the length of the spar. A fourth-order polynomial is fitted to the measured strains (with a zero-strain condition imposed at the free tip). The polynomial approximation to the surface bending strain \( (\varepsilon_s) \) is then integrated twice according to the linearized beam bending equation,

\[ \varepsilon_{s} = -\frac{t}{2} \frac{\partial^2 Y}{\partial Z^2} \]

where \( Y \) is the lateral deflection of the beam, and \( Z \) is the coordinate along the beam axis from the fixed root.

Boundary conditions are imposed at the root, consisting of a zero-displacement constraint and a torsional spring model to account for slight compliance of the PVC at the clamped root. The result is a polynomial approximation to the bending displacement of the spar.

Two such spars are embedded inside of the hydrofoil, as shown in Figure 1, yielding two non-collinear reconstructions of bending displacements for each foil section along the span. Assuming chordwise rigidity, these measurements are transformed into sectional bending and twisting deflections. Calibration and characterization of the spars demonstrated that
bending and twisting measurements has RMS error bounds of 0.9 mm and 0.25 degrees, respectively, making the method competitive with camera-based approaches such as digital image correlation (DIC). The advantage of the SS spar, however, is that the spar requires no optical access to the test specimen, which is critical for in-situ measurements in multiphase flows. The kinematic model can also be run in near real-time to observe static and dynamic deflections in-situ, including the visualization of operating deflection shapes (ODSs) in the frequency-domain – a very useful capability for on-line discovery of approximate mode shapes during vibration testing and for in-situ health monitoring.

**IN SITU MODAL IDENTIFICATION**

Modal parameters, including undamped natural frequencies, damping ratios, and non-reciprocal mode shapes, were identified from experimentally measured frequency response functions (FRFs). Compliance FRFs were computed from displacements measured by the shape-sensing spars. Inertance FRFs were computed from measured tip accelerations from two single-axis accelerometers placed near the end of each shape sensing spar. In all cases, spectra were segmented into 8 to 32 windowed segments for smoothing. Transfer functions were fitted to FRFs and decomposed in terms of a partial fraction expansion to yield separate modal FRF contributions (Richardson and Formenti, 1982). This application of experimental modal analysis (EMA) to FSI has been described in (Harwood, 2016; Harwood et al, 2016a;c; Harwood et al, 2017; Ward et al, 2016, 2018) and in forthcoming journal articles (Harwood, et al., 2019 a,b).

Figure 4 shows an example of multi-degree of freedom (MDOF) identification on closely coupled modes (modes 2 and 3) in dry conditions. The inertance (acceleration) FRFs are shown for two degrees of freedom, each of which are decomposed into modal contributions, shown as blue and green lines. Each identified mode is re-cast as a single-degree of freedom (SOF) dynamical model, represented in the Argand plane by a circle that passes through the origin. Each is characterized by a residue and a pole; the latter defining the undamped natural frequency and damping ratio of the respective mode, while the former interpreted as a mode-shape “participation” scalar. Undamped natural frequencies and damping ratios are global quantities, independent of the DOF on which they are measured, so independent estimates on each compliance and inertance FRF for a given mode yields a population of independent estimates, from which means and standard deviations are calculated.

**VARIATION OF THE NATURAL FREQUENCIES WITH SUBMERGENCE**

Figure 5 shows the measured modal frequencies for modes 1, 2, and 3 as functions of the immersed aspect ratio. The fourth mode, which is a lead-lag mode, was not measured, due to rejection of purely axial strains by the shape sensing spars. The data were collected with the hydrofoil’s tip immersed to increasing depths in a large water-filled drum. All of the modal frequencies decrease as the foil is more deeply immersed, indicating that the fluid inertia becomes increasingly important as a greater proportion of the hydrofoil is submerged. Fluid inertial forces arise from the local accelerations of the fluid-structure interface, so the cumulative modal added mass depends strongly upon the mode shape. Thus, each mode is affected differently by the partial immersion.
The natural frequencies were also measured in the PMB at the University of Michigan, as well as in the variable pressure free surface cavitation channel at CNR INM (part of a separate experimental campaign not described in this paper), for cases with forward speed in both FW, FV, PC, and PV conditions (Harwood et al., 2016a,c; 2017, 2019a,b; Harwood, 2016). The results show negligible difference between natural frequencies in still water condition between the various facilities. A slight variation of the FW natural frequencies with forward speed was observed (Harwood et al., 2017, 2019a,b; Harwood, 2016). As shown in Figure 5, for ARh ≥ 2 and in FW still water conditions, modes 2 and 3 coalesce. Such coalescence can lead to flutter, which typically happens when two modal frequencies merge at high speeds while the damping of one of the modes goes to zero. The total damping was measured to be greater than zero for all cases in the present study, however, so coalescence did not lead to flutter. Details of the measured structural and hydrodynamic damping for varying flow regimes will be discussed in upcoming journal papers (Harwood et al. 2019a,b).

The natural frequencies are higher in FV flow compared to FW flow due to reduction in added mass caused by replacement of water with gas on the suction side and free tip of the foil. Experimental measurements of the natural frequencies in FV flow are shown as open symbols in Figure 5 for ARh = 1.0 only. From the CNR INM data, an empirical model has been developed to estimate the FV natural frequencies from measured FW natural frequencies by rescaling of the fluid inertia and disturbing force. A simplified version of the model, which neglects the fluid disturbing force is given as,

$$\omega_0|_{FV} \approx \omega_0|_{FW} \sqrt{\frac{a_{FW}+1}{a_{FW}+1}},$$

where

$$a = \frac{m_{fluid}}{m_{solid}}$$

is the modal added mass coefficient, estimated from the correlation,

$$a = \begin{cases} 
C_1 & \text{FW Flow} \\
C_1 - C_2 \left( \frac{1}{\alpha F_N} \right) C_3 & \text{FV Flow}.
\end{cases}$$

Note that to ensure static stability and to avoid material failure, the fluid disturbing force is typically limited to be much less than the structural elastic restoring force, which justifies the simplification. The coefficients C1,2,3 in Equation (5) are given in Error! Reference source not found.. These empirical correlations were used to predict FV natural frequencies, shown as dashed lines in Figure 5 for Froude numbers of Fn_h = 1.5, 2.0, & 2.5 with an assumed angle of attack of α = 15°. More details of this correlation are to be published in forthcoming journal articles (Harwood, et al., 2019a,b). The correlation correctly predicted the FV frequencies at ARh = 1.0, but data are lacking for other immersion depths, so the validity is difficult to assess for ARh ≠ 1.0. Presently, the correlation accounts for changing immersion depth only through the indirect effect that it has upon the Froude number, as Fn_h \(\propto h^{-1/2}\) at a fixed speed. Interestingly, modes 2 and 3 are predicted to reverse order in FV flow for ARh ≥ 1.75. This is a meaningful result, as it suggests that the modes 2 and 3 frequencies may separate in FV flow, which has important implications on the dynamic response of the hydrofoil, as will be demonstrated later in this paper. Though not shown here, the hydrodynamic damping is also different in FV flow compared to FW flow (Harwood et al. 2019a,b).

**Figure 5.** Undamped modal frequencies for mode 1 (first bending), mode 2 (first twisting), and mode 3 (second bending + twist). FW data obtained in a quiescent container of water. FV frequencies at ARh = 1.0 were obtained during a separate experimental program at CNR INM. Predictions of FV frequencies are made from Equations (4) and (5) for three values of Fn_h and a yaw angle of α = 15°. Modes 2 and mode 3 appear to coalesce together in FW flow when ARh ≥ 2. Equation 4 predicts that modes 2 and 3 will reverse order in FV flow for ARh ≥ 1.75. In addition, modes 2 and 3 will separate in FV flow.
Table 2. Correlation constants for Equation (5). The correlation was produced from experimental measurements at $AR_h = 1.0$.

<table>
<thead>
<tr>
<th>Mode</th>
<th>$C_1$</th>
<th>$C_2$</th>
<th>$C_3$</th>
</tr>
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<tbody>
<tr>
<td>1</td>
<td>2.55</td>
<td>0.0106</td>
<td>-1.11</td>
</tr>
<tr>
<td>2</td>
<td>1.12</td>
<td>0.0134</td>
<td>-0.887</td>
</tr>
<tr>
<td>3</td>
<td>0.599</td>
<td>4.3 x 10^{-5}</td>
<td>-1.84</td>
</tr>
<tr>
<td>4</td>
<td>0.428</td>
<td>6.4 x 10^{-4}</td>
<td>-1.29</td>
</tr>
<tr>
<td>5</td>
<td></td>
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</tr>
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EFFECTS OF WAVES ON THE DYNAMIC HYDROELASTIC RESPONSE

To keep the paper succinct, results are shown in this section for $AR_h = 2$ and $Fn_h = 1.5$ only. The measured time histories of the carriage velocity ($V_c$) and incident wave profile are shown in Figure 6. The incident wave free surface elevation is measured at wave probe 3, which is located at 6.5c upstream of the hydrofoil’s midchord position, as shown in Figure 3. Figure 6(a) corresponds to commanded monochromatic incident waves with frequency of $f_e = 0.68$ Hz and amplitude of $A_w = 5$ cm; the corresponding encountered wave frequency at $\alpha = 5^\circ$ is $f_e = 1.73$ Hz. Figure 6(b) corresponds to commanded monochromatic incident waves with frequency of $f_e = 0.68$ Hz and amplitude of $A_w = 5$ cm; the corresponding encountered wave frequency at $\alpha = 20^\circ$ is $f_e = 1.67$ Hz. Note that the depth-based Froude number, $Fn_h$, is defined using the target steady-state speed $U$. A summary of the incident wave and encountered frequencies, as well as natural frequencies in fully wetted (FW) and fully ventilated (FV) conditions are summarized in Table 3. Also shown in Table 3 is the measured vortex shedding frequency, $f_{vs}$, at $\alpha = 5^\circ$ and $20^\circ$. As noted in Harwood et al. (2019a), the blunt trailing edge lead to formation of von Kármán vortex streets in the hydrofoil wake. The vortex shedding frequency increased linearly with speed, and was found to yield a constant Strouhal number of $St_c = \frac{f_{vs}T}{U} = 0.275$, where $T$ is the foil trailing edge thickness.

Snapshots of underwater videos taken by the GoPro° cameras of the hydrofoil at $AR_h = 2$, $Fn_h = 1.5$, and $\alpha = 5^\circ$ in calm water condition is shown in Figure 7(a), and in waves with $f_e = 0.68$ Hz and $A_w = 3$ cm is shown in Figure 7(b). The corresponding time histories of the measured lift coefficient ($C_l$), drag coefficient ($C_d$), and moment coefficient ($C_m$) are shown in Figures 8(a) and 8(b). The hydrodynamic load coefficients are non-dimensionalized using the steady-state speed $U$ and the submerged projected foil area $hc$.

To illustrate the influence of waves, the steady-state values from the calm water case in Figure 8(a) are shown as dashed lines in Figure 8(b) for the wave case. The black and orange boxes indicate the portion of the time histories used in the FFT analysis for the calm water and the wave cases, respectively. The resulting frequency spectra of the hydrodynamic load coefficients are shown in Figures 9(a) and 9(b). The time histories and frequency spectra of the corresponding measured tip bending ($\delta_{tip}$) and twisting ($\theta_{tip}$) deflections are shown in Figures 10 and 11, respectively.

For $AR_h = 2$, $Fn_h = 1.5$, and $\alpha = 5^\circ$ in both calm water and wave conditions, the flow is fully wetted, as shown in Figure 7. The results in Figure 8 suggest that the mean lift and drag coefficients in the steady-state region are approximately the same in calm water and in wave conditions, although the mean moment coefficient appears to be slightly lower in wave condition. The latter is probably caused by wave effects on the fluctuations of the size of the separated region on the suction side of the hydrofoil, which have a nonlinear effect upon the moment due to the change in center of pressure. Note that while flow separation on the suction side could not be observed in Figure 7, flow visualization using paint streak method did confirm the existence of a separated region in calm water condition of the surface-piercing hydrofoil (Harwood et al., 2016b).

Table 3. Measured incident wave frequency $f_w$ and encountered frequency $f_e$, as well as vortex shedding frequency $f_{vs}$ behind the blunt foil trailing edge. Also shown are the measured modal frequencies $f_{nFW}$ for FW regime and $f_{nFV}$ for FV regime for the first three modes, $n = 1, 2, 3$, at $AR_h = 2$ and $Fn_h = 1.5$. Experimentally determined modal frequencies are given for the FW regime, while empirically calculated modal frequencies are shown for the FV regime, estimated with Equations (4) and (5).

<table>
<thead>
<tr>
<th></th>
<th>FW ($\alpha = 5^\circ$)</th>
<th>FV ($\alpha = 20^\circ$)</th>
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<tbody>
<tr>
<td>$f_w$ [Hz]</td>
<td>0.68</td>
<td>0.68</td>
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<tr>
<td>$f_e$ [Hz]</td>
<td>1.73</td>
<td>1.67</td>
</tr>
<tr>
<td>$f_{1FW}$ [Hz]</td>
<td>3.56</td>
<td>3.99</td>
</tr>
<tr>
<td>$f_{1FV}$ [Hz]</td>
<td>3.99</td>
<td>3.99</td>
</tr>
<tr>
<td>$f_{2FW}$ [Hz]</td>
<td>26.84</td>
<td>29.73</td>
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<td>$f_{2FV}$ [Hz]</td>
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<td>$f_{3FW}$ [Hz]</td>
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<tr>
<td>$f_{vs}$ [Hz]</td>
<td>34.96</td>
<td>34.98</td>
</tr>
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</table>
Figure 6. Time histories of the carriage speed $V_C$ and free surface elevation $\zeta$ for $AR_h = 2.0$ and $Fn_h = 1.5$, in waves of frequency 0.68 Hz and amplitude $A_w$ of (a) 3 cm and (b) 5 cm.

Figure 7. Snapshot from GoPro® videos of the hydrofoil at $AR_h = 2.0$, $Fn_h = 1.5$, and $\alpha = 5^\circ$ (a) in calm water, and (b) in waves with $f_w = 0.68$ Hz and $A_w = 3$ cm. Both photos show the suction surface of the hydrofoil in FW flow.
Figure 8. Time histories of $C_D$, $C_L$, and $C_M$ of the hydrofoil at $AR_h = 2.0$, $Fn_h = 1.5$, and $\alpha = 5^\circ$ shown in Figure 9(a), the frequency spectra of the hydrodynamic loads have no distinct peaks for frequencies less than 10 Hz. For the lift and moment coefficients, $C_L$ and $C_M$, the strongest peak is observed near 27 Hz, which is where the second ($f_{2,FW}$) and third ($f_{3,FW}$) natural frequencies practically coincide in FW flow, as shown in Figure 5 and Table 3. The coalescence of the second and third modes is believed to be responsible for dynamic load amplification observed in Figure 8, presumably excited by broad-band vibrations that occurred during operation of the towing-tank carriage.

For the wave case shown in Figure 9(b), the most significant peak is observed at the encountered frequency, $f_e$, which is expected. The second most significant peak remains near 27 Hz, where $f_{2,FW}$ and $f_{3,FW}$ practically coincide in FW flow. Additional peaks can also be observed at the second and third harmonics of the wave encountered frequency, $2f_e$ and $3f_e$. Note that $2f_e$ is very near the first natural frequency in FW condition, $f_{1,FW}$, which explains the noticeable peak near 3.6 Hz in the wave case, which is absent in the calm water case.

High-frequency, large amplitude dynamic load fluctuations can be observed in the steady-speed region of Figure 8 for both the calm water and wave cases, which is suspected to be due to frequency coalescence of the second and third mode. This behavior will be explained later via the frequency spectra. For the wave case, the encountered wave frequency of $f_e = 1.73$ Hz (0.58 s) is also clearly visible once the waves arrived in the steady-speed portion of load time history in Figure 8(b). The encountered wave frequency reduces in the deceleration region due to reduction in carriage velocity.

For the calm water case with $AR_h = 2.0$, $Fn_h = 1.5$, and $\alpha = 5^\circ$ shown in Figure 9(a), the frequency spectra of the hydrodynamic loads have no distinct peaks for frequencies less than 10 Hz. For the lift and moment coefficients, $C_L$ and $C_M$, the strongest peak is observed near 27 Hz, which is where the second ($f_{2,FW}$) and third ($f_{3,FW}$) natural frequencies practically coincide in FW flow, as shown in Figure 5 and Table 3. The coalescence of the second and third modes is believed to be responsible for dynamic load amplification observed in Figure 8, presumably excited by broad-band vibrations that occurred during operation of the towing-tank carriage.

For the wave case shown in Figure 9(b), the most significant peak is observed at the encountered frequency, $f_e$, which is expected. The second most significant peak remains near 27 Hz, where $f_{2,FW}$ and $f_{3,FW}$ practically coincide in FW flow. Additional peaks can also be observed at the second and third harmonics of the wave encountered frequency, $2f_e$ and $3f_e$. Note that $2f_e$ is very near the first natural frequency in FW condition, $f_{1,FW}$, which explains the noticeable peak near 3.6 Hz in the wave case, which is absent in the calm water case.

Note that the frequency spectra for the drag coefficient is magnified by 10 times in Figure 9 to better showcase the peaks. For both the calm water and wave cases, significant peak can be observed near the vortex shedding frequency ($f_{vs}$) of 35 Hz. The encountered frequency can also be observed in the drag coefficient in the wave case. The other peaks in the drag coefficient spectra correspond to $f_{vs}/2 = 17.5$ Hz, and the subharmonic of the lead-lag or 4th mode natural frequency, which is $93.3/2 = 47$ Hz obtained based on finite element predictions. The drag acts parallel to the inflow, and hence is most sensitive to streamwise changes, i.e. lead-lag mode vibrations and vortex shedding from the blunt trailing edge.

The time histories and frequency spectra of the tip bending and twisting deflections shown in Figures 10 and 11 follow the same trends as the lift and moment coefficients, respectively, of Figures 8 and 9, indicating a predominantly linear elastic structural response. Similar to the moment coefficients shown in Figure 8, the mean value of the twisting deflection in the steady state region is lower in waves compared to in calm water, as illustrated in Figure 10. Although the time histories of the foil bending and twisting seem less noisy than the lift and moment coefficients, the
trend and the location of the peaks of the frequency spectra are very similar.

Overall, the results in Figures 8-11 suggest that frequency coalescence between the second and third mode near 27 Hz is responsible for the significant dynamic amplifications observed in the time histories of the lift and moment coefficients, and tip bending and twisting deformations.

Snapshots of underwater videos taken by the GoPro cameras of the hydrofoil at $AR_h = 2.0$, $Fn_h = 1.5$, and $\alpha = 20^\circ$ in calm water condition are shown in Figure 12(a), and in waves with $f_w = 0.68$ Hz and $A_w = 5$ cm is shown in Figure 12(b). The corresponding time histories of the measured lift, drag, and moment coefficients are shown in Figures 13(a) and 13(b). The frequency spectra of the hydrodynamic load coefficients in calm water and in waves are shown in Figures 14(a) and 14(b), respectively. The time history and frequency spectra of the measured tip
bending ($\delta_{tip}$) and twisting ($\theta_{tip}$) deflections are shown in Figures 15 and 16, respectively.

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{figure11}
\caption{Frequency spectra of the tip bending $\delta_{tip}$ and twisting $\theta_{tip}$ deflections of the hydrofoil at $AR_h = 2.0$, $Fn_h = 1.5$, and $\alpha = 5^\circ$ (a) in calm water and (b) in waves with $f_w = 0.68$ Hz and $A_w = 3$ cm.}
\end{figure}

For $AR_h = 2$, $Fn_h = 1.5$, and $\alpha = 20^\circ$ in both calm water and wave conditions, the flow is fully ventilated in the steady-speed region, as shown in Figure 12. Compared to the results shown in Figure 8 for $\alpha = 5^\circ$, the slope of the hydrodynamic load coefficients for $\alpha = 20^\circ$ are nonlinear in the carriage acceleration and deceleration stages due to transition from FW to FV flow and from FV to FW flow, respectively. Note that the time histories during the acceleration and deceleration stages are not symmetric because hysteresis delayed flow re-attachment during deceleration (Harwood, et al. 2016b).

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{figure12}
\caption{Snapshot from GoPro video of the hydrofoil at $AR_h = 2.0$, $Fn_h = 1.5$, and $\alpha = 20^\circ$ (a) in calm water, and (b) in waves with $f_w = 0.68$ Hz and $A_w = 5$ cm. The photographs confirm that the hydrofoil is in FV flow regime.}
\end{figure}

The hydrodynamic lift and drag coefficients are much higher for $\alpha = 20^\circ$ than for $\alpha = 5^\circ$. The moment coefficients are approximately the same between the two cases. In FV flow, the center of pressure/lift shifts towards the midchord, which significantly reduces the moment arm between the
elastic axis and the center of pressure, and counteracts the increased magnitude of the lift. The drag coefficient is much higher for \( \alpha = 20^\circ \) than for \( \alpha = 5^\circ \) because additional energy is dissipated through separation of the flow and generation of the ventilated cavity closure line to the ventilated cavity wall in the wake.

Comparison of Figures 13 and 8 indicate that the relative amplitude of the dynamic load fluctuations are much lower for FV flow at \( \alpha = 20^\circ \) than for FW flow at \( \alpha = 5^\circ \). This is also true when comparing the fluctuations in the time history of the tip bending and twisting deflections between Figures 15 and 10. The higher level of dynamic load amplification and flow-induced vibrations for FW flow at \( \alpha = 5^\circ \) is caused by coalescence of the modes 2 and 3 in FW flow, as

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**Figure 13.** Time histories of \( C_D \), \( C_L \), and \( C_M \) of the hydrofoil at \( AR_h = 2.0 \), \( Fn_h = 1.5 \), and \( \alpha = 20^\circ \) (a) in calm water and (b) in waves with \( f_w = 0.68 \text{ Hz} \) and \( A_w = 5 \text{ cm} \).

**Figure 14.** Frequency spectra of \( C_D \), \( C_L \), and \( C_M \) of the hydrofoil at \( AR_h = 2.0 \), \( Fn_h = 1.5 \), and \( \alpha = 20^\circ \) (a) in calm water and (b) in waves with \( f_w = 0.68 \text{ Hz} \) and \( A_w = 5 \text{ cm} \).
indicated in Figures 5 and Table 3. In general, the nature frequencies of all modes tend to increase with the onset of ventilation because added mass is substantially reduced. At the same time, the relative increase varies from mode to mode, and it was predicted that modes 2 (first twisting) and 3 (second bending + twisting) would be affected very differently by ventilation. Hence, the single large peak existed near 27 Hz for FW flow at \( \alpha = 5^\circ \) is replaced by two smaller spectral peaks for FV flow at \( \alpha = 20^\circ \), indicating the separation of modes 2 and 3 in FV flow. Such a conclusion is supported by comparison of the frequency spectra between Figures 14 and 9, and between Figures 16 and 11. Note the location of the peaks agree fairly well with the predicted values of the natural frequencies and the vortex shedding frequencies indicated in vertical dashed lines in Figures 14 and 9, and in Figures 16 and 11, which are the same as those shown in Figures 5 and Table 3.

Comparison of the frequency spectra between Figures 14 and 9, and Figures 16 and 11, showed more low amplitude but broad-band spectral noise for FV flow case at \( \alpha = 20^\circ \) than FW flow at \( \alpha = 5^\circ \). Noisy response in FV flow is expected due to additional load fluctuations caused by the turbulent ventilated cavity closure, oscillation of the large ventilated tip vortex, as well as their interactions with the foil deformations and free surface.

**CONCLUSIONS AND FUTURE WORK**

In summary, experimental studies have been conducted in the Physical Model Basin (PMB) at the University of Michigan to study the effects of waves and ventilation on the dynamic response of a canonical surface-piercing hydrofoil. Experimental modal analysis techniques were adapted to the analysis of multi-degrees-of-freedom fluid-structure systems. Using data collected from a variety of sources, including the 2018 tests in the PMB, tests in a large water-filled container, and prior tests at CNR INM, the first three modal frequencies were shown to decrease monotonically with increasing submerged depth of the surface-piercing hydrofoil. The relative decrease in frequency with increasing submergence was different for the different modes because of the directional dependency of the fluid added mass. Using empirical correlations from prior tests at CNR-INM, the natural frequencies in the bucket of water were used to predict the frequencies in FV flow by modeling the change in the modal fluid added mass. In general, the natural frequencies increased in FV flow compared to FW flow due to reduction in added mass caused by replacement of liquid with gas on the foil suction side and free tip vortex. In addition, mode 2 (first twisting) and mode 3 (second bending + twisting) experienced frequency coalescence in FW flow at an immersed aspect ratio of \( AR_h = 2.0 \). In FV flow, the empirical model predicted that the two modes would separate and reverse order.

Experimental data are reported in this paper for the surface-piercing hydrofoil at \( AR_h = 2 \) and \( Fr_n = 1.5 \) in FW flow at \( \alpha = 5^\circ \) and in FV flow at \( \alpha = 20^\circ \), in both calm water and regular wave conditions. Compared to the response in calm water, the frequency response of the hydrodynamic loads in waves showed peaks corresponding to the encountered wave frequency and its harmonics, as well as to the foil natural frequencies and vortex shedding frequencies. Dynamic load amplification was observed when the harmonics of the wave excitation frequency neared the first natural frequency of the foil. In fully wetted conditions at a submerged aspect ratio of \( AR_h = 2.0 \), the second and third resonant modes practically coalesce, which lead to significant dynamic load amplifications and increased flow-induced vibrations in both calm water and wave conditions. For the FV flow case at \( \alpha = 20^\circ \), however, modes two and three separated, which lead to significant reduction in amplitude of the dynamic load fluctuations and flow-induced vibrations compared to the FV flow case at \( \alpha = 5^\circ \).

Future work will be aimed at refinement of experimental techniques and the models they enable. The correlations of Equations (4) and (5) would benefit from additional data at \( AR_h \neq 1.0 \). Detailed analyses of modal fluid added mass, fluid damping, and hydrodynamic stiffness ratios are shown in Harwood et al. (2019a,b). Additional results for varying \( AR_h, Fr_n \) and wave steepness ratios will be presented in upcoming papers. Additional tests in irregular waves will be conducted in the next round of experimental studies.

**ACKNOWLEDGEMENTS**

The authors would like to acknowledge the support of Dr. Ki-Han Kim under ONR grant number N00014-16-1-2433.
Figure 15. Time histories of tip bending $\delta_{\text{tip}}$ and twisting $\theta_{\text{tip}}$ deflections of the hydrofoil at $AR_h = 2.0$, $Fn_h = 1.5$, and $\alpha = 20^\circ$ (a) in calm water and (b) in waves with $f_w = 0.68$ Hz and $A_w = 5$ cm.

Figure 16. Frequency spectra of tip bending $\delta_{\text{tip}}$ and twisting $\theta_{\text{tip}}$ deflections of the hydrofoil at $AR_h = 2.0$, $Fn_h = 1.5$, and $\alpha = 20^\circ$ (a) in calm water and (b) in waves with $f_w = 0.68$ Hz and $A_w = 5$ cm.

REFERENCES


DISCUSSIONS AND AUTHORS’ REPLY

Discussions by each of the reviewer are shown first, followed by the authors’ response in italic font.

Written Discussion

Dr. Christopher P. Kent
Naval Surface Warfare Center – Carderock Division

The authors should be congratulated for the very interesting paper on the interesting loading and structural phenomena that occur on ventilated and non-ventilated hydrofoils.

The authors noted that there was a bias of the mean twisting deflection in waves relative to calm water shown in figure 10 and twisting moment in figure 8. Can they comment on what physical phenomena they believe is introducing this while still maintaining the mean lift? This is not as apparent in the fully ventilated flow at higher angles of attack and I’m wondering if there is an explanation.

Author Response

Thank you for the positive comments. The difference between the mean values of the moment coefficient between the wave case and calm water case in FW flow is probably caused by wave effects on the fluctuations of the size of the separated region on the suction side of the hydrofoil, which have a nonlinear effect upon the moment – and a greater impact than that on the lift coefficient – due to changes in the center of pressure. In FV flow, the entire suction side is covered by a gas cavity, and hence no significant difference between the mean hydrodynamic load coefficients between the calm water and wave cases. Note that while flow separation on the suction side could not be observed in Figure 7, flow visualization using paint streak method did confirm the existence of a separated region in calm water condition of the surface-piercing hydrofoil (Harwood et al., 2016b). To illustrate, an example of the result from paint streak test for $AR_h = 1, F_{th} = 2.5, \alpha = 14^\circ$ that was shown in Harwood et al. (2016b) is reproduced in Figure A1.

On-Site Discussion

Dr. Arthur Reed,
Naval Surface Warfare Center – Carderock Division

Thank you for an excellent paper on an interesting subject. You present the reduction of lift and drag as the foil goes from fully wetted to ventilated and show that this reduction in lift and drag varies with Froude number. It would be useful to see how lift over drag (L/D) changes, so that the change in “efficiency” of the foil can be assessed and understood – does the lift decrease more than the drag or vice versa?

Author Response

Thank you for the positive comments. The lift to drag ratio affected very strongly by the onset of ventilation. Depending on the submerged aspect ratio and Froude number, the transition from fully wetted to fully ventilated flow is often accompanied by a reduction in the lift of 25% - 50%. At the same time, the drag is very nearly unaffected because of near balance of decrease in lift-induced drag and increase in form drag – a behavior discussed at some length in Harwood, et al. (2016b). The result is a huge drop in the foil’s efficiency. This behavior can also be generalized by using a cavitation parameter $\sigma_c / \alpha$, where $\sigma_c$ is a general cavitation number valid in both ventilated and vaporous cavitation flows, which correlates quite well with the flow regime. This approach was used by Harwood (2016) and Harwood, et al. (2019a) to collapse lift and drag data onto curves that spanned wetted, partially ventilated, partially cavitating, and fully ventilated flows.
Figure A2 shows the lift to drag ratio plotted against the cavitation parameter. Note that these data are specific to an immersed aspect ratio of $h/c = 1.0$ in calm water. Despite some scatter due to variations with Froude number ($F_{nh} = 0.5 - 3.5$) and yaw angle ($\alpha = 0^\circ - 25^\circ$), it is clear that the efficiency of the foil is greatest in wetted flows, and partially cavitating flows (where a vaporous nose cavity introduces virtual camber), and lowest in fully ventilated flow.

**Figure A2.** Lift-to-drag ratio (or efficiency) plotted against the effective cavitation parameter $\sigma_c/\alpha$ for runs on rigid and flexible surface-piercing hydrofoils at $AR_h = 1.0$. 