Impact of Composite Layup on Hydrodynamic Performances of a Surface Piercing Hydrofoil

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ABSTRACT

Composite materials are good candidates for hydrofoils manufacturing, ensuring a good balance between strength and weight. In the high performances sailing yacht domain, hydrofoils are thin structures, highly loaded that experience significant displacements. This study investigates experimentally and numerically the influence of the laminate layup on the hydrodynamic performances of a surface piercing hydrofoil. Four hydrofoils with a constant chord, geometrically identical with different composite layups are mechanically characterized and tested in a hydrodynamic flume. The foils are designed to have a significant tip displacement of 5 to 10% of the span. Experimental results highlight a bending-twisting effect that leads to significant change in the hydrodynamic performances of the structures. Two different FSI numerical approach: from a potential code coupled with beam theory to the full coupling of a shell structural code and a VOF hydro model with free surface are presented and the first one is compared to the experiments with great results. The two approaches are two complementary bricks in the design process to compute the effect of passive deformation on hydrodynamic performances of the foils and therefore the yacht stability.

Key words: Bending-twisting coupling, Composite materials, Fluid Structure Interactions, Hydrofoils.

NOTATION

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>$\alpha$, AoA</td>
<td>Angle of Attack $[^\circ]$</td>
</tr>
<tr>
<td>DSBT</td>
<td>Differential Stiffness Bend-Twist</td>
</tr>
<tr>
<td>$E_f$</td>
<td>Young modulus of the fiber</td>
</tr>
<tr>
<td>$E_l$</td>
<td>UD’s Young modulus in longitudinal direction [MPa]</td>
</tr>
<tr>
<td>$E_m$</td>
<td>Young modulus of the resin</td>
</tr>
<tr>
<td>$e_p$</td>
<td>UD’s thickness [mm]</td>
</tr>
<tr>
<td>$E_t$</td>
<td>UD Young modulus in transverse direction [MPa]</td>
</tr>
<tr>
<td>FFT</td>
<td>Fast Fourier Transform</td>
</tr>
<tr>
<td>FSI</td>
<td>Fluid structure Interactions</td>
</tr>
<tr>
<td>$G_{lt}$</td>
<td>Shear modulus of the UD in direction $lt$ [MPa]</td>
</tr>
<tr>
<td>$G_{mn}$</td>
<td>Shear modulus of the resin [MPa]</td>
</tr>
<tr>
<td>LT</td>
<td>Laminate Theory</td>
</tr>
<tr>
<td>$\nu_f$</td>
<td>Poisson coefficient of the fiber</td>
</tr>
<tr>
<td>$\nu_m$</td>
<td>Poisson coefficient of the resin</td>
</tr>
<tr>
<td>$\nu_{lt}$</td>
<td>Poisson coefficient of the UD in direction $lt$</td>
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<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>PAC</td>
<td>Passive Adaptive Composites</td>
</tr>
<tr>
<td>POM</td>
<td>Polyoxymethylene material</td>
</tr>
<tr>
<td>UD</td>
<td>Unidirectional ply made of resin and fiber</td>
</tr>
<tr>
<td>U</td>
<td>Velocity [m/s]</td>
</tr>
<tr>
<td>VOF</td>
<td>Volume of Fluid</td>
</tr>
<tr>
<td>$V_f$ [%]</td>
<td>Volume of fiber in the composite</td>
</tr>
<tr>
<td>VAM</td>
<td>Variational Asymptotic Method</td>
</tr>
<tr>
<td>VLM</td>
<td>Vortex Lattice Method</td>
</tr>
<tr>
<td>$Y$</td>
<td>normal displacement of the foil [mm]</td>
</tr>
<tr>
<td>$Z$</td>
<td>foil dimension in spanwise [mm]</td>
</tr>
</tbody>
</table>
INTRODUCTION

For the top high performances foiling yacht design, the design process is a complex combination of free surface hydrodynamic simulation coupled with highly loaded composite structure analysis. The hydrodynamic loading leads to significant deformations of the structure that must be consider in the design. These deformations can have an impact on the hydrodynamic performance and the global yacht equilibrium (Balze et al., 2017). Within the available literature, a part of the described problem has been studied as the performance of a surface piercing straight foil (Young and Brizzolara, 2013), the coupled effect of fluid structure interaction FSI on cavitation (Ducoin et al., 2012) or the Velocity Program Prediction due to the apparent wind seen by the foiler (Hagemeister and Flay, 2017).

Figure 1: type of asymmetric lay-ups required to produce (a) bending-twist and (b) tension-twist coupling. (Veers et al., 1998)

The structural deformations can have a devastating effect on structure and are highlighted by the fluttering effect on the keel. The bend-twist coupling in the structure can be a good approach to control the lifting force through FSI with a passive adaptive method. Composite structures are extensively used for the appendages in the sailing yacht domain, putting more complexity in the behavior prediction.

Bend twist coupling in composites has been extensively used for wind turbine blade applications. Works on wind turbine have described that the phenomenon depends on the plies orientations in the layup Fig 1 (Veers et al., 1998) or more recently (Capellaro, 2012), where bend-twist coupling is evaluated with the plies orientation for glass and carbon fibers.

Figure 2: Effect of deformation of the hydrofoil on the hydrodynamic resultant on the 747 flying Mini.

Regarding hydrofoil application, (Giovannetti, 2017) has recently described the use of different techniques for bending-twist coupling as Passive Adaptive Composites (PAC) that tailors the response of a structure by changing the orientation of the composite plies (Veers et al., 1998) and Differential Stiffness Bend-Twist (DSBT) that utilizes the internal stiffness of a structure to change the aero-hydrodynamic response to fluid load (Raither et al., 2013). They have shown that the PAC can be used to control the lift response to hydrodynamic load in the case of a composite structure (Giovannetti et al., 2018) including by decreasing the tip load. (Young et al., 2018) also studied the effect of bend-twist coupling on the hydro-elastic response of a composite hydrofoil in a cavitation tunnel.
Indeed, bend twist coupling is a target effect for highly loaded structures to reduce the loads by reducing the incidence at the tip. (Gozcu et al., 2015),(Rohde et al., 2015) investigated the effect of this coupling on natural frequencies and mode shape to improve the design and the control of composite structures. The present study investigates the impact of composite layup on hydrodynamic performances of a hydrofoil piercing the free surface in the case of large displacements up to 10% of the span, at the tip. The application test case, the so called 747 flying mini 6.50 (Figure 2) indeed experiences such important deformation. As a first approach, the problem is simplified by a foil piercing the free surface at $45^\circ$, four composite hydrofoils made up of carbon or glass fibers are built for the experiments.

The first chapter focuses on the mechanical characterization of the composite hydrofoils through a comparison of different experimental techniques. These results are also compared with theoretical models based on Laminate theory (LT) on one hand and with numerical approaches on another hand: the in house "FS6R", based on a Variational Asymptotic Method (VAM) applied to a non linear composite beam theory (Hodges, 2006) and the commercial software ABAQUS.

The second chapter is dedicated to the hydrodynamic experiments carried out in a flume for the four composite hydrofoils, with FSI and free surface interaction. The foils displacements and hydrodynamic loads are measured and compared to a reference case corresponding to a rigid body computation. Vibration analysis is also performed to quantify the resonance frequencies in water. Last part described the FSI numerical simulations based on two different approaches. The first tool, "FS6R", is a coupled approach between the potential flow code AVL, and an internal code based on beam theory by finite elements that integrates cross section properties calculation. "FS6R" aims to compute FSI on a hydrofoil during the pre-design process. The second tool, much more time CPU consuming, is a coupled approach between the composite shell structure code ASTER and the viscous VOF flow solver interFOAM. Numerical and experimental comparison are presented based on the potential simplified approach when the the second one is a work in progress.

The principal objective of this paper is first to find out is to quantify and measure the effect of composite lay-up on bend-twist coupling and the hydrodynamic performances and to validate the different numerical approaches.

Hydrofoils characterization

This chapter describes briefly the manufacturing process and the mechanical characterization of the hydrofoils in air.

Hydrofoils Manufacturing

Four hydrofoils are manufactured with the help of SEAir, the industrial partner of this research work. The foils are straight structures of $1.35m$ span and $0.114m$ constant chord. The section is a NACA 0013 sandwich structure made of an AIREX web and laminated skin illustrated in Fig 3. In the cut-section, the green AIREX is clearly visible, the black part is the glue Spabond 345 (shows up as orange in CAD figure) and the white part is the glass fiber.

![Mechanical structure of the composite foils and cut section of the $P_2$ foil. The orange glue in the sketch is visible in black in the section.](image)

The four different layups and associated materials are given in Table 1. The manufacturing of the foils is realized by a vacuum lamination method and the overall process goes through several steps including: stratification in the mold, glueing of the web on the skin, intrados-extrados glueing, demolding, foil base manufacturing and finishing. The foils base (Fig 4) is directly manufactured on the foils by molding using a 3D printed piece to position the hydrofoil in the mold. Laminate plates of $200 \times 200 \, mm^2$ with the layup of Table 1 are manufactured simultaneously for the tensile specimens.
Table 1: Laminates Layups and Mechanical properties of the unidirectional plies, the matrix is made up of Epoxy. The layup $P_1$ uses the Glass2 when layup $P_2$ and $P_3$ uses the Glass1.

The results expected with these layup configurations are:

- P1: Important bending and no bend twist coupling
- P2: Bend twist coupling with a negative twist and higher bending when negative incidences are investigated
- P3: Bend twist coupling with a positive twist and smaller bending when negative incidences are investigated
- P2: Small bending and no bend twist coupling

Figure 4: Hydrofoils equipped by strain gauges and foil base in black (Dimensions are in mm). The white coat represents the wetted surface, the foil base ’s mold is in black and the connection piece are shown on the right.

Mechanical characterization

The straight foils are made of an extruded constant chord and a constant composite layup along the span. The structure will be then considered as a beam, with an equivalent EI modulus constant along the span. Three different methods are used here to experimentally determine the EI modulus of the 4 composite structures.

Three different methods are used here to experimentally determine the EI modulus of the 4 composite structures:

- A cantilever bending test with known masses
- A natural frequency vibration test
- A tensile test on representative specimens
Experimental setup

For the cantilever bending test and the vibration test, the clamping system used during the hydrodynamic test is mounted on a rigid structure to reproduced the same boundary conditions. Displacements of the structure are measured by a laser telemeter, also used during the hydro test, at the tip of the foils.

Bending tests: Four different calibrated masses \( M_1 = 518 \, g \), \( M_2 = 1018 \, g \), \( M_3 = 2018 \, g \) and \( M_4 = 3018 \, g \) are applied to the considered center of hydrodynamic load and the laser measures the vertical displacement \( Y \) of the foil at the tip at distance \( X \) from the root (See Figure 5). The bending stiffness \( EI \) of the structure is calculated with the elastic straight beam relation (1).

\[
EI = \frac{FL^2(3X - L)}{6Y} \tag{1}
\]

Vibration tests: the structure is manually displaced and release; the laser measures the vertical displacement over time at the tip. For each case, two tests are performed and the average is presented in the results. The bending stiffness \( EI \) of the structure from vibration analysis is then calculated by (2).

\[
EI = \left( \frac{(2\pi f_i L^3)^2}{\lambda_i^4} \right) m \tag{2}
\]

\( f_i \): the natural frequency of the mode \( i \)
\( \lambda_i \): a number associated to the proper mode \( i \), depending of the boundary conditions
\( m \): the structural weight per unit of length
\( L \): the length of the structure.

The natural frequency is obtained by a FFT on the temporal signal recorded by the laser as illustrated in Fig 6. In this case the first bending mode is \( 3.9 \, Hz \) giving a bending stiffness of \( EI = 157.87 \, N.m^2 \).

Tensile tests: For each layup, three specimens complying with the ISO 527 standard are used and a tensile test is carried out on each. A specimen is fixed in the jaws of the machine (Fig 5) and the displacement speed is fixed. As output of the machine, the temporal evolution of the force and deformation are recorded. The Young’s modulus \( E \) is evaluated as the slope of the elastic stress versus the elastic strain. The specimens of each layup gives good agreement results and \( E \) is the average value. The bending stiffness \( EI \) of the structure is obtained by the product of the measured \( E \) and the inertial moment \( I \) calculated on the hydrofoil section, taking into account the skin thickness of the laminates.
Calculation of composite structural properties

Three different approaches are used to calculate the mechanical properties of the composite foils:

- An in house tool based on laminates theory
- An in house tool "FS6R" based on a Variational Asymptotic Method (VAM)
- The commercial code ABAQUS

Laminates theory describes a ply in its membrane plane and calculates its properties \(E_l, E_t, G_{lt}, \nu_{lt}, e_p\) as a function of the fibers and resin properties \(E_f, E_m, G_m, \nu_f, \nu_m\).

(Gay, 1991) describes the theory and shows the functions calculating the plies properties in the membrane plane and the equivalent properties of a stratified structure in all the directions.

\(EI\) is obtained by the product of \(E\) from the laminates theories by the inertial momentum \(I\) calculated with the real skin thickness.

The commercial software ABAQUS uses its meshed beam cross-sections function which allows the description of a beam cross-section including multiple materials and complex geometry.

The mechanical part of "FS6R" is presented in the next chapter.

Comparison of the hydrofoil stiffness

Figure 6: Vibration response of P1.

Figure 7: Comparison of the EI measured vibration test, bending test with the mass M1, M2, M3, M4 and tensile and the EI calculated with laminate theory LT, ABAQUS ABQ and the in house code FS6R.
Figure 7 compares the bending stiffness $EI$ obtained experimentally with the numerical results from the laminates theory, ABAQUS and FS6R for all the layups $P_1$, $P_2$, $P_3$ and $P_4$ (see Table 1). The y-axis gives the EI values and the x-axis represents the different methods:

- \textit{vib}: the mean value of the vibration tests.
- \textit{Mi}: The bending test using the mass $M_i$ as load.
- \textit{LT}: Laminate theory using the properties defined in Table 1.

The different experimental techniques give EI modulus close in a range of 13% when comparing a test to another. The high differences observed with the vibration tests may be due to the stiffness of the clamping support. Table 2 gives the mean value of EI computed as an average of the experimental tests.

<table>
<thead>
<tr>
<th>Foil Layup</th>
<th>$P_1$</th>
<th>$P_2$</th>
<th>$P_3$</th>
<th>$P_4$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$EI \ [N.m^2]$</td>
<td>155.6</td>
<td>153.8</td>
<td>145</td>
<td>420.5</td>
</tr>
</tbody>
</table>

Table 2: Bending stiffness of the different foils

A comparison of the analytic and numerical results to these reference values show low discrepancies up to 10% with \textit{LT} (reached with $P_1$), 3% with \textit{ABQ} (reached with $P_4$) and 5% for \textit{FS6R} (reached on $P_2$, $P_3$). The results show a good confidence in the different approaches. With these simple extruded structures, one should prefer the very simple \textit{FS6R} tool when ABAQUS is time consuming for no gain.

**FS6R numerical model**

FS6R is a code dedicated to the preliminary stages of foils design to model and to analyze the structure through fluid structure interactions simulations. It stands on the lifting line method and beam theory by finite elements and its algorithm is presented in Figure 9.

After defining the foil geometry, the materials and the configurations to simulate, the fluid flow is solved by the open source tool AVL which performs a VLM in-viscid 3D calculations on the whole surface and provides the hydrodynamic forces. A viscous correction is then realized with XFOIL which performs 2D viscous simulations and the structural analysis is performs by an in-house code standing on beam theory by finite elements. This code uses the Variational Asymptotic Method describes in (Hodges, 2006) to calculate the properties of a given section associated to a material such as: the shear center, torsional center, young modulus, inertia, shearing stiffness $GJ$, etc.

As an output of FS6R calculation, we get the efforts applied on the structure and the distorted shape. More details on FS6R can be found in (V. Temtching and D.R, 2018) where a validation on a 3D trapezoidal foil in POM material is shown. This part presents the implementation of bend twist coupling in structural analysis of the code.

![Figure 8: DOF of a 1 element beam. (Hodges, 2006)](image)

Figure 8 shows a 1-element beam with the DOF on the two nodes. We have 3 DOF in translation and 3 DOF in rotation for each node which are the solution of the system (3).
\[ K \times U = F \] (3)

K: the stiffness Matrix in the global reference, the matrix is symmetric.
F: Forces applied to the structure in the global reference.
U: DOF of the structure.

To consider bend twist coupling in FS6R, coupling terms usually neglected for quasi-isotropic or orthotropic materials, and set as zero in the stiffness matrix, are replaced by bend twist coupling terms \( c_{ij} \), described in (Capellaro, 2012) as presented in Fig 10.

\( c_{ij} \) depend on the coupling percentage \( \alpha[\%] \) described in (Capellaro, 2012) for glass and carbon fibers, that varies with the plies orientation, the torsional stiffness and the bending stiffness (see Fig 11). The proportion of plies in the laminate responsible of this coupling is also consider.

The coupling terms are calculated by (4)

\[
C_{ij} = \sum_{\theta=0}^{90} \alpha_{\theta} \times A_{\theta} \times \sqrt{k_{ii} \times k_{jj}}
\] (4)
Figure 10: Global stiffness matrix of the structure with the added bend-twist coupling terms $c_{ij}$ in red and torsional stiffness in yellow. $k_{1010} = k_{44}$ is the torsional stiffness.

Figure 11: Limit of bend twist coupling percentage with the plies orientation for glass and carbon fibers. (Capellaro, 2012)

$\theta$: is the ply orientation [°].

$\alpha_\theta$: The maximum percentage of bend twist coupling induced by a ply, oriented at $\theta$ degrees.

$A_\theta$: the percentage of the ply oriented at $\theta$ degrees in the layup of the structure.

These new terms of the stiffness matrix will induce a twist angle when the structure is loaded by a bending for or a bending moment and therefore a bending motion when the structure is loaded by a torsional moment.

**Flume experimental setup**

The described foils are tested in a flume in order to measure the impact of the composite layup on their hydrodynamic performances.

Hydrodynamic tests are carried out at IFREMER Lorient’s flume in a working section of 2.5m in the flow direction and 1.5m depth with a maximum velocity of 1m/s (see Fig 12). The foil is clamped to a 6-DOF balance measuring the efforts and moments express in the references frame of the foil and displacements are measured with a laser telemeter through an underwater window (See Fig 12).

The lateral displacements of the foils are measured at 3 different heights. The laser sweeps 10 consecutive times the foil in chord wise at the different height in order to average the distance. This method is preferred to a single point measurement to distinguish the bending from the twist. A calibration process is used to correct the diffraction effect due to the different interfaces that the laser encounters.

As shown on Fig 12 and 13, the foils are piercing the free surface with an angle of 45° and mounted cantilevered on the
Figure 12: Left: Experimental setup of the hydrofoil test mounted at a 45° angle in a flume. The water level depends of the flow speed. Dimensions are in mm. Right: Calibration set up for the Laser telemeter system. Three defined geometries made up of 3D printed material (shows up in orange) are placed at the level of the path of the laser balance.

Figure 13: Hydrofoil tested in IFREMER Lorient flume at 0.9ms and AoA= −9°. Picture on the right shows the balance, the clamping system and the strain gauges wires.

The clamping conditions are the same used for the mechanical characterization tests; the flow is aligned with X-axis. Most of the tests are done with a negative AoA to enhance the laser telemeter measurement and to prevent the tip to get to close to the side of the flume (Fig 10). The 45° tilt angle is chosen to represents the interaction of the MINI 747 foil with the free surface and to analyze its effects on the hydrodynamic performances. It also has the advantage of maximizing the immersed surface with a low confinement effect. The foils are equipped by 2 full bridges strain gauges installed at respectively at 350mm and 250mm from the embedding. They are placed on the dry part of the foil so they do not impact the flow. The use of two full bridges is chosen to get the hydrodynamic load resultant position on the span. Two speeds 0.7m/s, 0.9m/s and several angles of attack ranging from [−9°, +3°] are investigated. As most of flume, the free surface height varies with the velocity: 1.435m for 0.7m/s and 1.42m for 0.9m/s.
Hydrodynamic test results

The foil displacements recorded with the laser and the hydrodynamic loads measured with the 6-DOF balance are compared with FS6R simulations.

Foil displacements

Foil displacements are measured at three different span positions with the laser telemeter. The displacements for $-5^\circ$ and $-7^\circ$ at $0.9 m/s$ are represented in Fig 14 and in Fig 15, they respectively compare experiments to a FS6R simulations without and with bending twist coupling terms in the stiffness matrix for all the layups.

Without the coupling, the plies orientation sign of $\pm 45^\circ$ in the layups is not taken into account and leads to the same simulation results for $P_2$ and $P_3$ (See Fig 14). In FS6R computations: $P_2$, $P_3$ are very close to $P_1$. They see the same load and the displacement at the top is around 3% higher due to the different stiffness.

When bend-twist coupling is considered, displacements of $P_3$ decreases and $P_2$ increases experimentally and numerically. We observed at the tip displacements up to $77 mm$ (5.8% of span) for $P_3$ glass fiber foil and only $68 mm$ (5.01%) for the $P_2$ with the same type of laminate ($P_2$ and $P_3$ only differs by the plies orientation of $-45^\circ$ for $P_2$ and $+45^\circ$ for $P_3$ ) when looking at $-7^\circ$ configuration.

As expected in the design process and in agreement with its highest bending stiffness (depicts in Figure 7), the carbon foil $P_4$ experiences the lowest displacement with $40 mm$ at the tip (3% of its span length).

![Figure 14: comparison of the measure Z displacement of the foils with a no bending twisting coupling calculation.](image1)

![Figure 15: comparison of the measure Z displacement of the foils with a bending twisting coupling calculation.](image2)

These measurements highlight different behaviors that are linked both to the composite layup and the materials of the tested foil. First, the carbon foil, with the same number of plies is significantly more rigid than the glass fiber foils. The second behavior is directly linked to the subject of this paper, meaning the bending-twisting coupling. As illustrated in the Fig 13, the coupling coefficient $g$ in the mechanical matrix, described in chap 3, differentiates significantly the simulated displacement of $P_2$ and $P_3$ and leads to a good fit of the experimental results. The $-45^\circ$ orientation of the plies in the
P2 composite structure leads to a negative twist of the structure, with the direct consequence of loading the tip (negative incidence investigated). The opposite phenomenon is expected on P3 where the structure contains +45° plies.

**Effects on hydrodynamic**

![Figure 16: Experimental forces compared to FS6R calculations.](image)

The efforts recorded during the experiments and the computations from FS6R tool for all the incidences at velocity 0.9m/s are presented in figure 16. Lift magnitude and its projections on Y-axis ($F_y$) and Z-axis ($F_z$) are shown for the 4 foils and for a rigid case (only fluid calculation, no FSI) simulated with FS6R. Many observations can be made with this curves.

Experimental and FS6R results have the same trend and the values fits perfectly excepts for Z projection of the lift force. This value is very small and does not affect the lift force nor the moment leading to a good agreement of both approaches. The maximum discrepancy observed is around 5% on the lift force for $P_3$ with experiments smaller than FS6R (same observation on the displacements).

The lift force clearly exhibits the different behaviors of the layups: $P_1$ and $P_4$ overlap with the rigid case as expected, highlighting no bend twist effect. $P_3$ is smaller than the rigid case and $P_2$ is higher. This behavior is exactly what we expected, because negative incidence is investigated, $P_2$ with the negative twist is the most loaded when $P_3$ with the positive twist become the less loaded.

We also observe that when $P_1$ and $P_4$ lift magnitude are the same, their projections $F_y$ and $F_z$ are different due to their different bending deflection.

Experimental lift projection on Z-axis does not have a well defined trend which may be due to the balance precision when having small values as for the case.

$F_z$ from FS6R computations has the same trend as the displacements, the force is transferred to the span wise direction due to the foil deflection, modifying the hydrodynamic behavior.

We observe that $F_y$ match very well in both approaches and experimental values are slightly higher. The bending momentum $M_x$ being an image of $F_y$, we have the same agreement.
We are thus able to estimate the impact of bend twist coupling on lift force distribution.

Since potential flow theory gives good trends for Drag coefficient with underestimated values, Drag force is not presented in this part.

**Strain gauges measurements**

The measures from the strain gauges don’t allow at this stage to recover the resultant force position due to some measurement inconsistencies described in the chapter introduction. The possibility of replacing the 6-DOF balance on the foil by inexpensive and easy to use of strain gauges as full bridges will however be explored. The cross correlation of the balance $M_x$ and the gauges signal is close to one. The dynamic and the resolution of such equipment are perfectly adapted. Transverse deformations and calibration process are the main direction for an upgrade of the system.

**Natural frequencies in water**

![Comparison of water and Air frequencies](image)

Figure 17: Hydrofoil natural frequencies in air and water.

Figure 17 shows the natural frequencies measured in water in condition identical to the mechanical characterization in air. The natural frequency is, indeed, impacted by the added mass due to the acceleration of the water around the foil which leads to a reduction of the frequency. The added mass can be computed from the natural frequency formula (from Eq. 2) by adding the added mass $m_a$ in the mass term as depicted by (5).

$$f_i = \frac{\lambda_i^2}{2\pi L^2} \times \sqrt{\frac{EI}{m + m_a}}$$  

(5)

Table 3 shows the added masses calculated for each case.

<table>
<thead>
<tr>
<th>Foil Layup</th>
<th>$P_1$</th>
<th>$P_2$</th>
<th>$P_3$</th>
<th>$P_4$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$m_a$ [kg/m]</td>
<td>10.03</td>
<td>10.43</td>
<td>9.73</td>
<td>10.84</td>
</tr>
</tbody>
</table>

Table 3: calculated added mass in water

The first bending mode frequency is very low and could be subjected to external solicitations at resonance frequency (wave, ...).

**Work in progress: ASTER-InterFOAM coupling**

The fluid structure interaction is simulated with OpenFoam for the fluid aspects and code Aster for the composites structure modeling. The fluid simulation is carried out with the volume of fluid phase-fraction based interface capturing approach named interFoam, with a steady mesh. The composites structured is solved with a shell composites layup model. The coupling is done by:
• Launching OpenFoam runs on the deformed foil computed by code Aster
• Launching code Aster runs with loads computed by OpenFoam and projected on the structural mesh (wall shear stress tangential to the shape and pressure normal to the shape)

Going back and forth between fluid and structural solver typically converges in 4 to 6 iterations. Fig 18, shows a first FSI

Figure 18: Deformed Hydrofoil computed with InterFoam coupled to code Aster, presented in the flow field. The bottom is the cantilevered part and the tip is at the top in the picture.

simulation computed with the coupling of code ASTER and InterFoam solver, depicting encouraging results.
This approach will allow to take into account, the free surface interaction, to evaluate the drag and to consider more complex structures.

Conclusion

This work presented an experimental and numerical study of the impact of composite layup on the hydrodynamic performances of a surface piercing hydrofoil.
The structural and mechanical characterization of 4 foils from an identical mold but having different layup and material has shown small discrepancies between the different approaches. The comparison, based on the EI modulus of the straight constant section foil, has demonstrated the ability of the Laminate Theory, as well as the FS6R structural module to well estimate the bending stiffness of the composite structure. Discrepancies may be significantly higher with more complex composite layups.
The foils are tested in hydrodynamics flume and show different behaviors due to the layup and the materials. The carbon foil $P_4$ is significantly stiffer, as expected during the design process. No bend twist coupling is observed for $P_4$ and $P_1$, showing the same lift force that only differs in the projections. The bending motion does not modify the force magnitude but its distribution on the axis changing the balance between the lift and the side force component.
The experiments investigated with negative incidences clearly highlight the impact of bending-twisting coupling. The foil $P_2$, with oriented plies at $-45^\circ$ in the laminate, experiences a significantly higher displacement and hydrodynamic loads due the negative twist loading the tip.
The comparison of experimental displacements with FS6R computations is very good (discrepancies are less than 10% with experiments values higher in most of the cases excepts $P_3$), showing the ability of the code to compute the bending-twisting coupling.
The first step of a FSI coupling between the structural code ASTER and the VOF hydrodynamic solver interFOAM is eventually presented. The coupling, significantly more CPU demanding, brings the final bricks of the design process for foil with FSI, where FS6R has proven to estimates greatly the tendencies and ASTER-interFOAM would provide accurate results on the performances.
REFERENCES


