Application of System-based Modelling and Simplified-FSI to a Foiling Open 60 Monohull

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ABSTRACT

The increasing number of foiling yachts in offshore and inshore races has driven engineers and researchers to significantly improve the current modelling methods to face new design challenges such as flight analysis and control (Heppel, 2015). Following the publication of the AC75 Class Rules for the 36th America’s Cup (RNZYS, 2018) and since the brand new Open 60 Class yachts are all equipped with hydrofoils, the presented study will propose a system-based modelling coupled with a simplified FSI (fluid-structure interaction) method that leads to better understand the dynamic behavior of monohulls with deformable hydrofoils.

The aim of the presented paper is to establish an innovative approach to assess appendage behavior in a dynamic VPP (velocity prediction program). For that purpose, dynamic computations are based on a 6DOF mathematical model derived from the general non-linear maneuvering equations (Horel, 2016). The force model is expressed as the superposition of 7 major force components expressed at the center of gravity of the yacht: gravity, hydrostatic, maneuvering, damping, propulsion (wind), control (rudders, daggerboard, foils …) and wave (Froude-Krylov and diffraction phenomenon).

As test cases, course keeping simulations are performed on an Open 60 yacht with control loops to simulate the wing trimmer, helmsman and foil trimmer when finding the optimal foil settings is needed. In first hand, IMOCA’s polar diagrams are used as reference.

In calm water and in waves, the influence of foil’s shapes (foil with shaft pointing downward and tip pointing upward, foil with shaft pointing upward and tip pointing downward) and stiffness (non-deformable, realistic, flexible) on the global behavior of the yacht is presented.

NOTATION

\[ A_v \] Transverse section area (for air resistance) (m²)
\[ A_{w0} \] Waterplane area at zero speed (m²)
\[ B_{wl} \] Beam of waterline (m)
\[ B \] Center of buoyancy
\[ c \] Chord of the section (m)
\[ C_p \] Prismatic coefficient
\[ G \] Center of gravity of the yacht
\[ g \] Gravity acceleration (m.s \(^{-2}\))
\[ I_r \] Roll moment of inertia (kg.m\(^2\))
\[ I_p \] Pitch moment of inertia (kg.m\(^2\))
\[ I_y \] Yaw moment of inertia (kg.m\(^2\))
\[ I_{xy} \] Roll and pitch product of inertia (kg.m\(^2\))
\[ I_{xz} \] Roll and yaw product of inertia (kg.m\(^2\))
\[ I_{yz} \] Pitch and yaw product of inertia (kg.m\(^2\))
\[ LCB \] Longitudinal position of the center of buoyancy to fpp (m)
\[ LCF \] Longitudinal position of the center of flotation to fpp (m)
\[ L_{wl} \] Length of waterline (m)

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\( p \) Roll angular rate (rad.s\(^{-1}\))
\( q \) Pitch angular rate (rad.s\(^{-1}\))
\( r \) Yaw angular rate (rad.s\(^{-1}\))
\( Re \) Reynolds number of the yacht
\( S_w \) Wetted surface (m\(^2\))
\( S_{w0} \) Static wetted surface (m\(^2\))
\( T \) Draft (m)
\( TWA \) True wind angle (m.s\(^{-1}\))
\( TWS \) True wind speed (m.s\(^{-1}\))
\( V \) Yacht speed (m.s\(^{-1}\))
\( u \) Surge velocity (m.s\(^{-1}\))
\( v \) Sway velocity (m.s\(^{-1}\))
\( w \) Heave velocity (m.s\(^{-1}\))
\( \rho \) Density of water (kg.m\(^{-3}\))
\( \rho_a \) Density of air (kg.m\(^{-3}\))
\( \eta \) Incident wave amplitude (m)
\( V_c \) Volume of displacement of canoe body (m\(^3\))

**INTRODUCTION**

Recent studies have shown that system-based modelling can be applied to foiling yacht in order to evaluate their maneuvering abilities for a panel of standard maneuvers (Hor el, 2017) but calculations with complex foil geometries were still under development. Since that, improvements have been done to existing models in order to take into account a large panel of foil geometries and simple structural deformations such as hydrofoil’s flexion.

In the foil’s FSI prediction, regarding the main FSI methods (direct integration, finite elements and modal analysis) and based on experimental validation, a simplified model of dynamic shear force and bending moment is implemented. The accuracy of such a model was investigated by performing experimental test in rough waves on a segmented containership’s hull in the 50m×30m×5m hydrodynamic and ocean engineering tank of Ecole Centrale de Nantes.

In this context, the main objective of this paper is to evaluate through simplified modelling whether basic structural deformations of hydrofoils can affect the global yacht’s behavior for several sea conditions.

In calm water, numerical results are compared with polar diagrams on conventional IMOCA yacht. In waves, only numerical results are presented and commented regarding the relevant effects of the hydrofoils on the yacht motions.

The application of structural deformations in dynamical simulations is a recent field of research. Also, Open 60 monohulls are complex yachts and the presented work is a first step to better understand the physical phenomena that drive the yacht’s performances.

**DYNAMICAL VPP**

Velocity prediction programs (VPP) aim to provide performance data for sailing yachts which can be expressed as polar diagrams where the target speed is given as a function of the true wind angle (TWA) for several true wind speeds (TWS). These tools are solving a quasi-static equilibrium at each time step and the degree of accuracy depends on the accuracy of each force model. Analytical method are commonly used but recent studies have been done on coupling VPP with Computational Fluid Dynamics (CFD) (Roux et al., 2008) (Viola et al., 2015) in order to improve the accuracy of the predictions and to reduce the computing time.

When dealing with dynamical motions, static VPPs are no longer suitable for predicting the performances of the yacht. Then, in order to solve dynamical problems a so-called dynamical VPP (DVPP) is used. This DVPP is based on a 6 degrees of freedom (DOF) model derived from the Newton’s second law. In this study, the hull of the yacht is assumed to be symmetric in \((x, O, y)\) and \((y, O, z)\) planes. However, the canting keel and sails make the moments of inertia of the yacht very asymmetric. Knowing the total force components \((X, Y, Z, K, M\) and \(N)\), the accelerations of the yacht can be predicted by solving the following general differential equations:

\[
X = m[\dot{u} + qw - rv - x_g(q^2 + r^2) + y_g(pq - \dot{r}) + z_g(pr + \dot{q})] \\
Y = m[\dot{v} + ru - pw - y_g(r^2 + p^2) + z_g(qr - \dot{p}) + x_g(qp + \dot{r})] \\
Z = m[\dot{w} + pv - qu - z_g(p^2 + q^2) + x_g(rp - \dot{q}) + y_g(rq + \dot{p})] \\
K = I_x\ddot{q} - I_{xy}(\dot{q} - rp) - I_{xz}(\dot{r} + pq) + (I_x - I_y)qr + I_{yz}(r^2 - q^2) + m[y_g(\dot{w} + pv - qu) - z_g(\dot{v} + ru - pw)]
\]
\[ M = l_y \ddot{q} - l_{yz} (\dot{r} - \dot{p}q) - l_{xy} (\dot{p} + qr) + (l_z - l_x) r p + l_{xz} (p^2 - r^2) + m [z_G (\ddot{u} + qw - rv) - x_G (\ddot{w} + pv - qu)] \]  
\[ N = l_x \ddot{r} - l_{xz} (\dot{q} - qr) - l_{yx} (\dot{q} + rp) + (l_y - l_z) q p + l_{xy} (q^2 - p^2) + m [x_G (\ddot{v} + ru - pw) - y_G (\ddot{u} + qw - rv)] \]

with:

\[ O_G = \begin{bmatrix} x_G \\ y_G \\ z_G \end{bmatrix} \]  

\[ p = -\sin \theta \dot{\psi} + \dot{\psi} \]  
\[ q = \sin \phi \cos \theta \dot{\psi} + \cos \phi \dot{\theta} \]  
\[ r = \cos \phi \cos \theta \dot{\psi} - \sin \phi \ddot{\theta} \]

Previous equations are expressed in the yacht’s reference frame as defined in Figure 1.

**Figure 1 - Reference frames: earth \( (b_0) \) and yacht \( (b_y) \)**

Equations 1 to 6 are solved by applying a 4th order Runge-Kutta integration scheme. This leads to calculate the yacht’s accelerations and then to predict the yacht’s motions in surge, sway, heave, roll \( \phi \), pitch \( \theta \) and yaw \( \psi \). The accuracy of the predicted motions is governed by the accuracy of the implemented force models.

**FORCE MODELLING**

In system-based modelling, the complex behavior of the yacht is calculated from taking into account the local phenomena and the global effect of their interactions when mathematical models are available.

This simplified modelling expresses the total loads \( F \) on the yacht as the superposition of force components acting on the hull, on the appendages and on the sails. Then, since \( O_b \) and \( G \) are combined, 7 major force components have been identified and expressed at the center of gravity \( G \) of the yacht: gravity \( (F_{Grav}) \), hydrostatic \( (F_{HS}) \), maneuvering \( (F_{Man}) \), damping and radiation \( (F_{Damp}) \), propulsion due to the wind \( (F_{Prop}) \), control such as rudders, daggerboards, hydrofoils or keel \( (F_{Ctrl}) \) and waves including Froude-Krylov and diffraction phenomenon \( (F_w) \).

\[ F = F_{Grav} + F_{HS} + F_{Man} + F_{Damp} + F_{Prop} + F_{Ctrl} + F_w \]  

with:
\[ F = \begin{bmatrix} X \\ Y \\ Z \\ K \\ M \\ N \end{bmatrix}_{(G,x_b,y_b,z_b)} \]  

(12)

Gravity forces

An Open 60 is actually composed with at least a bare hull, a canting keel, daggerboards or hydrofoils, rudders and a mast and a boom that have a significant influence on the total weight of the yacht. Since each of these \( N_{\text{element}} \) elements has its own center of gravity, the total yacht’s weight distribution can be written as follows:

\[ F_{\text{grav}} = \sum_{i=1}^{N_{\text{element}}} \left[ F_{\text{grav}_i} \right] \]

(13)

Hydrostatic forces

In order to take into account the strong geometrical non-linearities when the yacht experiences large amplitude motions in calm water and in waves, a nonlinear method was developed to compute the hydrostatic loads on the bare hull and underwater appendages. This method is based on the use of a surface mesh from a STL file and the integration of the hydrostatic pressure \( p_{\text{HS}} \) on the total instantaneous immersed surface \( S_w \) composed with \( N_f \) elementary surfaces.

\[ F_{\text{HS}} = \sum_{i=1}^{N_f} \left[ F_{\text{HS}_i} \right] \]

(14)

with:

\[ F_{\text{HS}_i} = -p_{\text{HS}}(z_i) dS_w \]

\[ p_{\text{HS}}(z_i) = -\rho g z_i \]  

(15)  

(16)

In equation 16, \( z_i \) is the vertical position of the center \( B_i \) of the elementary surface compared to \( O \), the origin of the earth’s reference frame.

Maneuvering forces

Maneuverability is the study of stable and transient states of the yacht motion in calm water and in waves at low encounter frequency. It consists in evaluating the abilities of a yacht to keep a desired heading or to change her heading after the action of the control appendages. In this context, 3DOF mathematical models in surge, sway and yaw are used to study complex maneuvers and several maneuvering scenario.

In surge, the resistance of the yacht can be estimated following the ITTC recommendations 7.5-02-05-01 (ITTC, 2017) and using the regressions based on the Delft Systematic Yacht Hull Series (DSYHS) (Keuning et al., 1998). The Resistance \( R_T \) is expressed in the non-dimensional form using the resistance coefficient \( C_T \).

\[ C_T = \frac{R_T}{\frac{1}{2} \rho w^2 v^2} \]  

(17)

This total resistance coefficient is expressed as the superposition of 4 identified factors:

- Residuary resistance coefficient of the bare hull, \( C_R \)
- Frictional resistance coefficient, \( C_F \)
- Wind resistance coefficient, \( C_{AA} \)
- Appendage resistance coefficient, \( C_{APP} \)

\[ C_T = C_R + \frac{S_w}{3w_0} C_F + C_{AA} + C_{APP} \]  

(18)

Empirical formula from ITTC-57 is used to calculate the values of \( C_F \). ITTC mentioned that the frictional coefficient is associated with a form factor \( k \) but for high speed marine vehicles with transom stern as Open 60, it is recommended to assumed that \((1 + k) = 1.0\).
Wind resistance coefficient formulation is given by using the aerodynamic drag coefficient $C_D$ in hydrodynamic form:

$$ C_{AA} = \frac{\rho_a h_s}{\rho S_w} C_D $$

In first approximation, as mentioned in ITTC-78, $C_D$ is assumed to be equal to 0.8 as default value and the transverse section area $A_t$ is calculated from the values of the air draft and the mean width of the hull.

As mentioned in previous studies (Raymond, 2009) (Huetz et al., 2011), the most consistent formulation for residuary resistance coefficient of the bare hull $C_R$ is the DSYHS formulation.

$$ C_R = \frac{2gT_c}{S_w^3} C_{R_{DSYHS}} $$

The expression of $C_{R_{DSYHS}}$ can be found in previous work on mathematical model for the tacking maneuver of a sailing yacht (de Ridder et al., 2004):

$$ C_{R_{DSYHS}} = a_0 + \left( a_1 \frac{L_C}{L_{wl}} + a_2 C_p + a_3 \frac{v^2}{S_{wo}} + a_4 \frac{\rho_{wl}}{S_{wl}} + a_5 \frac{v^2}{S_{wo}} + a_6 \frac{L_C}{L_{wl}} + a_7 \left( \frac{L_C}{L_{wl}} \right)^2 + a_8 C_p^2 \right) \frac{v^2}{L_{wl}} $$

In expression 22, values of the coefficients $a_1$ to $a_8$ are determined from tables for Froude numbers from $F_n = 0.10$ to $F_n = 0.60$. Also, additional resistance due to the heeling and the sway velocity of the yacht can be added to expression 22.

In previous study on maneuvering models on conventional ship, identification of hydrodynamic derivatives in sway and yaw from several experimental data sets leads to establish empirical expressions as functions of the ship features: draft, width, length between perpendicular, block coefficient, etc. (Tjoswold, 2012) (Clarke, 1983). But for sailing yacht, previous mentioned work from de Ridder et al. (2004) also propose polynomial regressions for the calculations of forces in sway and moments in yaw due to heelf angle, sway, roll and yaw velocities and their coupled effects.

**Damping forces**

Damping forces on the bare hull are known to be due to 2 phenomena: frictional damping and radiated waves. The former can be modeled using Ikeda’s formulation as explained in ITTC recommended procedures (2011) while the radiation forces from radiated waves are evaluated in time domain by using Cummins formulation based on linear theory. Added mass and damping coefficients at zero forward speed are calculated using the boundary element methods code NEMOH (Babarit et al., 2015). Then, these coefficients are used for the yacht with forward speed by using a formulation based on a first order development of the slip boundary condition on the hull as described by Delhommeau et al. (1987).

**Control forces**

The modelling of control forces includes the maneuvering forces from the rudder and the forces on the appendages such as the keel, the daggerboards and the foils. These forces tend to modify the dynamic equilibrium of the yacht while sailing on a desired heading. In this section, particular focus will be paid to the modelling of the forces acting on the hydrofoils.

In our work, a foil is defined using the following parameters:

- General shape (position of the nodes),
- Foil sections,
- Chord of the sections,
- Linear mass and added mass,
- Local stiffness.

As can be noticed in Figure 2 for a non-conventional modern Dali’s foil 2018, complex geometries are taken into account by discretizing the foil with $N_{elem}$ elements. Then, a local reference frame is associated to each element.
The global hydrodynamic forces $F_{Hyd}$ on the foil are evaluated from the computation of the lift and drag on each element assuming that the element is encountering a 2D flow. The effective angles due to the yacht motion are used in order to evaluate the angle of the incident flow at each time step. Knowing the position of each element in the ship reference frame, the moment can be expressed at the center of gravity $G$ of the yacht. Also, since the yacht accelerations are known, the added mass forces $F_{AM}$ on the foil can be calculated. In first approximation, a simple linear added mass coefficient $m_{yy}$ is used.

$$m_{yy} = \frac{\pi}{4} \rho c^2$$  \(23\)

The global force on the foil $F_{foil}$ is expressed from the local forces on each element.

$$F_{foil} = \sum_{i=1}^{N_{elem}} \left( F_{Grav_i} + F_{Hyd_i} + F_{AM_i} + F_{Inertials} \right)$$  \(24\)

The effect of the interactions between the foil and the free surface are taken into account by Faltinsen’s formulation (2005) where a submergence Froude number is calculated in the modelling of the lift coefficient reduction near the free surface.

When sailing in waves, the velocity of the incident flow is modified according to the orbital velocity of the water particles.

**Wave forces**

Wave loads on the bare hull are calculated from potential theory. The total wave potential is known from the superposition of a diffracted wave potential $\phi_d$ and an incident wave potential $\phi_i$. The former is evaluated by using the previous mentioned boundary element methods code NEMOH which gives the complex form $F_d(\omega)$ of the diffraction forces. In time domain, the diffraction forces $F_{Diff}$ can be expressed from wave frequency $\omega$ and encounter frequency $\omega_e$ as follows:

$$F_{Diff} = \text{Re}\{\eta F_d(\omega_e)e^{-i(\omega t)}\}$$  \(25\)

The incident wave forces are computed under the so-called Froude-Krylov assumption where the yacht has no effect on the velocity field around her. Then, from Bernouilli-Lagrange pressure relation, diffraction forces can be estimated by integrating the pressure expressed at the center $B_i$ of each elementary surface $dS_w$ on the total wetted surface $S_w$.

$$F_{FK} = \sum_{i=1}^{N_f} GB_i \cap F_{FK_i}$$  \(26\)

with:

$$F_{FK_i} = -p_{FK}(z_i)dS_w$$

$$p_{FK}(z_i) = -\rho \left( \frac{\partial \phi_i}{\partial t} + \frac{1}{2}(\nabla \phi_i)^2 \right)$$  \(27\) \(28\)

The total wave forces $F_w$ are expressed as the superposition of the diffraction and Froude-Krylov components.
\[ F_w = F_{Diff} + F_{FK} \]  

**Propulsion forces**

The effect of the wind is modelled through a simplified quasi-steady model whose lift \( C_{l,\text{max}} \) and drag \( C_D \) coefficients of each individual sail, mainsail, jib and spinnaker are identified from the IMS measurement procedure (ORC, 2016).

As mentioned in the procedure, the reduction of heeling force by the crew trimming and changing sails is modelled by the \textit{Flat} and \textit{Reef} parameters. The former is used to simulate the reduction of the lift coefficient, while the latter is used to simulate the reduction of sail area.

Also, in order to reflect the fact that as sails are depowered, the height of the center of effort is reduced, a \textit{Twist} function depending on the amount of flat is introduced (Jackson, 2001).

\[ C_L = \text{Flat}.\text{Reef}^2.C_{l,\text{max}}(\beta_{eff}) \]
\[ C_D = \text{Reef}^2.[C_D(\beta_{eff}) + \kappa_0 C_L^2(1 + c_1\text{Twist}^2)] \]  

In first approximation, the dynamics of the sails is taken into account by adding a simple added mass coefficient \( m_{yy\text{Sail}} \) calculating from strip theory (Newman, 1977).

\[ m_{yy\text{Sail}} = \frac{\pi}{4} \rho_a c^2 dz \]  

Also, a log wind profile can be used in order to evaluate the true wind speed at the center of effort height above the water (Flay et al., 1995).

**STRUCTURAL MODELLING**

In order to investigate the effect of simple local deformation on the global behavior of the yacht, a simplified structural model based on beam theory was implemented to quantify the flexion of the hydrofoil.

**Direct integration**

As previously mentioned, the complex geometry of the foil is discretized with a finite number of elements. It is assumed that the element in contact with the hull is clamped. Then, the dynamical transverse loads \( F_{\text{Beam}_i} \) supported by the elementary beam, that are, loads that act perpendicular to the longitudinal axis of the beam are expressed as the superposition of gravity, inertial, added mass and hydrodynamic loads.

\[ F_{\text{Beam}_i} = F_{\text{Grav}_i} + F_{\text{Hyd}_i} + F_{\text{AM}_i} + F_{\text{Inertia}_i} \]  

Inertial loads are calculated as the mass of each element times the linear acceleration. This force is expressed at the center of gravity of the element and is taken into account in the calculation of the bending moment.
Figure 4 - 2D configuration of the loads on the hydrofoil

Since the hydrofoil’s tip is free, the problem to solve is similar to a cantilever problem. As shown in Figure 5, the unknown reaction $F_0 = [0, Y_0, Z_0, M_0, 0]$ is considered in the intersection $O$ between the foil and the hull. When considering all the elements and according to the dynamic equilibrium, the reaction can be computing as follows:

$$F_0 = -\sum_{i=1}^{N_{\text{elem}}} \left[ \begin{align*} & F_{\text{Grav}_i} + F_{\text{Hyd}_i} + F_{\text{AM}_i} + F_{\text{Inertial}_i} \\ & OB_i \wedge (F_{\text{Hyd}_i} + F_{\text{AM}_i}) + OG_i \wedge (F_{\text{Grav}_i} + F_{\text{Inertial}_i}) \end{align*} \right]$$

(34)

For each element, the bending moment $M_I$ in any section of the beam is calculated from a 2D point of view. A coordinate of the sections $x$ is introduced and the beam is sectioned as shown in Figure 5.

Figure 5 - 2D configuration of the loads on 1 element

For a given stiffness $EI$ of the beam, the 2nd order derivative $y''$ of the deflection of the foil along $y_{F_i}$ in $G_i$ is calculated from the formulation of the bending moment on the segment $O_{i-1}G_i$.

$$y'' = -\frac{M_{F_i}}{EI}$$

(35)

with, for $x_{G_i} > x_{B_i}$:

$$M_{F_i}(x) = -\left[ M_0 + (O_{i-1}.z_{F_i}) \cdot (F_0 \cdot y_{F_i}) + \sum_{j=1}^{i-1} (G_jO_{i-1}.z_{F_i}) \cdot ((F_{\text{Grav}_j} + F_{\text{Inertial}_j}) \cdot y_{F_i}) + \sum_{j=1}^{i} (B_jO_{i-1}.z_{F_i}) \cdot ((F_{\text{Hyd}_j} + F_{\text{AM}_j}) \cdot y_{F_i}) \right]$$

(36)

When an element is above the water, the forces are equal to zero, but its deflection is a function of the forces acting on the other elements.

Experimental investigation

The formula used in beam theory to calculate the shear force and bending moment were tested during an experimental campaign carried out in the hydrodynamic and ocean engineering tank of Ecole Centrale de Nantes to study the variations of internal forces on a notional hull form in rough waves (Horel et al., 2019). Figure 6 shows a comparison between the measured value of the vertical bending moment and the analytical reconstruction.
Figure 6 - Comparison between measured bending moment and analytical reconstruction

It can be noticed that analytical formulation reach to approximate with a relatively good accuracy the time evolution of the bending moment in the amidship section.

APPLICATIONS

A conventional and a foiler Open 60 are used to evaluate the ability of the presented models to predict the yacht behavior in calm water and in waves. According to IMOCA Class Rules (2018), their main features are given in Table 1.

<table>
<thead>
<tr>
<th>Feature</th>
<th>Conventional / Foiler</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length overall, LOA (m)</td>
<td>18.28</td>
</tr>
<tr>
<td>Width, B (m)</td>
<td>5.7</td>
</tr>
<tr>
<td>Water draft, T (m)</td>
<td>4.5</td>
</tr>
<tr>
<td>Total mass of the yacht, m (t)</td>
<td>7.7</td>
</tr>
<tr>
<td>Keel’s bulb mass, m_{bulb} (t)</td>
<td>3.1</td>
</tr>
<tr>
<td>Ballast and crew mass, m_{crew} (t)</td>
<td>0.5</td>
</tr>
<tr>
<td>Longitudinal position of the center of gravity of the hull from stern, x_{CoG} (m)</td>
<td>5</td>
</tr>
<tr>
<td>Vertical position of the center of gravity of the hull from baseline, z_{CoG} (m)</td>
<td>1.5</td>
</tr>
<tr>
<td>Roll radius of gyration, k_{x} (m)</td>
<td>2.035</td>
</tr>
<tr>
<td>Pitch radius of gyration, k_{y} (m)</td>
<td>4.57</td>
</tr>
<tr>
<td>Yaw radius of gyration, k_{z} (m)</td>
<td>4.57</td>
</tr>
<tr>
<td>Mainsail area, A_{MS} (m²)</td>
<td>149.2</td>
</tr>
<tr>
<td>Jib area, A_{Jib} (m²)</td>
<td>150</td>
</tr>
<tr>
<td>Spinnaker area, A_{Spi} (m²)</td>
<td>400</td>
</tr>
</tbody>
</table>

Table 1 - Open 60 main features

The features of the non-conventional modern Dali’s hydrofoil 2018 of the foiler Open 60 are given in Table 2. The coordinates X, Y and Z of the origin of the elements (El^i) are given in the foil reference frame (O, x_F, y_F, z_F) as described in Figure 2.
Table 2 - Foil main features

Lift and drag coefficients of the foil derived from an asymmetrical H105 profile whose section shape is given in Figure 7 below. The H105 hydrofoil section was designed to avoid laminar separation and ventilation when operating at low speeds and moderate angles of attack, while still having low velocities at small angles of attack to avoid cavitation at high speeds (Speer, 2001).

Calm water

Polar diagrams for non-deformable (rigid) foils

First, in order to evaluate whether the models get to predict with a reasonable accuracy the performance of the yacht, numerical results from ECN simulations are compared with reference speed polar diagram of the PRB Open 60. Even if the features of the simulated and real yachts are not exactly the same, this comparison reveals that the patterns of the speed polar diagram are similar.

Figure 8 shows the predicted performances of the conventional and foiler Open 60 for non-deformable appendages and true wind speed ($TWS$) of 20 knots. The cant of the keel is set to 30 degrees and no tilt is set.

From Figure 8 it can be noticed that the unconventional modern Dali’s foils 2018 are helping the yacht to sail faster for a large range of true wind angle ($TWA$), except for downwind conditions. However, these performances are highly depending on the chosen appendages features. Also, for a reverse foil, with the tip pointing downward, the gain in speed compared to a
conventional daggerboard Open 60 seems to be significant for a range of TWA from 10 to 140 degrees. But with the chosen foil feature, the gain is smaller than the gain of the modern Dali’s foils 2018.

From the Polar diagram of total lateral and vertical forces on the foil (Figure 9), it can be noticed that the reverse foil acts more like a daggerboard by mainly creating anti-drift force while the modern Dali’s foil 2018 is creating much more vertical lift force that helps the yacht to increase her performances.

Figure 9 - Polar diagram of forces on the foil (values in tons) (TWS=20knots)

Constant acceleration with deformable modern Dali’s foil 2018
This case is performed for 3 different stiffness (non-deformable, realistic and flexible) in order to evaluate the ability of the structural model to predict the time evolution of the deflection of the foil. The yacht is fixed in sway, heave, roll, pitch and yaw. The acceleration in surge is remained constant and equal to 0.5m.s$^{-2}$ up to a forward speed of 10m.s$^{-1}$. Above that, the speed is kept constant. The rake angle of the foil is set to 0 degree.

The lateral and vertical deflection of the tip can be respectively evaluated from Figure 10 and Figure 11.

Figure 10 - Lateral deflection of the tip

Figure 11 - Vertical deflection of the tip

With realistic stiffness whose values are given in Table 2, the lateral deflection of the tip at 10m.s$^{-1}$ reaches a value of 75mm. When the stiffness is reduced, the deflection is increasing.
In Figure 12 and Figure 13, it can be noticed that the stiffness and the deflection of the foil seem to have a low influence on the drag and lift forces.

![Figure 12 - Longitudinal drag force on the foil](image1)

![Figure 13 - Vertical lift force on the foil](image2)

However, as can be observed in Figure 14, the lateral force on the flexible foil seems to be highly influenced by the deflection of the foil. This deflection changes the loading distribution on the foil and modifies the resulting force. The global deflection and the loading distribution on the foil are presented in Figure 15.

![Figure 14 - Lateral drift force on the foil](image3)
The global effect of the flexion of the foil on the yacht’s performance is evaluated from the speed polar diagram given in Figure 16.

It can be noticed that for true wind angles from 50 degrees to 130 degrees, the presented method get to capture the variations of the performances between the non-deformable foil and the flexible foil. In the particular case of this study, the maximum gap in speed is equal to 1 knot and is experienced for a true wind angle of 110 degrees. This shows that taking into account the effects of the foil’s deformation is an important step in the design stage.

This loss in speed can be partly explained from the fact that the flexible foil generate less lift than the non-deformable one, creating less righting moment and then does not allow to trim the sails to their best settings.

The differences in the vertical force can be seen in Figure 17.
These previous results show that for the same optimum targets, the stiffness has an influence on the reached speed. In the next section the influence of the deflection of a flexible foil on the performance of the yacht sailing in quartering head waves is studied. In order to reach a similar speed, the target \( TWA \) is 70 degrees for the conventional Open 60 and 50 degrees for the foiler.

Regular sea state

Simulations in regular waves are the most penalizing conditions, since the motions of the yacht are excited with a unique encounter frequency. However, they are given precious information about the dynamic behavior of the yacht. In this test case, the true wind speed is 20 knots and the wave is a regular Airy wave with a wavelength that is twice the yacht length and a wave height to wavelength ratio equals to 0.02. The heading between the yacht’s longitudinal axis and the wave direction is 70 degrees. Since the tests are performed in 6DOF, only the rudder deflection is controlled using a PD controller. Sails deflections and rake angle are maintained to constant values.

In these conditions, for a similar target speed between the conventional Open 60 and the foiler, the dynamic behavior can be evaluated from phase diagrams and amplitude spectra.

Figure 18 is a phase diagram in pitch motion that shows the dynamic evolution of the pitch angular rate according to the pitch angle. It can be noticed that the foiler seems to sail with the bow up (negative pitch angle) and experiences less pitch motion than a conventional Open 60. Also, by being influenced by the wave particle orbital velocities, the flexible foil seems to increase the amplitude of the pitch motion.

![Figure 18 - Phase diagram in pitch (TWS=20knots)](image)

The attitudes of the conventional Open 60 and the foiler Open 60 with flexible foils can also be evaluated and qualitatively compared from Figure 19, where the yachts have the same position compared to the wave crest.

![Figure 19 - Attitude of the yacht in waves, a) conventional, b) foiler Open 60 (TWS=20knots)](image)

In Figure 20, it can be noticed that the foiler seems experiencing higher amplitude variation than the conventional.
Moreover, the foil seems to act like a low pass filter on the yacht speed variations since the second harmonic of the speed variation is smaller. In our study, the foiler’s performances seem to be affected by the wave encounter frequency. Since the true wind angles are different between the foiler and the conventional Open 60, this test case helps to evaluate the motions and to capture the variations of the performance.

**Figure 20 - Speed amplitude spectrum (TWS=20knots)**

**Irregular sea state**

The simulation of irregular wave conditions is performed in order to evaluate the influence of the foil structural deformation on the global behavior of the yacht in a more realistic sea state than regular waves. The wave characteristics are chosen according to conditions encountered off the shore of Auckland, New-Zealand (Pickrill et al., 1979). The wave spectrum is a Bretschneider spectrum with a significant wave height of 1 meter and a wave peak period of 8 seconds. The same wave phases are chosen for each yacht setup. The wave amplitude spectrum is shown in Figure 21.

**Figure 21 - Wave amplitude spectrum**

According to this wave spectrum, the motions of the yacht are evaluated in frequency domain. The amplitude spectrum of the pitch and heave motions are given in Figure 22 and Figure 23. As for regular waves, the flexible foil tends to increase the amplitude of the motions. Also, from the pitch amplitude spectrum, it can be noticed that for all the stiffness, the pitch motion of the foiler Open 60 seems to be also governed by the low frequency content of the wave spectrum.
Unlike for regular waves, the amplitude spectrum of the forward speed of the yacht sailing in irregular waves shown in Figure 24, show that the amplitude of the speed variations are similar for conventional and foiler Open 60. The peak frequency is sharper and the maximum value of the peak is higher for foilers. However, the use of a flexible foil tends to decrease the peak amplitude.

The attitude of the yacht with flexible foil in irregular waves and the qualitative aspects of the foil flexion deformation are also shown in Figure 25.
CONCLUSION

The objective of the paper was to evaluate through simplified modelling the influence of basic structural deformations of hydrofoils on the global yacht’s behavior for several sea conditions.

The presented method has the ability to capture the variations of the performance of an Open 60 according to the stiffness of the appendages. The gain and loss were discussed for different test cases in calm water and in waves. Depending on the sea conditions, the effects of the simple deformation such as flexion of the foil were highlighted.

The application of system-based modelling coupled with dynamical structural deformations has proven that it can help to understand the global behavior of a yacht with deformable foils. The presented work is a first step to better understand the physical phenomena that drive the yacht’s performances.

Then, based on this work, further studies will be carried out in order to evaluate the effect of other modes of deformation, such as torsion of the foil that could be responsible for higher variations in the predicted yacht performances.

Also, validation data from sea trial testing or model tests are required in order to validate such a simplified method.

ACKNOWLEDGMENTS

Authors want to warmly acknowledge PRB sailing team for providing the speed polar diagrams that made possible comparisons presented in this paper.

Authors also want to acknowledge Antoine Connan and his colleagues for their last year student project at ECN on trying to couple the FSI solver Cast3m with a DVPP.

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