Design and analysis of folding propulsion mechanism for hybrid-driven underwater gliders

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**A B S T R A C T**

This paper presents a design of foldable propellers for a hybrid-driven underwater glider. The design ensures that the propellers are fully closed when the glider is working in buoyancy-driven gliding mode, and become fully open to provide propulsion when necessary. The hydrodynamical moments during the folding and unfolding processes are analyzed and computed using computational fluid dynamics (CFD) methods. Torsion springs are used as key components in the folding and unfolding mechanism. The stiffness of the torsion springs are designed to achieve balance between the mechanical and hydrodynamical moments acting on the propellers. It is shown that comparing to a conventional unfoldable design, the foldable propellers may achieve a significant reduction in drag force when the glider is operating in the gliding mode. Pool experiment results demonstrate the effectiveness of the folding mechanism when installed on an underwater glider.

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1. Introduction

In the past decade, Autonomous Underwater Gliders (AUGs) are widely used in oceanographic research (Eriksen et al., 2001; Claus et al., 2012). AUGs are a class of underwater vehicles that change their volume and buoyancy to cycle vertically in the ocean and use lift forces generated by a pair of symmetrical wings attached to the glider hull to convert this vertical velocity into forward motions (Sherman et al., 2001; Yu et al., 2011, 2013). As a result, the glider usually follows a sawtooth motion pattern in the vertical plane. AUGs can also generate helical motion using a rudder or a rolling mass system (Graver, 2005; Zhang et al., 2013; Hussain et al., 2011). AUGs have an excellent endurance up to a few months or more (Webb et al., 2001). In comparison, the autonomous underwater vehicles (AUVs) in common use today perform missions measured by days (Paull et al., 2013; Hyakudome et al., 2002). However, the speed of existing AUGs is lower than that of AUVs (Wang et al., 2011). Thus the slow speeds of AUGs present challenges for navigation in areas of strong current. The vehicle maneuverability will be greatly affected leading to failure in performing observational operations (Merckelbach et al., 2008). In addition, the zigzag trajectory does not allow leveled flight hence preclude observation of dynamic processes in horizontal plane or in shallow water column (Hobson et al., 2012; Zhu et al., 2012; Tian et al., 2014).

Hybrid-driven underwater glider (HDUG) combines the strength of conventional buoyancy-driven AUGs with the propeller-driven AUVs, thus may offer a relatively higher endurance and a better capability to overcome strong current. Some work (Caffaz et al., 2010; Claus et al., 2010b; Isa and Arshad, 2012). Claus et al. (2010a) developed a HDUG with aeronautical foldable propeller placed in the stern of a Slocum glider to achieve leveled flight. Vehicle Control Technologies, Inc. (VCT) has developed a lower-cost, expendable glider (xGlider) that will last sufficiently longer so that recovery is not required. It is worth mentioning that this kind of underwater glider can be easily upgraded to be an AUV with a maximum speed of 2 knots by using modular design method. The ALCEN company have developed a SeaExplorer glider that can operate in hybrid ‘AUV/glider’ mode if necessary. Designs of HDUG have also been reported by other researchers (Wang et al., 2011).

Most existing HDUGs use conventional fixed-blade propellers. This design is simple, but has drawbacks. The vehicles will experience parasitic drag due to trailing propellers and associated appendages when operating in the glider buoyancy-driven mode, leading to reduced efficiency in operation (MacKenzie and Forrester, 2008; Tian et al., 2012). In order to reduce the impact of drag by the propellers, some researchers have proposed a foldable propeller design (Claus et al., 2010a). If the increase is taken as a
percentage of the overall glider drag, it would add to the drag force of a glider by 15–18% when the folding propulsion module is not in use (Claus, 2009). But the blades of the folding propeller are unable to close entirely if not in use in water. An improved folding propeller must require the blades fully folded in when the propeller is not in use, and unfolded out when the propeller is in use. Foldable propellers have been used on sailing yachts to reduce drag while under sail for improved performance in competitions (MacKenzie and Forrester, 2008). However, the blades resetting mechanism of the foldable propellers in sailing boats have a large size which does not fit in underwater vehicles. Foldable propellers are also used on model sailplanes to reduce air drag when the power is turned off (Patel et al., 2008). Since the propellers are usually installed in vertical direction, the blades can be easily folded by the force of gravity without any resetting mechanism. So far foldable propellers are rarely used in autonomous underwater vehicles, because the propellers are usually installed at the stern in horizontal direction. The blades cannot be perfectly folded together without enough resetting force when not in use. A foldable propeller applied in HDUG will greatly improve vehicle maneuverability and endurance. Folding propulsion mechanism dedicated for HDUG will be designed and analyzed in the paper. Similar work has not been found in published literature.

In our work, the foldable propeller is aft located on the vehicle hull, its two blades can be opened or closed according to the requirements of operational mission, illustrated in Fig. 1. Results imply that the HDUG with the designed foldable propeller shows advantage in low drag performance over that with fixed-blade propeller.

This paper is organized as follows. Section 2 introduces the foldable propeller design. In Section 3, we built the mathematical model. And in Section 4 presents the verification of the folding mechanism based on CFD data. Section 5 presents the experimental data and analysis. The last section draws conclusions.

2. Mechanical design

A resetting mechanism has been adopted to fold the blades when the HDUG is traveling in the gliding mode at low speed. The mechanism is based on a resetting spring that keeps the blades in folding states when the motor shaft is not rotating. When the drive motor starts to spin the shaft, the blades will experience centrifugal force at high spinning speed to unfold the blades, shown in Fig. 1. The effect of the hydrodynamic forces on the folding mechanism is examined in the next section.

Two or more blades are usually chosen for a ship propeller. It is generally believed that if the plate area and diameter of the propeller is given, when the number of blades increases, the blades interference effect increases, resulting in a lower efficiency. For best efficiency, it is therefore essential to keep the number of blades as low as possible (Bellingham et al., 2010). In addition, propeller with less number of blades has benefit to avoiding vacuoles. Therefore, according to existing AUG’s hydrodynamic drag and some other main parameters estimated (Zhang et al., 2012), we chose a two-blade design by applying the propeller atlas design method.

The propulsion mechanism designed in this paper includes the following components: a drive motor, a reset unit (torsion spring), two blades, and a hub, shown in Fig. 2. Four torsion strings are installed inside the hub. The stiffness of the torsion string should be designed carefully. The blades cannot successfully unfold from its folding position if the torsion coefficient is too high (i.e., the springs are too stiff). On the contrary, the blades cannot successfully fold completely from its unfolding position if the torsion coefficient is too low (i.e., the springs are too soft). Therefore, mathematical model and analysis are needed to determine the feasible range of torsion coefficient.

3. Balance of moments

Much of our work has been examining the balance of moments during the process of folding and unfolding of the propeller blades. According to this analysis, the required stiffness of the torsion springs has been determined.

3.1. Balance of moments during blades unfolding

When the propulsion mechanism needs to work to generate thrust, the drive motor starts to spin and drives the hub to rotate at a constant speed. Then the blades unfold from its folding position due to the centrifugal force and the hydrodynamic thrust force acting on the blades face. Therefore, the blade opening angle \( \theta \) varies approximately from 0° to 90°. The blades will stop at a position where the whole propulsion mechanism reaches a dynamic equilibrium by the action of the centrifugal moment, the hydrodynamic thrust moment, the torsion spring torque, and the gravitational moment on the blades.

Through the above analysis, the moment balance equation is obtained in the blade unfolding process as:

\[-M_L - M_G + M_T + M_C < 0\]  \( (1) \)

where \( M_L \) is the centrifugal moment, \( M_G \) is the hydrodynamic thrust moment, \( M_T \) is the torsion spring torque, and \( M_C \) is the gravitational moment, all acting on the blade shaft. The positive directions of the moment are defined to follow the right hand rule, as shown in Fig. 3(a).
where two functions are defined as follows: \( f(\omega) = m_p \omega^2 |R_C| \) and \( f(\theta_p) = \left[ \frac{\omega}{2} \cos(2\theta_p) + r_1 \sin \theta_p \right] - 1 \). Then the derivative of \( f(\omega) \) is obtained with respect to \( \omega \) by \( \frac{df}{d\omega} = 2m_p \omega |R_C| \). Assuming the propeller rotation is right-handed, \( \omega > 0 \). Therefore \( f(\omega) \) is a monotone increasing function with respect to \( \omega \). The derivative of \( f(\theta_p) \) with respect to \( \theta_p \) is \( \frac{df(\theta_p)}{d\theta_p} = \frac{\omega}{2} \sin(2\theta_p) \). Therefore, \( f(\theta_p) \) is defined as \( \theta_n = \arcsin \left( \frac{1}{2} \left( \frac{(\frac{\omega}{2})^2 + 8 - \frac{r_1}{|R_C|}}{1} \right) \right) \), then the following conclusions are obtained.

For \( \frac{df(\theta_p)}{d\theta_p} > 0 \), then
\[ 0 < \theta_p < \theta_m \]
For \( \frac{df(\theta_p)}{d\theta_p} < 0 \), and then
\[ \theta_m < \theta_p < 90° \]

Therefore, the function \( f(\theta_p) \) monotonically increases for \( \theta_p \in [0, \theta_m] \), and monotonically decreases for \( \theta_p \in [\theta_m, 90°] \). It is easy to find out that the two functions \( f(\omega) > 0 \) and \( f(\theta_p) > 0 \) in \( \theta_p \in [0, 90°] \), so the centrifugal moment values are also greater than 0. Therefore, \( M_c \) has the same monotonicity as \( f(\theta_p) \) for a given \( \omega \).

From the above analysis, it should be noted that the centrifugal moment increases firstly and reduces afterward, and reaches the maximum at \( \theta_p = \theta_m \) during the process when the blade opening angle increases from 0 to 90°.

**3.1.2. Hydrodynamic thrust moment**

In sailboat propeller analysis, formulas for the hydrodynamic moment and coefficients are derived (MacKenzie and Forrester, 2008). In addition, the hydrodynamic thrust moment \( M_h \) is also a driving moment for unfolding the blades, so it can be derived by the following equations:

\[ M_h = f(\rho, n, D, J) = K_{mh} \rho n^2 D^5 \]

\[ K_{mh} = K_{m1} + K_{m2} J + K_{m3} J^2 \]

where \( \rho \) is the water density, used by ship propeller design \( \rho = 104.6 \, \text{kgf} \, \text{s}^2/\text{m}^4; \) \( J \) is the advance ratio; \( K_{m1} \) is the hydrodynamic moment coefficient; \( K_{m2} \) and \( K_{m3} \) are all hydrodynamic coefficients which will be calculated in Section 4.

**3.1.3. Gravitational moment**

The value of the gravitational moment \( M_g \) depends on the rotational position (i.e., the angle \( \epsilon \) between \( z_1 \)-axis and the plumb line.) of the propeller as shown in Fig. 3(b). When the propeller spins to the position as in Fig. 3(b), then the function is obtained as follow:

\[ M_g = m_p g x_{1C} \cos \epsilon \]

Therefore, if \( |\epsilon| \in [0, 90°] \), then \( M_c > 0 \) acts as a retarding moment for blades unfolding. If \( |\epsilon| \in [90°, 180°] \), then \( M_c < 0 \) acts as a driving moment for blades unfolding. But in order to ensure reliability, the maximum \( M_{c,max} = m_p g x_{1C} \) at \( |\epsilon| = 0 \) should be considered as a retarding moment.

**3.1.4. Torsion spring torque**

Unlike the centrifugal moment and hydrodynamic thrust moment, the torsion spring torque \( M_t \) is a retarding moment when the blades unfold. It counteracts \( M_c \) and \( M_h \). The value of \( M_t \) can be calculated by the following formula:
\[ M_b = K_s(\theta_p + \theta_{p0}) \]  

(9)

\[ K_s = \frac{E_s I_b}{n d^2 N_s} \]  

(10)

where \( E_s \) is the Young's modulus with unit MPa, \( I_b \) is the area momentum of inertia of the torsion spring wire with unit mm\(^4\), and \( k = n d^3/64 \) for circular cross-section; \( d_{c1} \) is the diameter of the torsion spring wire with unit mm; \( \theta_{p} \) is the angle of deflection from rest position equaling to the blade opening angle with unit radian; \( d_{c2} \) is the torsion spring mid diameter with unit mm; and \( N_s \) is the number of active windings. \( \theta_{p0} \) is a preset blade opening angle so that \( M_s \) when \( \theta_p = 0 \) to provide enough torque to balance the gravity moment \( M_c \).

3.2. Balance of moments during blades folding

When the driving motor stops spinning, the blades are folded by the moments generated by the water pressure acting on the back surface of a blade and the torsion spring force. The moment balance equation is

\[ M_0 + M_s + M_c > 0 \]  

(11)

where \( M_0 \) is the hydrodynamic moment acting on the blade shaft generated by the water pressure on the back surface. \( M_0 \) is a driving moment during the process of blades folding and can be obtained by using CFD software packages. Unlike the process of blades unfolding, \( M_s \) is a driving moment during the process of blades folding. Here, in order to ensure reliability, the minimum \( M_c \) should be considered.

3.3. Torsion coefficient

From the description in Sections 3.1 and 3.2, the necessary and sufficient conditions that should be satisfied to successfully unfold and fold the blades are:

\[
\begin{align*}
- M_c - M_0 + M_s + M_c_{\text{max}} &< 0 \\
(M_0 + M_s + M_c)_{\text{min}} &> 0
\end{align*}
\]  

(12)

then the torsion coefficient \( K_s \) is determined by substituting Eqs. (4), (6), (9) and (8) into inequality (12).

\[
\left\{ \begin{array}{l}
\frac{1}{\theta_p} \left( -m_p g R_{1c} \cos \theta_p \cos \epsilon - M_0 \right)_{\text{max}} < K_s \\
\left\{ \begin{array}{l}
\frac{1}{\theta_p} \left( m_p R_{1c} n^2 \left( \frac{R_{1c}}{2} \sin(2\theta_p) + r_{c1} \cos \theta_p \right) + K_{m0} m^3 D^5 \right) \\
- m_p g R_{1c} \cos \theta_p \cos \epsilon \end{array} \right\}_{\text{min}} \end{array} \right.
\]  

(13)

4. Verification of foldable propeller

4.1. Model of the propulsion mechanism

When fluid flows past the vehicle underwater, viscous turbulent flow happens at the fluid vehicle boundary layer. The Reynolds Averaged Navier–Stokes (RANS) equations method is for hydrodynamic analysis of underwater vehicles. A computational fluid dynamics (CFD) software package, the CFX, that implements RANS to calculate the hydrodynamic coefficients. CFD simulation requires knowledge of the Reynolds number, proper selection of a turbulence model, and proper settings for grid generation and boundary conditions.

The Reynolds number is computed as

\[ R_e = \frac{\rho V D}{\mu} \]  

(14)

where \( \rho = 1025 \text{ kg/m}^3 \) is the water density, \( V \) is the propeller velocity, \( D \) is the propeller diameter, and \( \mu \) is fluid viscosity coefficient. The propeller velocity ranges from 0.28 m/s to 1.68 m/s. Hence the Reynolds number ranges from \( 6.7 \times 10^4 \) to \( 4 \times 10^5 \). So low Reynolds turbulence model \( k-\epsilon \) is used in our work (Jagadeesh et al., 2009). This turbulence model has high accuracy in the prediction of flows which involve both velocity decrease and flow separation astern underwater vehicle hull. In addition, it is proved that the \( k-\epsilon \) model can accurately simulate the flow parameters in the proximity of the propeller plane (Hayati et al., 2012). The automatic wall function switches between standard low Reynolds formulation and wall functions on the basis of the grid spacing of the near-wall cell. Usage of this kind of wall treatment leads to improving the accuracy of the flow simulation near the walls surfaces. The turbulence intensity of the inflow is set to 5%. The absolute conversion criteria for the residuals of the equations including continuity, momentum, turbulence kinetic energy and turbulence specific dissipation rate are set to \( 10^{-5} \) throughout to obtain the results with acceptable accuracy.

Multi Reference Frame (MRF) method is used to create a mesh for the flow domain in this paper (Hayati et al., 2013). A rectangular shape control volume is assumed around the folding

![Image](a)
propeller with inlet and outflow boundary conditions, as shown in Fig. 4(a). A moving mesh zone is generated around the folding propeller for better prediction of the flow between the propeller blades in Fig. 4(b). A no-slip wall condition is used on all solid surfaces, including the propeller blades. The global domain is surrounded by external walls located far from the propeller. The domain dimensions are considered large enough to avoid external effects such as reversed flows and induced pressure on the performance of the propeller. The stationary domain boundary is 28D x 14D x 10D and the moving domain boundary is 1.5D for diameter and 1D for length. The rotating speed of the propeller shaft is set to 600 rev/min (revolutions per minute) throughout this simulation. The inflow velocity varies between 0.28 m/s and 1.68 m/s (Fig. 5).

Based on the main parameters of Sea-Wing glider (Zhang et al., 2013), the final main physical parameters of the folding propulsion mechanism obtained are given in Table 1 using standard procedures for propeller design (Huang, 2012).

<table>
<thead>
<tr>
<th>List of items</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Propeller diameter, D</td>
<td>0.28 m</td>
</tr>
<tr>
<td>Pitch/propeller diameter ratio, d_p/D</td>
<td>0.786</td>
</tr>
<tr>
<td>Hub/propeller diameter ratio, d_h/D</td>
<td>0.14</td>
</tr>
<tr>
<td>Single blade mass, m_p</td>
<td>0.022 kg</td>
</tr>
<tr>
<td>Number of blades, N_b</td>
<td>2</td>
</tr>
<tr>
<td>Torsion spring wire diameter, d_s1</td>
<td>1 mm</td>
</tr>
<tr>
<td>Torsion spring mid diameter, d_s2</td>
<td>8 mm</td>
</tr>
<tr>
<td>Young’s modulus of torsion, E_S</td>
<td>93.2 GPa</td>
</tr>
<tr>
<td>Number of active windings of torsion spring, N_s</td>
<td>4</td>
</tr>
<tr>
<td>Number of torsion springs per blade, Z_s</td>
<td>2</td>
</tr>
</tbody>
</table>

4.2. Open water performance

The propeller performance is needed to establish the kinematic and dynamic models of HDUG (Nemati Hayati et al., 2013; Broglio et al., 2012). CFD method is then used to compute the hydrodynamic performance for the designed propeller. Simulations are carried to obtain the open water performance for the propulsion unit and the low drag performance for the HDUG’s hull with propeller attached.

4.2.1. Parameters for propulsion generation

In this paper, the open water performance is described by the follow two forward-performance characteristics of propellers:

(i) Torque coefficient:

\[ K_{M_B} = \frac{M_B}{\rho n D^5} \]  \hspace{1cm} (15)

where \( D \) is the propeller diameter, \( M_B \) is the torque on the blade shaft, \( n \) is the spinning speed.

(ii) Advance ratio:

\[ J = \frac{\sqrt{V^2 - n^2 D^2}}{n D} \]  \hspace{1cm} (16)

where \( V \) is the forward speed of the propulsion mechanism.

In folding propulsion design practice, the relationship between these two Eqs. (15) and (16) is used routinely to characterize individual designs of propulsion devices by fixing \( n \), one would be able to obtain \( J \) for any desired value of velocity \( V \). By using the CFD, the values of \( M_B \) can be determined for a given \( J \). Then the corresponding \( K_{M_B} \) can be determined using Eq. (15).

The results of open water performance include the torque coefficient at the blades opening angle 90° (i.e., the blades are fully open) and the hydrodynamic moment \( M_B \) when the blades opening angle increases from 0 to 90°.

4.2.2. Hydrodynamic thrust moment at \( \theta_p \in [0, 90°] \)

The drive motor rotational speed is selected at \( n = 600 \) rev/min. Then the calculation is performed respectively at different blade opening angles \( \theta_p \in [0, 90°] \) (with 15° increments) and different advance coefficients \( J \in [0.1, 0.6] \) (with 0.1 increments, corresponding to \( V \in [0.28, 1.68] \) m/s). The configurations are illustrated in Fig. 4. The \( K_{M_B} \) computed are plotted in Fig. 6.

As shown in Fig. 6, at any fixed \( \theta_p \), the smaller \( J \) is, the bigger \( K_{M_B} \) is. When the vehicle advance speed \( V \) equals zero, the hydrodynamic moments on the blade reaches the maximum value. According to Eq. (1) in this case, it is best for unfolding the blades. Results indicate that the larger \( J \) is, the more difficult it is to unfold the blades. But at the same value for \( J \), it is found that \( K_{M_B} \) is not strictly monotonous with the increasing \( \theta_p \). This is because if the two blades are too close, the hydrodynamics will be complex and irregular due to the interference. So in the propulsion design, the peak \( K_{M_B} \) across the range of \( \theta_p \) should be noted.

4.3. Hydrodynamic resistant moment

In order to analyze the folding process, it is necessary to calculate the hydrodynamic resistant moment \( M_D \) in Eq. (11) in the similar way as calculating the hydrodynamic thrust moment. But in this case, the driving motor spinning speed is set to zero. The blade opening angles are also set as \( \theta_p \in [0, 90°] \) (with 15° increments). And the CFD results are shown in Fig. 7.

At a given speed, the results show that the hydrodynamic resistant moment \( M_D \) increases with the increasing of blade opening
angle $\theta_p$, and then decreases, as shown in Fig. 8. The reason for this phenomenon is that the hydrodynamic resistant moment $M_D$ is generated not only from the forward flow drag force, but also from the lift force. So according to airfoil theory, when the flow direction is fixed, the angle of attack increases with the increase of $\theta_p$, resulting in the increasing lift force. But when the opening angle exceeds the stall angle of attack, the lift force begins to decrease. Although during the whole process, the drag force increases as shown in Fig. 9, $M_D$ increases firstly and reduces afterward with the increase of $\theta_p$ (see Fig. 8). Figs. 8 and 9 show that the stall angle of attack is approximately $50^\circ$. This means that during the blades changing from fully open status to fully closed status, the maximum hydrodynamic resistant moment occurs not at $\theta_p = 90^\circ$ but at $\theta_p = 50^\circ$.

4.4. Determine the spinning speed of the driving motor

From Fig. 6, the hydrodynamic thrust moment $M_H$ reduces with the increasing of advance ratios. Therefore, an operation with an initial gliding velocity should be considered for the design of folding mechanism. When the HDUGs driving mode is changed from buoyancy-driven mode to propeller-driven mode, the forward speed is maintained. So a normal gliding velocity equaling 0.5 knots is selected to find the relationship between the hydrodynamic thrust moment and the blade opening angle at different spinning speeds $n$ shown in Fig. 10. The figure shows that the hydrodynamic thrust moment $M_H$ increases with the increasing of spinning speed. So higher spinning speeds are beneficial to blades unfolding.

Set $M_U = -M_M - M_L + M_S + M_G$, herein $M_M$ is the net moment during the dynamic process of blades unfolding. Then according to inequality (12), the blades can be fully expanded for any $\theta_p \in [0, 90^\circ]$ if and only if $M_M < 0$. The form of $M_M$ is described as follows:

$$M_M = -M_H - \frac{m_p R_C n^2}{100} \left( \frac{R_C}{2} \sin(2\theta_p) + r_{11} \cos \theta_p \right) + \frac{E_{11} \theta_p}{\pi S N_S^2} + m_p g R_{1c} \cos \theta_p$$

After putting the relevant parameters into Eq. (17), $M_M$ is calculated at the spinning speeds $n \in [600, 1000]$ rev/min and plotted in Fig. 11. The figure shows that some portion of the $M_M$ curve lies in the upper half-plane at $n < 800$ rev/min, which does not meet the requirements. This results indicate that the designed blades of the propulsion mechanism should be able to open completely at $n > 800$ rev/min.

4.5. Determine the preset blade opening angle

Set $M_F = M_H + M_L + M_G$, herein $M_F$ is the net moment during the dynamic process of blades folding:

$$M_F = M_D + \frac{E_{11} \theta_p}{\pi S N_S^2} - m_p g R_{1c} \cos \theta_p$$

Then according to inequality (12), the blades can be fully folded for any $\theta_p \in [0, 90^\circ]$ if and only if $M_F > 0$. After putting the relevant
The solid lines describe the upper bound in Fig. 12. The net moment $M_f$ during the dynamic process of blades unfolding as a function of the increasing of blade opening angle $\theta_p$ at different hub rotational speeds $n$. The results show that the average drag reductive ratio of the HDUG of the folding propulsion mechanism. The drag reductive ratio $\tau$ is defined as follows:

$$\tau = \frac{D_U - D_F}{D_U}$$

where $D_U$ is the hydrodynamic drag when the blades are fully open, and $D_F$ is the hydrodynamic drag when the blades are fully closed.

The blades will be fully folded when the HDUG operates as a glider. Since the gliding velocity is usually slower than 2 knots, the whole system is 45%, and this value can reach higher than 60% at $V = 0.2$ m/s. This shows that the method using the folding propeller instead of traditional fixed-wing propeller effectively reduces the gliding drag of the HDUG.
5. Pool experiment and analysis

In-water experiments have been carried out in the indoor testing pool of Shenyang Institute of Automation (SIA) in order to validate the folding propulsion mechanism. In some experiments, the propeller is mounted on an HDUG, as illustrated in Fig. 15. For another set of the experiments, the folding mechanism alone has been tied to a fixed location underwater. The blade folding angle at different spinning speeds is recorded by a camera. During the experiments where the folding mechanism is installed at a fixed location, the motor speed is increased from 0 to 1000 rev/min at intervals of 100 rev/min. Fig. 16 demonstrates the unfolding process underwater. Fig. 17 plots the experimental data of the blade opening angle related to the spinning speed during the experiment and the comparison between experimental data and theoretical prediction.

The results show that the blades of the propeller can fully unfold when spinning speed exceeds 8000 rev/min based on the designed parameters of the mechanism propulsion. From Figs. 11 and 17, the pool experiment results and the theoretical prediction have a good agreement up to some numerical errors. The experimental data curve increases slowly at small speed and then rapidly increase after the opening angle exceeds $\theta_p = 45^\circ$ approximately. It indicates that the largest counter-acting torque during the unfolding process happened near $\theta_p = 45^\circ$, which has been predicted by the theoretical results in Fig. 11.

In addition, the exploration of the option of using a softer torsion spring with wire diameter $d = 0.8 \text{ mm}$ instead of the original design with $d = 1 \text{ mm}$ in the propulsion mechanism is conducted in the pool experiment. The experimental results are shown in Fig. 18. As mentioned earlier in Eq. (10), if the torsion spring wire diameter is smaller, then the torsion coefficient is smaller. So it shows from the figure that the blades can unfold more easily when the torsion coefficient becomes smaller. However, the blades cannot fold fully when the motor stopped spin in this case.

During the experiments while the HDUG is free to move, it is found that if the preset blade opening angle $\theta_p < 10^\circ$, then one of the two blades cannot fully fold when the HDUG is moving

Fig. 13. The torsion coefficient as the function of the increasing of blade opening angle at different preset blade opening angle $\theta_p$. (a) $\theta_p = 5^\circ$. (b) $\theta_p = 10^\circ$.

Fig. 14. The CFD results for low drag performance of the HDUG hull with respect to the blade opening angle at 90° and 0° respectively. (a) Towing drag. (b) Drag reductive ratio.

Fig. 15. The folding propulsion mechanism is installed in the tail of an HDUG and being tested in the pool of SIA.
This actually justifies the theoretical results in Fig. 13 where the preset blade opening angle is set at $\theta_p = 10^\circ$. It is found that both blades can fully fold when $\theta_p = 10^\circ$.

6. Conclusion

A foldable propeller is designed for a hybrid-driven underwater glider that can be fully closed when the glider is operating in buoyancy driven gliding mode. CFD analysis shows that such design significantly reduces the hydrodynamic drag comparing to un-foldable design. In order to ensure that the propeller can be fully open or closed, a mechanism is also designed where torsion springs are used to provide mechanical moments to balance the hydrodynamical moments. Such mechanism is feasible with the range of stiffness chosen. Pool experiment results confirm that the design is feasible for potential application to HDUG.

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