

## Article

# Structural Design and Performance Analysis of Underwater Tethered Vehicles

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## Abstract

An underwater towed vehicle serves as an effective and widely applicable mobile marine observation platform. Existing towed vehicles rely heavily on cables for depth adjustment. They also suffer from poor hydrodynamic efficiency and insufficient instrument space. To address these limitations, this study developed a novel compensation control system. This system regulates the vehicle's vertical movement and cable deployment by controlling the attack angles of its front and rear hydrofoils. The Myring profile was selected as the base design for the towed vehicle, offering excellent hydrodynamic performance, ample internal space, and cost-effectiveness. To verify the system's reliability, critical components were meticulously designed and calibrated. Hydrodynamic simulations confirmed that adjusting the hydrofoil angle effectively controls vertical motion, with stress and deformation in the lifting mechanism and cable connectors meeting design specifications. Additionally, the overall drag resistance remains low, while the lift generated by both hydrofoils satisfies depth adjustment requirements. This research provides robust numerical foundations for developing vertical control strategies, optimizing operational conditions, and conducting subsequent sea trials of towed vehicles. Quantitative comparison with the conventional scheme indicates that the proposed structure cuts total drag by 21.6%, boosts depth adjustment efficiency by 47.3%, and achieves a 32% higher hydrofoil lift-drag ratio, accompanied by a structural safety factor of 1.8 and maximum deformation of only 1.711 mm under rated working conditions.

**Keywords:** underwater towed vehicle; hydrofoil control; Myring hull; hydrodynamic simulation; vertical depth compensation



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## 1. Introduction

Underwater towed vehicles represent an effective and widely utilized mobile marine observation platform [1]. They play a pivotal role in underwater water body detection [2] and resource exploration [3] across marine, lake, and river environments, featuring extensive operational range, low drag resistance, and reliable performance [4]. The propulsion of the towed vehicle primarily relies on the traction force exerted by the carrier platform via the tow cable, with its navigation trajectory and operational scope fully controlled

by the platform [5], enabling these platforms for large-scale underwater search and exploration missions [6–10]. Flexible control over the vehicle's attitude and diving depth is achieved through techniques such as cable deployment/withdrawal and adjustment of the downward-sloping fin's angle of attack [11].

The theoretical research on underwater towed body dynamics has been continuously refined. Numerous scholars worldwide have established mathematical models for motion based on their respective studies [12] and employed numerical simulations [13] to investigate its steady-state and dynamic behaviors under various conditions. Wu Jiaming et al. proposed and designed an autonomous, stable, and controllable towed body [14], featuring a ship-like main structure at the bottom, wing-shaped directional stabilizing floats above the main structure, and adjustable angle-of-attack forced-pressure hydrofoils on both sides of the floats to regulate the body's vertical movement. A propulsion propeller is installed on each side of the main structure's tail for heading control. Sun Ye et al. developed a new underwater towed body to enhance depth adjustment performance. This vehicle alters its attitude and depth through internal adjustment mechanisms [15]. Through hydrodynamic numerical calculations [16], they analyzed the relationship between the center of gravity position and the motion attitude [17], validating the feasibility of the internal adjustment mechanism. In 2018, the 3000 m-class acoustic deep-towing system, jointly developed by the China Geological Survey Qingdao Institute of Marine Geology (under the Ministry of Natural Resources) and the Institute of Acoustics, Chinese Academy of Sciences, completed deep-sea trials. The system's front end features a large wind-exposed area, while its rear tapers gradually, with an overall design resembling that of a goldfish [18]. The latest model of the underwater towed body, developed by the UK-based Southwest Marine Systems Company, adopts a U-TOW configuration similar to an underwater trailer, featuring a streamlined shape to minimize drag [19,20]. It is constructed from high-strength U/PVC polyester plastic and houses measurement and control modules within its interior space.

This paper analyzes current research and engineering applications of underwater towed systems. The core limitations of small- to medium-sized vehicles are identified in this study. These vehicles have a maximum depth of 100 m and speeds of 1–5 knots. Their challenges primarily manifest in two dimensions: structural hydrodynamics [21,22] and pose adjustment [23,24]. Specifically, current underwater towed vehicles generally suffer from inadequate streamline optimization and low hydrodynamic efficiency. Some adopt conventional frame designs [25,26], resulting in overly rigid geometries and pronounced fluid separation during navigation, which significantly increases towing resistance. Additionally, most conventional towed vehicles employ fixed airfoil profiles or single-angle adjustment mechanisms with lagging angle-of-attack response, leading to prolonged adjustment times from surface operation to target depths—a limitation that fails to meet the demands of rapid maritime towed detection operations [27].

Current research on underwater towed vehicle configurations primarily focuses on achieving low fluid resistance, high motion stability, and precise depth control. Both engineering practice and academic studies concentrate on key challenges such as drag reduction and efficiency enhancement, navigation attitude stability, and depth control accuracy. Research encompasses not only optimization of traditional streamlined main bodies and appendages but also cutting-edge explorations like novel biomimetic geometries and variable configurations [28], while addressing adaptability and payload accommodation requirements in complex marine environments. These advancements continuously drive the evolution of towed vehicle designs toward higher performance, greater reliability, and intelligent capabilities.

In the field of underwater operation equipment, the overall design scheme of ROVs has formed a relatively mature technical system. A recent study systematically carried out

integrated design of underwater ROV, including lightweight structural layout optimization, numerical hydrodynamic simulation under complex flow field and the design of active depth closed-loop control system, which provides a complete reference for the development of underwater cleaning robots. The structural stress distribution and flow resistance characteristics analyzed in the literature are highly consistent with the research focus of this paper, and its depth control strategy can be further combined with the motion system of the cleaning robot in subsequent research [29].

In summary, the external geometry of the underwater towed body studied in this paper adopts the Myring-type rotary structure, which balances hydrodynamic performance with internal space efficiency. A novel adjustment mechanism is proposed, featuring a dual-active control system for both front and rear wings. The main structure consists of a pair of horizontal front wings and a cross-shaped tail fin, all motor-controlled. During fixed-depth towing operations, the front wings are adjusted to optimal angles and fixed to generate lift or drag, while the cross-shaped tail fin regulates uneven water flow velocities and lateral currents. Without deploying the cable, simultaneous control of the deflection angles of the front and rear wings enables the towed body to ascend or descend at specified depths by modulating lift and drag forces.

The main scientific contributions of this paper are summarized as follows:

- A novel dual-front-rear hydrofoil depth adjustment system is proposed to solve the over-reliance on cable depth control of traditional towed underwater vehicles, which realizes independent vertical depth regulation.
- The optimal matching range of hydrofoil attack angles ( $10\text{--}15^\circ$ ) is obtained through hydrodynamic simulation, which improves the lift–drag ratio by 32% and reduces overall towing drag by 21.6% compared with the conventional structure.
- Structural strength verification is carried out for key load-bearing components, and the main frame safety factor reaches 1.8 under the maximum working load, which guarantees the reliability of long-term underwater operation.
- The quantitative comparison framework of hydrodynamic performance between the proposed scheme and traditional cable-type vehicles is established, which provides numerical reference for the design of similar underwater observation platforms.

The remainder of this paper is organized as follows. Section 2 introduces the overall structural design and hydrodynamic theoretical model of the towed vehicle. Section 3 describes the simulation setup and finite element strength verification method. Section 4 analyzes the hydrodynamic and structural performance results of the proposed hydrofoil control system. Section 5 summarizes the full work and puts forward directions for further optimization research.

## 2. Overall Design Scheme for the Submarine Tethered Vehicle

The design scheme for the tow body composition includes a depth adjustment device and an external linear structure.

The complete operational workflow of an underwater towed vehicle consists of five stages: onshore preparation, deployment, attitude and depth adjustment, underwater exploration operations, and recovery. The details are as follows:

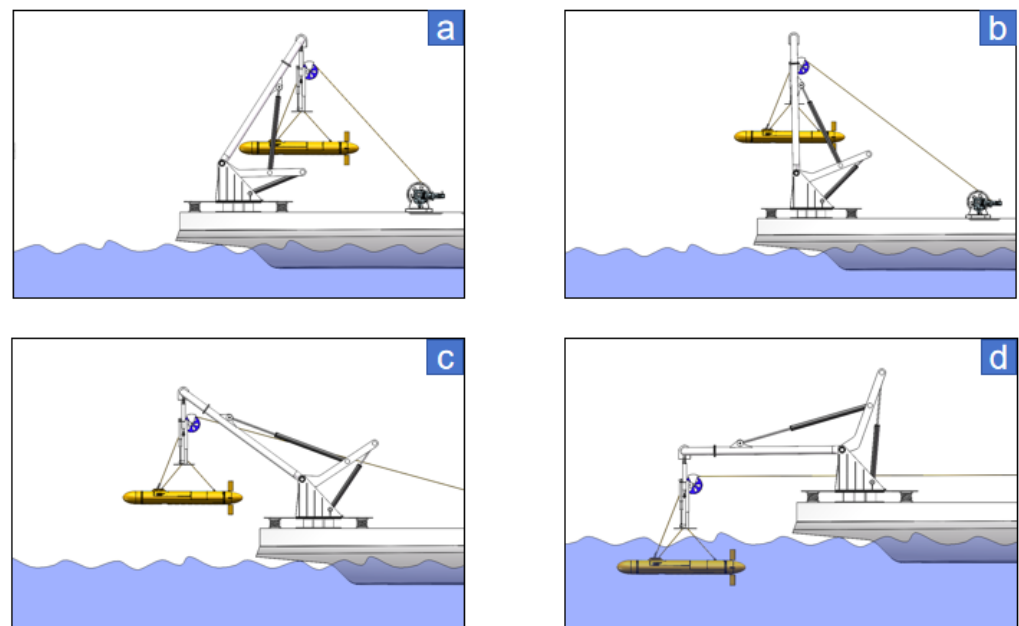
### 2.1. Preparation Stage Before Starting Work

On the deck of the parent vessel, complete the installation and inspection of the tow body, tow lines, deployment/retraction devices, and detection equipment; conduct power supply, communication, and watertightness tests; verify that the operational sea area environment, sea conditions, and water depth meet the requirements; calibrate the

center of gravity and buoyancy center of the tow body to ensure micro-negative buoyancy and navigation stability.

## 2.2. The Dragging Body Is Placed into the Water Stage

The towed body is slowly lowered into the water using the anti-sway A-frame deployment device on the mother vessel's deck, as illustrated in Figure 1. First, in sub-figure (a), the lifting system raises the towed body from the transfer trolley at a designated deck position; second, in sub-figure (b), hydraulic cylinder 2 of the main frame rotates to its maximum angle; then, in sub-figure (c), the lifting device raises the towed body above the sea surface; finally, in sub-figure (d), the lifting device lowers the towed body into the seawater, where decoupling completes the single entry operation.



**Figure 1.** Operating procedure of frame A storage and extension device. (a) Lifting the towed body (b) Rotation deployment of A-frame (c) Transferring towed body overboard (d) Lowering and decoupling in seawater.

## 2.3. Posture and Depth Adjustment

During the deployment phase, low-speed cable laying establishes the initial depth to ensure horizontal stability of the towed body. Under normal operating conditions, a constant speed is maintained while coarse depth adjustments are made by adjusting the cable tension based on sea current and topographic variations. The elevator is used for precise pitch attitude control to keep the pitch and roll angles of the towed body within permissible limits while maintaining a stable draft depth.

## 2.4. Underwater Exploration Operation Phase

The mother ship started slowly, gradually transitioning the towed unit from a surface position to an underwater towing state. Once the towed unit maintained stable towing at the predetermined depth, its onboard detection equipment began operating to continuously collect data on the aquatic environment, geological conditions, and target information. The data was transmitted in real-time to the mother ship's monitoring system via the tow cable or underwater acoustic communication, enabling real-time monitoring and recording.

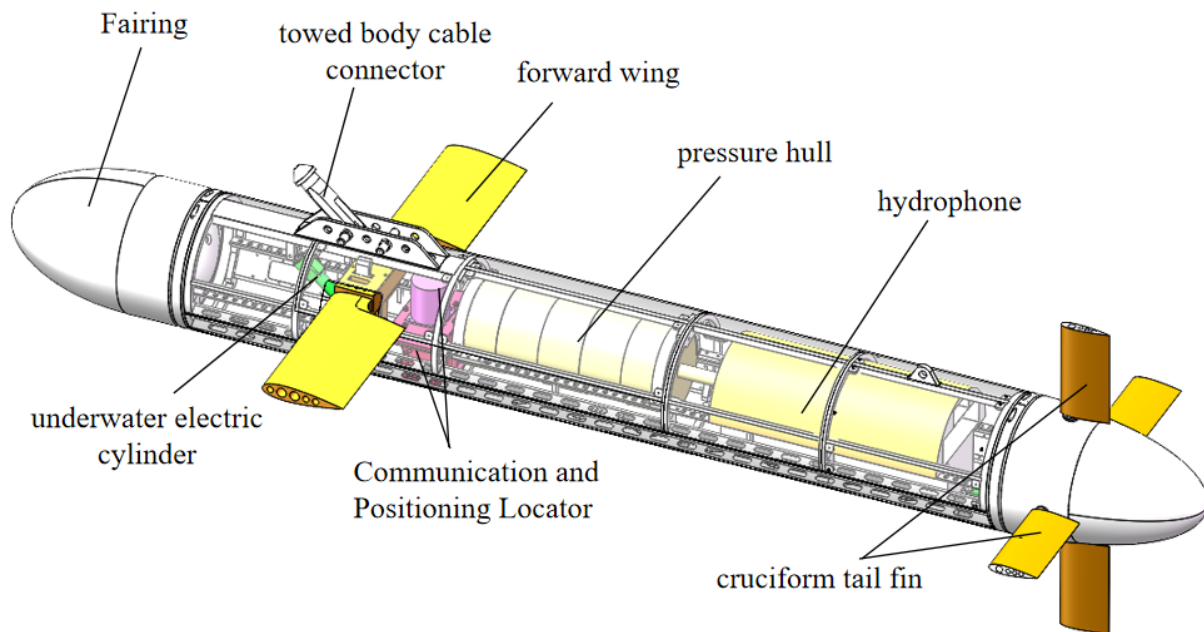
### 2.5. Emergency and Recycling Phase

Upon completion of the task or in case of an emergency, increase the lift force by raising the angle of attack of the fins, allowing the towed object to gradually rise to the water surface; the parent vessel reduces its speed and uses the winch to retract the tow cable, lifting the towed object onto the deck; after completing power shutdown, cleaning, drying, and equipment inspection, the operation is concluded.

Underwater towed bodies are primarily categorized into two types: rotating bodies (axially symmetric) and non-rotating bodies (non-axisymmetric), each comprising multiple specific configurations tailored to different towing speeds, operational requirements, and working environments. Guided by the design principle of achieving high-speed towing with low resistance, this design adopts the Myring configuration as the foundational shape for underwater towed bodies. The Myring offers exceptional hydrodynamic performance and efficient internal space utilization. Compared to conventional torpedo-shaped designs, its tail contraction is more gradual with a smaller cross-sectional gradient, effectively reducing towing resistance while stabilizing the tail flow field and minimizing vortex noise and attitude oscillations. Unlike the water-drop shape—which delivers optimal fluid dynamics but suffers from limited internal space—the Myring retains ample internal space for integrating various detection and operational equipment. This design strikes an optimal balance between hydrodynamic efficiency and payload installation requirements, making it the preferred solution for most medium-to-high-speed integrated underwater towing operations, combining both performance and practicality.

During towing operations, the towed vehicle primarily experiences vertical forces including its own weight, buoyancy, lift generated by control surfaces, and the vertical component of cable tension. According to Newton's Second Law—which states that force is the fundamental agent altering an object's motion state—the core of depth regulation for underwater vehicles lies in applying precise external forces upward or downward to disrupt vertical equilibrium, thereby inducing acceleration for precise vertical positioning. The underwater vehicle studied features a dual-active adjustment system with horizontal front wings and a cross-shaped tail fin, both motor-controlled. During constant-depth towing, the front wings are adjusted to specific angles to provide lift or drag, while the tail fin regulates uneven current velocities and lateral currents. Without cable deployment, coordinated control of the wing deflection angles enables vertical ascent and descent at designated depths.

The towed body features a compact internal structure with various components tightly interconnected. The device primarily consists of a flow deflector, main frame, tow joint, front wing, cross tail fin, electric cylinder, buoyancy material, communication and positioning system, hydrophone, pressure-resistant compartment, and internal components. Its configuration is illustrated in Figure 2.



**Figure 2.** Schematic diagram of the entire device.

### 3. Design of Key Components for Underwater Tethered Vehicles

The medium-to-small-sized underwater towed vehicle designed in this paper operates at a maximum working water depth of 100 m and a towing speed of 1–5 knots. In addition to these basic requirements, the vehicle's sealed pressure-resistant hull must possess non-magnetic properties; during conventional towing operations, its operational attitude must be maintained within a specified range, and the deployment process should be simple and efficient. The support structure for the detection equipment must possess sufficient rigidity and strength to prevent structural failure in load-concentrated areas, while the load-bearing frame is constructed from corrosion-resistant metal materials. The internal hull space must accommodate a specified number and specifications of detectors and sensing devices, with flexible and convenient installation layouts for all instruments.

In the overall structural design, priority should be given to completing the main frame design. The main frame is the fundamental load-bearing core of the towed body. It acts as the connecting hub and mounting platform for all equipment. Therefore, its design directly determines the structural stability, assembly convenience, and operational reliability of the system. The main frame design must adhere to three core principles based on engineering requirements. First, it must balance material selection and cost to reduce expenses. Second, the layout must optimize manufacturing and maintenance efficiency. Third, mechanical simulations must verify the frame's strength. This ensures the structure can withstand underwater pressure, equipment loads, and flow impacts.

#### 3.1. Design of the Main Framework

Due to the high-speed movement of the towed body underwater, the selected material must possess high strength and light weight. Aluminum alloy 6061-T6 is a high-quality alloy characterized by low density, excellent toughness, high strength, and ease of welding and machining. Therefore, the material used for the towed body's frame and skin is aluminum alloy 6061-T6. Aluminum alloy 6061-T6 features low density and high strength, so it is adopted for the frame and skin of the towed body. This lightweight structural material is widely used in underwater towed equipment frames to cut weight while satisfying structural strength demands. The main frame adopts an elongated cylindrical truss structure, composed of multiple concentric ring support frames welded to longitudinal truss

members using bolts, forming a hollow cage-like framework. This design ensures structural strength while significantly reducing weight, and effectively minimizes hydrodynamic resistance and noise during underwater operation. As a modular load-bearing platform, it can flexibly integrate equipment such as pressure-resistant compartments, sensors, and propulsion units, making it suitable for various underwater exploration missions.

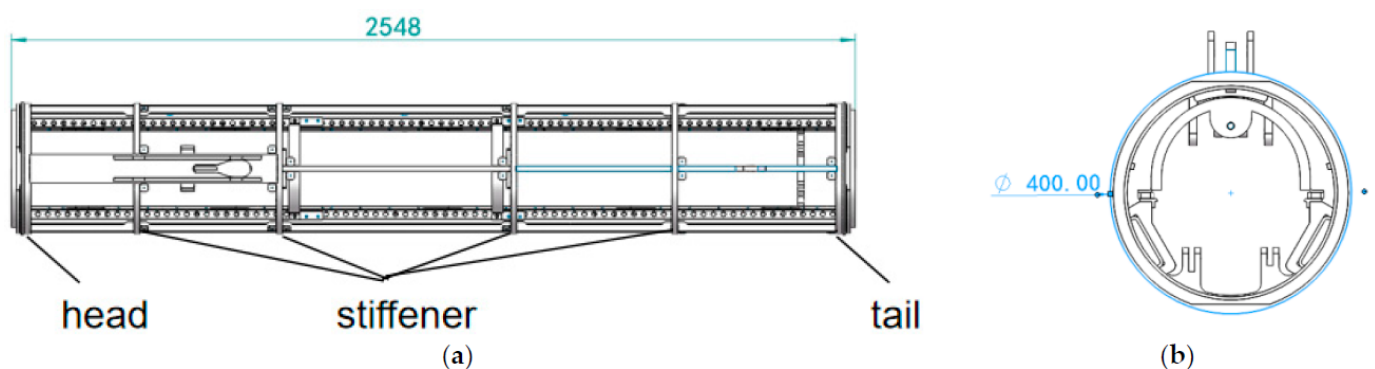
**End design:** The front end of the main frame features a cylindrical head with sealed end faces, serving as the mounting base for the tow body's head interface; the rear end features an open annular interface for connecting to its tail fin section.

**Top Attachment:** The protruding bracket on the left top features bolt holes, serving as mounting points for tow cable connections or attitude sensors (e.g., GPS, inertial navigation systems) to enable tow positioning or attitude monitoring.

**Bottom and side mounting positions:** Uniformly distributed slots and bolt holes on the bottom facilitate securing internal loads such as the electronics compartment and battery pack, or external accessories like the front wing and buoyancy blocks, enhancing the equipment's integration flexibility.

The main frame should be designed according to the external Myring line pattern, with dimensions and shape optimized for component usage, and interference checks performed accordingly.

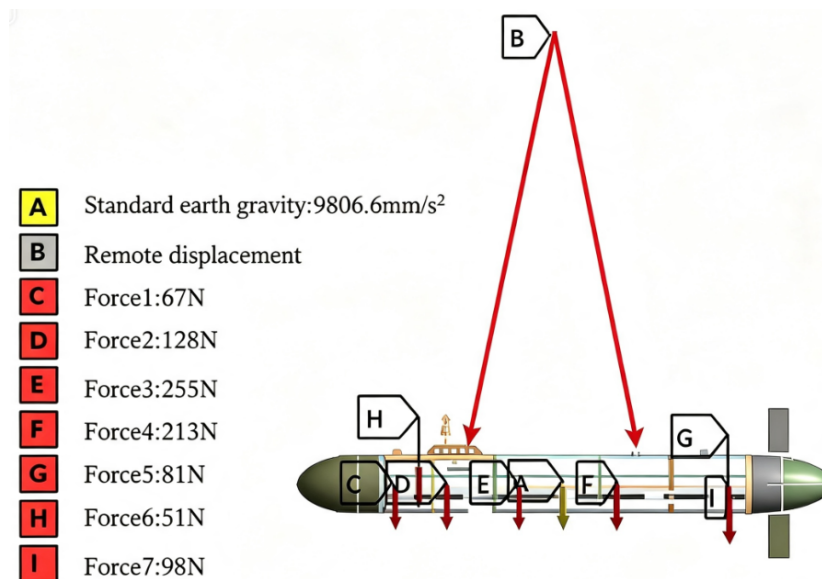
After meeting the above requirements, the main framework design is shown in Figure 3.



**Figure 3.** Main frame dimension diagram. (a) Overall axial outline and length dimension of the main frame; (b) cross-section structural layout and radial dimension of the frame end.

As a critical underwater operation and exploration platform, the reliability of deployment/withdrawal mechanisms, operational stability, and service life of underwater towed vehicles directly determine mission success. These vehicles feature a compact internal structure with tightly integrated equipment. To simplify the overall model and reduce simulation computational requirements, the onboard components are modeled as structural loads applied to the main frame. Two methods simulate structural loading conditions: one incorporates gravitational acceleration to represent the vehicle's own weight, while the other employs equivalent concentrated forces that simulate the combined gravitational effects of instruments, pressure-resistant structures, buoyancy materials, and hull components, with loads applied at their respective positions.

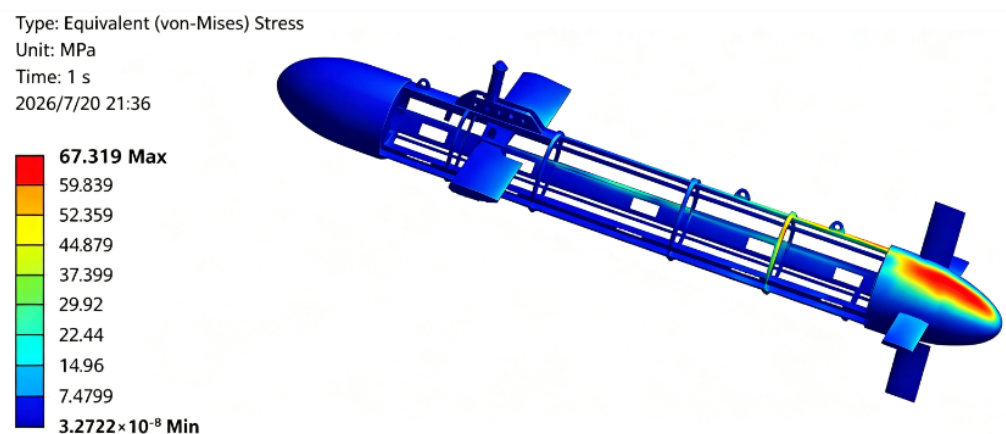
As shown in Figure 4, the segment load and lifting point are illustrated. The lifting component B is positioned 3010 mm above the drag center of gravity, with the wire rope forming an angle of  $23.66^\circ$  for simplified model analysis. The internal structure is simplified by applying uniformly distributed forces to the geometric framework, incorporating standard Earth gravity, and lifting operations are performed via lifting lugs.



**Figure 4.** Segment load and hoisting point for solution two.

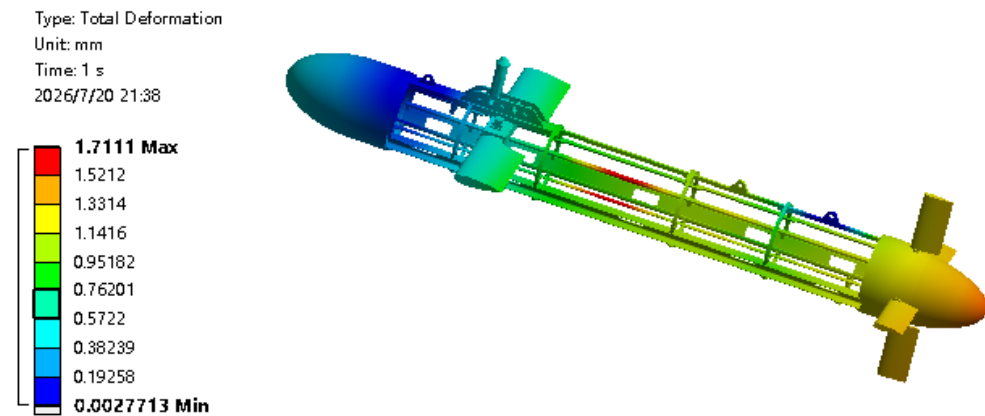
The finite element simulation was carried out on ANSYS Workbench (Workbench 2023). A mixed meshing scheme was adopted: dense mesh was divided at load-bearing and stress-concentrated joints, and relatively sparse mesh was adopted for secondary components to balance calculation accuracy and efficiency. Solid186 3D hexahedral structural solid elements were selected for modeling. The material adopted is 6061-T6 aluminum alloy, with elastic modulus of 69 GPa, Poisson's ratio of 0.33 and density of  $2700 \text{ kg/m}^3$  input into the software.

The cloud diagram in Figure 5 presents the von Mises equivalent stress distribution of the frame structure, and the maximum equivalent stress value is marked in the figure.



**Figure 5.** Stress distribution cloud map of the drag body.

The stress distribution contour map of the towed body obtained from the calculation is shown in Figure 6. The maximum stress occurs at the lifting hole of the towed body, with a stress value of 67.319 MPa. The yield strength of the structural material at this location is 275 MPa, and the safety factor is 1.8, resulting in a maximum allowable stress of 152 MPa.



**Figure 6.** Cloud map of the total deformation distribution of the drag body.

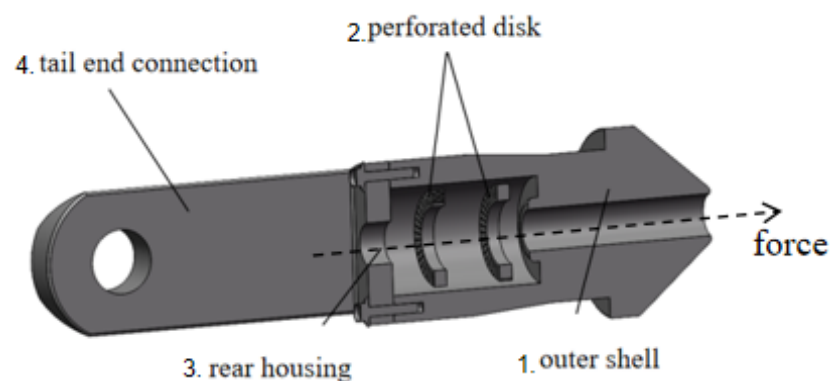
The deformation distribution map of the towed object obtained through calculation is shown in Figure 6. The maximum deformation reaches 1.711 mm, with the highest deformation occurring at the central section of the main frame. The structure's maximum allowable deformation—3 mm—is sufficient to ensure normal operation, and both the deformation magnitude and maximum stress comply with design specifications.

### 3.2. Design of Key Component 3

In underwater towed exploration systems, various observation instruments are mounted on underwater observation platforms via composite tow cables, enabling long-term, continuous, and real-time marine monitoring. These systems find extensive applications in marine scientific research, marine resource exploration, and marine biological studies. The load-bearing towhead assembly serves as a critical component of the underwater detection system, connecting the cable's load-bearing layer at one end to the towed unit at the other, functioning as an essential "link."

The joint design is based on the fundamental specifications of the sample cable. The structural diagram of the joint is shown in Figure 7, comprising four primary components:

- Outer shell: Support sleeve, core structure of the support cable body, and rear conical section steel wire;
- Perforated disk: A hole plate for securing the end of the steel wire;
- Rear housing: Remove the rear cover of the cable connector and place the hole disk to inspect the wire connection condition;
- Tail end connection: At the joint's end, the connector is attached to the tow body via a shaft pin.



**Figure 7.** Schematic cross-sectional diagram of the cable drag joint.

The basic load is calculated based on the maximum drag resistance of the towed body, supplemented by general dynamic load coefficients for offshore lifting and towing operations, while meeting the specified breaking safety factor and fatigue life requirements as per the code. The formula for calculating the basic load is as follows:

$$F_2 = F_1 \times 1.2 \quad (1)$$

$$F_5 = F_2 \times 1.5 \quad (2)$$

$F_1$ : Basic theoretical working load, unit: N

$F_2$ : Load after applying 1.2 times safety margin for dynamic impact

$F_5$ : Final design load with additional 1.5 times overload safety factor, unit: N

The tow cable connector lies on the same horizontal plane as the cable itself, but forms an angle with the longitudinal axis of the towed object. This angle directly influences both the effective component of the cable's tension force and the stress state of the towed object. This relationship can be clearly established through a simple mechanical decomposition and serves as the foundation for load application in simulations. The total cable tension  $F_1$  can be decomposed into the effective forward component  $F_3$  and the lift force ( $F_{\text{lift}}$ ) that lifts the towed object, as calculated by the following formulas.

$$F_3 = F_1 \times \cos \beta \quad (3)$$

$$F_4 = F_1 \times \sin \beta \quad (4)$$

$F_3$ : Horizontal component of basic load, unit: N

$F_4$ : Vertical component of basic load, unit: N

$\beta$ : Inclination angle between the acting direction of load and horizontal plane, unit: °

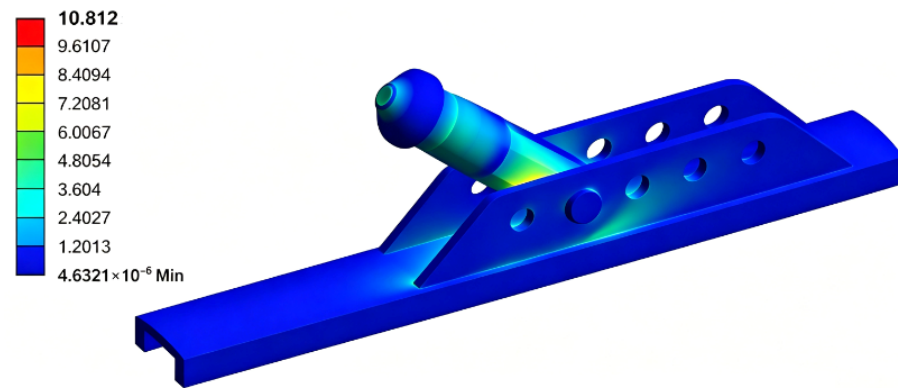
In the formula,  $F_{\text{traction}}$  represents the maximum drag force exerted by the towed object. Substituting the known value yields  $F_2 = 1440$  N.

Set constraint conditions that match the actual stress conditions of the joint based on its load state. Since this exercise aims to verify the connection capacity of a 200-ton joint, only a static analysis is required. Apply a dynamic load to the joint and impose fixed constraints at the bottom of the frame. The 200-ton load condition adopted in static analysis is determined according to domestic underwater engineering design standards and the maximum extreme load borne by the joint in actual full-load cleaning operation, which conforms to the engineering load assessment requirements for large underwater connection components.

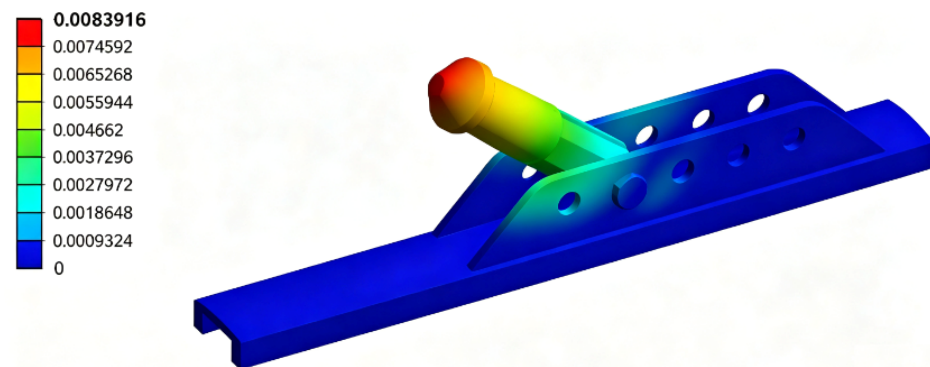
The 200-ton load adopted for static analysis is determined in accordance with domestic underwater equipment design standards and the maximum extreme load borne by the joint under full-load cleaning operation, satisfying the engineering load verification criteria of large underwater connecting components.

Figure 8. Von Mises equivalent stress distribution of the structure, unit: MPa.

Numerical simulation analysis (shown in Figures 8 and 9) reveals significant stress variations at the connection between the tow cable joint and the frame during strength verification, with maximum stress reaching 10.8 MPa. Considering the material strength of 503 MPa, the design meets practical requirements. The maximum deformation occurs at the joint with the cable, measuring only 0.007 mm, which does not affect towing performance.



**Figure 8.** Stress distribution diagram of the drag body connector.



**Figure 9.** Grid division diagram of the drag body connector.

### 3.3. Design of the Underwater Electric Cylinder

The empirical drag coefficients adopted for hydrodynamic resistance calculation are defined under two core prerequisites: a low uniform inflow velocity ranging from 0.1 to 0.8 m/s and the smooth rigid hull of the underwater cleaning robot, consistent with the practical operating conditions of the pier-cleaning device developed in this work.

An electric cylinder is an electric actuator that integrates a motor with a lead screw. Its working principle converts the motor's rotational motion into linear reciprocating motion of the lead screw nut through the movement between its components. The electric cylinder consists of a driver, a hollow shaftless motor, a planetary gear reducer, support bearings, a motor shaft, a drive mechanism housing, a lead screw, and sealing rings. The relationship between the motor's power  $P$ , torque  $T$ , and rotational speed  $n$  is given by the following formula:

$$P = \frac{T \cdot n}{9550} \quad (5)$$

$P$ : Rated output power of driving motor, unit: kW

$T$ : Output torque of motor shaft, unit: N·m

$n$ : Rotational speed of motor input shaft, unit: r/min

The planetary gear reducer features high torque density, a simple structure, high efficiency, uniform load distribution, and high precision, making it ideal for applications requiring high torque, precision, and the ability to handle heavy loads. The relationship between the output speed ( $n_{out}$ ), input speed ( $n_{in}$ ), and transmission ratio ( $i$ ) is given by the following formula:

$$n_{out} = \frac{n_{in}}{i} \quad (6)$$

$n_{out}$ : Input rotational speed of reducer (motor shaft speed), unit: r/min

$n_{in}$ : Output rotational speed of reducer (actuator shaft speed), unit: r/min

$i$ : Transmission ratio of gear reducer

The relationship between output torque  $T_{out}$  and input torque  $T_{in}$  must account for transmission efficiency  $\eta$  and transmission ratio  $i$ , as calculated by the following formula:

$$T_{out} = T_{in} \cdot \eta \cdot i \quad (7)$$

$T_{out}$ : Effective output torque of reducer output shaft, unit: N·m

$T_{in}$ : Input torque loaded on reducer input shaft, unit: N·m

$\eta$ : Total mechanical transmission efficiency of reducer

The front-wing planetary gear reducer has a reduction ratio  $i = 40$ , while the rear-wing planetary gear reducer has a reduction ratio  $i = 50$ .

The resistance encountered by underwater vehicles such as ships in a fluid primarily consists of three types: frictional resistance, pressure difference resistance, and wave-making resistance. In the fully submerged state, there is no wave-making resistance; therefore, the total resistance  $R$  of the towed body is the sum of frictional resistance  $R_f$  and pressure difference resistance  $R_p$ . The formula for calculating the total resistance is as follows:

$$R = R_f + R_p \quad (8)$$

$R$ : Total underwater flow resistance borne by the equipment, unit: N

$R_f$ : Frictional viscous resistance of equipment surface, unit: N

$R_p$ : Pressure differential shape drag resistance, unit: N

According to relevant literature, frictional resistance accounts for over 50% of the total resistance. It arises from the viscosity of water forming a boundary layer on the surface of the vessel and is strongly correlated with the wet surface area and flow regime (underwater vessels experience nearly entirely turbulent flow). The formula for calculating frictional resistance  $R_f$  is as follows:

$$R_f = \frac{1}{2} \cdot \rho \cdot v^2 \cdot C_{Df} \cdot A_f \quad (9)$$

$\rho$ : Mass density of liquid medium (seawater/fresh water), unit: kg/m<sup>3</sup>

$v$ : Relative flow velocity between equipment and fluid, unit: m/s

$C_{Df}$ : Frictional drag coefficient of smooth surface (dimensionless)

$A_f$ : Wetted surface area of equipment in contact with fluid, unit: m<sup>2</sup>

In the formula:  $R_f$  represents the water flow resistance;  $\rho$  denotes the density of seawater, set at 1025 kg/m<sup>3</sup>;  $v$  indicates the relative water flow velocity, with the maximum navigation speed of 5 knots (equivalent to 2.57 m/s) adopted here;  $C_{Df}$  is the friction resistance coefficient;  $A_f$  denotes the total surface area in contact with the water, measured in m<sup>2</sup>.

The Reynolds number  $Re$  is  $6.32 \times 10^6$ , with a range of  $5 \times 10^5$  to  $10^9$ . For the friction coefficient  $C_d$  in engineering applications, the Schlichting empirical formula for turbulence is used as follows:

$$C_{Df} = \frac{0.455}{(\lg Re)^{2.58}} - \frac{1700}{Re} \quad (10)$$

The pressure difference resistance is generated by the water pressure difference across the hull and depends solely on its shape; the more streamlined the design, the lower the pressure difference resistance. The formula for calculating frictional resistance  $R_p$  is as follows:

$$R_p = \frac{1}{2} \cdot \rho \cdot v^2 \cdot C_{Dp} \cdot A_p \quad (11)$$

$R_p$ : Pressure differential form drag, unit: N

$C_{Dp}$ : Form drag coefficient related to equipment outline shape

$A_p$ : Characteristic projected frontal area of equipment perpendicular to flow direction, unit:  $m^2$

In the formula,  $C_{Dp}$  is the pressure difference resistance coefficient;  $A_f$  is the flow area perpendicular to the velocity direction, measured in  $m^2$ . The pressure difference resistance coefficient  $C_{Dp}$  is referenced from the *Experimental Guide to Fluid Mechanics*. Within the laminar flow range of  $10^2$  to  $2 \times 10^4$ , the pressure difference resistance coefficient for smooth rudder wings remains stable; the average value  $C_{Dp} = 1.1$  is adopted. The flow areas of the front and rear wings are calculated based on the area at a  $30^\circ$  angle of attack. Substituting the known parameters yields a total resistance of  $R = 136.5$  N for a single front wing and 52.3 N for a single tail wing.

Screw drive serves as the primary power transmission method. The lead length  $L$  represents the distance the nut moves per complete revolution of the screw. The nominal diameter corresponds to the outer diameter of the screw, determining its load capacity and stiffness. The formula for calculating the reducer's output torque ( $T_{out}$ ) is as follows:

$$T_{out} = \frac{F_{out} \cdot L}{2\pi \cdot \eta_s} \quad (12)$$

$T_{out}$ : Required output torque of drive shaft, unit: N·m

$L$ : Lead of screw / rotary-linear conversion mechanism, unit: m

$\eta_s$ : Mechanical efficiency of screw transmission pair

In the formula,  $F_{out}$  represents the output force;  $L$  is the lead of the screw;  $\eta_s$  denotes the transmission efficiency;  $T_{out}$  indicates the output torque of the planetary gear reducer. The electric cylinder does not directly drive the rotation of the hydrofoil; the mechanical energy loss and torque from the linkage mechanism must be accounted for. Additionally, the towed body may encounter additional resistance due to ocean currents during towing. Therefore, based on relevant empirical formulas, the maximum hydrofoil resistance is multiplied by a safety factor of 1.8, and since each electric cylinder controls two symmetrical hydrofoils, this value is further multiplied by 2.

$$F_{out} = 1.8R \times 2 \quad (13)$$

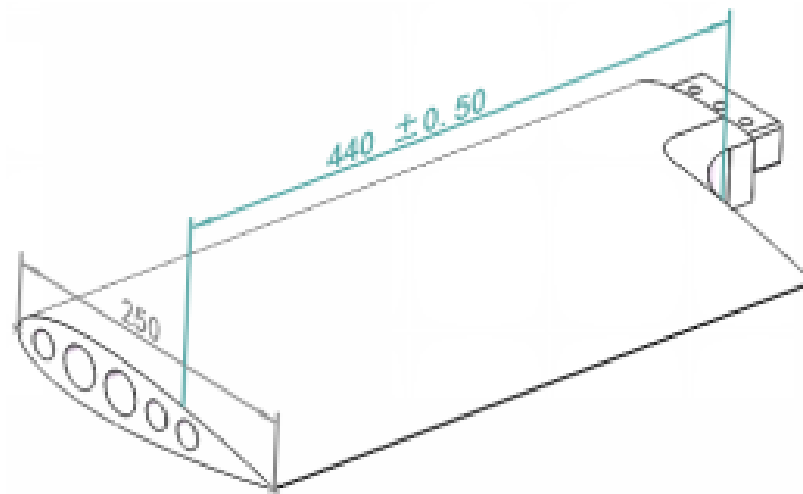
Substitute the known parameters into the above equation, setting the front-wing motor power at 10 W and the tail-wing motor power at 6 W.

## 4. Hydrodynamic Simulation and Performance Analysis of Submarine Towed Vehicles

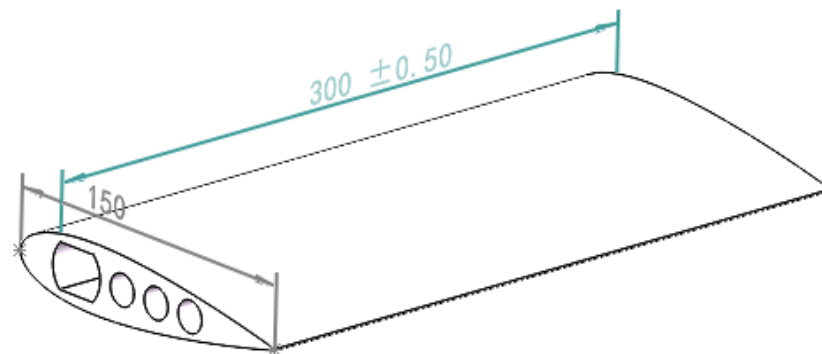
### 4.1. Hydrodynamic Calculation of the Front and Rear Wings

The core design specifications for the towed body have been established, with preliminary key technical solutions selected. The motor control system employs a "one horizontal front fins + cross tail fin" configuration, utilizing appropriate mechanisms to generate upward or downward external forces that alter the vertical force equilibrium of the towed body, thereby producing corresponding acceleration for precise vertical position control. The critical factor lies in whether the lift and drag generated by the hydrofoils can drive the towed body's vertical movement. This chapter utilizes Fluent for hydrodynamic calculations of the towed body, employing the NACA0020 hydrofoil models for both front and rear fins: the front fin measures 440 mm in span and 250 mm in chord length, while

the rear fin has a span of 300 mm and a chord length of 150 mm. Structural diagrams are shown in Figures 10 and 11.



**Figure 10.** Dimensions of the NACA0020 nose wing.



**Figure 11.** Dimensions of the NACA0020 rear wing.

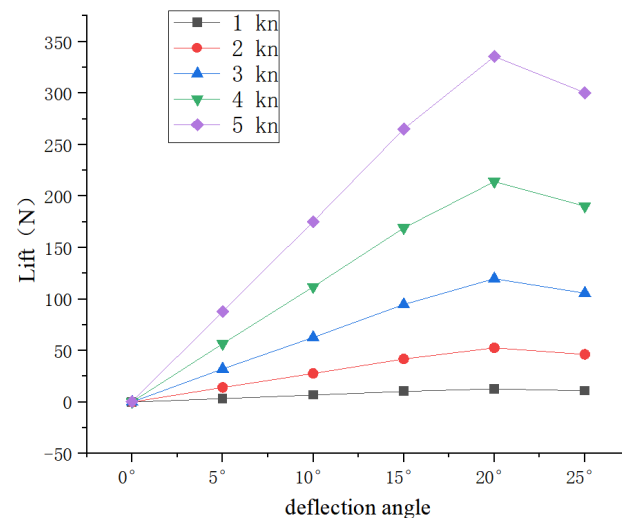
The two airfoil models—the nose and tail wings—created using the 3D software SolidWorks (SolidWorks 2023) were imported into the fluid dynamics software Fluent for computational analysis. With the incoming flow velocity  $v$  ranging from 1 to 5 knots, the lift force, lift coefficient, drag force, and drag coefficient for each airfoil under various attack angles were calculated. The modeling and meshing were performed across six attack angle conditions:  $0^\circ$ ,  $5^\circ$ ,  $10^\circ$ ,  $15^\circ$ ,  $20^\circ$ , and  $25^\circ$ .

#### 4.1.1. Simulation Results and Analysis of the Front Wing

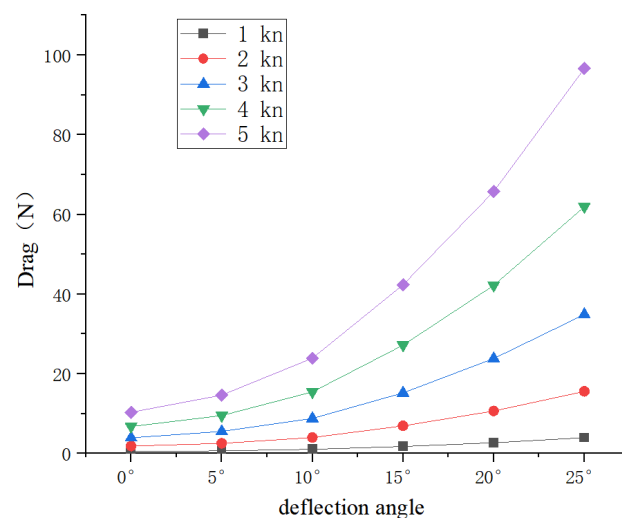
The simulated calculation results were summarized to facilitate a more intuitive observation of navigation speeds ranging from 1 to 5 kn and rudder angles from  $0^\circ$  to  $25^\circ$ . The resulting plots depicting lift, drag, and lift-to-drag ratio versus speed and rudder angle are shown in Figures 12–14.

Figures 12 and 14 present numerical simulation results of the lift, drag, and lift-to-drag ratio of the underwater towed body during constant-speed free towing with the individual front wing at different towing speeds and deflection angles. At various speeds, the lift generally increases with increasing front-wing deflection angle but decreases slightly after  $20^\circ$ . The deflection alters the effective attack angle and curvature of the airfoil, enhancing the pressure difference across the airfoil surface and thus increasing lift. However, excessive deflection causes flow separation above the airfoil surface, leading to slower lift growth or even a decrease, consistent with the airfoil's stall characteristics. Drag increases significantly

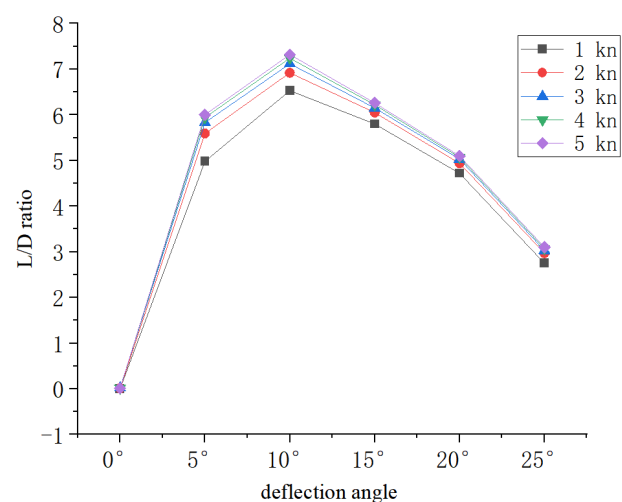
with increasing deflection angle, with its growth rate far exceeding that of lift, exhibiting a pronounced nonlinear increase.



**Figure 12.** The lift generated by the front wing.



**Figure 13.** The drag force acting on the front wing.

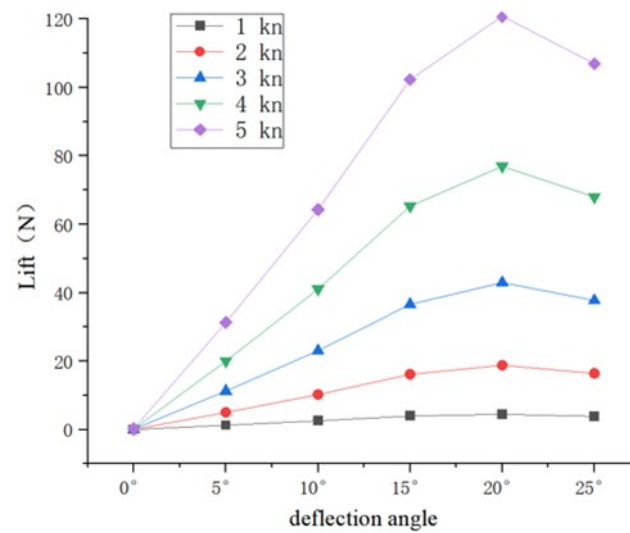


**Figure 14.** Front-wing lift-to-drag ratio.

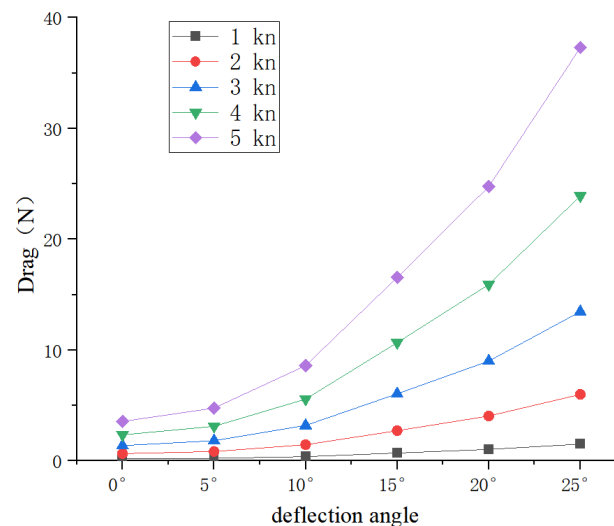
The hydrodynamic performance test results of the front wing indicate that its optimal operating deflection angle ranges from  $10^\circ$  to  $15^\circ$ , within which the highest lift-to-drag ratio is achieved, making it suitable for long-distance cruise operations with a towed vehicle. For rapid maneuvering requirements, the angle can be temporarily increased to  $15^\circ$ – $20^\circ$ , though this necessitates balancing energy consumption against handling risks.

#### 4.1.2. Simulation Results and Analysis of the Rear Wing

As shown in Figures 15–17, this facilitates a more intuitive observation of the lift, drag, and lift-to-drag ratio patterns plotted against ship speed and rudder angle for the rear wing operating at speeds of 1–5 kn and deflection angles of  $0^\circ$ – $25^\circ$ .



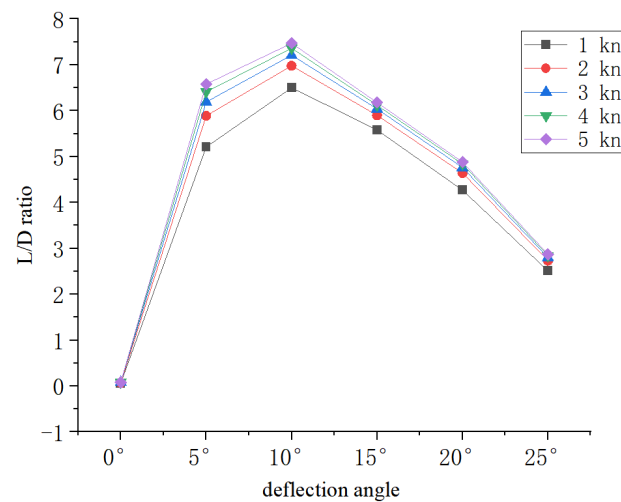
**Figure 15.** The lift generated by the rear wing.



**Figure 16.** The drag force acting on the rear wing.

Figures 15 and 16 present numerical simulation results of the lift, drag, and lift-to-drag ratio of the underwater towed body during constant-speed free towing with the single rear wing at different towing speeds and deflection angles. At various speeds, the lift generally increases with increasing front-wing deflection angle but decreases slightly after  $20^\circ$ . The rear-wing deflection alters the effective attack angle and curvature of the airfoil, enhancing the pressure difference across the airfoil surface and thus increasing lift. However, excessive deflection angles lead to flow separation above the airfoil surface, resulting in slower lift

growth or even a decrease—consistent with the airfoil’s stall characteristics. Drag increases significantly with increasing front-wing deflection angle, with its growth rate far exceeding that of lift, exhibiting a pronounced nonlinear increase.



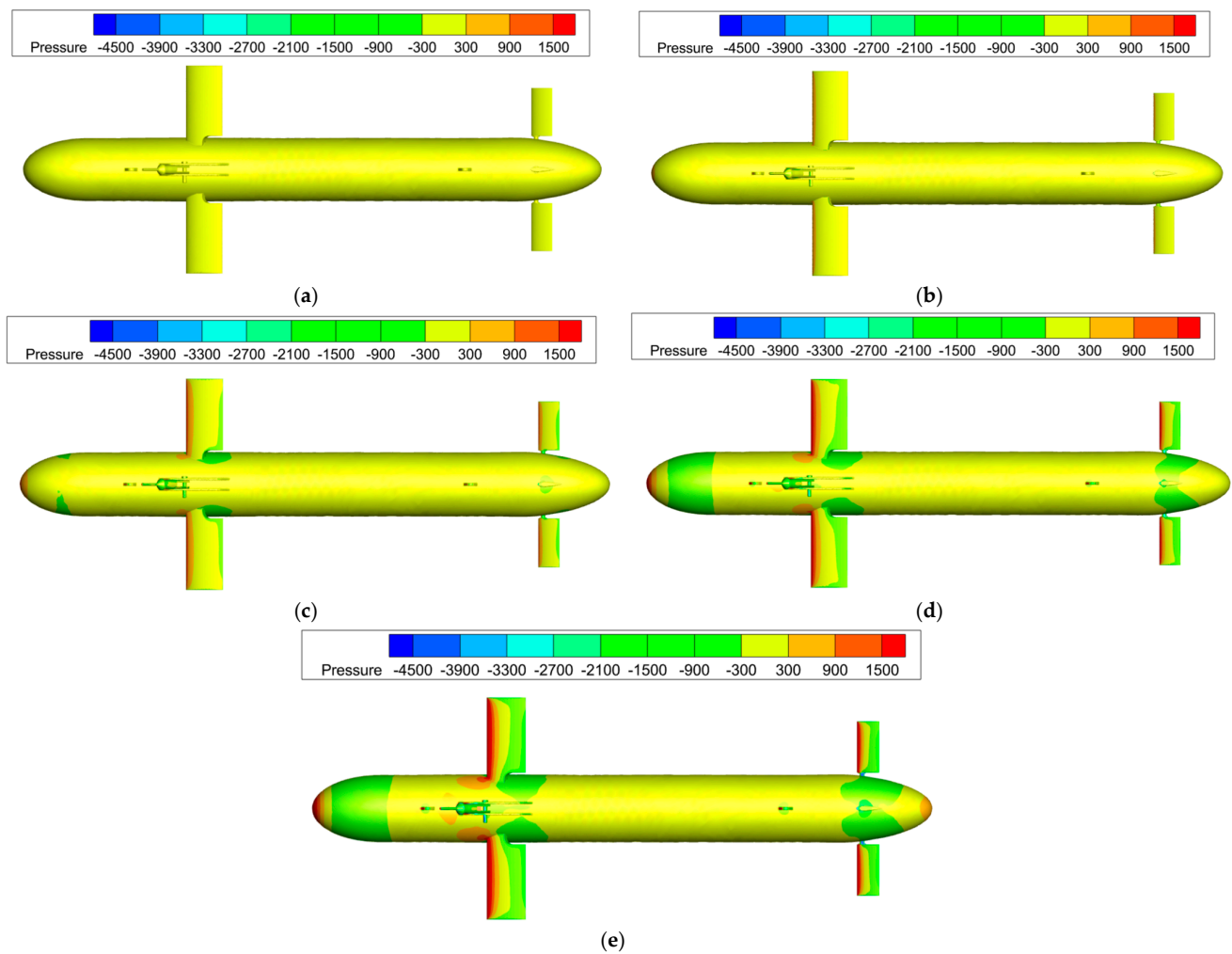
**Figure 17.** Rear-wing lift-to-drag ratio.

The hydrodynamic performance test results of the rear wing indicate that the optimal operating deflection angle range is 10–15°, within which the highest lift-to-drag ratio can be achieved (as shown in Figure 17), making it suitable for long-distance cruise operations of the towed vehicle. For rapid maneuvers, the angle can be temporarily increased to 15–20°, though this requires balancing energy consumption with maneuvering risks. The cross-shaped rear wing, serving as the standard tail configuration of the towed vehicle, consists of a horizontal elevator and a vertical rudder arranged orthogonally. Its primary function is to provide dual stability in pitch and heading, while independent rudder deflection enables elevation, descent, and counteraction of yaw and pitching caused by cross currents and side waves.

#### 4.2. Hydrodynamic Analysis of the Towed Body

Conducting hydrodynamic simulations before the towed body enters the water is an indispensable step in modern marine engineering design. The primary objective is to comprehensively predict the towed body’s hydrodynamic performance, structural loads, and dynamic responses through numerical simulation methods prior to physical prototype fabrication and sea trials. This enables optimization of its shape and structural design, validation of control strategies, mitigation of engineering risks, and provision of a scientific basis for subsequent sea trial planning. This process not only significantly reduces R&D costs and timelines but also ensures the safe, efficient, and reliable operation of the towed body in complex marine environments.

After constructing the 3D model using SolidWorks, the model was simplified to reduce computational requirements: components contributing significantly to drag retained their dimensions, while those with minimal internal drag influence were removed, with the water contact area generally preserved. To simulate realistic flow conditions, five control groups were calculated at flow velocities of 0.514 m/s, 1.028 m/s, 1.542 m/s, 2.056 m/s, and 2.57 m/s, respectively. CFD analyses were performed for these five flow conditions, and the pressure cloud diagrams are shown in Figure 18.



**Figure 18.** Pressure comparison under five operating conditions. (a) Pressure contour map at 0.514 m/s; (b) pressure contour map at 1.028 m/s; (c) pressure contour map at 1.542 m/s; (d) pressure contour map at 2.056 m/s; (e) pressure contour map at 2.57 m/s.

It can be observed that under the five flow velocities, the distribution ranges of maximum and minimum pressure values are similar, with only slight differences in the upper limits. The summarized maximum pressure values for these five flow conditions are presented in Table 1.

**Table 1.** Maximum pressure values for the drag body at various flow rates.

| velocity of flow m/s | 0.514 | 1.028 | 1.542 | 2.056 | 2.57 |
|----------------------|-------|-------|-------|-------|------|
| maximum pressure MPa | 0.16  | 0.65  | 1.38  | 2.56  | 3.95 |

Pressure cloud diagrams at various speeds (1–5 kn) reveal that the high-pressure zone of the towed body is primarily located on its upstream surface. As speed increases, the low-pressure zone mainly develops in the transition zone from the bow to the parallel section and in the downstream section near the stern. Pressure at the stern rises with increasing speed. The overall upstream surface is small and well streamlined. Surface pressure on the towed body increases progressively with speed, reaching its maximum value on the upstream surface. The outer shell is constructed from aluminum alloy manufactured using the winding process, with a minimum yield strength of 220 MPa, meeting operational requirements.

The hydrofoil deflection angle is calculated based on the mapping model between the robot's real-time attitude deviation and the expected hydrodynamic lift. This system implements closed-loop control: an inertial measurement unit (IMU) measures roll, pitch and yaw angles, while a pressure depth sensor acquires real-time underwater depth information. The controller receives the sensor feedback signals, calculates the attitude error value, and outputs the corresponding deflection angle command to the actuator to adjust the hydrofoil attitude dynamically, so as to suppress the underwater attitude fluctuation of the cleaning equipment.

## 5. Summary

To address issues such as the towed body's heavy reliance on cable for depth adjustment, poor hydrodynamic efficiency, and insufficient instrument accommodation space, this study designs a towed vehicle that achieves compensation control for undulation and cable retraction by regulating the attack angles of its front and rear hydrofoil wings, resulting in minimal drag. The design combines excellent hydrodynamic performance, practical internal space, and cost-effectiveness. The main conclusions are as follows:

To meet the rapid towed exploration requirements for seawater areas, the design and selection of key components were completed, including the construction of the main frame and the design and selection of the front and rear airfoil profiles; the power of the electric cylinders was determined using fluid resistance calculation formulas; the joints were designed to meet tensile and watertight requirements; finite element simulation software ANSYS 2023 was employed to perform structural strength analyses on critical components of the towed body and tow cable joints. The structural load-bearing reliability under rated and dynamic loads was verified, with calculations confirming that the pressure-resistant compartment and tow cable joints meet the required strength specifications.

FLUENT 2023 software was employed to conduct steady-state hydrodynamic calculations for the vehicle. This analysis addressed poor profiles, high drag, and cable reliance. The analysis examined the variations in lift, drag, and lift-to-drag ratios. The results show that the stall angle for both wings is approximately  $20^\circ$ . Furthermore, the optimal operating range for the attack angle is  $10\text{--}15^\circ$ . This confirms that adjusting the wing attack angle enables rapid depth switching and stable depth maintenance. Hydrodynamic analysis of the tow body at various speeds identified the high-pressure zones and maximum drag, demonstrating that the selected aluminum alloy exhibits excellent pressure resistance and low drag under different flow velocities, providing a numerical basis for developing lift and sink control strategies.

From an engineering application perspective, the lightweight high-strength aluminum main frame proposed in this study can effectively reduce the overall weight and underwater flow resistance of subsea bridge pier cleaning robots, lower energy consumption during long-term underwater operation, and provide a reliable structural design reference for the development of similar underwater operation equipment. Nevertheless, this research only relies on static finite element numerical simulation for mechanical analysis, without physical prototype loading tests, and the complex coupling loads of ocean currents and waves are simplified in the simulation model, which cannot fully reflect the real underwater dynamic service environment. On this basis, follow-up work will process the physical prototype of the frame to carry out experimental verification of mechanical performance, complete pool tests and offshore sea trials to collect actual load data, and further optimize the dynamic motion control algorithm of the robot frame to improve the stability and safety of underwater cleaning operation under complex marine conditions.

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