

Article

Investigation into the Internal Flow Characteristics of an Axial-Flux Canned Motor Pump

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Abstract

Canned motor pumps are widely utilized due to their distinct advantage of a completely leakage-free structure. Among them, an integrated impeller–rotor configuration is employed in the axial-flux canned motor pump, resulting in a shorter axial length and higher power density. This novel configuration allows for easy integration into space-constrained systems, such as electric vehicles, aerospace applications, and liquid-cooled servers. However, research on the internal flow characteristics of these pumps remains scarce. To address this gap, the present study investigates the internal flow across various flow rates. Numerical simulations are validated against experimental data. The average error remains below 2%. The pump achieves a peak efficiency of 68.6% at the design condition, but experiences efficiency drops of 15.0 and 25.2 percentage points under $0.5Q_d$ and $1.5Q_d$, respectively. Results demonstrate that flow rates significantly govern internal characteristics. These include pressure, velocity, and entropy distributions, along with vortex structures and pressure fluctuations. Notably, operating at off-design conditions can intensify the internal pressure fluctuations by up to a factor of 29.4. Entropy analysis identifies major losses on blade suction sides and diffusers. These findings provide crucial hydrodynamic guidelines for low-noise thermal management systems in electric vehicles and ensuring high-reliability cooling loops in aerospace and liquid-cooled servers.

Keywords: centrifugal pump; flow rate; energy performance; internal flow; pressure fluctuation

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1. Introduction

Canned motor pumps are widely applied in modern industrial production and civilian sectors due to their hermetically sealed and leakage-free structure [1–3], which ensures the zero-pollution transport of the working medium. As illustrated in Figure 1, the axial-flux canned motor pump adopts a unique architecture characterized by an axial parallel alignment of the stator and rotor, with the impeller and rotor consolidated into a unified assembly. Consequently, this pump exhibits a compact axial dimension and high power density, making it suitable for space-constrained fluid circulation systems in applications such as electric vehicles, aerospace, and liquid-cooled servers.

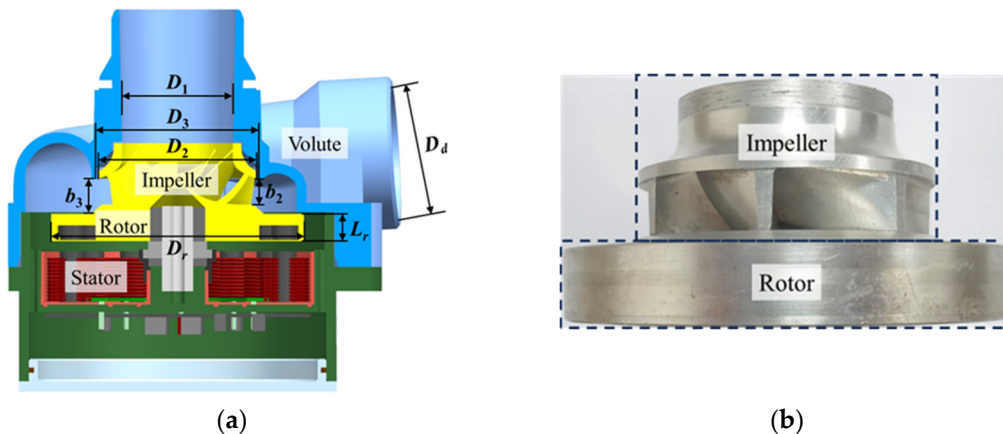


Figure 1. Axial-flux canned motor pump: (a) meridional section; (b) prototype of the integrated impeller–rotor.

To date, research on axial-flux canned motor pumps has predominantly concentrated on the design and optimization of the motor. FenerciOGLu [4] introduced a dual axial-flux motor design for axial blood pumps, numerically analyzing its magnetic parameters, axial force, and torque. By developing a wirelessly controlled axial-flux motor for micro-pumps, Pang et al. [5] demonstrated through electromagnetic and mechanical evaluations that this design outperforms radial-flux equivalents in both output torque and efficiency. For cryogenic liquid pumps, Ai et al. [6–9] developed an axial-flux motor-pump structure incorporating super-conducting magnetic bearings to compare its rotor suspension force against that of a radial-flux design. They further investigated the motor's axial and radial electromagnetic force characteristics and the relationship between driving torque and key structural parameters.

Other integration efforts include the work of Li et al. [10], who combined a radial piston pump with an axial-flux motor, analyzing the pump's internal flow field and optimizing the motor structure. Similarly, Nishanth et al. [11] created an integrated hydraulic pump and axial-flux motor, exploring the optimal design through multiphysics analysis. An experimental assessment was conducted by Eker et al. [12] to contrast the operational characteristics of axial-flux and radial-flux permanent magnet synchronous motors, focusing on their efficiency, power factor, and dynamic responses across diverse loading scenarios. In the context of blood pumps, Karabulut et al. [13] examined an axial-flux permanent magnet synchronous motor, contrasting different stator core materials based on output torque and core loss. Yim et al. [14] designed an axial-flux high-temperature superconductor field coil motor for liquid hydrogen centrifugal pumps, analyzing the electromagnetic performance of different coil types and conducting a mechanical stress analysis. More recently, Zhang et al. [15] analyzed how geometric variables affect viscous frictional dissipation within axial-flux motors designed for electronic coolant pumps. Their findings indicated that the axial-flux architecture yields superior torque output compared to radial-flux designs, provided that thermal constraints remain identical.

Despite the extensive research on the motor component, the axial-flux canned motor pump remains a novel structure, and there is a notable gap in the understanding of its internal flow characteristics. Unlike traditional radial-flux pumps with cylindrical rotors, this axial-flux design employs a disk-type rotor. Its rotation generates strong disk friction loss and complex three-dimensional secondary flows. Moreover, the proximity of the rotor disk to the volute tongue significantly alters the rotor–stator interaction (RSI). These structural differences fundamentally change the flow physics and pressure fluctuation mechanisms within the pump. Although structural deformations in narrow clearances under coupled loads can influence the volumetric efficiency [16], this preliminary study

focuses on the rigid-body fluid dynamics. Specifically, a systematic analysis of internal phenomena such as pressure fluctuations and the distribution of turbulent kinetic energy dissipation is absent from the current literature.

Pressure fluctuation is a critical parameter for characterizing unsteady flow in pumps. Its spectral properties and amplitude evolution are directly linked to transient mechanisms like RSI and vortex shedding, making it essential for understanding broadband vibration and turbulent energy dissipation. Numerous studies have investigated pressure fluctuations in various pump types. For instance, Luo et al. [17] examined the energy loss and fluctuation mechanisms in jet pumps arising from non-uniform inflow. For pump-turbines, Zhang et al. [18] combined simulations and experiments to model the full flow passage, analyzing pressure fluctuations in the S-shaped region across multiple operating modes. Regarding high-speed centrifugal pumps, Zhang et al. [19] discovered that pressure pulsations at elevated discharge levels are primarily governed by the blade passing frequency. Conversely, to counteract the intense low-frequency oscillations typical of near-no-load operation, Fu et al. [20] introduced a refined guide vane closure methodology. Advanced signal processing techniques have also been employed; Li et al. [21] used wavelet analysis to study pressure variations in a bulb tubular pump during deceleration.

The influence of geometric design has also been a key focus. Li et al. [22] compared transient pressure fluctuations in regenerative pumps with symmetric versus staggered blade arrangements. Lu et al. [23] identified that pump outlet pressure could act as a dynamic indicator for flow instability near the hump point. Sonawat et al. [24] explored the synergistic suppression of fluctuations in double-suction pumps using staggered impellers and split volutes. Methodologically, Jiao et al. [25] integrated entropy production theory with Computational Fluid Dynamics (CFD) to analyze energy loss and pressure fluctuations in low-head pumping stations. Other components, such as the flap valve, have also been studied for their contribution to energy loss and fluctuations by Zhang et al. [26]. Recent research has delved into complex operational modes and analysis methods. Zhang et al. [27] addressed grid connection difficulties in reversible pump-turbines by using Variational Mode Decomposition, revealing intense low-frequency and broadband fluctuations in the vaneless region and draft tube. Song et al. [28] explored how the configuration of the impeller outlet influences the flow patterns and pulsations at the trailing edge. Tu et al. [29] analyzed cavitation-induced unsteady flow and pressure responses in a large double-volute pump. Additionally, the low-frequency components of pressure oscillations in centrifugal pumps were investigated by Wang et al. [30] using the complete ensemble empirical mode decomposition with adaptive noise and Hilbert transform techniques. However, most of these current fluctuation mitigation techniques, such as complex geometric modifications, often compromise peak hydraulic efficiency or significantly increase manufacturing complexity. Furthermore, their applicability and effectiveness in integrated disk-type configurations remain largely unexplored.

However, the analysis of pressure fluctuation characteristics within the axial-flux canned motor pump remains a research gap. To address this gap, this manuscript combines numerical simulations and experiments to reveal the internal pressure fluctuation characteristics of the axial-flux canned motor pump and their link to energy dissipation. This manuscript is organized as follows.

The design specifications of the axial-flux canned motor pump and the computational framework are introduced in Section 2. Subsequently, Section 3 details the experimental apparatus and verifies the numerical model using empirical data. Finally, Section 4 provides an in-depth analysis of how flow rate modulates internal flow behavior, focusing on variables such as pressure distributions, vortex topologies, and entropy generation.

2. Numerical Simulation

2.1. Pump Parameters

The axial-flux canned motor pump investigated here is illustrated in Figure 1a. Its design flow rate (Q_d) and rotational speed are $9 \text{ m}^3\cdot\text{h}^{-1}$ and $5200 \text{ r}\cdot\text{min}^{-1}$, respectively. The primary hydraulic components of this pump are the impeller and the volute, whose main geometric parameters are listed in Table 1. The diameters of the suction and discharge pipes are equal to the impeller inlet and volute outlet diameters, respectively. The length of each extension reaches five times its respective diameter [31].

Table 1. Main geometrical parameters.

Component	Parameter	Value
Impeller	blade inlet diameter D_1	32 mm
	blade outlet diameter D_2	53 mm
	blade outlet width b_2	7.8 mm
	number of blades z	7
	rotor diameter D_r	72.5 mm
Volute	rotor axial length L_r	11 mm
	based circle diameter D_3	55 mm
	inlet width b_3	12 mm
	outlet diameter D_d	32 mm

2.2. Mesh Generation

As illustrated in Figure 2, the computational domain of the axial-flux canned motor pump comprises six distinct sub-regions: the suction and discharge pipes, the front and rear chambers, the impeller, and the volute.

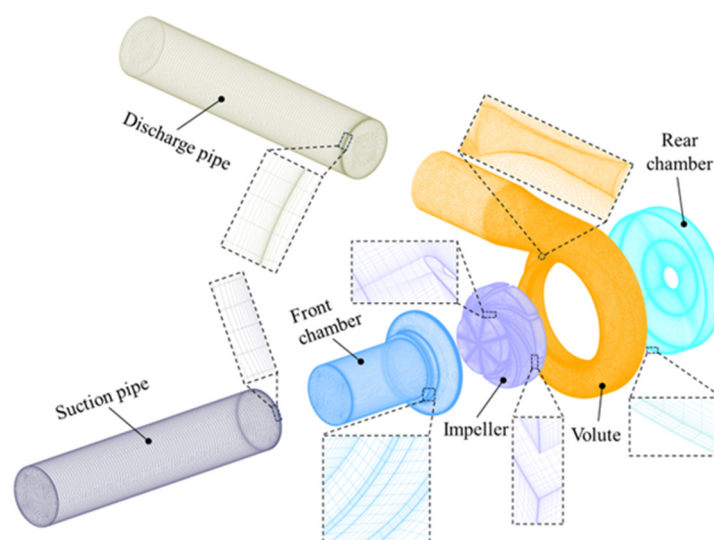


Figure 2. Mesh of the computational domain.

The pipes, chambers, and impeller are discretized with hexahedral elements via ANSYS ICEM CFD 2020R2, employing block-structured topologies like O-grid, C-grid, and Y-grid to accurately delineate flow passages. These structured grids maintain high quality [32], evidenced by Jacobian determinants greater than 0.5 and minimum angles surpassing 18° . Conversely, due to the intricate 3D spatial profile and sharp tongue features of the volute, a fully hexahedral mesh is impractical. Consequently, a mixed-element strategy is implemented within ANSYS Meshing, combining tetrahedral and hexahedral cells with local refinements in high-gradient regions. This hybrid approach yields an

orthogonal quality exceeding 0.7 and an aspect ratio under 3. Collectively, the mesh metrics fulfill the rigorous standards necessary for precise internal flow field evaluations [33].

2.3. Computational Setup

Numerical investigations in the present work are conducted via ANSYS CFX 2020R2, adopting the Unsteady Reynolds-Averaged Navier–Stokes (URANS) approach to resolve turbulence [34]. The corresponding mathematical framework is delineated in Equations (1) and (2).

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_i}(\rho u_i) = 0 \quad (1)$$

$$\frac{\partial}{\partial t}(\rho u_i) + \frac{\partial}{\partial x_j}(\rho u_i u_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left[\mu_{eff} \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \right] + F_i \quad (2)$$

where ρ is the density, t is the time, u is the velocity, x is the coordinate, p is the pressure, μ_{eff} is the effective viscosity, F_i is the external force.

In the numerical simulation, the head of the axial-flux canned motor pump is defined as:

$$H = \frac{p_{2,tot} - p_{1,tot}}{\rho g} \quad (3)$$

where $p_{1,tot}$ is the total pressure at the pump inlet and $p_{2,tot}$ is the total pressure at the pump outlet.

To ensure consistency across diverse operating regimes and minimize the influence of physical scale and rotational frequency, the head is normalized according to similarity laws. The resulting dimensionless head coefficient is expressed as follows:

$$\psi = \frac{2gH}{u_2^2} \quad (4)$$

where u_2 is the circumferential velocity at the impeller outlet, $u_2 = 14.43 \text{ m}\cdot\text{s}^{-1}$.

The efficiency of the pump is defined as:

$$\eta_e = \frac{\rho g H Q}{(T_f + T_r + T_b) \omega} \quad (5)$$

where Q is the flow rate, T_f is the friction torque on the front shroud of the impeller, T_r is the friction torque on the outer surface of the rotor, T_b is the torque on the blades, and ω is the angular velocity of the impeller.

At the design condition, five different turbulence models are tested. The resulting efficiency and head coefficients are compared with experimental data. According to Figure 3, the highest level of consistency with the empirical data is observed in the Shear Stress Transport (SST) $k-\omega$ model. Specifically, their relative errors for the head coefficient all exceed 5%. Therefore, the SST $k-\omega$ model is employed alongside the URANS equations to close the system and accurately capture the primary flow characteristics [35,36].

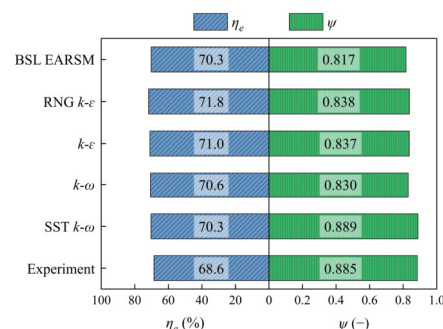


Figure 3. Performance comparison of turbulence models.

The numerical computation is conducted within an absolute pressure reference of 0 Pa. Boundary conditions involve a total pressure of 101,325 Pa at the inlet and a specified mass flow rate at the outlet to match the target conditions. A rotating reference frame is assigned to the impeller domain, while the others remain static. Regarding the numerical schemes, high-resolution spatial discretization for advection and a second-order backward Euler formulation for transient terms are utilized. The working fluid is treated as an incompressible medium, with its density and dynamic viscosity fixed at $997 \text{ kg}\cdot\text{m}^{-3}$ and $8.899 \times 10^{-4} \text{ kg}\cdot\text{m}^{-1}\cdot\text{s}^{-1}$, respectively. All solid boundaries are defined as no-slip walls.

To capture unsteady flow behaviors with high temporal precision, the transient rotor–stator interface is utilized, with each time step set to an angular increment of 0.2 degrees [37]. To guarantee numerical robustness, the mean Courant–Friedrichs–Lewy number is kept under 0.4 across the global domain [38]. Convergence is established once the Root Mean Square (RMS) residual levels [39] for every governing equation reach a threshold of less than 10^{-5} .

2.4. Mesh Sensitivity

To verify grid independence, six mesh configurations with densities spanning from 1.42 to 7.85 million counts are evaluated at Q_d . As depicted in Figure 4, the head coefficient and efficiency served as the primary evaluation indicators. Both parameters exhibit a stabilizing trend once the element count surpasses 5.67 million, confirming that the numerical solution has reached a state of grid insensitivity [40]. Computational times and y^+ values for different mesh counts are listed in Table A1 of Appendix A. The average y^+ for the impeller domain remains constant at 2.6. For the 5.67 million mesh, the average y^+ values for all other domains are below 28 [41]. Consequently, to optimize the trade-off between solution precision and resource requirements, the 5.67 million mesh count is selected for all further investigations [42].

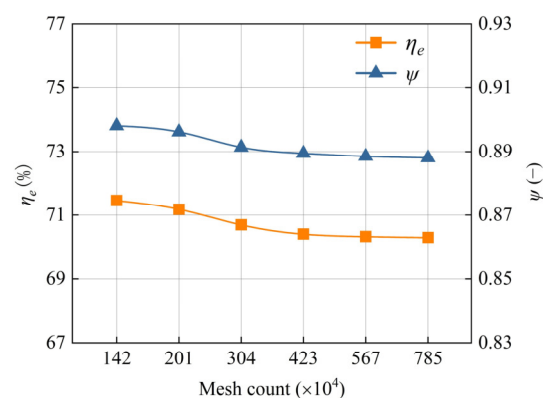


Figure 4. Mesh sensitivity.

2.5. Monitor Position

Figure 5 illustrates the locations of the monitoring points within the impeller and volute. Within the rotating frame of the impeller, two sets of six monitoring points are arranged along the streamwise direction on the blade surfaces at the mid-span. They are designated as PS1 to PS6 on the pressure side and SS1 to SS6 on the suction side. Conversely, all monitoring points in the volute are located in the stationary frame of reference. To capture the RSI, four points (TP1 to TP4) are positioned in the critical region near the tongue. An additional eight points (VP1 to VP8) are distributed circumferentially at the center of the volute cross-sections, arranged in the clockwise direction with an angular spacing of 45° between adjacent points.

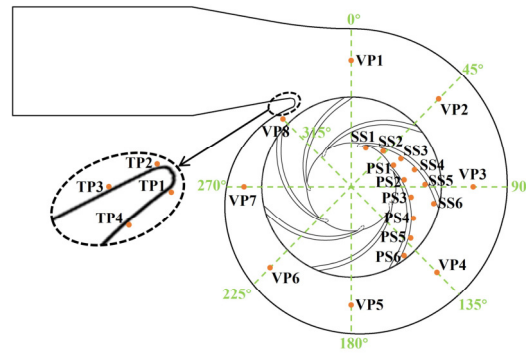


Figure 5. Monitor position.

3. Experimental Setup and Validation

3.1. Experimental Setup

Figure 6 illustrates the experimental test rig used in this research, which consists of the pump under test, a servo motor, pressure sensors, a dynamic torque sensor, a turbine flowmeter, valves, a tank, and a data acquisition system.

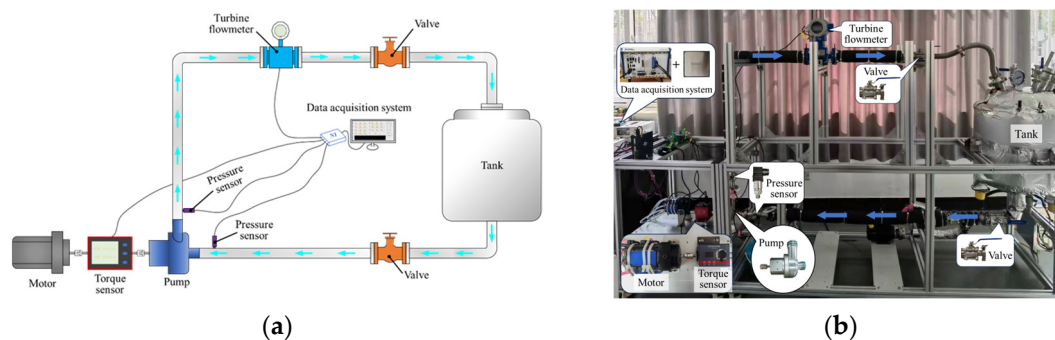


Figure 6. Test rig: (a) schematic layout; (b) experimental setup.

The test rig is compliant with the Grade 1 accuracy specifications for rotodynamic pump performance tests outlined in the ISO 9906:2012 standard [43]. Pump inlet and outlet pressures are measured using two absolute pressure sensors, each boasting an accuracy of $\pm 0.1\%$ of the full-scale (FS) range. The flow rate is quantified by the turbine flowmeter with $\pm 0.5\%$ FS accuracy, while a dynamic torque sensor tracks the pump torque at $\pm 0.2\%$ FS in real-time. For head and efficiency, the systematic uncertainties are calculated to be $\pm 0.14\%$ and $\pm 0.56\%$ [44], respectively. Each sensor is designed to transmit a 4–20 mA current signal to a data acquisition chassis. These inputs are subsequently managed by a LabVIEW 2016 platform using a fixed 1000 Hz sampling frequency, allowing for the continuous observation of performance metrics. During trials, the inlet valve is kept in a wide-open position to reduce hydraulic dissipation, whereas the discharge is controlled by modulating the outlet valve.

3.2. Experimental Validation

The validity of the numerical model is confirmed by benchmarking the predicted head coefficient and efficiency against experimental data across a range of flow conditions, as illustrated in Figure 7. In this analysis, the flow rate is normalized as $Q^* = Q/Q_d$. To maintain the integrity of the comparison, the hydraulic profiles of the physical test pump are strictly consistent with the numerical geometry.

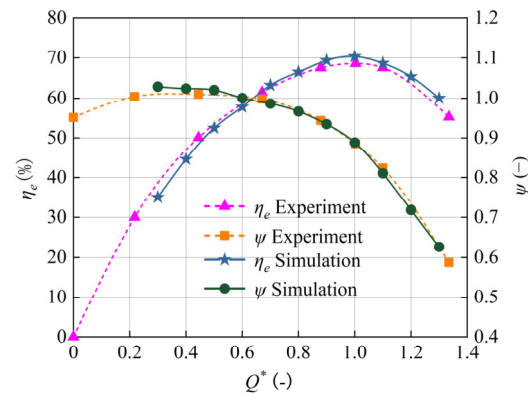


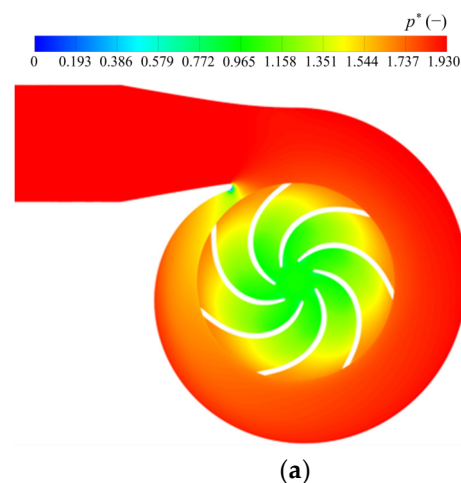
Figure 7. Validation of CFD results with experimental data.

A comparison between the numerical and experimental data reveals good agreement across the tested flow range of $0.3Q_d$ to $1.5Q_d$. Specifically, the predicted head coefficient at Q_d is 0.89. This value shows a relative deviation of only 0.4% from the measured data. At the same point, the simulation predicts an efficiency of 70.3%. This result is 1.7 percentage points higher than the experimental value. Across the full flow range, the average relative error for the head coefficient is 1.5%. The maximum deviation reaches 4.1% at $1.4Q_d$. For efficiency, the mean absolute error is 1.9%. The largest discrepancy is a negative deviation of 3.7% at $0.3Q_d$. Such strong consistency confirms the reliability of the current numerical approach. It justifies the further study of the pump's unsteady flow dynamics [45,46].

4. Results and Discussion

4.1. Pressure Field

Figure 8 presents the time-averaged pressure field on the cross-section of the axial-flux canned motor pump across different flow rates, non-dimensionalized by the pressure coefficient $p^* = p/(0.5\rho u^2)$. A clear trend is observable: the overall pressure level within the pump decreases significantly with increasing flow rate. Under all flow rates, the pressure inside the impeller shows a distinct stepwise increase from inlet to outlet, which reflects the energy addition by the blades.



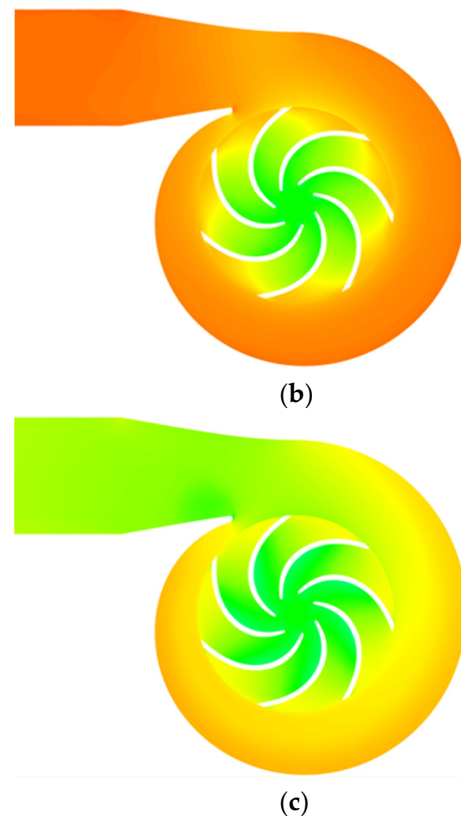
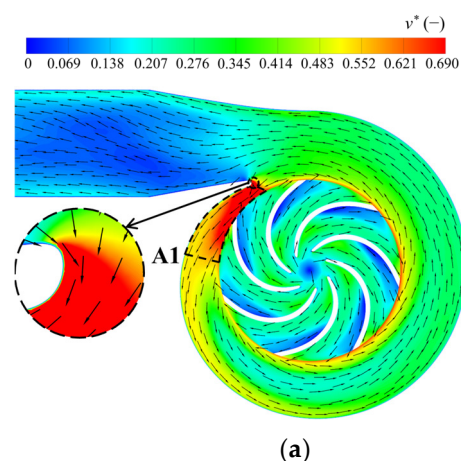


Figure 8. Average pressure field: (a) $0.5Q_d$; (b) Q_d ; (c) $1.5Q_d$.

Under the design condition, a smooth and gradual pressure transition is observed within the volute along the spiral path from the impeller outlet to the volute exit. At $0.5Q_d$, however, a pronounced low-pressure zone forms near the tongue, which in turn generates a strong local pressure gradient. Conversely, at $1.5Q_d$, the flow within the diffuser becomes unstable. Flow separation likely occurs, hindering the efficient conversion of kinetic energy to pressure energy and resulting in a noticeable pressure decay along this section.

4.2. Velocity Field

Figure 9 illustrates the time-averaged velocity field on the cross-section across various flow rates, visualized through both velocity contours and vectors. The velocity is presented in its non-dimensional form, $v^* = v/u_2$, where v represents the velocity magnitude.



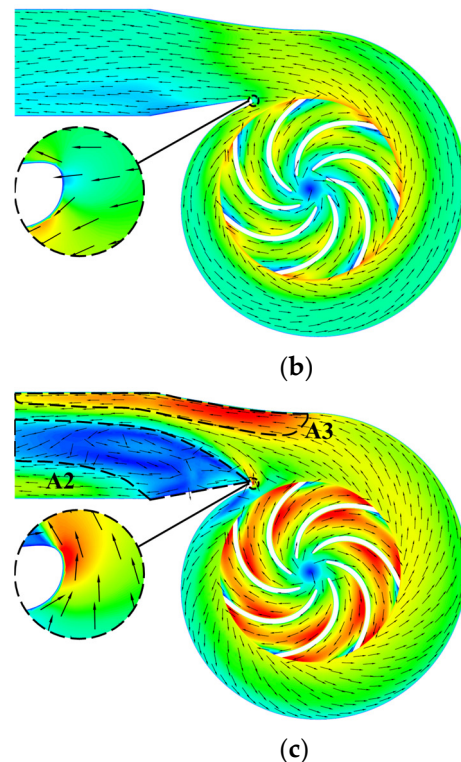


Figure 9. Average velocity field with vectors: (a) $0.5Q_d$; (b) Q_d ; (c) $1.5Q_d$.

At all flow rates, the flow remains well-guided by the blade profiles as it transitions smoothly into the volute, with the vectors aligning with the passage geometry. Under the design condition, as revealed in Figure 9b, the flow within the impeller passage undergoes a smooth and uniform acceleration. Due to the Coriolis force, a cross-passage flow is induced, directed from the suction side towards the pressure side. This secondary flow accumulates near the trailing edge on the pressure side, resulting in a region of locally accelerated flow. Furthermore, at Q_d , the flow exiting the impeller is highly uniform. Within the volute diffuser, the velocity exhibits a smooth and gradual deceleration along the streamwise direction, conforming to ideal diffusion characteristics. The circumferential velocity distribution near the tongue is also uniform, with no significant transverse pressure gradients. This combination of features indicates an excellent hydraulic match between the impeller and the volute.

At $0.5Q_d$, the reduced absolute velocity at the impeller inlet, combined with the constant rotational speed, results in a significant deviation in the relative flow angle. This large positive incidence causes the flow to separate from the suction side of the blade near the trailing edge, which is visible as a distinct low-velocity zone in Figure 9a. As this separated flow exits the impeller and enters the volute, its angle is severely mismatched with the volute's design angle. This leads to the flow impinging on the volute wall rather than aligning smoothly with it, triggering further instability and separation. Furthermore, the smaller cross-sectional area at A1 reduces the effective flow-through area, thus increasing the flow velocity in this region. The velocity vectors reveal that the fluid carries significant rotational momentum as it enters the diffuser, with some of the fluid migrating towards the inner wall. This results in a reverse flow phenomenon at the tongue, where fluid flows backward from the diffuser side towards the impeller side.

At $1.5Q_d$, the flow accelerates along the middle of the blade suction side. This acceleration, combined with the Coriolis effect, creates a distinct high-velocity jet. As the flow enters the volute, its radial momentum is substantially higher. Consequently, when this high-momentum flow impinges on the tongue, its inertia prevents it from smoothly

following the volute's curvature. This causes a portion of the flow to be deflected from the impeller side of the tongue towards the diffuser, leading to a significant velocity increase at the tongue itself, while simultaneously forming a low-velocity region inside A1. At the diffuser inlet, the high-inertia fluid is centrifuged towards the outer wall, creating a high-velocity band at A3. Conversely, the fluid near the inner wall of the diffuser possesses insufficient kinetic energy to overcome the adverse pressure gradient and wall friction. This results in an intensified vortex at A2, which entrains the surrounding low-energy fluid and develops into a pronounced low-velocity zone.

4.3. Vortex Structures

The instantaneous vortex structures and the distribution of turbulent kinetic energy (TKE) within the volute are analyzed using the Q-criterion, with the results for different flow rates presented in Figure 10. The vortex structures are visualized using a fixed isovalue of 0.005. At Q_d , significant vortex structures are observed near the impeller exit and along the volute wall. These structures coincide with regions of elevated TKE, particularly at the impeller outlet.

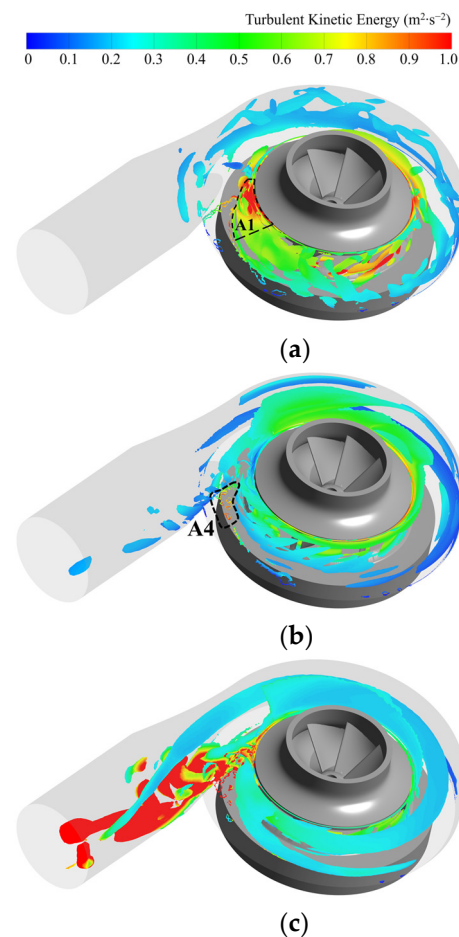


Figure 10. Instantaneous vortex structures with TKE: (a) $0.5Q_d$; (b) Q_d ; (c) $1.5Q_d$.

At $0.5Q_d$, a backflow of high-pressure fluid is established, originating from the diffuser side and recirculating past the tongue towards the impeller side. The interaction between this backflow and the main flow from the impeller outlet leads to a local accumulation of vortices in A1. Through the turbulent energy cascade, these large-scale backflow vortices rapidly break down into highly randomized, smaller eddies. This vortex breakdown transfers kinetic energy to smaller scales, resulting in a continuous build-up

of TKE in A1. Furthermore, the flow field within the volute is characterized by the distribution of discrete, smaller vortices.

At $1.5Q_d$, two distinct spiral vortex bands are generated in the volute. The one near the wall is a low-energy vortex band, which dissipates after migrating for a certain distance along the flow passage. The vortex band in the middle of the volute, however, extends all the way into the diffuser. Meanwhile, the local secondary flow at the tongue interacts strongly with the main flow. This intense interaction destabilizes the shear layer, forming an aggregation zone of broken, high-TKE vortices on the diffuser side.

4.4. Entropy Production

This study adopts the entropy production theory within a URANS-based framework. The governing equations are presented as Equations (6)–(9):

$$\Phi_{\bar{D}} = \frac{\mu}{2} \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right)^2 \quad (6)$$

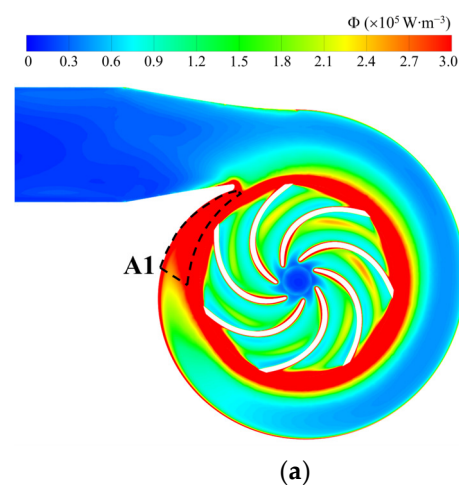
$$\Phi_{D'} = \rho \varepsilon \quad (7)$$

$$\Phi = \Phi_{\bar{D}} + \Phi_{D'} \quad (8)$$

$$P_{EP} = \int_V \Phi dV \quad (9)$$

where $\Phi_{\bar{D}}$ is the viscous dissipation, $\Phi_{D'}$ is the turbulent dissipation, ε is the TKE dissipation rate, Φ is the total entropy production dissipation, P_{EP} is the total entropy production power, and V is the volume.

Figure 11 illustrates the time-averaged entropy production patterns on the pump cross-section under various flow conditions. Elevated energy dissipation is observed at the impeller discharge and near the volute tongue. This is primarily caused by the jet-wake phenomenon and the sharp geometric transitions in these areas. Therefore, the impeller outlet and the tongue are high entropy production areas at all flow rates. At Q_d , the flow in the volute precisely matches the geometry, and dissipation is effectively suppressed. No significant high entropy production concentration zones are formed, which exhibits a characteristic of uniform, low-loss energy transfer. Under off-design flow conditions, the overall entropy production increases relatively.



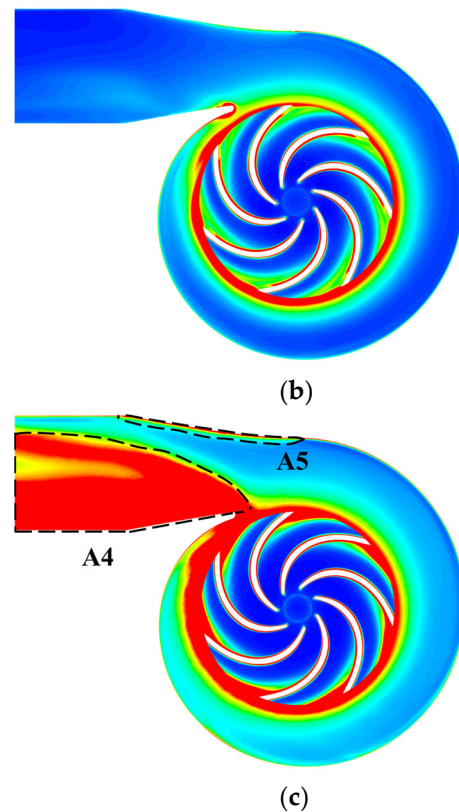


Figure 11. Average entropy production: (a) $0.5Q_d$; (b) Q_d ; (c) $1.5Q_d$.

At $0.5Q_d$, the entropy production inside the impeller is larger than at other flow rates. The breakdown of vortices at the impeller outlet significantly increases TKE dissipation. Local backflow injects high-TKE fluid back into the trailing part of the passage, causing the high entropy production region at the impeller outlet to exhibit a feature of gradual reverse-extension along the impeller passage. In region A1, the high energy dissipation is driven by intense shear-layer interactions. Under $0.5Q_d$, the low-speed backflow from the diffuser collides with the high-speed discharge jet from the impeller. This collision creates an unstable shear layer with steep velocity gradients, triggering rapid vortex shedding and turbulent dissipation. The intense shear stress between these two opposing flows generates extremely strong energy dissipation. Given that A1 is a high Reynolds number region where inertial forces significantly outweigh viscous effects, the energy loss is primarily governed by turbulent processes. The strong spatial correlation between the entropy production peaks and the high TKE levels in Figure 10 further confirms that the energy dissipation in A1 is predominantly driven by turbulence.

At $1.5Q_d$, due to the strong flow separation induced by the high-velocity flow, the entropy production at the trailing edge of the blade suction side increases significantly. In A1, a peak zone of viscous dissipation power density is formed due to the continuous work done by viscous shear stress. A4 contains a cluster of broken vortices. The rapid disintegration of these vortices increases the local TKE dissipation rate, converting turbulent kinetic energy into irreversible thermal loss. This energy cascade is the fundamental scientific reason for the high entropy production characteristic of A4. Energy dissipation in A5 is driven by intense wall shear within the high-velocity band, as shown in Figure 9c, with entropy production primarily originating from TKE dissipation in the high-gradient region.

4.5. Pressure Fluctuations

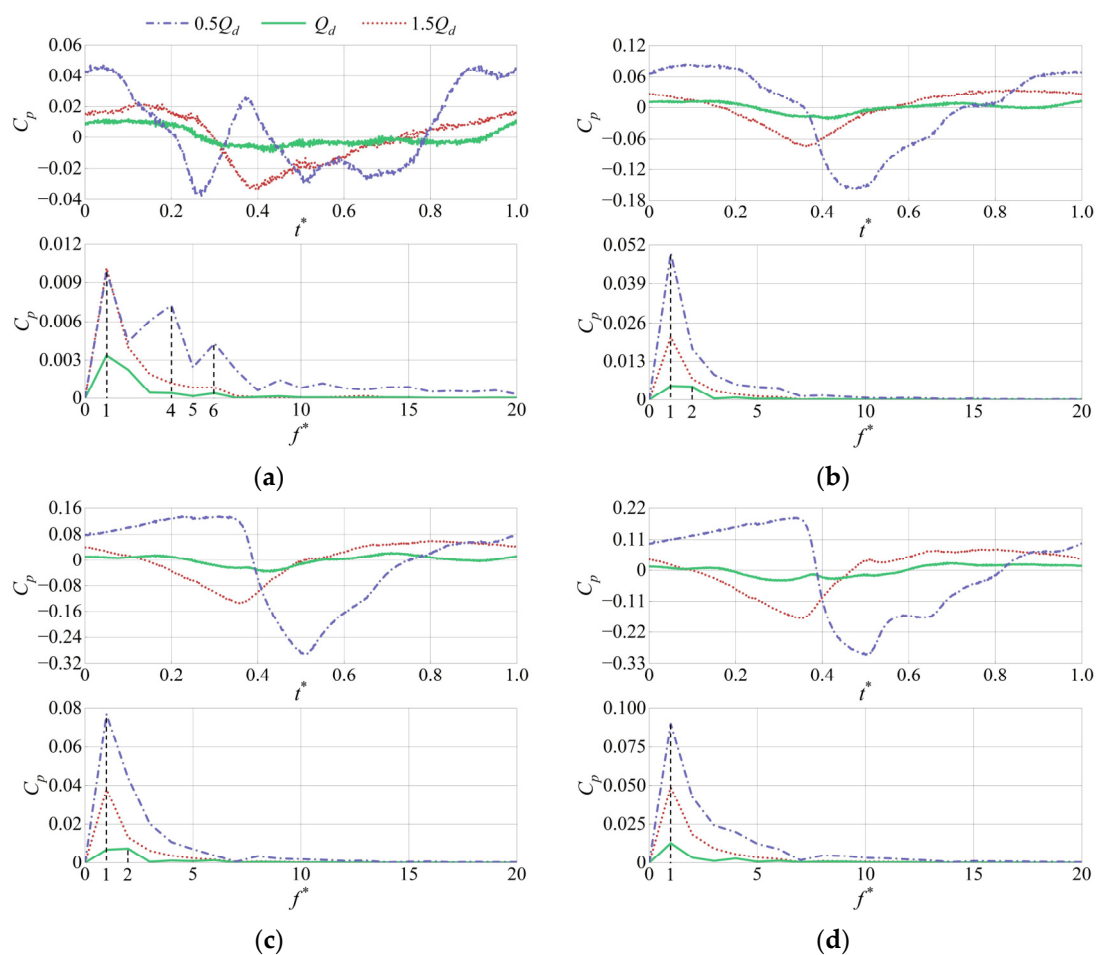
4.5.1. Blade Pressure Side

For a standardized comparison of flow characteristics across various operating conditions, the pressure fluctuations are non-dimensionalized using the pressure fluctuation coefficient, C_p . By eliminating the effects of dimensional quantities like fluid velocity and density, C_p provides a common baseline that enhances the generality and reliability of the findings. It is defined as:

$$C_p = \frac{p_i - \overline{p_\theta}}{0.5\rho u_2^2} \quad (10)$$

where p_i is the instantaneous pressure, and $\overline{p_\theta}$ is the time-averaged pressure.

Figure 12 presents the time-domain and frequency-domain analysis profiles of pressure fluctuations at PS1 through PS6 on the blade pressure side. The time-domain results are illustrated against the dimensionless time t^* , defined as $t^* = t/T$, where T is the time required for an impeller revolution.



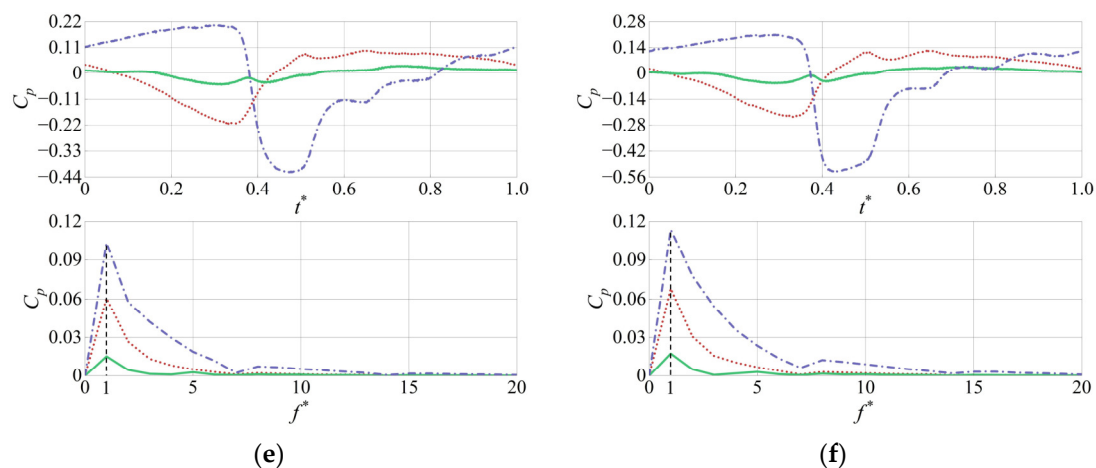


Figure 12. Time- and frequency-domain characteristics: (a) PS1; (b) PS2; (c) PS3; (d) PS4; (e) PS5; (f) PS6.

The spectral results from various monitoring points must be comparable. Therefore, the signals are examined in the frequency domain. This analysis utilizes the Fast Fourier Transform method. A two-step normalization process is also applied to standardize the data. First, the single-sided spectra are normalized to eliminate dependencies on the sampling duration and operating parameters. Second, the frequency axis is non-dimensionalized by the shaft frequency (f_s) to yield the dimensionless frequency, $f^* = f/f_s$.

The statistical analysis of the pressure fluctuations at these points is summarized in Table A2, located in Appendix A. The maximum (Max) and minimum (Min) values delineate the dynamic range of fluctuations. The RMS value quantifies their average intensity.

Time-domain plots show that pressure fluctuations on the blade pressure side are stable at Q_d . However, under off-design conditions, both the fluctuation amplitudes and RMS values increase significantly. At $1.5Q_d$, the fluid velocity within the impeller increases markedly, as shown in Figure 9c, which enhances the RSI as the blades pass the volute tongue. Consequently, the average fluctuation amplitude and RMS value reach 3.4 and 3.7 times the levels at Q_d , respectively.

At $0.5Q_d$, PS2 to PS6 exhibit a sharp pressure drop induced by high TKE vortices at region A1, as shown in Figure 10a. Their average fluctuation amplitude surges to approximately 7.5 times that of Q_d , while the RMS values show a more pronounced increase, averaging 8.2 times the Q_d level. In contrast, PS1 is situated near the impeller inlet, a region with relatively stable flow and weak influence from the RSI with the tongue; its fluctuation amplitude and RMS values are approximately 3.8 and 4.5 times those at Q_d .

Frequency-domain comparison reveals that at Q_d , the dominant frequencies of pressure fluctuations at PS2 and PS3 consist of f_s and $2f_s$ with comparable amplitudes. For the other points, the dominant frequency is solely f_s . Under off-design conditions, the dominant frequencies for PS1 to PS6 are all f_s alone, with significantly increased amplitudes. At $0.5Q_d$, the amplitude of f_s at PS3 reaches up to 11.1 times the value at Q_d . Meanwhile, secondary frequencies of $4f_s$ and $6f_s$ emerge at PS1, with amplitudes approximately 73.7% and 43.5% of the f_s amplitude, respectively. These secondary frequencies stem from specific flow instabilities under $0.5Q_d$. As described in Section 4.2, the large positive incidence angle at the impeller inlet induces unsteady flow fluctuations, acting as the physical source of the $4f_s$ component. On the other hand, the $6f_s$ component represents the pressure modulation caused by flow separation. As shown in Figure 9a, severe flow separation occurs on the blade suction side near the trailing edge, visible as a distinct low-velocity zone. This localized blockage periodically restricts the passage flow, modulating the pressure

field of the 7-blade impeller ($z = 7$) and shifting the excitation frequency to $(z - 1)f_s = 6f_s$ in the rotating frame.

4.5.2. Blade Suction Side

Figure 13 illustrates the time-domain and spectral characteristics of pressure fluctuations. These results correspond to monitoring points SS1 through SS6. All of these locations are situated on the blade suction side. The statistical analysis of the pressure fluctuations at these points is summarized in Table A3, located in Appendix A.

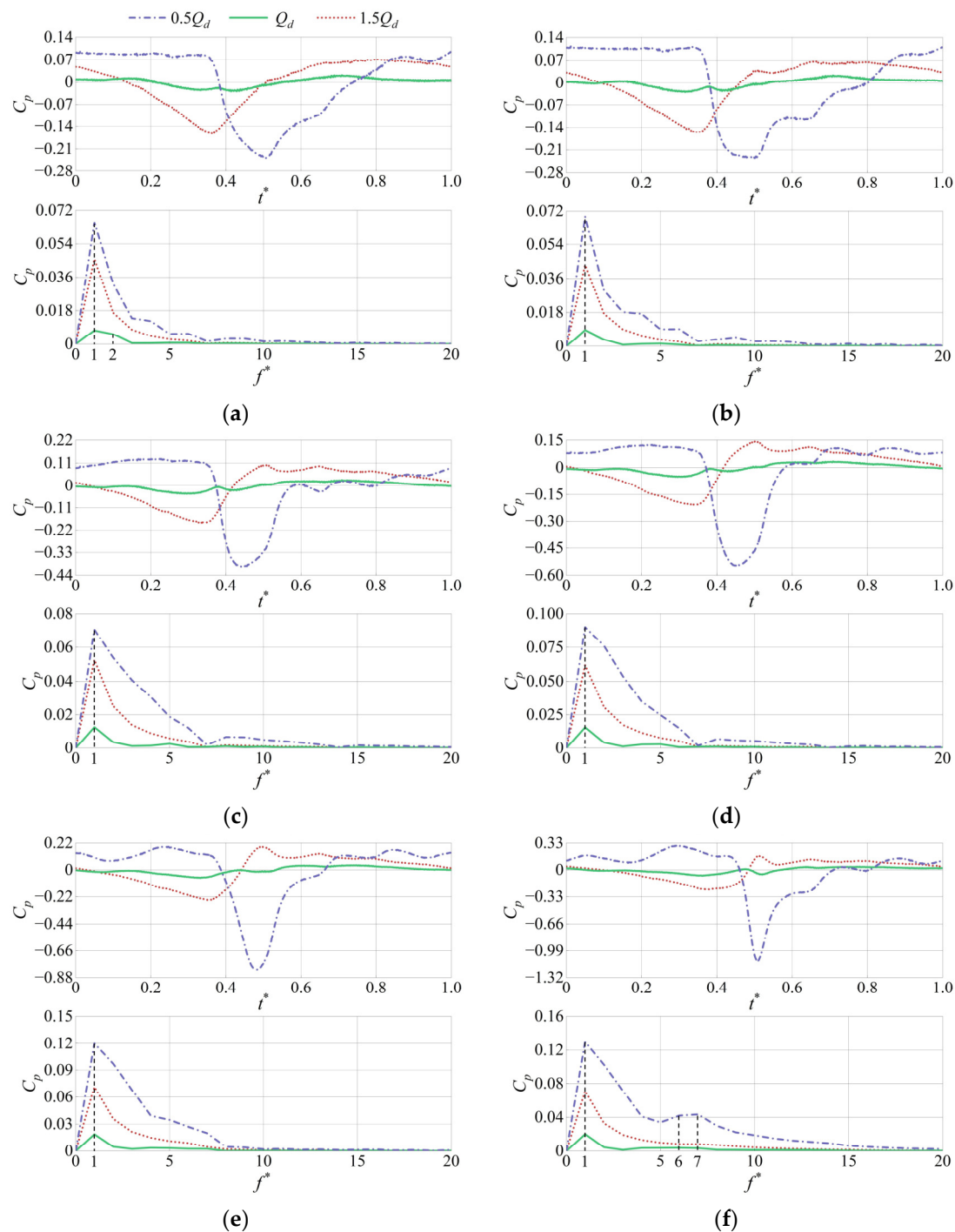


Figure 13. Time- and frequency-domain characteristics: (a) SS1; (b) SS2; (c) SS3; (d) SS4; (e) SS5; (f) SS6.

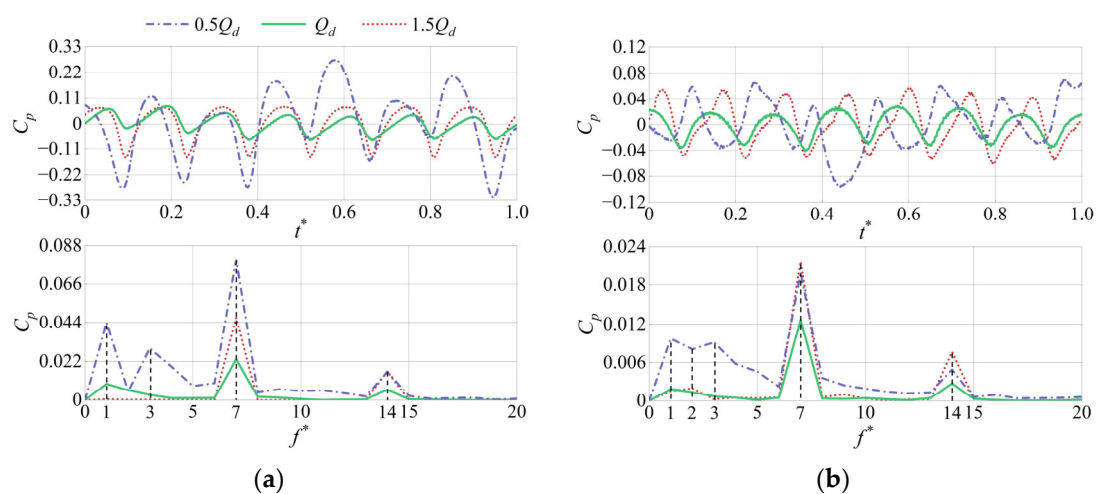
At $0.5Q_d$, the average RMS value of pressure fluctuations for SS1 to SS6 increases to 8.2 times the Q_d level due to high TKE vortices at region A1, as shown in Figure 10a. Simultaneously, their time-domain curves exhibit a distinct pressure drop, which leads to an average increase in the fluctuation amplitude on the suction side to 8.5 times that at Q_d . The synchronization between these pressure drops and the blade passage through A1 confirms the causal link between high-TKE vortices and fluctuations. Notably, SS6 is the most significantly affected, with its amplitude reaching up to 12.5 times the value at Q_d . At $1.5Q_d$, enhanced RSI causes the average amplitude and RMS value on the suction side to increase to 4.2 and 4.5 times those at Q_d , respectively.

Frequency analysis of pressure fluctuations from SS1 to SS6 shows that f_s remains the dominant frequency under all flow rates. Under off-design conditions, its amplitude increases significantly, with more pronounced growth observed at $0.5Q_d$. At $0.5Q_d$ and $1.5Q_d$, the average amplitudes of the dominant frequency on the suction side increase to 6.9 and 4.5 times those at Q_d , respectively. Specifically, the growth of the f_s amplitude is most prominent at SS1, reaching 8.9 and 6.3 times the values at Q_d .

Furthermore, at $0.5Q_d$, flow separation on the blade suction side excites several intermediate frequency components within the range from f_s to the blade passing frequency (BPF), i.e., f_s to $7f_s$. Notably, at SS6, the spectrum exhibits a distinct broadband plateau between $6f_s$ and $7f_s$. The amplitude of this broadband plateau is approximately 33% of the dominant frequency amplitude. This broadband characteristic originates from the unsteady vortex shedding and the stochastic breakdown of the high-TKE vortices in A1. These turbulent structures do not possess a single periodic signature but rather distribute energy across a frequency range adjacent to the BPF.

4.5.3. Tongue

Figure 14 presents the time-domain and frequency-domain analysis profiles of pressure fluctuations at TP1 through TP4 on the tongue. The statistical analysis of the pressure fluctuations at these points is summarized in Table A4, located in Appendix A. As the critical region governing the geometric transition from the impeller to the volute, the pressure characteristics at the tongue are of paramount importance to the pump's operational stability.



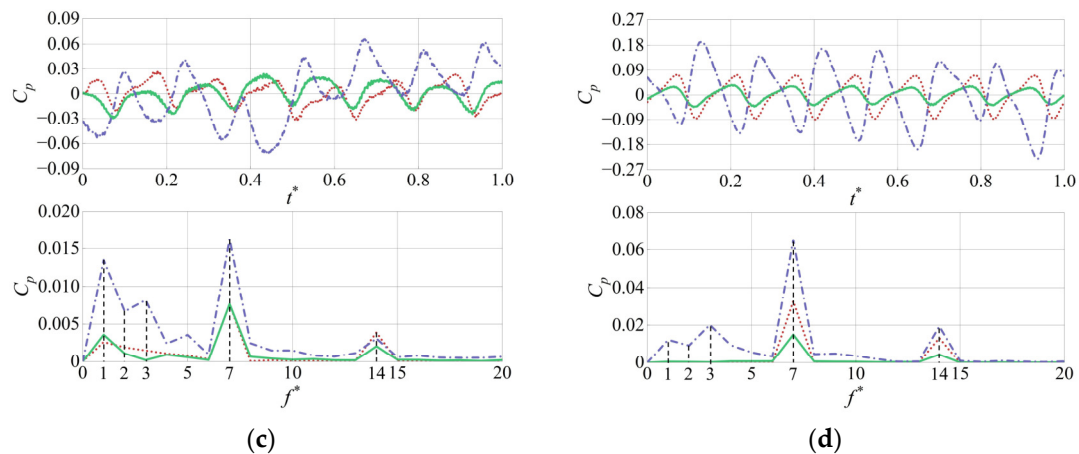


Figure 14. Time- and frequency-domain characteristics: (a) TP1; (b) TP2; (c) TP3; (d) TP4.

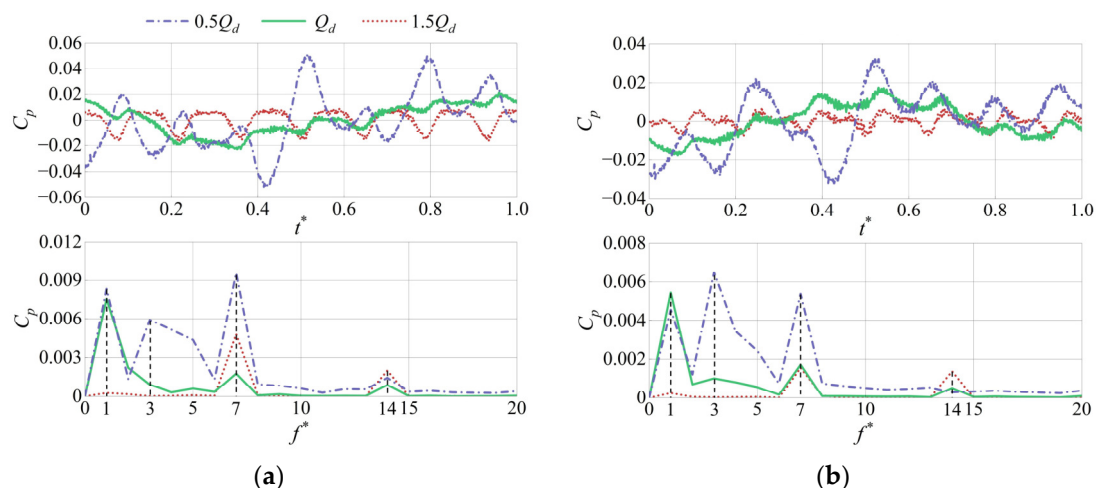
Across all investigated flow rates, the time-domain signals exhibit a distinct and well-defined periodic behavior. The number of pressure cycles per impeller revolution precisely aligns with the number of blades, identifying the BPF as the dominant excitation source. Furthermore, the statistical data in Table A4 reveal a significant disparity in fluctuation intensity between the impeller and diffuser sides of the tongue. The pressure fluctuation intensity is substantially higher on the impeller side (TP1 and TP4), a direct consequence of the periodic pressure pulses generated as each blade sweeps past.

At $0.5Q_d$ and $1.5Q_d$, the pressure fluctuation amplitudes on the impeller side increase to 4.6 and 1.8 times those at Q_d , respectively. The corresponding RMS values also show a synchronized increase to 5.3 and 2.1 times the Q_d levels. TP4 exhibits the most significant growth among these monitoring points. Specifically, at $0.5Q_d$, its fluctuation amplitude and RMS value reach 6.8 and 5.3 times the values at Q_d , respectively.

BPF remains the dominant frequency of pressure fluctuations at the tongue across all flow rates. At $0.5Q_d$, the BPF amplitude reaches its maximum, averaging approximately 2.9 times the value at Q_d . At this condition, the influence of f_s and its harmonics on the pressure fluctuations is also most pronounced. Furthermore, the amplitude of 2BPF increases significantly under off-design conditions.

4.5.4. Volute

Figure 15 presents the time-domain and frequency-domain analysis profiles of pressure fluctuations at VP1 through VP8, situated at the center of the volute cross-sections along the circumferential path. The statistical analysis of the pressure fluctuations at these points is summarized in Table A5, located in Appendix A.



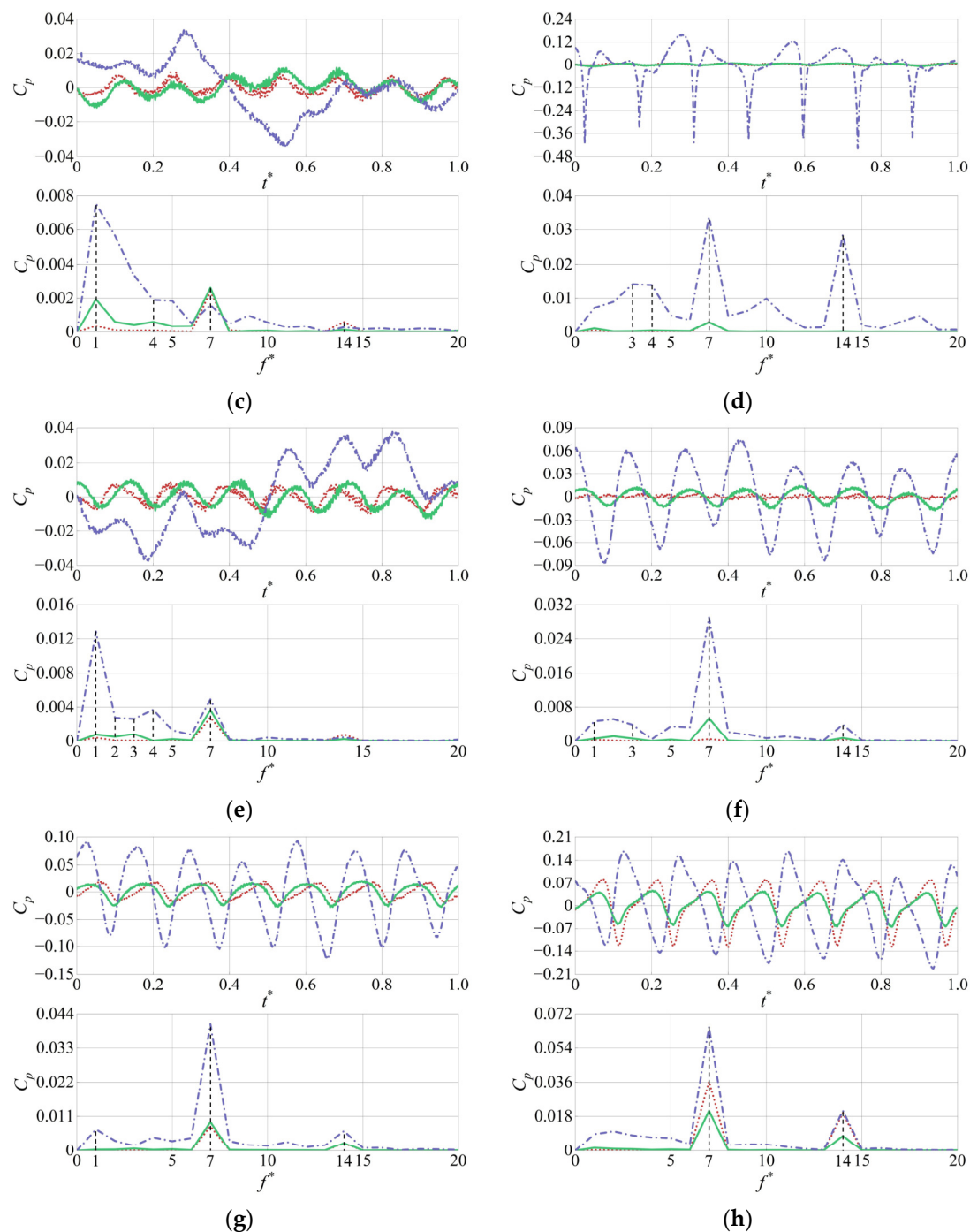


Figure 15. Time- and frequency-domain characteristics: (a) VP1; (b) VP2; (c) VP3; (d) VP4; (e) VP5; (f) VP6; (g) VP7; (h) VP8.

At $0.5Q_d$, VP4 exhibits a distinct pressure drop due to the influence of vortex [47], with its pressure fluctuation amplitude reaching 29.4 times the value at Q_d . In contrast, the fluctuation amplitude at $1.5Q_d$ remains nearly identical to that at Q_d . Notably, the RMS value at VP4 remains consistent across all investigated flow rates. For other locations in the volute, both the fluctuation amplitudes and RMS values increase significantly at $0.5Q_d$, averaging 6.6 and 3.0 times those at Q_d , respectively. At $1.5Q_d$, the overall fluctuation amplitude and RMS value within the volute generally decrease. Only at VP8, which is situated near the tongue, do both parameters increase to 1.9 times the levels at Q_d . At the remaining locations, the amplitude and RMS value decrease by an average of 31% and 35%, respectively.

BPF remains the characteristic frequency of pressure fluctuations within the volute under all operating conditions. At Q_d , the BPF amplitude decreases sequentially along the flow direction from VP8 to VP1. VP1 to VP3 are significantly influenced by f_s , with the f_s amplitude increasing along the flow direction. At $0.5Q_d$, the amplitudes of f_s and BPF increase substantially, reaching up to 16.4 times the values at Q_d . Moreover, the pressure fluctuations in the volute are markedly affected by the harmonics of f_s and the 2BPF at this flow rate. At $1.5Q_d$, BPF remains the dominant frequency at all monitoring points. Amplitude growth is only observed at VP1 and VP8, averaging approximately 2.2 times the levels at Q_d .

5. Conclusions

The present study examined an axial-flux canned motor pump. This pump utilizes a unique integrated impeller–rotor configuration. A numerical approach has been detailed, and it was verified using experimental tests. The results confirm that the computational model is highly reliable. A detailed analysis of the pump’s internal flow characteristics, including the pressure field, velocity field, vortex structures, entropy production distribution, and pressure fluctuations, was conducted at three representative flow rates. The main findings are summarized as follows:

(1) The pressure distribution is optimal at Q_d , featuring a smooth volute pressure transition and uniform pressure at the tongue. In contrast, off-design conditions show significant instabilities. At $0.5Q_d$, a pronounced low-pressure zone forms at the tongue, while at $1.5Q_d$, flow separation causes a significant pressure decay in the diffuser. Overall, the pressure level decreases as the flow rate increases.

(2) The velocity field is highly dependent on the flow rate. The design condition exhibits a well-behaved flow with smooth acceleration in the impeller and uniform deceleration in the volute. At $0.5Q_d$, a large incidence angle causes flow separation on the blade suction side and disrupts the volute flow with impacts and secondary currents. At $1.5Q_d$, a high-velocity jet forms in the impeller, leading to an inertia-dominated high-velocity band and a low-energy recirculation zone in the diffuser.

(3) Vortex structures and TKE distribution are highly sensitive to the flow rate. At Q_d , the flow is well-organized, with vortices confined to the impeller exit and volute wall. At $0.5Q_d$, tongue backflow creates a high-TKE vortex concentration in A1 and widespread fragmented eddies throughout the passage. Conversely, at $1.5Q_d$, the flow field is dominated by two spiral vortex bands that break down into a distinct high-TKE region in the diffuser.

(4) The sources of hydraulic loss, identified by entropy production, are highly sensitive to the flow rate. At Q_d , losses are minimal and uniformly distributed. At $0.5Q_d$, major losses are concentrated at the impeller exit and in A1, driven by backflow and intense flow impingement. At $1.5Q_d$, losses are dominated by flow separation, creating distinct high-entropy zones on the blade suction side and in the diffuser.

(5) Pressure fluctuations intensify significantly under off-design conditions, reflecting an overall rise in turbulent energy that peaks at $0.5Q_d$. At this flow rate, the accumulation and breakup of highly turbulent kinetic energy vortices transfer massive unsteady energy to the blade surfaces. The most dramatic surge is observed in the volute, where fluctuation amplitudes reach up to 29.4 times those at Q_d . Enhanced rotor–stator interaction remains the primary driver of instability at $1.5Q_d$. Across all conditions, BPF remains the dominant frequency, though f_s harmonics become markedly more pronounced during off-design operation.

Despite these valuable findings, this study has certain limitations. Using isothermal room-temperature water ignores thermodynamic effects, cavitation, and viscosity changes present in actual cryogenic or electronic coolant applications. Future research will

incorporate these variable fluid properties and thermal effects. Furthermore, fluid–structure interaction analyses will be conducted to evaluate the structural deformation of the disk-rotor. Additionally, future research trajectories will focus on numerical and experimental investigations using non-Newtonian or cryogenic fluids under actual operating conditions.

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Nomenclature

The following nomenclature is used in this manuscript:

Symbols

b_2	blade outlet width, mm
b_3	volute inlet width, mm
C_p	non-dimensional pressure coefficient
D_1	blade inlet diameter, mm
D_2	blade outlet diameter, mm
D_d	volute outlet diameter, mm
D_r	rotor diameter, mm
L_r	rotor axial length, mm
f	external force, N
f^*	dimensionless frequency
f_s	shaft frequency, Hz
g	standard gravity, m/s ²
H	head, m
p	pressure, Pa
p^*	dimensionless pressure
$p_{1,tot}$	total pressure at the pump inlet, Pa
$p_{2,tot}$	total pressure at the pump outlet, Pa
P_{EP}	total entropy production power loss, W
Q	flow rate, m ³ /h
t	time, s
T_b	blades torque, N·m
T_f	front shroud disk friction torque, N·m
T_r	rotor disk friction torque, N·m

u	velocity, m/s
u_2	circumferential velocity of blade trailing edge, m/s
v^*	dimensionless velocity
V	volume, m ³
x	coordinate, m
z	number of blades
Greek letters	
ϕ	cutwater angle, °
η_e	efficiency
η_{fc}	total friction loss ratio
η_{ffc}	friction loss ratio of the impeller front shroud
η_{rc}	friction loss ratio of the rotor outer surface
μ_{eff}	effective viscosity, kg/(m·s)
ρ	density, kg/m ³
ψ	head coefficient
ω	angular velocity of impeller, rad/s
δ_r	radial clearance between rotor and casing, mm
$\mathcal{E}$	turbulent kinetic energy dissipation rate, m ² ·s ⁻³
Φ	total entropy production dissipation, W·m ⁻³
Φ_D	turbulent dissipation, W·m ⁻³
$\Phi_{\bar{v}}$	viscous dissipation, W·m ⁻³
Subscripts	
d	design condition
i, j, k	index
Abbreviations	
AFM	Axial Flux Motor
BPF	Blade Passing Frequency
CFD	Computational Fluid Dynamics
Max	Maximum
Min	Minimum
PMSM	Permanent Magnet Synchronous Motor
URANS	Unsteady Reynolds-Averaged Navier–Stokes equations
RMS	Root Mean Square
RSI	Rotor–Stator Interaction

Appendix A

Table A1. Mesh parameters and y^+ values.

Mesh Count ($\times 10^4$)	Computational Time (min)	Mean y^+ Value			
		Impeller	Volute	Chambers	Pipes
142	52	2.6	50	35	6.9
201	118	2.6	32	33	6.5
304	330	2.6	28	28	7.0
423	445	2.6	25	28	7.2
567	575	2.6	20	27	7.1
785	750	2.6	18	27	7.0

Detailed statistical parameters, including maximum (Max), minimum (Min), and Root Mean Square (RMS) values for the pressure fluctuation coefficient (C_p) at various monitoring points, are presented below.

Table A2. Pressure fluctuation statistics on the blade pressure side.

Flow Rate Parameter		Monitoring Point					
		PS1	PS2	PS3	PS4	PS5	PS6
0.5 Q_d	Max	0.048	0.085	0.135	0.187	0.206	0.211
	Min	−0.037	−0.158	−0.294	−0.300	−0.420	−0.532
	RMS	0.027	0.076	0.130	0.150	0.186	0.221
Q_d	Max	0.012	0.013	0.024	0.027	0.032	0.036
	Min	−0.010	−0.022	−0.037	−0.038	−0.049	−0.056
	RMS	0.006	0.009	0.015	0.020	0.023	0.026
1.5 Q_d	Max	0.022	0.035	0.061	0.077	0.099	0.125
	Min	−0.033	−0.074	−0.135	−0.172	−0.215	−0.234
	RMS	0.016	0.032	0.058	0.076	0.097	0.110

Table A3. Pressure fluctuation statistics on the blade suction side.

Flow Rate Parameter		Monitoring Point					
		SS1	SS2	SS3	SS4	SS5	SS6
0.5 Q_d	Max	0.104	0.113	0.131	0.125	0.191	0.299
	Min	−0.239	−0.235	−0.401	−0.550	−0.817	−1.114
	RMS	0.109	0.117	0.149	0.150	0.252	0.288
Q_d	Max	0.021	0.024	0.027	0.034	0.040	0.042
	Min	−0.029	−0.030	−0.041	−0.052	−0.065	−0.071
	RMS	0.013	0.014	0.019	0.023	0.028	0.032
1.5 Q_d	Max	0.072	0.068	0.104	0.146	0.193	0.179
	Min	−0.163	−0.157	−0.185	−0.211	−0.248	−0.245
	RMS	0.070	0.068	0.086	0.103	0.120	0.118

Table A4. Pressure fluctuation statistics on the tongue.

Flow Rate	Parameter	Monitoring Point			
		TP1	TP2	TP3	TP4
0.5 Q_d	Max	0.274	0.073	0.066	0.194
	Min	−0.316	−0.096	−0.074	−0.231
	RMS	0.142	0.039	0.035	0.150
Q_d	Max	0.078	0.029	0.025	0.036
	Min	−0.073	−0.041	−0.030	−0.044
	RMS	0.038	0.019	0.013	0.022
1.5 Q_d	Max	0.075	0.058	0.028	0.075
	Min	−0.146	−0.062	−0.034	−0.090
	RMS	0.071	0.033	0.013	0.050

Table A5. Pressure fluctuation statistics in the volute.

Flow Rate	Parameter	Monitoring Point							
		VP1	VP2	VP3	VP4	VP5	VP6	VP7	VP8
0.5 Q_d	Max	0.052	0.032	0.034	0.162	0.038	0.075	0.093	0.168
	Min	−0.052	−0.032	−0.035	−0.456	−0.037	−0.088	−0.123	−0.193
	RMS	0.022	0.015	0.015	0.150	0.021	0.044	0.061	0.101
Q_d	Max	0.021	0.018	0.012	0.010	0.010	0.015	0.021	0.046
	Min	−0.023	−0.018	−0.012	−0.011	−0.012	−0.017	−0.028	−0.065
	RMS	0.012	0.008	0.005	0.150	0.006	0.008	0.014	0.032
1.5 Q_d	Max	0.010	0.007	0.009	0.010	0.010	0.006	0.021	0.082
	Min	−0.017	−0.008	−0.008	−0.010	−0.011	−0.005	−0.021	−0.128
	RMS	0.007	0.003	0.004	0.150	0.004	0.002	0.012	0.060

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