

Investigation of Tandem-Blade Impellers on the Performance and Flow Characteristics of the Centrifugal Pump

Vom Fachbereich Maschinenbau und Verfahrenstechnik
der Rheinland Pfälzischen Technischen Universität Kaiserslautern-Landau
zur Erlangung des akademischen Grades

Doktor-Ingenieur (Dr.-Ing.)

genehmigte

Dissertation

von

Herr

Gang Yang, M.Eng.

aus Nanjing, China

Tag der mündlichen Prüfung:

17.02.2026

Dekan:

Prof. Dr. rer. nat. Roland Ulber

Vorsitzender:

Jun. Prof. Dr.-Ing. Leyu Lin

Berichterstatte:

Prof. Dr.-Ing. Martin Böhle

Prof. Dr.-Ing. habil. Alexander Wiedermann

Acknowledgements

This dissertation was carried out at the Institute of Fluid Mechanics and Turbomachinery (SAM), RPTU University Kaiserslautern-Landau, during my doctoral studies. I would like to express my sincere gratitude to all those who have supported me throughout my doctoral journey and contributed, in different ways, to the completion of this dissertation. First and foremost, I would like to extend my deepest appreciation to my supervisor, Prof. Dr.-Ing. Martin Böhle, for his invaluable guidance, continuous encouragement, and patient support throughout this research. His academic insight, rigorous attitude, and trust have been essential to the development of this work and to my growth as a researcher.

I am also profoundly grateful to Prof. Dr.-Ing. habil. Alexander Wiedermann and Jun. Prof. Dr.-Ing. Leyu Lin for offering me the opportunity to conduct part of my doctoral research in Germany as a doctoral student. His generous support, helpful advice, and inspiring discussions greatly broadened my academic perspective and enriched this dissertation.

My sincere thanks also go to the China Scholarship Council (CSC) for providing financial support for my doctoral training abroad. Without this support, my research stays in Germany and the valuable academic experience it brought would not have been possible. I would further like to express my heartfelt gratitude to Jiangsu University, which has provided me with an excellent academic platform and strong institutional support throughout my doctoral studies. In particular, I am especially grateful to Prof. Desheng Zhang and Prof. Fan Zhang for their special guidance, encouragement, and generous help in both my research and personal development. Their trust and support have been of great importance to me.

I am also thankful to all the teachers, colleagues, and staff members at Institute of Fluid Mechanics and Turbomachinery (SAM) and National Research Center of Pumps for their assistance, professional support, and the stimulating academic environment they provided. Their help, whether in research, experiments, administration, or daily communication, is sincerely appreciated.

Special thanks are due to my Chinese friends in Germany, especially Yi Zhong, Mingyue Liu, Zhenhua Han, Wenjie Peng, Tianyi Jiang, and Yujia Wu who offered me tremendous support during my life abroad. Their companionship, kindness, and practical help made my time in a foreign country much easier and warmer. I am deeply grateful for the encouragement and friendship we shared during that period.

I would also like to thank my best friends, including Chengzhen Guan, Yan Zhang, and Jianyu Geng for their understanding, encouragement, and support throughout these years. Their presence, whether through discussions or companionship in daily life, has meant a great deal to me.

Finally, and most importantly, I would like to express my deepest gratitude to my family, especially my parents, for their unconditional love, endless support, and constant belief in me. Their sacrifices, encouragement, and understanding have always been the strongest foundation of my journey.

My most heartfelt thanks go to my fiancée, Jiayu Mao, for her love, patience, and unwavering support. She has stood by me through every challenge and every moment of self-doubt, offering me strength, comfort, and confidence. Her understanding and companionship have been invaluable throughout this journey.

Kaiserslautern, February 2026

Gang Yang

Contents

List of Figures.....	V
List of Tables.....	XI
Nomenclature	XIII
Abstract.....	XIX
Kurzfassung.....	XXI
1 Introduction.....	1
1.1 Overview and Motivation	1
1.2 Current Research Progress	2
1.2.1 Approaches for Centrifugal Pump Performance Enhancement	3
1.2.2 Internal Flow Characteristics and Unsteady Phenomena in Centrifugal Pumps	5
1.2.3 Tandem Blade Technology in Pumps and Turbomachinery	8
1.3 Research Objectives and Report Structure.....	9
2 Theoretical Basis and Research Methods	11
2.1 Nomenclature and Notation	11
2.2 Operating Principle and Performance Indicators of Centrifugal Pumps.....	12
2.2.1 Principle and Performance Parameters of the Centrifugal Pump	13
2.2.2 External Performance Curves of the Centrifugal Pump	15
2.3 Operating Principle of the Tandem Impeller.....	17
2.4 Unstable Flow Phenomena in the Impeller	18
2.4.1 Unstable Flow Structures.....	18
2.4.2 Vortex and Transport Characteristics	22
2.4.3 Pressure Pulsations and Excitation Forces.....	26
3 Hydraulic Design of Centrifugal Pump and Tandem Impeller Design Methods. 29	
3.1 Hydraulic Design of Centrifugal Pump	29
3.1.1 Hydraulic Design of Impeller	30
3.1.2 Hydraulic Design of Volute	34
3.2 Hydraulic Design of Tandem Impeller.....	37
3.3 Optimization of Tandem Impeller Rear Blades	40
4 Experimental Setup and Test Methods	43
4.1 Experimental Loop Design	43
4.2 Structural Design of the Test Pump	44
4.3 Measurement Parameters and Instruments	46

4.4 Experimental Measurement Methods	48
4.5 Data Processing Methods and Uncertainty Analysis	48
5 Numerical Simulation and Validation.....	51
5.1 Numerical Calculation Theory	51
5.1.1 Governing Equations	51
5.1.2 Turbulence Model	52
5.2 Numerical Calculation Model Construction	55
5.3 Grid Research.....	56
5.3.1 Grid Generation	56
5.3.2 Grid Independence Research	57
5.4 Numerical Calculation Settings	60
5.4.1 Boundary Conditions	61
5.4.2 Time Step Verification	62
5.5 Numerical Simulation Accuracy Verification	63
6 Comparison Analysis of Centrifugal Pump Performance.....	67
6.1 Influence of Impeller Design Parameters on Performance	67
6.1.1 Influence of Blade Number on Performance	67
6.1.2 Influence of Blade Wrap Angle on Performance	71
6.1.3 Comparison of Flow Patterns with Different Conventional Impellers	74
6.2 Performance Analysis of Tandem Impeller.....	81
6.2.1 Influence of Axial Offset on Performance	82
6.2.2 Influence of Rear Blade Angle on Performance	83
6.2.3 Performance Comparison between Tandem Impeller and Conventional Impeller.....	86
6.3 Adaptability Analysis of Tandem Impeller with Different Blade Numbers.....	89
6.3.1 Comparison of Head	89
6.3.2 Comparison of Efficiency	90
6.3.3 Comparison of Shaft Power	91
6.3.4 Performance Improve Evaluation of Tandem Impeller	92
7 Comparison Analysis of Internal Flow Characteristics of the Centrifugal Pump 	95
7.1 Comparative Analysis of Mainstream Flow Patterns.....	95
7.1.1 Pressure and Velocity Distribution.....	95
7.1.2 Potential Rothalpy Distribution	108
7.2 Comparison Analysis of Vortex Characteristics.....	118
7.2.1 Vorticity Distribution and Intensity	119
7.2.2 Vortex Evolution and Transport Characteristics.....	123
7.3 Comparison Analysis of Unsteady Excitation Force Characteristics.....	132

7.3.1 Pressure Pulsation Characteristics	133
7.3.2 Axial Force and Radial Force	159
8 Discussion	169
8.1 Performance Improvement of Tandem Impeller	169
8.2 Application Prospects and Feasibility Analysis	172
9 Conclusion and Outlook.....	175
9.1 Main Research Conclusion	175
9.2 Outlook.....	177
References.....	181
Publications and Conferences.....	193
List of Supervised Student Thesis	195
Curriculum Vitae	197

List of Figures

2.1. Main dimensions	11
2.2. Nomenclature	12
2.3. Coordinate system.....	12
2.4. Velocity triangle	14
2.5. Pump performance curves.....	15
2.6. Forms of the Q - H curve	16
2.7. Principle of the tandem impeller	17
2.8. Secondary flow at the impeller outlet of a centrifugal impeller [83].....	19
2.9. Principle of flow separation on blade surfaces [82].....	20
2.10. Backflow phenomenon at the impeller inlet and outlet	21
2.11. Jet wake phenomenon at the impeller outlet	22
2.12. Axial force distribution diagram of centrifugal pump impeller	27
3.1. The axial projection design of the centrifugal pump impeller	32
3.2. The blade angle design of the centrifugal pump impeller.....	33
3.3. The 3D blades design of the centrifugal pump impeller	34
3.4. The cross sections design of the volute.....	36
3.5. The diffusion section design of the volute	37
3.6. The hydraulic design of the volute.....	38
3.7. Main dimensions of the centrifugal pump	39
3.8. The gap design of the tandem impeller	40
3.9. The offset design of the tandem impeller rear blade	41
3.10. The inlet blade angle design of the tandem impeller rear blade	42
4.1. 3D experimental system loop.....	44
4.2. 2D experimental loop and instruments layout diagram	44
4.3. Test pump general structure	45
4.4. Test pump structural decomposition	46
4.5. Impeller structure models.....	46
5.1. Numerical simulation computational domains	56
5.2. Mesh discretization of the full model	57
5.3. Pressure distribution at midspan of impeller blades under different mesh schemes	60
5.4. Average y^+ value distribution on impeller blades.....	60
5.5. Boundary condition settings.....	62
5.6. Time step validation	62
5.7. Comparison of experimental and simulation performance curves of the	

conventional impeller	64
5.8. Comparison of experimental and simulation performance curves of the tandem impeller.....	65
6.1. Geometry of impellers with different blade numbers	68
6.2. Comparison of head curves for impellers with different blade numbers	69
6.3. Comparison of efficiency curves for impellers with different blade numbers.....	70
6.4. Comparison of shaft power curves for impellers with different blade numbers ...	71
6.5. Geometry of impellers with different blade wrap angles	71
6.6. Comparison of head curves for impellers with different blade wrap angles.....	72
6.7. Comparison of efficiency curves for impellers with different blade wrap angles	73
6.8. Comparison of shaft power for impellers with different blade wrap angles	74
6.9. Comparison of velocity streamlines for impellers with different blade numbers .	75
6.10. Comparison of total pressure ratio for impellers with different blade numbers .	77
6.11. Comparison of velocity streamlines for impellers with different blade wrap angles	79
6.12. Comparison of total pressure ratio for impellers with different blade wrap angles	81
6.13. Tandem impellers with different axial offset.....	82
6.14. Comparison of performance curves for tandem impellers with different axial offset	83
6.15. Tandem impellers with different rear blade angle	84
6.16. Comparison of head for tandem impellers with different rear blade angle.....	84
6.17. Comparison of efficiency for tandem impellers with different rear blade angle	85
6.18. Comparison of shaft power for tandem impellers with different rear blade angle	86
6.19. Comparison of head curves between conventional impeller and tandem impeller	87
6.20. Comparison of efficiency curves between conventional impeller and tandem impeller.....	88
6.21. Comparison of shaft power curves between conventional impeller and tandem impeller.....	88
6.22. Geometry of tandem impellers with different blade numbers.....	89
6.23. Comparison of head curves between conventional and tandem impellers with different blade numbers.....	90
6.24. Comparison of efficiency curves between conventional and tandem impellers with different blade numbers.....	91
6.25. Comparison of shaft power curves between conventional and tandem impellers with different blade numbers.....	92
6.26. Comparison of ΔH_{re} with different blade numbers	93

6.27. Comparison of ψ_H with different blade numbers	94
7.1. Comparison of total pressure ratio of conventional and tandem impellers with different blade numbers.....	97
7.2. Comparison of velocity streamlines of conventional and tandem impellers with different blade numbers.....	99
7.3. Comparison of pressure distribution of conventional and tandem impellers with different blade numbers.....	101
7.4. Comparison of blade surface load distribution of conventional and tandem impellers with different blade numbers.....	103
7.5. Comparison of $c_{u,22}/u_{22}$ distribution of conventional and tandem impellers with different blade numbers.....	104
7.6. Comparison of flow angle of conventional and tandem impellers outlet with different blade numbers.....	106
7.7. Comparison of flow angle of conventional and tandem impellers middle with different blade numbers.....	108
7.8. Two force actions under ideal flow pattern in the impeller.....	110
7.9. Comparison of potential rothalpy of conventional and tandem impellers with different blade numbers.....	113
7.10. Comparison of PRG distribution of conventional and tandem impellers with different blade numbers (white lines-PRG contour lines, and the dyed lines-relative velocity streamlines)	116
7.11. Comparison of average PRG value from conventional and tandem impellers inlet to outlet with different blade numbers	118
7.12. Comparison of vorticity in conventional and tandem impellers with different blade numbers	122
7.13. Transient of vorticity in conventional and tandem impellers under different moments	123
7.14. Comparison of omega vortex identification in conventional and tandem impellers with different blade numbers.....	125
7.15. Transient of omega vortex identification in conventional and tandem impellers under different moments	127
7.16. Comparison of vortex transport terms in conventional and tandem impellers with different blade numbers.....	130
7.17. Transient of RVS term in conventional and tandem impellers under different moments	131
7.18. Transient of CORF term in conventional and tandem impellers under different moments	132
7.19. Distribution of monitoring points.....	134
7.20. Comparison of pressure pulsation intensity in the pump with 6 blades conventional	

impeller under different conditions	135
7.21. Comparison of pressure pulsation intensity in the pump with 6 blades conventional and tandem impellers under different conditions	136
7.22. Comparison of pressure pulsation intensity in the volute under different conditions	137
7.23. Comparison of pressure pulsation intensity in the pump with conventional and tandem impellers under different blade numbers ($1.0Q_{des}$)	138
7.24. Comparison of pressure pulsation intensity in the volute with conventional and tandem impellers under different blade numbers ($1.0Q_{des}$)	139
7.25. Comparison of pressure pulsation intensity in the gap of tandem impellers under different blade numbers	140
7.26. Comparison of pressure pulsation time domain signals in the pump with 6 blades conventional and tandem impellers under different conditions.....	141
7.27. Comparison of pressure pulsation time domain signals in the pump with conventional and tandem impellers under different blade numbers ($1.0Q_{des}$) .	142
7.28. Comparison of pressure pulsation time domain signals in the volute with conventional and tandem impellers under different blade numbers ($1.0Q_{des}$) .	143
7.29. Comparison of pressure pulsation time domain signals in the gap of tandem impellers under different blade numbers.....	144
7.30. Comparison of pressure pulsation frequency domain signals in the pump with 6 blades conventional and tandem impellers under different conditions	145
7.31. Comparison of pressure pulsation frequency domain signals in the middle of SS with conventional and tandem impellers under different blade numbers ($1.0Q_{des}$)	146
7.32. Comparison of pressure pulsation frequency domain signals in the middle of PS with conventional and tandem impellers under different blade numbers ($1.0Q_{des}$)	147
7.33. Comparison of pressure pulsation frequency domain signals in the V_1 with conventional and tandem impellers under different blade numbers ($1.0Q_{des}$) .	148
7.34. Comparison of pressure pulsation frequency domain signals in the V_2 with conventional and tandem impellers under different blade numbers ($1.0Q_{des}$) .	148
7.35. Comparison of pressure pulsation frequency domain signals in the gap with tandem impellers under different blade numbers.....	150
7.36. Time-frequency characteristics of pressure pulsations in the middle of SS with conventional and tandem impellers under different flow conditions	151
7.37. Time-frequency characteristics of pressure pulsations in the middle of PS with conventional and tandem impellers under different flow conditions.....	152
7.38. Time-frequency characteristics of pressure pulsations in the middle of SS with conventional and tandem impellers under different blade numbers ($1.0Q_{des}$) .	153

7.39. Time-frequency characteristics of pressure pulsations in the middle of PS with conventional and tandem impellers under different blade numbers ($1.0Q_{des}$) .	154
7.40. Time-frequency characteristics of pressure pulsations in the V_1 with conventional and tandem impellers under different blade numbers ($1.0Q_{des}$)	155
7.41. Time-frequency characteristics of pressure pulsations in the V_2 with conventional and tandem impellers under different blade numbers ($1.0Q_{des}$)	156
7.42. Time-frequency characteristics of pressure pulsations in the gap of 6 blades tandem impeller under different flow conditions	157
7.43. Time-frequency characteristics of pressure pulsations in the gap of tandem impellers under different blade numbers ($1.0Q_{des}$).....	158
7.44. Peak to peak values of axial thrust with conventional and tandem impellers under different blade numbers.....	159
7.45. Comparison of axial thrust time and frequency domain signals with conventional and tandem impellers under different blade numbers.....	162
7.46. Time-frequency characteristics of axial thrust with 6 blades conventional and tandem impellers under different flow conditions	163
7.47. Time-frequency characteristics of axial thrust with conventional and tandem impellers under different blade numbers ($0.7Q_{des}$).....	164
7.48. Peak to peak values of radial force with conventional and tandem impellers under different blade numbers	165
7.49. Comparison of radial force time and frequency domain signals with 6 blades conventional and tandem impellers under different flow conditions	166
7.50. Comparison of radial force time and frequency domain signals with conventional and tandem impellers under different blade numbers ($0.7Q_{des}$)	166
7.51. Comparison of radial force vectors with 6 blades conventional and tandem impellers under different flow conditions	167

List of Tables

3.1. Primary parameters of the centrifugal pump.....	38
4.1. Measurement instrument list on the test rig	47
4.2. Results of measurement uncertainty calculations	50
5.1. Mesh independence research scheme information. (Unit: million)	59
5.2. Mesh discretization uncertainty analysis.....	59
6.1. Blade angle setting information at Span=0.5	84

Nomenclature

Symbols

Symbols	Meaning	Unit
D_i	Shaft diameter	mm
D_h	Hub diameter	mm
D_j	Impeller inlet diameter	mm
D_2	Impeller outlet diameter	mm
b_2	Impeller outlet width	mm
D_3	Volute base circle diameter	mm
D_4	Volute outlet diameter	mm
ψ_0	Volute tongue placement angle	°
b_3	Volute inlet width	mm
H	Head	m
Q	Flow rate	m ³ /h
η	Efficiency	%
P	Power	kW
u	Impeller circumferential velocity	m/s
c_u	Circumferential component of the absolute velocity	m/s
g	Gravitational acceleration	m/s ²
c	Absolute velocity	m/s
w	Relative velocity	m/s
H_{th}	Theoretical head	m
c_m	Meridional velocity	m/s
β_1	Blade inlet angle	°
β_2	Blade outlet angle	°
E_d	Energy per unit weight of the liquid at the pump outlet	-
E_s	Energy per unit weight of the liquid at the pump inlet	-
p_1	Static pressures at the pump inlet	Pa
p_2	Static pressures at the pump outlet	Pa
Q_m	Mass flow rate	kg/s
ρ	Fluid density	kg/m ³
M	Torque	N·m
ω	Angular velocity	rad/s
P_e	Pump output power	kW
n	Rotational speed	r/min
i, j, k	Unit vectors in the x , y , and z directions	-

R_{ω}	Vorticity rotational component	s^{-1}
S_{ω}	Vorticity non-rotational component	s^{-1}
$\nabla \mathbf{V}$	Velocity gradient tensor	-
$\mathbf{B}$	Rate-of-rotation tensor	-
$\mathbf{A}$	Rate-of-strain tensor	-
ε_{ω}	An infinitesimal positive value	-
ω	Total vorticity	s^{-1}
$\mathbf{V}$	Velocity vector	-
ν	Kinematic viscosity	m^2/s
f	Body force	N
V_{α}	Angular velocity	rad/s
ν_m	Kinematic viscosity	m^2/s
ν_t	Turbulent viscosity	m^2/s
z	Number of blades	-
$F(t)$	Excitation force	N
$p(t,r)$	Instantaneous pressure distribution	-
n_v	Surface normal vector	-
S	Physical wall surface of the impeller	-
F_A	Axial force	N
F_R	Radial force	N
n_q	Specific speed	-
Q_{des}	Design flow rate	m^3/h
H_{des}	Design head	m
D_{min}	Minimum shaft diameter	mm
τ	Shear stress of the material	MPa
SF	Safety factor	-
D_i	Shaft diameter at the impeller mounting position	mm
D_0	Equivalent impeller inlet diameter	mm
k_D	Impeller outlet diameter coefficient	-
k_{D2}	Correction coefficient for D_2	-
k_b	Impeller outlet width coefficient	-
k_{b2}	Correction coefficient for b_2	-
c_{m1}	Radial velocity at the impeller inlet	m/s
c_{m2}	Radial velocity at the impeller outlet	m/s
k_2	Correction factor for the outlet flow channel blade area	-
β_{22}	Relative fluid flow angle of impeller outlet	$^{\circ}$
T_1	Leading edge thickness	mm
T_m	Midsection thickness	mm
T_2	Trailing edge thickness	mm

A	Volute cross-sectional area	mm^2
r_c	Curvature radius	mm
k_{D4}	Volute outlet diameter coefficient	-
b_s	Meridional chord lengths from the LE to TE along the shroud	mm
b_h	Meridional chord lengths from the LE to TE along the hub	mm
l_s	Meridional distances from the LE to the gap along the shroud	mm
l_h	Meridional distances from the LE to the gap along the hub	mm
Δr	Radial gap size	mm
D	Circumferential offset of the rear blade	-
Δu	Length of the circumferential offset of the rear blade	mm
r_m	Radius at the gap position	mm
β_{21}	Inlet attack angle of the rear blade	$^\circ$
c_{m21}	Meridional velocity of the flow entering the rear blade	mm
A_F	Flow area at the rear blade inlet section	mm^2
b_m	Channel width at the gap position	mm
t_B	Blade thickness at the gap	mm
U_D	Local circumferential pitch at the gap radius	-
n_{sp}	Design rotational speed	r/min
Q_T	Transfer flow rate	m^3/h
H_T	Transfer head	m
η_T	Transfer efficiency	%
$s_{\bar{x}}$	Random errors	%
$b_{\bar{x}}$	Systematic errors	%
$U_{\bar{x}}$	Total uncertainty of the experimental measurement	%
N	Sample count	-
X_j	Sample variable value for each independent measurement	-
M	Number of factors contributing to systematic error	-
$b_{\bar{x}_m}$	Standard deviation estimate for the Mth error source	-
t	Time	s
u_i	Velocity tensor	m/s
S_i	Generalized source in the momentum conservation equation	-
μ	Kinetic viscosity	$\text{Pa}\cdot\text{s}$
μ_t	Turbulent viscosity	$\text{Pa}\cdot\text{s}$
k	Turbulent kinetic energy	m^2/s^2
δ_{ij}	Kronecker delta	-
ω	Turbulence frequency	Hz
P_k	Turbulent kinetic energy production rate	-
$P_{kb}, P_{\omega b}$	Turbulent kinetic energy rates due to buoyancy forces	-
S	Strain rate tensor	-

φ_1	Internal constant	-
φ_2	External constant	-
L_t	Turbulence length scale	-
$\Delta_{\max}$	Grid spacing	-
F_{DES}	Detached eddy simulation factor	-
$\Delta_{\max}$	Locally maximum grid size	-
D_k	Turbulent kinetic energy dissipation term	-
e_a^{32}	Relative error between the medium and coarse meshes	%
e_{ext}^{32}	Extrapolated relative error	%
GCI_{medium}^{32}	Fine-grid convergence index	-
h	Grid scale parameter	-
r_{21}, r_{32}	Grid refinement ratios	-
ΔV_i	Volume of the i -th mesh cell	-
N	Total number of mesh cells	-
ϕ_k	Computational results of flow parameters based on k -th grids	-
y^+	Height index of grid layer near wall	-
Δt	Unsteady simulations time-step	s
p_{tot}	Total pressure variable	-
Ω	Total pressure ratio	-
$\Delta\varphi$	Rear-blade circumferential rotation angles	°
β_{12}	Outlet setting angle of the front blade	°
$\Delta H_{\max}$	Maximum head rise	m
ΔH_{re}	Relative head increment percentage	%
H_{tandem}	Experimental head of the tandem impeller	m
H_{conv}	Experimental head of the conventional impeller	m
ψ_H	Unit energy consumption index	kW/m
C_p	Pressure coefficient	-
P_{ave}	Average pressure over the final two revolutions	Pa
P_{ref}	Average pressure at the impeller inlet	Pa
ΔC_p	Pressure coefficient difference between the PS and SS	-
$c_{u,22}$	Circumferential absolute velocity of the impeller outlet	m/s
I_p	Potential rothalpy	J/kg
Q_r	Second invariant	-
$\nabla \mathbf{V}_r$	Relative velocity gradient tensor	-
F_c	Coriolis force	N
$\mathbf{V}_r$	Relative velocity	m/s
$-\nabla I_p$	Potential rothalpy gradient	m/s ²
T_n	One full rotation period of the impeller	s
A_P	Standard deviation of the pressure pulsation	-

C_{PP}	Dimensionless pressure pulsation coefficient	-
f_n	Shaft frequency	Hz
f_{BPF}	Blade passing frequency	Hz
ΔF_A	Peak-to-peak value of the axial force	N
ΔF_R	Peak-to-peak value of the radial force	N

Abbreviations

Symbol	Name
RVS	Relative Vortex Stretching
CORF	Coriolis Force
RSI	Rotor-Stator Interactions
PS	Pressure Surface
SS	Suction Surface
CFD	Computational Fluid Dynamics
WA	Wrap Angle
DOE	Design of Experiments
CCD	Central Composite Design
LHD	Latin Hypercube Design
PIV	Particle Image Velocimetry
BPF	Blade Passing Frequency
RANS	Reynolds-Averaged Navier-Stokes
LES	Large Eddy Simulation
LE	Leading Edge
POD	Proper Orthogonal Decomposition
DMD	Dynamic Mode Decomposition
TE	Trailing Edge
Ω	Omega
RVD	Relative Vortex Dilatation
BT	Baroclinic Torque
VISD	Viscous Diffusion
FFT	Fast Fourier Transform
DNS	Direct Numerical Simulation
N-S	Navier-Stokes
SST	Shear Stress Transport
DES	Detached Eddy Simulation
GCI	Grid Convergence Index
RPTU	Rheinland-Pfälzische Technische Universität Kaiserslautern-Landau
AHRP	Alliance for High Performance Rhineland-Palatinate

FVM	Finite Volume Method
RMS	Root Mean Square
EXP	Experimental Measurement
PRG	Potential Rothalpy Gradient
IBFV	Inlet Backflow Vortex
LEV	Leading Edge Separation Vortex
SSV	Suction Surface Separation Vortex
TEV	Trailing Edge Vortex
TED	Turbulent Energy Dissipation
CWT	Continuous Wavelet Transform
<i>O</i>	Origin
TR-PIV	Transient Particle Image Velocimetry
LDV	Laser Doppler Velocimetry

Abstract

The hydraulic performance and operational stability of centrifugal pumps are closely governed by the internal flow characteristics of the impeller. Under off-design conditions, conventional impellers tend to experience flow separation, vortex-induced energy loss, and intensified pressure pulsations, which impair performance and induce vibration. To improve these limitations, this study introduces a tandem-blade impeller configuration, where the front and rear blades cooperate to redistribute aerodynamic loading and enhance energy transfer.

Performance tests and high-fidelity numerical simulations analyses were conducted on conventional and tandem impellers with varying blade numbers to evaluate hydraulic performance, vortex evolution and unsteady excitation forces. The results show that the tandem impeller realizes a notable head increase of approximately 1-2m, particularly evident at moderate and high flow rates. This improvement arises from more progressive pressure loading along the blade passages and a more uniform exit velocity field. Furthermore, the tandem impeller achieves two-stage pressure rise, which weakens abrupt loading zones and alleviates mid-span flow separation typically observed in conventional impellers.

Vortex identification based on the Ω -criterion reveals that the tandem impeller induces stronger suction surface separation vortex within the gap. The development of these vortices is dominated by vortex stretching and rotational inertia effects, leading to enhanced vortex intensity and broader spatial distribution. These flow structures significantly modify the unsteady pressure field, as confirmed by FFT and continuous wavelet transform analysis, where the tandem impeller exhibits higher amplitude low-frequency components and more pronounced time-varying spectral energy, particularly under part-load conditions.

In terms of force response, the tandem impeller shows improved radial force and axial force balance near the design and overload conditions. However, under part-load conditions, the enhanced low-frequency excitation leads to larger radial and axial force fluctuations, which may negatively impact mechanical stability.

Overall, the tandem impeller demonstrates clear advantages in improving hydraulic head and outlet flow uniformity, but its gap-induced unsteady flow behavior requires further optimization, especially for stable operation under part-load conditions.

Kurzfassung

Die hydraulische Leistung und Betriebsstabilität von Kreiselpumpen wird maßgeblich durch die Strömungsverhältnisse im Laufrad beeinflusst. Abseits des Auslegungspunktes neigen konventionelle Laufräder zu Strömungsablösungen, wirbelinduzierten Energieverlusten sowie verstärkten Druckpulsationen, was zu einem Leistungsabfall sowie Schwingungsanregungen führt. Zur Verbesserung dieser Effekte wird in dieser Studie ein Tandem-Laufrad mit einem vorderen und einem hinteren Schaufelsatz eingeführt, bei dem beide Schaufelreihen kooperativ wirken, um die hydraulische Belastungsverteilung zu optimieren und den Energieübertrag zu verbessern.

Experimentelle Leistungsprüfungen sowie hochauflösende CFD-Simulationen wurden an konventionellen und Tandem-Laufrädern mit variierender Schaufelzahl durchgeführt, um die hydraulische Leistung, die Wirbelentwicklung und die instationären Erregerkräfte zu analysieren. Die Ergebnisse zeigen, dass das Tandem-Laufrad eine deutliche Erhöhung der Förderhöhe um etwa 1-2m erzielt, insbesondere bei mittleren und hohen Förderströmen. Diese Verbesserung beruht auf einer progressiveren Druckbelastung entlang der Schaufelkanäle sowie auf einem gleichmäßigeren Strömungsaustritt. Darüber hinaus bewirkt das Tandem-Laufrad eine zweistufige Drucksteigerung, wodurch abrupte Lastspitzen abgeschwächt und Strömungsablösungen im mittleren Spannbereich reduziert werden.

Die Wirbelerkennung mittels Ω -Kriterium zeigt, dass das Tandem-Laufrad im Spaltbereich stärkere Ablösewirbel auf der Saugseite ausbildet. Deren Entwicklung wird hauptsächlich durch Wirbelstreckung und Rotationsinertialeffekte bestimmt, was zu erhöhter Wirbelintensität und größerer räumlicher Ausdehnung führt. Diese Strukturen verändern das instationäre Druckfeld signifikant, was durch FFT- und Wavelet-Zeit-Frequenz-Analysen bestätigt wird, bei denen insbesondere unter Teillastbedingungen amplitudenstarke niederfrequente Komponenten und zeitlich variierende spektrale Energie beobachtet werden.

Hinsichtlich der Kraftverteilung weist das Tandem-Laufrad nahe dem Auslegungs- und Überlastbereich eine verbesserte Balance von radialer und axialer Kraft auf. Unter Teillastbedingungen führen jedoch die verstärkten niederfrequenten Anregungen zu größeren Schwankungen dieser Kräfte, was sich nachteilig auf die Betriebsstabilität auswirken kann.

Insgesamt zeigt das Tandem-Laufrad klare Vorteile hinsichtlich der Förderhöhe und Strömungsverteilung am Laufradaustritt. Das durch den Spalt verursachte instationäre Strömungsverhalten erfordert jedoch eine gezielte Optimierung, insbesondere zur Gewährleistung eines stabilen Betriebs unter Teillastbedingungen.

1 Introduction

Centrifugal pumps have been widely applied in fields such as water supply, chemical engineering, and energy, owing to their simple structure, stable operation, and broad applicability. With the growing demand for energy saving and emission reduction, improving the hydraulic performance and operational stability of centrifugal pumps has become a major research focus in the field of fluid machinery. Due to the presence of complex three-dimensional viscous flows and unsteady interactions among hydraulic components, issues such as flow instabilities and performance degradation within pumps are particularly pronounced, especially under off-design conditions. Recently, tandem impeller configurations, originating from axial-flow machinery, have attracted increasing attention as a novel design concept, given their outstanding potential in controlling flow separation and optimizing load distribution. This chapter first introduces the background and significance of the study, then reviews recent advances in performance enhancement of centrifugal pumps, applications of tandem impeller technology in hydraulic machinery, and the current understanding of internal flow characteristics in centrifugal pumps. Finally, the research objectives and overall structure of this work are presented.

1.1 Overview and Motivation

As one of the most common and widely used power transmission devices in modern society, the centrifugal pump plays an indispensable role across numerous industrial and civil sectors. Its functional scope ranges from basic water supply and irrigation to high-end applications such as energy and power generation, chemical processing, oil and gas transportation, and even aerospace propulsion systems [1-5]. Statistical data indicate that centrifugal pumps can account for more than 20% of total electrical energy consumption in industrial sectors. Therefore, their operational efficiency, stability, and reliability are of critical importance. With the global push toward carbon-emission reduction and energy-efficient manufacturing, the high-efficiency and stable operation of centrifugal pumps directly affects overall energy consumption, operating costs, and system safety.

Although centrifugal pump design methods are well established and widely applied, significant challenges remain in achieving high efficiency, broad operating range, and enhanced stability. In practical operation, the internal flow within a centrifugal pump is extremely complex, characterized by strong three-dimensionality, viscosity, and unsteadiness. In conventional centrifugal pumps, due to the large blade curvature, strong rotational effects, and asymmetric rotor-stator interactions (RSI), unstable flow structures such as secondary flows, flow separation, jet-wake patterns, and pressure pulsations frequently occur within the flow field [6-9]. These phenomena become even more pronounced under off-design flow conditions, leading to reduced energy transfer efficiency and the generation of pulsating excitation forces, which may in severe cases

induce vibration and noise issues, posing risks to operational safety [10-12]. Therefore, how to effectively improve the internal flow characteristics, enhance energy transfer efficiency, and ensure operational stability has become a key scientific question in current research.

To address the aforementioned issues, extensive research has been conducted by scholars worldwide, leading to the development of various approaches for performance enhancement. These include advanced impeller geometry optimization methods, multi-component hydraulic matching designs, active flow control techniques, as well as coating and material modification technologies [13-16]. However, the performance improvements achievable through conventional geometric optimization of a single-stage impeller have approached their theoretical limits. This situation highlights the urgent need to explore new structural configurations and energy transfer mechanisms.

Against this background, tandem impeller technology has gradually attracted increasing attention as a promising and innovative solution. In this configuration, the conventional single blade is divided within the flow passage into two independently arranged blade rows, a front and a rear segment, allowing the fluid to experience re-acceleration and re-attachment between the pressure surface (PS) of the front blade and the suction surface (SS) of the rear blade. This design enables independent optimization of the loading and incidence angle of each blade segment, thereby suppressing flow separation, improving boundary layer development, and enhancing the overall performance and operational stability. Existing studies have demonstrated that tandem impellers can significantly improve flow stability and boost pressure-rise performance in turbomachinery [17, 18].

In recent years, researchers have introduced this configuration into radial fluid machinery and found that it offers potential advantages in reducing hydraulic losses and improving pump performance parameters [19]. However, compared with its relatively mature application in turbomachinery, the use of tandem impellers in centrifugal pumps is still in the exploratory stage. Comprehensive studies on their hydraulic performance, internal flow mechanisms, and unsteady characteristics under multiple operating conditions remain limited. The complex interactions between the front and rear blades, as well as the sensitivity of their geometric design parameters, present significant research challenges. Therefore, it is necessary to conduct an in-depth and systematic investigation of the overall performance and internal flow characteristics of tandem-impeller centrifugal pumps through a combination of experimental measurements and numerical simulations. The objective of this study is to reveal the energy transfer mechanisms and unsteady flow behaviors of tandem impellers in centrifugal pumps, thereby providing theoretical guidance and technical references for their broader application in pump design.

1.2 Current Research Progress

The pursuit of enhanced centrifugal pump performance has been a sustained research focus for scholars worldwide over the past several decades. Broadly speaking, current

research progress can be categorized into several main directions. The first involves improving the hydraulic performance of centrifugal pumps through structural design and optimization, including geometric refinements and matching designs of key components such as impellers, guide vanes, and volutes. The second focuses on in-depth analyses of the complex three-dimensional unsteady flow mechanisms within pumps, aiming to uncover the intrinsic causes of performance deterioration and to propose active and passive flow control strategies. These studies have provided a theoretical foundation and design guidance for achieving higher efficiency and stability in centrifugal pumps. This section reviews the major approaches and recent advancements in centrifugal pump performance improvement and further discusses the application and development of tandem impeller technology in the field of hydraulic machinery.

1.2.1 Approaches for Centrifugal Pump Performance Enhancement

In the field of centrifugal pump performance enhancement, traditional approaches have largely relied on empirical correlations and manual design experience [20, 21], leaving considerable room for further improvement. With the rapid advancement of Computational Fluid Dynamics (CFD) technology, CFD-based performance prediction and hydraulic optimization design have gradually become the main-stream methods, elevating the optimization of pump hydraulic performance to a new level [22-24].

The impeller and diffuser are the key hydraulic components that determine the hydraulic performance of a centrifugal pump, as their geometric parameters directly influence the energy transfer efficiency and flow stability. Regarding the impeller, researchers have optimized geometric parameters such as blade number, wrap angle (WA), outlet width, and installation angle to achieve a balance between performance improvement and energy loss reduction [25-28]. At the diffuser level, the geometric matching of guide vanes and volutes also plays a crucial role in the overall performance of centrifugal pumps. Through coupled optimization design of the impeller and volute, it is possible to maintain high performance while reducing pulsating excitation forces [29, 30].

With the maturation of CFD technology, the Design of Experiments (DOE) method has been increasingly applied to the hydraulic performance optimization of centrifugal pumps. DOE enables optimal planning of design experiments under limited resources, allowing the identification of optimal design schemes with a relatively small number of simulations or tests. Commonly used DOE approaches include the Central Composite Design (CCD) method [31], full factorial design, orthogonal experimental design, and Latin Hypercube Design (LHD) [32]. Furthermore, researchers have combined DOE with intelligent optimization algorithms to achieve multi-objective optimization of centrifugal pumps [33, 34]. These approaches allow simultaneous consideration of multiple performance indicators and the identification of globally optimal designs under various operating conditions. Some studies have also introduced surrogate models to reduce computational costs and enable efficient parameter screening and iterative design. Such models offer

superior global search capability compared to conventional DOE-based approaches. Common surrogate models include response surface models, Kriging models, radial basis function neural networks, and multilayer artificial neural networks [35-37]. Widely used modern optimization algorithms include genetic algorithms, ant colony optimization, simulated annealing, tabu search, and particle swarm optimization [38-40].

Wang et al. [41] selected four design parameters of a pump-turbine impeller as optimization variables and used efficiencies under pump and turbine operating modes as the objective functions. A genetic algorithm was employed for optimization. The optimized model exhibited an efficiency improvement of 0.27 percentage points in pump mode. In turbine mode, the efficiency improvement reached 16.3 percentage points. Pei et al. [42] employed a two-layer feedforward neural network to model the relationship between impeller design parameters and pump efficiency, and used a particle swarm optimization algorithm to enhance the centrifugal pump efficiency under the design condition. Shi et al. [43] conducted a multidisciplinary optimization of an axial-flow pump, taking impeller mass and efficiency under the design condition as objectives, while constraining the head, maximum stress, and maximum deformation under part-load conditions. These design strategies, based on simulation and optimization algorithms, mark a paradigm shift in centrifugal pump design from experience-driven to data-driven and intelligently optimized methodologies.

The overall performance of the centrifugal pump depends not only on the design quality of individual components but also on the matching and synergistic interactions among key hydraulic elements such as the impeller, diffuser, and volute. In terms of impeller-diffuser matching, studies have shown that the velocity distribution and rotational angular momentum at the impeller outlet directly influence the diffuser fluid diffusion capacity and pressure recovery characteristics [44]. By optimizing the phase relationship between the diffuser inlet angle and the impeller outlet angle, flow deviation and vortex formation can be effectively mitigated. The matching design between the impeller and the volute is equally critical. An excessively small clearance at the volute tongue can cause strong RSI, leading to periodic pressure pulsations and radial unbalanced forces, whereas an overly large clearance reduces energy conversion efficiency. To balance these effects, researchers have proposed design strategies such as asymmetric volutes, double volute configurations, and adjustable tongue structures to maintain uniform flow and balanced loading under varying operating conditions [45, 46]. In addition, some researchers have employed three-dimensional fluid-structure interaction and transient CFD simulations to reveal the unsteady interference characteristics between the impeller and volute and their contribution to excitation forces, thereby providing valuable insights for structural reinforcement design [47, 48]. This systematic optimization philosophy not only enhances the overall performance of centrifugal pumps but also significantly improves their operational stability and reliability.

1.2.2 Internal Flow Characteristics and Unsteady Phenomena in Centrifugal Pumps

The flow pattern in the centrifugal pump is highly complex and exhibits pronounced three-dimensional and unsteady characteristics. Although the external performance parameters of the pump can be measured experimentally, these macroscopic indicators often obscure the intricate transient flow details within the flow pattern. To better understand the origins of performance variation and flow instability mechanisms, researchers have gradually shifted their focus toward the internal fluid motion characteristics and unsteady phenomena in pumps.

Flow instabilities in centrifugal pumps are diverse and complex, including leakage flow, backflow, flow separation, and stall phenomena [49-51]. In recent years, numerous experimental and numerical studies have been conducted to investigate the internal flow mechanisms and instability behaviors of centrifugal pumps. In early research, scholars mainly relied on experimental measurements and flow visualization techniques to reveal flow field characteristics inside pumps. Using high-speed photography and Particle Image Velocimetry (PIV) techniques, periodic jet-wake structures and non-uniform velocity distributions at the impeller outlet have been widely observed [3, 8]. These structures induce RSI at the inlet of the diffuser or the volute, generating periodic pressure pulsations whose dominant frequency typically corresponds to the blade passing frequency (BPF) [52, 53]. Pedersen et al. [54] employed PIV to identify stall phenomena within centrifugal pump impeller passages. Experimental results revealed that under part-load conditions, alternating stall occurs in adjacent passages. At the inlet of the stalled passage, the flow is highly turbulent with severe blockage, while strong backflow appears along the PS near the impeller outlet. Additionally, the number of vortex structures in the stalled passages increases markedly. Sinha et al. [55] conducted PIV experiments on a vaned centrifugal pump and captured the generation, development, and breakdown processes of unstable vortex structures in the pump. Pressure pulsation data indicated that the propagation velocity of these vortices was significantly lower than the impeller rotational speed, being only about 6.2% of it. Xue et al. [56] investigated the instability behavior of pump operating modes in pumped-storage units and pointed out that flow separation and rotating stall under part-load conditions are major causes of head fluctuation and flow reversal. These experimental findings were among the first to reveal the presence of complex unsteady structures in centrifugal pumps, providing valuable validation data for subsequent numerical studies.

With the rapid advancement of CFD technology, researchers have increasingly combined numerical simulations with experimental methods to investigate the unsteady flow mechanisms in centrifugal pumps. Zhang et al. [57] employed a hybrid Reynolds-Averaged Navier-Stokes/Large Eddy Simulation (RANS/LES) method based on the von

Kármán length scale and modified eddy viscosity to study unsteady flow structures and pressure pulsation characteristics in a centrifugal pump. They found that the interaction between different numbers of impeller and diffuser blades generates an impeller-diffuser interaction frequency within the diffuser and volute, thereby exciting higher-order harmonics of the impeller BPF. Ye et al. [58] developed a computational strategy that accounts for transition effects to capture the transient characteristics of stall vortices in a marine centrifugal pump. The transition model predicted a greater number of stall vortex cells with lower occurrence frequency. The study showed that as the flow rate decreases, the number of separation vortices near the leading edge (LE) increases, resulting in a sharp rise in surface friction losses, identified as the main cause of the hump-shaped feature in the Q - H curve. Zhang et al. [59] summarized recent research progress on rotating stall in pump-turbines and found that the propagation frequency of stall vortices is typically 0.5-0.7 times the shaft frequency. The formation of stall vortex clusters causes severe flow blockage and induces stronger pressure pulsations in the impeller or guide vanes, which is a primary cause of flow instability in pump-turbines. In addition, Backman [60] conducted unsteady numerical simulations on pump-turbines, focusing on the effect of RSI on pressure pulsation characteristics. The results showed that the amplitude of pressure pulsations reaches its maximum near the runner at the BPF. These studies collectively demonstrate that CFD technology has become a crucial tool for understanding unsteady flow phenomena in pumps, enabling accurate capture of internal vortex structure evolution under various operating conditions.

Building upon the understanding of unsteady phenomena within pumps, researchers have further sought to analyze the formation and evolution of flow structures from a physical-mechanism perspective. Wu et al. [11] reviewed the sources of flow-induced excitation in centrifugal pumps and emphasized the central role of RSI in the formation of unsteady pressure fields. Luo et al. [61] investigated the unsteady characteristics of cavitation-induced vortices near the volute tongue. During cavitation at the tongue, the compression of the fluid by vapor cavities promotes the formation of separation vortices, while the re-entrant jet induced by these vortices leads to cavity shedding. Analysis of the vorticity transport equation revealed that the baroclinic torque term dominates vorticity variation at the vapor-liquid interface. Wang et al. [62] proposed a rigid-body vorticity transport equation based on vorticity decomposition theory. Compared with the conventional vorticity transport equation, this new theoretical framework can distinguish between local rigid-body rotation and shear motion, providing clearer insight into the intuitive evolution of vortex structures. Liao et al. [63] analyzed the formation process of unsteady flow structures in a centrifugal pump impeller using the Proper Orthogonal Decomposition (POD) method. The POD results revealed that major unsteady structures, such as flow separation and jet-wake patterns, significantly influence the wake distribution at the impeller outlet. Lu et al. [64] employed the Dynamic Mode Decomposition (DMD)

method to study vorticity distributions in a centrifugal pump under part-load conditions. As the flow rate decreased, vortices were observed to diffuse continuously from the SS toward the PS of the blade, with increasing intensity and gradual expansion, eventually leading to flow passage blockage. These studies have broadened the theoretical foundation of unsteady flow research, marking a transition from mere phenomenological descriptions toward mechanistic modeling of flow instabilities in centrifugal pumps. Building upon the exploration of unsteady flow mechanisms, recent research on centrifugal pump optimization has increasingly focused on how to suppress unstable flow and improve internal flow distribution through flow control technologies. Depending on the mode of energy input and control strategy, flow control methods can generally be classified into passive flow control and active flow control. Passive flow control requires no external energy input and primarily regulates flow characteristics through geometric modifications, offering advantages in cost-effectiveness and simplicity. Ji et al. [65] employed four different impeller rim structures to suppress or delay the onset of stall, thereby mitigating the hump characteristics on the performance curve of mixed-flow pumps. The results showed that flanged and double-ribbed tip structures can effectively broaden the hump margin of the pump performance curve. Li et al. [66] proposed two optimization strategies for the high-pressure edge of impellers, demonstrating that increasing the outlet angle and radius near the shroud reduces stall vortices and decreases pressure fluctuations in the vaned diffuser. Saha et al. [67, 68] found that shallow grooves on the wall surface can effectively suppress rotating stall and improve the hump behavior in mixed-flow pump performance curves. Similarly, Kurokawa et al. [69] designed a J-shaped groove structure on the wall surface to control the angular momentum of the main flow, thereby weakening unstable flow modes. Ding et al. [70] reduced pressure pulsations by trimming the SS of the impeller blade trailing edge (TE), with results showing that linear trimming was more effective in reducing pulsations than curved trimming. Wu et al. [71, 72] improved the design of the blade PS to reduce pressure fluctuations in low specific speed centrifugal pumps. Their results indicated that actively unloading the PS of the blade led to a more uniform energy distribution at the impeller outlet, which is the fundamental reason for the reduced pressure pulsation amplitude within both the impeller and the volute. Active flow control, by contrast, requires additional energy input to perturb and regulate the internal flow patterns. Zhang et al. [73] introduced high-pressure injection at the impeller inlet and found that this method effectively weakened inlet backflow under part-load conditions and improved unstable flow structures in the centrifugal pump. Bloxham et al. [74] designed a flow-control scheme employing vortex generators, which successfully reduced wake losses by 69% and decreased area-averaged total pressure losses by 28%. Although active suppression methods are more complex and costly, limiting their widespread application in fluid machinery, these studies demonstrate that research on unsteady flow in centrifugal pumps

has evolved from passive identification to active control. This transition provides a new technical pathway for achieving high-performance and high-stability centrifugal pumps.

1.2.3 Tandem Blade Technology in Pumps and Turbomachinery

Based on previous research reviews, passive suppression methods have been shown to be a more economical and effective approach for mitigating unstable flow and improving the performance of fluid machinery. Consequently, an increasing number of studies have focused on structural optimization of bladed machines. Among various strategies, optimizing blade geometry and configuration is one of the most direct and effective approaches, and the tandem blade design represents a particularly promising method. In a tandem blade configuration, a gap is introduced in the middle of a conventional twisted blade, dividing it into two segments, a front blade and a rear blade. The presence of this intermediate gap allows the tandem impeller to generate a new thin boundary layer on the rear blade, thereby suppressing flow separation and maintaining a favorable pressure gradient in the flow passage.

Researchers have introduced tandem blade technology into various types of turbomachinery. This design concept originated from compressor cascade optimization in the aerospace field, where multistage loading and boundary-layer reconstruction were used to achieve higher aerodynamic loading capacity and lower loss coefficients [17, 18]. In axial-flow and mixed-flow fluid machinery, studies on tandem blades have made systematic progress. Roberts and Kacker [18] were among the first to numerically analyze the performance of tandem compressor cascades. When the ratio of blade spacing to chord length is properly designed, the flow from the TE of the front blade can reattach to the LE of the rear blade, forming a new boundary layer structure that significantly enhances pressure rise capability and reduces stall risk. McGlumphy et al. [75] investigated tandem airfoils in a subsonic, shock-free linear cascade. Both CFD and experimental results demonstrated that well-designed tandem airfoils outperform comparable single airfoils near the design condition. Mao et al. [76] optimized design strategies for high-loaded tandem blade cascades at the midspan and end wall regions. The midspan-optimized tandem blades effectively suppressed the interaction between low-energy fluid and the SS of the rear blade under the design condition, while the end wall-optimized configuration helped restrain secondary flow development and delay the onset of corner stall. Böhle et al. [77] analyzed the three-dimensional flow field in the sidewall region of two high-turning tandem cascades. Oil-film visualization and streamline patterns revealed detailed flow characteristics near the sidewall region of the tandem blades. Yuan et al. [78] applied tandem blade technology to a transonic ultra-high-loaded axial compressor. They developed an original design map for tandem blades and established selection criteria for tandem configuration parameters. Under the optimal parameter configuration, the tandem blades achieved a 0.83% increase in peak efficiency, a 2.16% improvement in stall margin, and a 0.30% increase in choking mass flow rate.

In comparison, research on tandem blade technology in centrifugal pumps remains in the exploratory stage. Due to the pronounced three-dimensional nature of centrifugal flow and the complexity of radial diffusion effects, the application of tandem blade designs in pumps faces challenges such as geometric parameter matching and mutual interference between blade segments. Shi et al. [79] conducted numerical and experimental studies on tandem impellers, demonstrating that tandem-bladed centrifugal impellers exhibit significant advantages in improving outlet flow uniformity, thereby reducing pressure fluctuations and hydraulic losses. By adjusting the gap position and offset between the front and rear blades, the study achieved a maximum efficiency improvement for the tandem impeller, albeit with a slight reduction in head. Prithvir et al. [80] found that a small shift of the rear blade toward the PS effectively enhanced pump efficiency, independent of the gap position. Li et al. [19] performed numerical simulations on tandem impellers and found that shifting the rear blade toward the SS led to reductions in both head and efficiency. However, under the design condition, the tandem configuration effectively attenuated pressure pulsations induced by RSI. Fensterseifer et al. [81] proposed a systematic optimization methodology for designing tandem impellers in a centrifugal pump. Both experimental and numerical results confirmed that the tandem impeller designed using this approach could significantly increase the pump head without compromising efficiency. Nicola et al. [82] compared the streamline distributions of conventional and tandem impellers using oil-film visualization experiments. The results indicated that tandem impellers can effectively suppress flow separation near blades SS.

1.3 Research Objectives and Report Structure

As revealed by the above review, research on improving centrifugal pump performance has primarily focused on impeller geometry optimization, flow passage matching design, and active flow control. These methods have enhanced pump efficiency and operational stability to some extent. However, conventional single-stage impellers still tend to exhibit unstable flow structures, such as flow separation, jet-wake patterns, and pressure pulsations, under overload and large flow-deflection conditions, thereby constraining further performance improvements. In contrast, tandem impeller technology has been demonstrated in turbomachinery to effectively suppress flow separation, enhance flow turning capability, and improve pressure rise performance. Nevertheless, its application in centrifugal pumps remains limited, with a particular lack of systematic studies on the mechanisms underlying its hydraulic performance enhancement and the unsteady flow characteristics in tandem impeller configurations.

Based on the above considerations, this study focuses on a tandem-impeller centrifugal pump, employing a combination of numerical simulations and experimental measurements to systematically investigate the effects of tandem impeller structures on pump performance and flow characteristics. The main objectives of this research are as follows, to propose and develop a systematic hydraulic design and optimization

methodology applicable to tandem impellers in centrifugal pumps. To reveal the working principles and hydraulic characteristics of tandem impellers through experimental and numerical analyses, thereby elucidating the mechanisms underlying improvements in energy transfer efficiency and flow structures. To evaluate the influence of tandem impellers on the external performance parameters of centrifugal pumps and identify their performance advantages and applicable operating ranges under various conditions. To comparatively analyze the three-dimensional unsteady flow characteristics and flow-induced excitation behavior in tandem impellers, clarifying the physical mechanisms responsible for enhanced flow stability. Overall, this study aims to establish a systematic theoretical framework and provide practical design references for the broader application and engineering implementation of tandem impeller technology in centrifugal pumps.

This dissertation is organized into nine chapters, structured as follows. Chapter 1 elaborates on the research background and significance, reviews the main research directions related to centrifugal pump performance enhancement, and clarifies the research objectives and overall structure of the work. Chapter 2 introduces theoretical foundations and research methods, including the basic operating principles and performance parameters of the centrifugal pump, the operating mechanism of the tandem impeller, and the unsteady flow phenomena in impellers, providing the theoretical basis for subsequent studies. Chapter 3 describes the hydraulic design process of both conventional and tandem impeller centrifugal pumps, with emphasis on the geometric configuration of the tandem impeller and the optimization method for the rear blade design. Chapter 4 presents the experimental system and testing methodology, covering the design of the experimental loop, the structure of the test pump, experimental procedures, and uncertainty analysis methods. Chapter 5 details the numerical simulations and validation processes, including the selection of turbulence models, grid sensitivity analysis, boundary condition settings, and verification of numerical results against experimental data. Chapter 6 compares and analyzes the external performance characteristics of different impellers, investigates the influence of design parameters on pump performance, examines the performance of tandem impellers under various structural configurations, and evaluates the adaptability and performance enhancement of tandem impellers with different blade numbers. Chapter 7 compares the internal flow characteristics of conventional and tandem-impeller centrifugal pumps, analyzing vortex distribution and evolution, pressure pulsation behavior, and excitation force variations to elucidate the mechanisms by which tandem impellers enhance flow stability. Chapter 8 provides a comprehensive discussion of the research findings, summarizing the mechanisms of performance improvement and flow control achieved by tandem impellers, and assessing their engineering applicability and feasibility. Finally, Chapter 9 concludes the dissertation by summarizing the main research outcomes, presenting key conclusions, and outlining prospects for future research directions.

2 Theoretical Basis and Research Methods

This chapter presents theoretical foundations and research methods related to the centrifugal pump and the tandem impeller. First, the symbols and terminology used throughout this study are defined and standardized. Subsequently, the fundamental operating principles and performance parameters of centrifugal pumps are introduced, and their operating characteristics are analyzed based on external performance curves. The following sections focus on the working mechanism of the tandem impeller and typical unsteady flow phenomena in the impeller, including secondary flow, vortex structure evolution, and pressure pulsation. The theoretical analyses presented in this chapter provide the foundation for the numerical simulations and experimental investigations discussed in the subsequent chapters.

2.1 Nomenclature and Notation

This section first defines appropriate terminology and symbols for various parameters and their characteristics in the centrifugal pump. Beyond describing the geometric parameters of the centrifugal pump, it also briefly discusses the analytical coordinate system and key diagrams used in this study.

The subject of this study is a typical centrifugal pump, whose primary energy conversion components are the impeller and the volute. Fig. 2.1(a) depicts the primary dimensions defined in the impeller for this study, including the shaft diameter D_i at the impeller mounting point, the hub diameter D_h , the inlet diameter D_j , the outlet diameter D_2 , and the outlet width b_2 . Fig. 2.1(b) depicts the primary dimensions defined in the volute, including the volute base circle diameter D_3 , the volute outlet diameter D_4 , the volute tongue placement angle ψ_0 , and the volute inlet width b_3 .

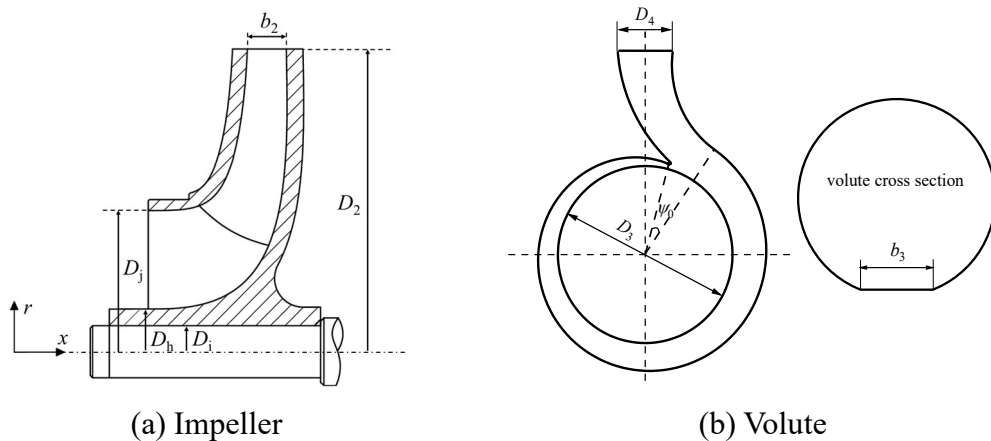


Fig. 2.1. Main dimensions.

This study primarily focuses on modifications to the impeller structure. Therefore, the following analysis is mainly concentrated in the impeller domain. Fig. 2.2 illustrates the

nomenclature of the principal geometric features in the impeller. The geometric descriptions include the blade LE, blade TE, shroud, hub, flow channel in the impeller, span, PS, and SS.

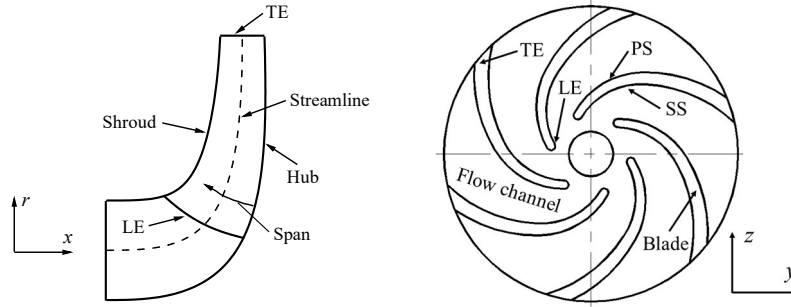


Fig. 2.2. Nomenclature.

In this study, all absolute reference coordinate systems adopt a Cartesian right-handed system (x, y, z) , with the x -axis defined as the rotational axis of the impeller. The meridional plane profile is obtained from the circumferential projection of the LE, TE, shroud, and hub. It is defined in the x - r plane as follows, where the radius corresponding to each axial position is calculated by:

$$r = \sqrt{z^2 + y^2} \quad (2.1)$$

Any number of streamlines can be generated between the shroud and hub of the impeller. When these streamlines are revolved around the x -axis, they form curved surfaces that represent different spanwise sections in the impeller. In this study, the definition of the rotational surface and the corresponding coordinate system is illustrated in Fig. 2.3. By slicing the impeller along specific positions on these curved surfaces, a series of blade-to-blade sections can be obtained.

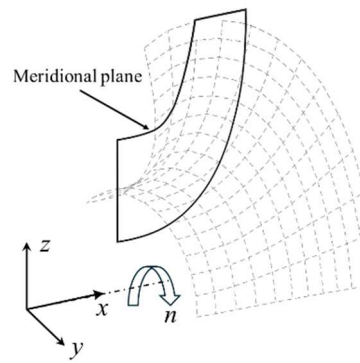


Fig. 2.3. Coordinate system.

2.2 Operating Principle and Performance Indicators of Centrifugal Pumps

This section introduces the operating mechanism and performance characteristics of the

centrifugal pump, laying the theoretical foundation for subsequent analysis. It first explains the basic working principle of the centrifugal pump and the definitions of key performance parameters, followed by an explanation of the relationship between head H , flow rate Q , efficiency η , and power P in their external characteristic curves.

2.2.1 Principle and Performance Parameters of the Centrifugal Pump

The centrifugal pump serves as the core power equipment for fluid transportation, and its operating principle is based on the conversion between mechanical energy and fluid kinetic energy. When the impeller rotates, the low pressure at the impeller center causes the fluid to be drawn axially into the impeller inlet. Guided by the blades, the fluid then moves radially toward the impeller outlet. In the impeller, the combined effects of centrifugal and Coriolis forces enable the fluid to gain both velocity energy and pressure energy. As the fluid exits the impeller and enters the volute or diffuser, part of its kinetic energy is further converted into pressure energy, thereby achieving energy transfer and pressurization. This energy conversion process can be quantitatively described by the Euler head equation, which is derived from the angular momentum theorem in fluid mechanics:

$$g \cdot H_{th} = u_2 \cdot c_{u,2} - u_1 \cdot c_{u,1} \quad (2.2)$$

where u denotes the circumferential velocity of the impeller, c_u represents the circumferential component of the absolute fluid velocity, and g is the gravitational acceleration. The term $u_2 c_{u,2} - u_1 c_{u,1}$ represents the change in angular momentum per unit weight of the fluid, which corresponds to the energy imparted to the fluid by the impeller. The impeller is the core component of the pump, and it is therefore crucial to analyze the motion of the liquid within it in detail. To illustrate the relative motion between the fluid and the impeller more intuitively, the velocity triangle is commonly used for analysis, as shown in Fig.2.4. At the impeller inlet and outlet, the fluid velocity can be decomposed into three components, the absolute velocity c , the circumferential velocity of the impeller u , and the relative velocity w , which satisfy the relationship $c = u + w$. In the velocity triangle, the circumferential components of velocity at the inlet and outlet, $c_{u,1}$ and $c_{u,2}$, determine the theoretical head H_{th} that the impeller can provide. The meridional velocity c_m represents the component of the fluid velocity along the meridional plane toward the impeller outlet and is related to the volumetric flow rate Q through the impeller. Properly designed blade inlet and outlet angles, β_1 and β_2 , ensure smooth fluid entry and exit from the impeller, minimizing flow separation and impact losses, thereby improving the efficiency and operational stability of the pump.

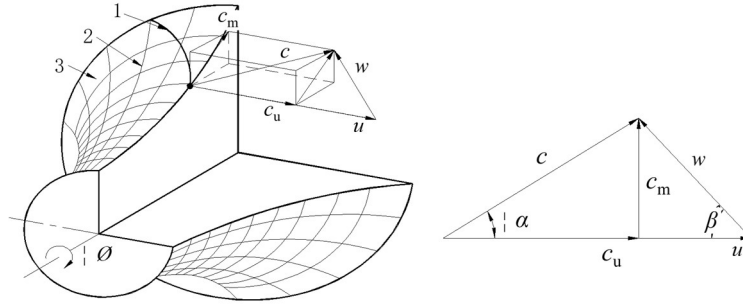


Fig. 2.4. Velocity triangle.

The H of pump represents the energy increase per unit weight of liquid from the pump inlet to the outlet, that is the effective energy gained by one newton of liquid as it passes through the pump. Its unit is $\text{N}\cdot\text{m}/\text{N}=\text{m}$. The actual H of a pump differs from the H_{th} due to hydraulic losses in the flow passages and mechanical friction losses in the shaft system. According to its definition, the H can be expressed as:

$$H = E_d - E_s \quad (2.3)$$

where E_d is the energy per unit weight of the liquid at the pump outlet, and E_s is the energy per unit weight of the liquid at the pump inlet. In hydraulics, the energy per unit weight of liquid is referred to as the H , which typically consists of three components, the pressure head $p/\rho g$, the velocity head $v^2/2g$, and the elevation head z . Therefore, according to Bernoulli's equation, the H can be expressed as:

$$H = \frac{p_2 - p_1}{\rho g} + \frac{v_2^2 - v_1^2}{2g} + (z_2 - z_1) \quad (2.4)$$

where p_1 and p_2 represent the static pressures of the liquid at the pump inlet and outlet, respectively; v_2 and v_1 denote the corresponding fluid velocities; and z_2 and z_1 are the distances from the inlet and outlet to an arbitrary reference datum. The H characterizes the intrinsic performance of the pump itself, it depends solely on the energy change of the liquid between the inlet and outlet flanges and is independent of the external pumping system.

The flow rate Q of pumps refers to the volume or mass of liquid delivered by the pump per unit time. The volumetric flow rate is denoted by Q , with common units such as m^3/s , m^3/h , or L/s . The mass flow rate is usually represented by Q_m , with units such as t/h or kg/s . The two are related by $Q_m = \rho Q$, where ρ is the fluid density.

The P of pumps typically refers to the input power. That is the power transmitted from the prime mover to the pump shaft, commonly referred to as the shaft power, denoted by P . It is generally calculated from the torque M applied to the pump shaft and the angular velocity ω , and is defined as:

$$P = M \cdot \omega = 2\pi n M \quad (2.5)$$

The effective power of the pump, also known as the output power, is denoted by P_e . It

represents the useful energy gained per unit time by the liquid as it passes through the pump. Since the H corresponds to the effective energy per unit weight of liquid delivered by the pump, the P_e can be expressed in terms of H as follows:

$$P_e = \rho g Q H \quad (2.6)$$

The difference between P and effective P_e represents the power loss in the pump, commonly measured by the pump η . Pump η is the ratio of effective P_e to P , expressed as:

$$\eta = \frac{P_e}{P} = \frac{\rho g Q H}{M \omega} \quad (2.7)$$

2.2.2 External Performance Curves of the Centrifugal Pump

There exists an intrinsic relationship among the motion parameters in the pump. According to the velocity triangles in the impeller, for a given pump operating at a specific rotational speed n , the circumferential velocity component c_u decreases as the meridional velocity c_m increases. Consequently, a corresponding correlation also exists among the external representations of these motion parameters. That is the pump performance parameters. When these relationships are expressed in graphical form, they are referred to as pump performance curves or characteristic curves.

The external characteristic curves of the centrifugal pump describe the macroscopic relationships among its main operating parameters, such as H , Q , η , and P under different operating conditions. These curves are typically obtained through experiments or numerical simulations by varying the flow rate and recording the corresponding changes in other parameters. They serve as essential tools for evaluating the overall performance, operational stability, and efficiency characteristics of the centrifugal pump. A typical example of pump performance curves is illustrated in Fig. 2.5.

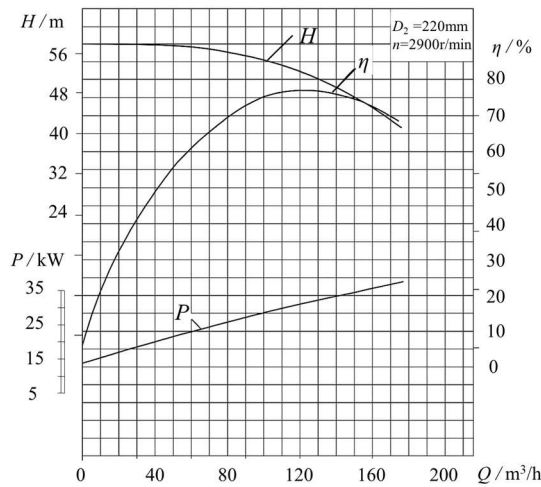


Fig. 2.5. Pump performance curves.

In a typical pump performance curve, the flow rate-head (Q - H) curve generally shows that the H decreases gradually as the Q increases, primarily due to the rise in hydraulic losses. The shape of the Q - H curve can vary and is generally classified into five characteristic forms, as illustrated in Fig. 2.6. In a smoothly decreasing curve, the H reaches its maximum when the $Q=0$. As Q increases, the H gradually decreases, and each H value corresponds to a unique Q , this represents a stable Q - H characteristic. In a steeply decreasing curve, the H drops rapidly as Q increases. In a flat curve, large variations in Q result in only small changes in H . In an unstable curve, the H equals H_0 at $Q=0$. As Q increases, the H first rises to a maximum value and then decreases with further increase in Q . In the region where $H>H_0$, each H corresponds to two different Q , indicating poor operational stability. The hump-type characteristic curve is often observed in axial-flow or mixed-flow pumps under part-load conditions, where the pump exhibits extremely poor operational stability.

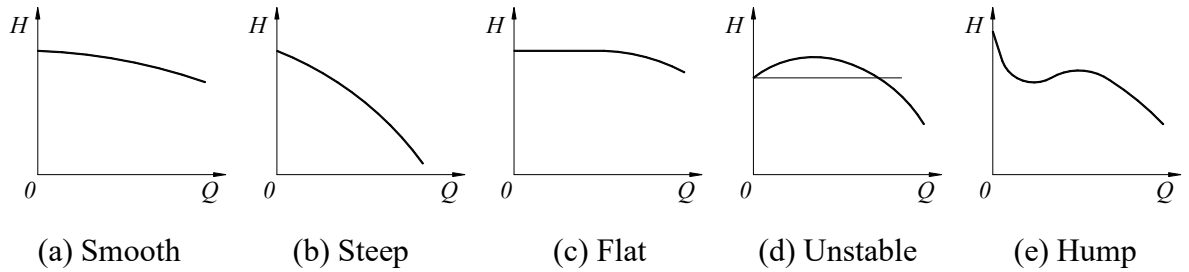


Fig. 2.6. Forms of the Q - H curve.

The flow rate-efficiency (Q - η) curve generally exhibits a single-peaked shape, where the peak corresponds to the best efficiency point, the operating condition under which the pump achieves optimal hydraulic performance and minimal energy loss. For centrifugal pumps with low specific speed, the Q - η curve is typically sharp, with a narrow peak region, indicating higher sensitivity to variations in Q . In contrast, mixed-flow and axial-flow pumps with higher specific speed display relatively flatter Q - η curves, suggesting better adaptability to flow fluctuations within a certain range.

For most centrifugal pumps, the flow rate-power (Q - P) curve increases monotonically with rising Q , since higher Q require greater energy input to overcome frictional and impact losses within the flow passages. Consequently, the Q - P curve of such pumps is usually a smooth, upward-sloping curve, reaching maximum P under the overload condition. However, for mixed-flow or axial-flow pumps with high specific speed, the Q - P curve may exhibit a hump-shaped or slightly declining trend after an initial rise. This behavior is closely related to changes in the internal flow structure under overload conditions, flow separation or backflow may occur near the impeller outlet, reducing the efficiency of energy transfer and causing a minor drop in the Q - P curve. Additionally, when the pump operates away from the design condition, local vortices or non-uniform flow patterns may form in some blade passages, further influencing the power

characteristics.

Differences in pump performance curves represent the external manifestation of varying fluid motion states in the pump, which are determined by the geometry of the pump flow-passing components. Consequently, the shape of the performance curve is significantly influenced by factors such as impeller geometry, blade count, outlet width, and flow channel configuration.

2.3 Operating Principle of the Tandem Impeller

The tandem impeller is a configuration consisting of two impellers arranged sequentially along the same axis, also referred to as a multi-blade-row impeller. The structure of a tandem impeller is illustrated in Fig. 2.7. Unlike a conventional single-stage impeller, the fluid in tandem configuration passes successively through the front and rear blade, achieving a staged energy transfer process that enables higher energy output at the impeller outlet. As shown in Fig. 2.7, the working principle of the tandem impeller can be described as follows: the fluid enters the front blade with an absolute velocity c_1 and a relative velocity w_1 , at an inlet flow angle β_1 . After passing through the front blades, the circumferential velocity component c_u of the fluid increases, and the flow angle changes accordingly. The fluid then passes through the clearance between the TE of the front blades and the SS of the rear blades. Due to the non-uniformity and residual vortices present in the wake of the front blades, the incoming flow at the rear blade inlet often exhibits a certain incidence angle deviation. Finally, the rear blades rectify and re-accelerate the flow, further increasing c_u and achieving secondary energy accumulation. Through this staged energy transfer mechanism, the tandem impeller enhances the pump H and improves the hydraulic loading capacity.

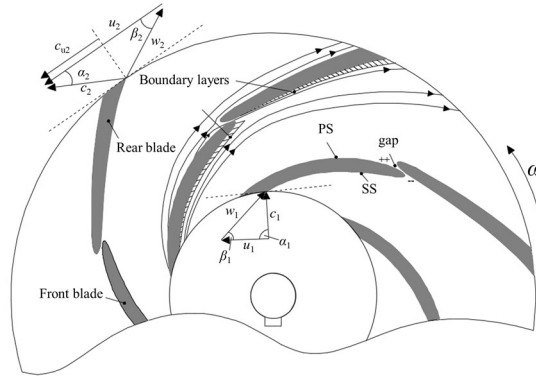


Fig. 2.7. Principle of the tandem impeller.

According to Euler head equation, the H_{th} of the tandem impeller is the sum of the H of the front and rear blades:

$$H_{th} = \frac{1}{g} (u_{2f} \cdot c_{u,2f} - u_{1f} \cdot c_{u,1f}) + \frac{1}{g} (u_{2r} \cdot c_{u,2r} - u_{1r} \cdot c_{u,1r}) \quad (2.8)$$

where the first term represents the energy contribution provided by the front blade, while

the second term corresponds to the energy gain from the rear blade. Under conditions of large flow deflection angles or overload, the tandem configuration can effectively delay or suppress flow separation in the impeller passages. As the fluid flows from the PS of the front blade through the gap toward the SS of the rear blade, a new boundary layer is formed and the flow is re-accelerated. This results in smoother flow deflection and more effective pressure recovery. Moreover, the staged energy transfer mechanism enables a more rational distribution of hydraulic loading between the front and rear blades, thereby reducing incidence losses and local stress concentrations.

2.4 Unstable Flow Phenomena in the Impeller

The flow pattern in the centrifugal pump impeller exhibits pronounced three-dimensional and unsteady characteristics. These instabilities primarily originate from the blade geometry, rotational motion, Coriolis effects, viscous shear forces, and flow interactions between the impeller and the volute. When the pump operates under off-design conditions, the flow pattern in the impeller passages becomes highly non-uniform, easily inducing complex phenomena such as secondary flows, vortex structures, and pressure pulsations. These flow instabilities not only reduce hydraulic efficiency but also affect the dynamic stability of the entire pump system. The following sections analyze the main types of unsteady flow in the impeller, including the evolution and transport characteristics of vortex structures, as well as the pressure pulsations and excitation forces induced by flow interactions.

2.4.1 Unstable Flow Structures

In the centrifugal pump impeller, the flow is inherently three-dimensional and non-uniform due to the curvature of the blade passages and the rotational motion of the fluid. As fluid particles are subjected to centrifugal forces, Coriolis forces, forces induced by the curvature of the meridional plane, and viscous shear stresses, a variety of complex unsteady flow structures are formed in the impeller passages. The interactions among these flow structures not only affect the efficiency of energy transfer but also play a crucial role in determining the operational stability and flow-induced vibration characteristics of the centrifugal pump.

As the fluid passes through the impeller passages, boundary layers are formed near various physical surfaces, such as the SS and PS of the blades, as well as the shroud and hub surfaces. Due to the strong curvature and rotational effects associated with the SS and PS of the blades, the velocity distribution of fluid particles exhibits distinct three-dimensional characteristics. The combined influence of the axial velocity gradient along the main flow direction and the centrifugal force in the radial direction induces transverse flow within the boundary layer. This transverse motion alters the momentum distribution and, in severe cases, may lead to flow separation in specific regions. The three-dimensional boundary layer in the impeller exhibits several typical features. Near the SS

of blades or shroud surfaces, the boundary layer thickness gradually increases. Under off-design conditions, particularly under part-load conditions, the viscous effects of the fluid become more pronounced, which may lead to the formation of localized separation vortices within the boundary layer. Consequently, the development of the three-dimensional boundary layer directly influences the efficiency of energy transfer and determines the uniformity of the downstream flow pattern.

One of the most prominent features of impeller internal flow is the presence of secondary flow, which refers to the cross-sectional flow superimposed on the main flow direction in the passage. Under the combined influence of blade curvature and unbalanced centrifugal forces, the fluid experiences deflection in both the radial and circumferential directions. Due to the non-uniform static pressure distribution along the blade height and curvature directions, a transverse pressure gradient is generated, driving the fluid from the PS toward the SS and forming a characteristic secondary flow pattern. For high-energy fluid in the main flow region, the strong centrifugal force dominates the motion, allowing the fluid to overcome the transverse pressure gradient and move toward the SS. In contrast, the low-energy fluid in the boundary layer possesses weaker centrifugal force and cannot resist the transverse pressure gradient, thus being pushed toward the PS. As a result, a pair of counter-rotating vortices symmetrically distributed across the passage cross-section, known as secondary flow vortices are formed. The typical flow path involves the fluid moving from the SS across the passage center toward the PS, and then returning to the SS along the shroud and hub surfaces. Secondary flow leads to a highly non-uniform velocity distribution across the impeller passage, causing significant kinetic energy losses. Under the influence of strong secondary flow, the non-uniformity of the impeller outlet flow field increases regions near the SS exhibit higher velocities, forming jet zones, while regions near the PS show lower velocities, forming wake zones. A typical example of impeller outlet secondary flow is illustrated in Fig. 2.8, where this flow imbalance at the volute inlet can further induce stronger pressure pulsations.

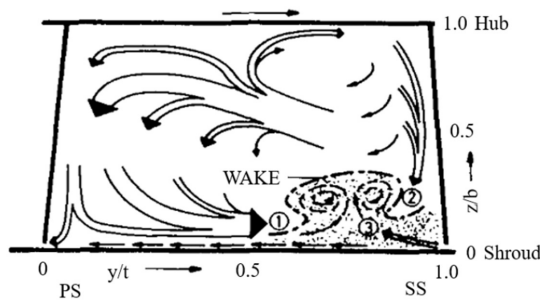


Fig. 2.8. Secondary flow at the impeller outlet of a centrifugal impeller [83].

Flow separation and backflow are also typical unsteady flow phenomena occurring in centrifugal pump impellers, particularly under off-design conditions. Flow separation arises when an adverse pressure gradient develops along the blade surface, causing the

fluid within the boundary layer to lose sufficient kinetic energy required to overcome the pressure rise, leading to a reversal in the velocity gradient, as illustrated in Fig. 2.9. In the centrifugal pump impeller, as the fluid flows along the SS of the blade, an excessive local diffusion rate or an increased blade incidence angle can cause the boundary layer to thicken progressively, eventually detaching from the blade surface and forming a separation vortex region. Flow separation commonly occurs near the SS at the blade inlet and in the diffusion section at the blade outlet. The separated region is characterized by low velocity, strong vorticity, and significant flow asymmetry. The large-scale vortex motion in the separation region dissipates mechanical energy through intense turbulence, converting it into thermal energy and resulting in substantial energy loss, which directly reduces pump η . Under part-load conditions, the adverse pressure gradient becomes stronger, and when separation further develops, it may cause the Q - H curve to exhibit an unstable “hump” region, impairing the stable operation of the pump. Separation vortices are typical unsteady flow structures, and their formation, development, and shedding can induce strong low-frequency pressure pulsations in the flow field.

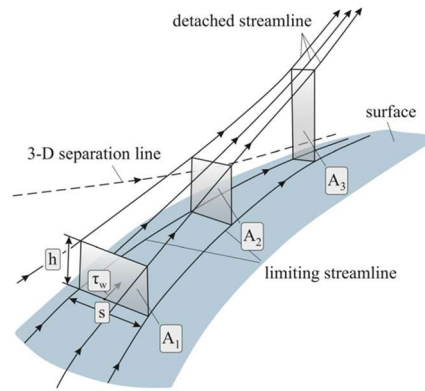


Fig. 2.9. Principle of flow separation on blade surfaces [82].

Backflow in the centrifugal pump impeller typically occurs at both the impeller inlet and outlet. Under part-load conditions, the c_u at the impeller inlet becomes relatively large, causing the absolute flow angle to increase. However, since the blade inlet installation angle remains unchanged, the relative flow angle decreases and may even reverse direction. As a result, part of the fluid cannot enter the blade passage smoothly, leading to the formation of the local backflow region near the LE of the impeller inlet. As shown in Fig. 2.10, this inlet backflow region initially appears near the shroud of the impeller and forms a spiral-shaped recirculation region. This backflow alters the velocity distribution and direction of the incoming flow, disrupting the uniformity of the inflow and potentially inducing inlet vortex structures. As the flow rate continues to decrease, the backflow region expands progressively to cover the entire inlet cross-section, causing fluid to flow backward from the impeller inlet into the inlet pipe. This phenomenon leads to a sharp decline and unstable fluctuations in the Q - H curve.

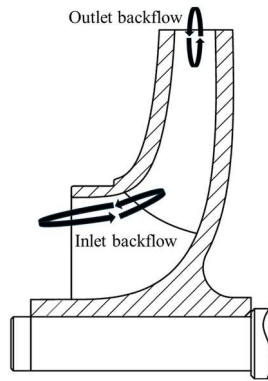


Fig. 2.10. Backflow phenomenon at the impeller inlet and outlet.

Corresponding to the inlet backflow, outlet backflow occurs when the pump operates under part-load or overload conditions. In this situation, the excessive diffusion at the impeller outlet causes part of the fluid, having lost too much kinetic energy to move in the opposite direction, forming the backflow region at the impeller outlet, as illustrated in Fig. 2.10. This backflow typically originates near the SS of the impeller blades at the outlet and propagates radially inward into the impeller passage. As shown in the velocity triangle in Fig. 2.4, the c_m at the impeller outlet decreases significantly as the flow rate is reduced, while the circumferential component c_u increases correspondingly. This change causes the relative velocity w to point toward the blade PS, resulting in a localized reverse flow namely the outlet backflow phenomenon. Under these conditions, the velocity distribution across the impeller outlet becomes highly non-uniform, with large velocity gradients in the circumferential direction, giving rise to strong vortices and regions of concentrated turbulent kinetic energy. The backflow at the impeller outlet not only reduces the effective head of the pump but also interacts with the recirculation region in the volute, inducing strong pressure pulsations and fluctuations in radial force, thereby compromising the operational stability of the pump.

At the impeller outlet of the centrifugal pump, after the fluid undergoes energy exchange with the rotating blades, a region of pronounced velocity and pressure non-uniformity develops near the blade TE. Spatially, this non-uniform flow manifests as a jet-wake structure, as illustrated in Fig. 2.11. The wake refers to the near-wall region on the blade SS, where viscous effects and flow separation lead to reduced velocity and increased turbulent kinetic energy. As the fluid passes the blade TE, the low-speed fluid in the boundary layer is entrained into the main flow, forming a distinct velocity deficit region. In the wake region, the c_u is significantly lower than that in the surrounding mainstream, accompanied by strong turbulent fluctuations and vortex structures. In contrast, the jet corresponds to regions where the fluid, after passing through the blade passages, possesses relatively high velocity and kinetic energy, typically occurring near the blade PS. The velocity difference between the PS and SS causes high-pressure fluid to

accelerate along the PS. Moreover, under the influence of centrifugal force, the fluid moves rapidly from the hub toward the shroud, forming a high-energy jet. At the impeller outlet cross-section, the jet and wake regions are alternately distributed. When operating away from the design condition, the periodicity of the jet-wake structure becomes more pronounced and may even evolve into large-scale unsteady vortices. The interaction between jet and wake regions in the circumferential and radial directions forms strong shear layers, which are one of the primary mechanisms responsible for the generation of unsteady vortex structures. The steep energy gradients in the alternating jet-wake regions can easily induce flow instabilities such as the rotating stall. Furthermore, when this unsteady fluid enters the volute, it generates asymmetric circumferential flow, resulting in periodic fluctuations of the pressure field in the volute, which in turn produce significant radial excitation forces and pressure pulsations acting on the pump casing.

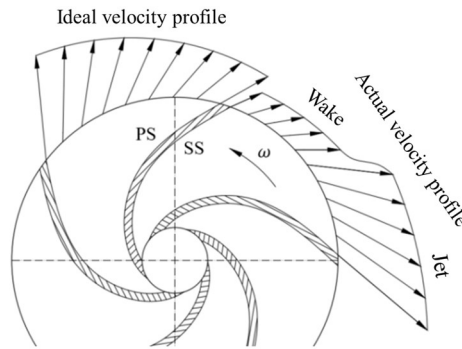


Fig. 2.11. Jet wake phenomenon at the impeller outlet.

2.4.2 Vortex and Transport Characteristics

As described in the previous section, in the impeller and flow passages of the centrifugal pump, the combined effects of centrifugal force, pressure gradients, and blade rotation give rise to various complex and unstable flow structures. These structures readily induce the formation of multiple types of vortices in the impeller passages. Common vortex structures include inlet backflow vortices (IBFV), leading edge separation vortices (LEV), secondary flow vortices, blade wake vortices, and impeller outlet vortices [84-86]. Together, these vortices govern the internal energy distribution, momentum transport, and unsteady flow characteristics in the impeller.

The formation of vortices in the impeller is primarily attributed to two mechanisms, velocity gradients and shear effects. Under the influence of the pressure difference between the SS and PS of the blades, distinct tangential velocity differences develop between adjacent fluid layers. When the shear intensity exceeds a critical threshold, local entrainment and rotational motion occur, leading to the generation of vortex structures. The non-uniform blade loading distribution between the PS and SS, the helical entrainment of the three-dimensional boundary layer, and the combined effects of

centrifugal and Coriolis forces constitute the three principal driving mechanisms for vortex formation in the impeller.

The aforementioned vortex structures typically exhibit periodic spatial distributions, with their core regions characterized by low static pressure and high vorticity. These regions represent focal points for unsteady energy dissipation and flow losses. Vorticity is a key variable for measuring local rotational intensity in centrifugal pumps, directly reflecting the rotational characteristics of fluid particles and unsteady phenomena such as flow separation and shear layer development. Since secondary flows and energy losses in centrifugal pump impellers are often closely associated with intense vortex structures, employing vorticity distribution as a comparative metric helps reveal how different impeller geometries influence complex flow patterns and energy dissipation mechanisms. In this research, relative vorticity is defined as the curl of the relative velocity vector field. The relative velocity vector is expressed as $\mathbf{V}_r = (u, v, w)$, and the relative vorticity is given by:

$$\text{Vorticity} = \nabla \times \mathbf{V}_r = \left(\frac{\partial w}{\partial y} - \frac{\partial v}{\partial z} \right) \mathbf{i} + \left(\frac{\partial u}{\partial z} - \frac{\partial w}{\partial x} \right) \mathbf{j} + \left(\frac{\partial v}{\partial x} - \frac{\partial u}{\partial y} \right) \mathbf{k} \quad (2.9)$$

where $\mathbf{i}$, $\mathbf{j}$, $\mathbf{k}$ denote the unit vectors in the x , y , and z directions, respectively. The magnitude of the relative vorticity characterizes the local rotational intensity of fluid particles, while its direction indicates the local rotation axis.

Vortex identification methods represent one of the most intuitive approaches to studying vortices. By employing different vortex identification criteria, vortex structures can be identified with relative precision and objectivity in complex turbulent flows. Consequently, the selection of vortex identification criteria becomes particularly crucial. Currently, several vortex identification criteria are commonly employed, including the vorticity method, Q -criterion, λ_2 -criterion, Ω (Omega) criterion, and *Liutex* method [87]. In recent years, the Ω criterion and *Liutex* method—both classified as third-generation vortex identification techniques, have been increasingly applied in the field of fluid machinery. These methods are capable of detecting vortical structures of varying intensities and spatial scales, with the most significant advantage being their reduced sensitivity to threshold selection, as the Ω criterion employs a relatively fixed threshold value. This greatly diminishes the influence of subjective parameter selection on the identification results [88, 89]. Therefore, in this study, the Ω -criterion-based vortex identification method is adopted to conduct a detailed analysis of the vortical structures in the impeller under the design condition.

The Ω -criterion was first proposed by Liu in 2016 [90, 91]. To address the challenge of threshold selection, Liu decomposed the vorticity $\boldsymbol{\omega}$ into two components: the rotational component $\mathbf{R}_\omega$ and the non-rotational component $\mathbf{S}_\omega$, defined as:

$$\boldsymbol{\omega} = \mathbf{R}_\omega + \mathbf{S}_\omega \quad (2.10)$$

The Ω criterion quantifies the proportion of rotational motion relative to the total local velocity gradient, providing a normalized measure of rotation dominance ranging from 0 to 1. The Ω expression is defined as:

$$\Omega = \frac{\|\mathbf{B}\|_F^2}{\|\mathbf{A}\|_F^2 + \|\mathbf{B}\|_F^2 + \varepsilon_\omega} \quad (2.11)$$

In the equation, $\mathbf{A}$ represents the symmetric component of the velocity gradient tensor $\nabla \mathbf{V}$, while $\mathbf{B}$ denotes its antisymmetric component. The symbol $\|\cdot\|_F$ denotes the Frobenius norm of a matrix. When $\Omega > 0.5$, it indicates that the antisymmetric part of the tensor dominates, signifying the presence of a rotational motion; thus, $\Omega > 0.5$ is commonly adopted as the criterion for vortex identification. Here, $\mathbf{A}$ is also referred to as the rate-of-strain tensor, and $\mathbf{B}$ as the rate-of-rotation tensor, which physically represents the local deformation and rotation of the flow field at a given point, respectively. Their mathematical expressions are defined as:

$$\mathbf{A} = \frac{1}{2}(\nabla \mathbf{V} + (\nabla \mathbf{V})^T) = \begin{bmatrix} \frac{\partial u}{\partial x} & \frac{1}{2}\left(\frac{\partial v}{\partial x} + \frac{\partial u}{\partial y}\right) & \frac{1}{2}\left(\frac{\partial u}{\partial z} + \frac{\partial w}{\partial x}\right) \\ \frac{1}{2}\left(\frac{\partial v}{\partial x} + \frac{\partial u}{\partial y}\right) & \frac{\partial v}{\partial y} & \frac{1}{2}\left(\frac{\partial w}{\partial y} + \frac{\partial v}{\partial z}\right) \\ \frac{1}{2}\left(\frac{\partial u}{\partial z} + \frac{\partial w}{\partial x}\right) & \frac{1}{2}\left(\frac{\partial w}{\partial y} + \frac{\partial v}{\partial z}\right) & \frac{\partial w}{\partial z} \end{bmatrix} \quad (2.12)$$

$$\mathbf{B} = \frac{1}{2}(\nabla \mathbf{V} - (\nabla \mathbf{V})^T) = \begin{bmatrix} 0 & \frac{1}{2}\left(\frac{\partial u}{\partial y} - \frac{\partial v}{\partial x}\right) & \frac{1}{2}\left(\frac{\partial u}{\partial z} - \frac{\partial w}{\partial x}\right) \\ \frac{1}{2}\left(\frac{\partial v}{\partial x} - \frac{\partial u}{\partial y}\right) & 0 & \frac{1}{2}\left(\frac{\partial v}{\partial z} - \frac{\partial w}{\partial y}\right) \\ \frac{1}{2}\left(\frac{\partial w}{\partial x} - \frac{\partial u}{\partial z}\right) & \frac{1}{2}\left(\frac{\partial w}{\partial y} - \frac{\partial v}{\partial z}\right) & 0 \end{bmatrix} \quad (2.13)$$

In Eq. 2.11, ε_ω in the denominator represents an infinitesimal positive value, intended to prevent the denominator from becoming zero. In the research by Dong et al. [92, 93], to avoid arbitrarily selecting the value of ε_ω , it is defined as:

$$\varepsilon_\omega = 0.001 \times \left(\|\mathbf{B}\|_F^2 - \|\mathbf{A}\|_F^2 \right)_{\max} \quad (2.14)$$

From the definition of the Ω vortex identification criterion, when $\Omega = 0$, the flow field corresponds to a non-rotational (irrotational) state. Given that the turbulent intensity in the centrifugal pump impeller in this study is relatively high and that strong rotational effects are present, selecting a threshold slightly greater than 0.5 yields a more accurate vortex identification. Following the approach of Liu et al. [94] and based on the existing applications of this method in centrifugal pumps [95, 96], a fixed threshold of $\Omega = 0.52$ is adopted in this study to perform detailed identification and analysis of the vortex structures in the pump.

The vortex structures in the impeller exhibit pronounced unsteady transport characteristics. On one hand, vortices continuously migrate, merge, or break down during the flow process, displaying multi-scale features in both time and space. On the other hand, the evolution of these vortices is accompanied by the redistribution and dissipation of kinetic energy, making them one of the primary sources of flow losses in the impeller. To gain a deeper understanding of the evolution characteristics of vortex identification structures, it is necessary to introduce an appropriate theoretical framework to describe the transport and development process of vortices. In this study, the three-dimensional viscous fluid total vorticity transport equation is introduced, and based on the vorticity decomposition theory, the relative vorticity transport equation is derived to characterize the vortex transport mechanism. For a three-dimensional viscous fluid, the complete form of the classical total vorticity transport equation is expressed as [97]:

$$\frac{D\boldsymbol{\omega}}{Dt} = (\boldsymbol{\omega} \cdot \nabla) \mathbf{V} - \boldsymbol{\omega}(\nabla \cdot \mathbf{V}) + \nu \Delta \boldsymbol{\omega} + \nabla \times \mathbf{f} - \nabla \times \left(\frac{1}{\rho} \nabla p \right) \quad (2.15)$$

where $\boldsymbol{\omega}$ denotes the total vorticity, $\mathbf{V}$ is the velocity vector, ν represents the kinematic viscosity, $\mathbf{f}$ is the body force, ρ is the fluid density, and p is the static pressure. By combining Eq. 2.15 with the mass conservation equation and the momentum conservation equation, the relative vorticity transport equation for a general viscous fluid can be derived as follows [98]:

$$\frac{D\boldsymbol{\omega}}{Dt} = (\boldsymbol{\omega} \cdot \nabla) \mathbf{V} - \boldsymbol{\omega}(\nabla \cdot \mathbf{V}) - 2\nabla \times (V_\alpha \times \mathbf{V}) + \frac{\nabla \rho_m \times \nabla p}{\rho_m^2} + (v_m + v_t) \nabla^2 \boldsymbol{\omega} \quad (2.16)$$

where V_α is the angular velocity, v_m is the kinematic viscosity, and v_t is the turbulent viscosity.

On the right-hand side of Eq. 2.16, the first term represents the relative vortex stretching term (RVS), which accounts for the bending and elongation of vortex lines induced by the velocity gradient. The second term denotes the relative vortex dilatation term (RVD), reflecting the influence of fluid element expansion or contraction on the vorticity. The third term corresponds to the Coriolis force term (CORF), describing the inertial deflection effects of the fluid mass caused by impeller rotation. The fourth term is the baroclinic torque term (BT), which represents the influence of density and pressure gradients on vortex evolution. The fifth term is the viscous diffusion term (VISD) associated with fluid viscosity. In the single-phase, high-Reynolds number flow of the centrifugal pump, the fluid density remains constant. Hence, the BT is zero. Meanwhile, the VISD is relatively small compared to the first two terms and can generally be neglected. This study focuses on the statistics of the first three terms of the relative vorticity transport equation.

2.4.3 Pressure Pulsations and Excitation Forces

During the operation of the centrifugal pump, significant pressure pulsations arise in the hydraulic components due to the RSI, non-uniform flow, and the periodic variation of vortex structures. These pressure fluctuations propagate through the pump casing and act upon the solid structure, generating hydraulic excitation forces and inducing pump vibrations. Pressure pulsation is a direct manifestation of the unsteady behavior in fluid dynamics and represents one of the key factors affecting the operational stability, noise level, and service life of centrifugal pumps.

Due to the pronounced non-uniformity of the flow field at the impeller outlet, when the fluid exits the impeller and enters the volute, it interacts with the stationary wall near the volute tongue, generating periodic pressure fluctuations. The dominant frequency of this RSI typically corresponds to the blade passing frequency (BPF) f_{BPF} , which is defined as:

$$f_{\text{BPF}} = z \times f_n \quad (2.17)$$

where z is the number of blades and f_n is the rotational frequency of the impeller. The f_{BPF} represents one of the most prominent periodic unsteady signals in centrifugal pumps. In addition, due to phenomena such as vortex shedding, secondary flow, and backflow, distinctive pressure pulsation characteristics often occur in the impeller passages. Particularly under part-load conditions, the intensification of flow instabilities such as flow separation leads to a significant increase in pulsation amplitude and the emergence of low-frequency pressure oscillations.

The spatial distribution of pressure pulsations in the centrifugal pump is highly non-uniform. Typically, the pulsation intensity reaches its maximum near the impeller outlet and the volute tongue. Along the radial direction of the impeller, the pressure amplitude gradually attenuates and transitions to a more stable state toward the pump outlet. In this study, the Fast Fourier Transform (FFT) is employed to extract the characteristic frequencies of the pressure pulsation signals in the pump, enabling the evaluation of the stability of the internal flow patterns. In the frequency domain, the fundamental frequency component generally corresponds to the rotational frequency f_n , while the f_{BPF} typically exhibits a pronounced amplitude. In addition, multiple harmonics of f_n can also be clearly observed in the spectrum.

The unstable flow patterns in the centrifugal pump can induce hydraulic excitation forces, which represent a more severe issue than mere unsteady pressure pulsations. When these pressure pulsations act on the physical boundaries of the pump, they generate unsteady fluctuating excitation forces, leading to significant structural vibrations and acoustic noise in the pump system. The excitation force $F(t)$ acting on the impeller surface as a result of pressure pulsations can be expressed as:

$$\vec{F}(t) = \iint_S p(t, \vec{r}) \vec{n}_v dS \quad (2.18)$$

where $p(t,r)$ represents the instantaneous pressure distribution, n_v denotes the surface normal vector, and S refers to the physical wall surface of the impeller.

In the centrifugal pump, the fluctuating excitation forces can generally be categorized into axial forces and radial forces, corresponding to the axial and radial directions, respectively. In this study, the axial force F_A on the impeller is obtained by integrating the static pressure acting on the shroud and hub, as well as on the blade surfaces, along the x -direction. The radial force F_R of the impeller consists of two components, derived from the integration of the static pressure on the impeller shroud, hub and blades in the y and z directions, respectively, yielding F_Y and F_Z . The resultant radial force vector of the impeller is then expressed as $F_R = [(F_Y)^2 + (F_Z)^2]^{0.5}$.

The axial force F_A acting on the centrifugal pump impeller primarily consists of three components, as illustrated in Fig. 2.12. The first component comprises the forces acting on the outer surfaces of the shroud and hub, denoted as F_{p1} and F_{p2} , respectively. Together, these constitute the shroud axial force. F_{p1} and F_{p2} are mainly caused by the pressure difference between the shroud and hub pump chambers and represent the dominant contribution to F_A . The second component includes the forces acting on the inner surfaces of the shroud and hub, denoted as F_{p3} and F_{p4} , respectively. These forces arise primarily from the non-uniform pressure distribution and pressure gradients induced by the unsteady flow patterns in the impeller passages. The third component, denoted as F_{p5} , corresponds to the blade axial force, which acts on the surfaces of the impeller blades. F_{p5} is mainly generated by the pressure difference between the PS and SS of the blades, the pressure variation between the LE and TE, and the pressure gradients associated with unsteady flow phenomena. However, F_{p5} is not a major contributor to the overall axial force. Finally, the total axial force F_A in this study can be expressed as $F_A = F_{p1} + F_{p2} + F_{p3} + F_{p4} + F_{p5}$.

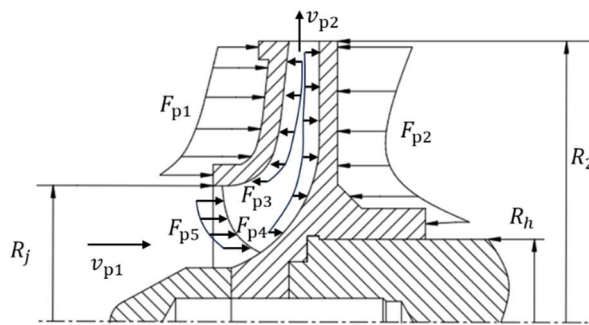


Fig. 2.12. Axial force distribution diagram of centrifugal pump impeller.

The unsteady flow patterns in the impeller also induce the generation of fluctuating radial forces F_R . In addition, the asymmetric annular geometry of the volute leads to a non-uniform circumferential pressure distribution at the impeller outlet, resulting in an unbalanced radial load. The F_R acts directly on the impeller, pump shaft, and bearings,

thereby increasing bearing loads and causing vibration, noise, and mechanical wear in the pump assembly. In practical centrifugal pump operation, both radial and axial excitation forces coexist and are often coupled. Their phase relationship and amplitude characteristics are determined by the unsteady nature of the internal flow structures. When the f_{BPF} or its harmonics approach the natural frequency of the pump structure, the excitation forces can be significantly amplified, leading to severe vibration, elevated noise levels, and in extreme cases, fatigue failure of mechanical components.

3 Hydraulic Design of Centrifugal Pump and Tandem Impeller Design Methods

The subject of this study is a typical single-stage, single-suction centrifugal pump. A rational hydraulic design forms the foundation for ensuring the high efficiency and stable operation of the centrifugal pump. Therefore, this chapter systematically presents the hydraulic design process and methodology of the investigated pump. First, the hydraulic design of the impeller and volute is described in detail, including the determination principles of key geometric parameters, the design of the flow passage structure, and the matching relationships between performance parameters. Subsequently, to further enhance the energy conversion efficiency and flow stability of the pump, the tandem impeller configuration is proposed and investigated. In addition, optimization methods tailored to the geometric characteristics of the tandem blades are explored, aiming to reduce flow losses and improve the overall hydraulic performance of the centrifugal pump.

3.1 Hydraulic Design of Centrifugal Pump

The primary objective of this study is to investigate the influence of impeller structural modifications, particularly the implementation of tandem blades on the performance and internal flow characteristics of the centrifugal pump. As discussed in Chapter 1, Fensterseifer et al. [81] successfully applied a tandem-blade impeller to a centrifugal pump with a specific speed $n_q=39$. To further examine the applicability and adaptability of tandem-blade impellers in different centrifugal pumps, the present study designs a pump with a lower specific speed of $n_q=25$ and conducts an in-depth analysis of the tandem-blade configuration. The specific speed n_q in this study is defined by Eq. 3.1. This focus is motivated by the fact that low specific speed centrifugal pumps exhibit more complex flow patterns, with phenomena such as flow separation and secondary flows occurring more readily in the impeller. Therefore, the practical effectiveness of tandem-blade impellers under such conditions has not yet been fully validated.

$$n_q = \frac{nQ_{\text{des}}^{1/2}}{H_{\text{des}}^{3/4}} \quad (3.1)$$

Based on the existing experimental test loop, the design flow rate is set to $Q_{\text{des}}=30\text{m}^3/\text{h}$. Considering both experimental conditions and model manufacturing constraints, the design rotational speed of the centrifugal pump is determined to be $n=2000\text{rpm}$. Since key components such as the impeller and volute are fabricated using rapid prototyping technology to facilitate rapid design iteration and geometric optimization, their material strength and stiffness are lower than those of conventional metallic pump casings. To reduce centrifugal stress and structural loads at high rotational speeds, and to prevent

deformation or damage of the impeller and volute, a relatively lower operational speed is selected for the experimental phase. Nevertheless, this speed ensures that the flow remains in a comparable Reynolds number range, thereby maintaining the engineering representativeness of the experimental results in terms of flow characteristics and energy transfer mechanisms. Based on the above-determined parameters and Eq. 3.1, the design head under the design condition can be calculated as $H_{des}=14.2\text{m}$.

Based on the aforementioned fundamental design parameters, the hydraulic design of the centrifugal pump can be carried out. The hydraulic design of the centrifugal pump primarily consists of two key components, the impeller and the volute. As the core energy conversion component, the geometry and flow passage structure of the impeller directly determine the H and η of the pump. The volute, on the other hand, is responsible for efficiently converting the kinetic energy at the impeller outlet into pressure energy, while ensuring a uniform and stable flow distribution. To achieve optimal overall pump performance, it is essential to consider the flow characteristics and energy matching relationship between the impeller and volute during the design process. The following sections introduce the hydraulic design methodology and key design parameters for both the impeller and the volute.

3.1.1 Hydraulic Design of Impeller

In this study, the main geometric parameters of the impeller were determined using the classical velocity coefficient method, with the empirical correlations and coefficient ranges adopted from the Modern Pumps Theory and Design [99], which summarizes well-established centrifugal pump design experience.

First, the minimum shaft diameter D_{min} was estimated based on the torque requirement. Given the allowable shear stress of the material $\tau=15\text{MPa}$, and considering additional factors such as stiffness and critical rotational speed, a safety factor of $SF=1.15$ was applied. The calculated minimum shaft diameter was $D_{min}=14.15\text{mm}$. Subsequently, considering the available manufacturing conditions and referencing previously designed pump shafts, the shaft diameter at the impeller mounting position was determined as $D_i=17.5\text{mm}$. Finally, the impeller hub diameter D_h was selected to ensure that sufficient wall thickness remains after machining the keyway in the shaft hole, providing adequate mechanical strength for the hub. Therefore, in this study, D_h is defined as:

$$D_h = (1.2 \sim 1.4) D_i = 23\text{mm} \quad (3.2)$$

when determining the impeller inlet diameter D_j , the influence of the hub must be excluded. Therefore, an equivalent impeller inlet diameter D_0 is introduced. Based on the velocity coefficient method, D_0 can be expressed as:

$$D_0 = k_0 \sqrt[3]{\frac{Q_{des}}{n}} = 72.44\text{mm} \quad (3.3)$$

where the coefficient k_0 is set to 4.5 to balance both the pump η and cavitation

performance [99]. Finally, the impeller inlet diameter D_j is determined based on the combined relationship between D_0 and D_h , as follows:

$$D_j = \sqrt{D_0^2 + D_h^2} = 76\text{mm} \quad (3.4)$$

when determining the impeller outlet diameter D_2 , the classical velocity coefficient correction method was also employed. First, the impeller outlet diameter coefficient k_D was introduced, which is defined as:

$$k_D = 9.35k_{D2} \left(\frac{n_q}{100} \right)^{-0.5} = 10.12 \quad (3.5)$$

where k_{D2} is the correction coefficient for D_2 , which is set to 0.54 based on Ref. [99]. Finally, the impeller outlet diameter D_2 is calculated based on k_D as follows:

$$D_2 = k_D \sqrt[3]{\frac{Q_{\text{des}}}{n}} = 163\text{mm} \quad (3.6)$$

when determining the impeller outlet width b_2 , a similar approach to the one described above is adopted. First, the impeller outlet width coefficient k_b is introduced, which is defined as:

$$k_b = 0.64k_{b2} \left(\frac{n_q}{100} \right)^{5/6} = 0.65 \quad (3.7)$$

where k_{b2} is the correction coefficient for b_2 , which is assigned a value of 3.23 based on Ref. [99]. Finally, the impeller outlet width b_2 is calculated based on k_b as follows:

$$b_2 = k_b \sqrt[3]{\frac{Q_{\text{des}}}{n}} = 10.5\text{mm} \quad (3.8)$$

After determining D_h , D_j , D_2 , and b_2 , the meridional projection of the impeller was constructed. By adjusting the control points of the B-spline curves representing the shroud and hub, the shape of the impeller meridional profile could be flexibly modified. During the design process, it was ensured that the curves of the shroud and hub remained smooth and continuous, while the flow passage cross-sectional area varied smoothly and monotonically from the impeller inlet to the outlet. Therefore, in this study, relatively large curvature radii were adopted for the shroud curves, and the shroud and hub were kept parallel near the impeller outlet. After multiple iterations of refinement, the final meridional projection of the impeller is shown in Fig. 3.1(a). Fig. 3.1(b) presents the distribution of the cross-sectional flow area from the impeller inlet to the outlet. In this design, the cavitation performance of the pump was also taken into consideration, resulting in a slightly convex shape in the area curve near the inlet region.

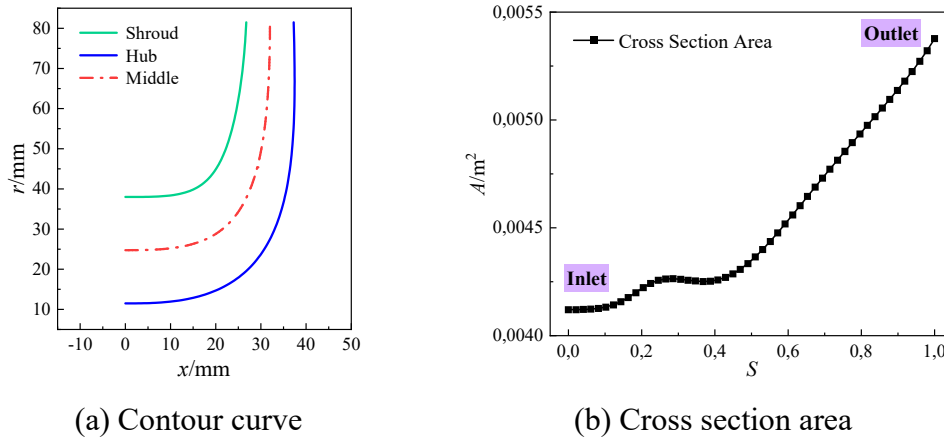


Fig. 3.1. The axial projection design of the centrifugal pump impeller.

During the twisted blade design stage of the centrifugal pump, the position of the blade LE was first determined. In this study, the blade LE was designed to be approximately perpendicular (90°) to the streamlines of the shroud and hub, with a slight extension toward the pump inlet to allow the fluid to engage with the blade earlier. This design helps to reduce the impeller outer diameter, thereby decreasing disk friction losses. The LE position is illustrated in Fig. 3.2(a). Next, in selecting the blade inlet setting angle β_1 , this study employed the velocity triangle matching principle based on the velocity coefficient method. The radial velocity at the impeller inlet c_{m1} was calculated from the design flow rate, inlet geometry, and rotational speed. Combined with the circumferential velocity of the impeller u_2 , the preliminary values of β_1 were determined along each streamline. Subsequently, considering the beneficial influence of a positive incidence angle on the inlet flow characteristics, the β_1 distribution along the streamlines was adjusted to achieve improved flow alignment and cavitation performance. Ultimately, the β_1 values in this study were determined to range between 10° and 20° across different spans.

The blade outlet setting angle β_2 plays a crucial role in determining the head generation and loss characteristics of the centrifugal pump. In practical design, the specified design head H_{des} corresponds to the actual delivered head of the pump, which is smaller than the ideal Euler head due to hydraulic losses (e.g., shock loss, friction loss, secondary-flow loss) and slip effects. Therefore, the outlet velocity triangle should be established based on the Euler head at the design point, $H_{E, des}$, rather than directly using H_{des} . To account for the influence of hydraulic losses, a hydraulic efficiency under the design condition $\eta_{h, des}$ is introduced, defined as:

$$\eta_{h, des} = \frac{H_{des}}{H_{E, des}} \quad (3.9)$$

Under the no pre-swirl assumption ($c_{u1}=0$), the circumferential component of the absolute velocity at the impeller outlet c_{u2} is obtained from the Euler equation:

$$c_{u2} = \frac{gH_{E,des}}{u_2} = \frac{gH_{des}}{\eta_{h,des}u_2} = 8.17 \text{ m/s} \quad (3.10)$$

The average meridional (radial) velocity at the impeller outlet c_{m2} is determined from the continuity equation. Considering that the flow rate passing through the impeller should reflect volumetric losses, the throughflow rate is expressed as $Q_{th}=Q_{des}/\eta_{v,des}$, where $\eta_{v,des}$ is the volumetric efficiency under the design condition. Meanwhile, the effective flow area is corrected by k_2 to account for blade thickness/blockage. Thus:

$$c_{m2} = \frac{Q_{th}}{2\pi r_2 b_2 k_2} = \frac{Q_{des}}{2\pi r_2 b_2 k_2 \eta_{v,des}} = 1.73 \text{ m/s} \quad (3.11)$$

where k_2 represents the correction factor for the outlet flow channel area of the blade, which is set to 0.9 based on Ref. [99]. Finally, based on the velocity triangle relationship at the impeller outlet, the theoretical outlet relative flow angle β_{22} is calculated as follows:

$$\tan \beta_{22} = \frac{c_{m2}}{u_2 - c_{u2}} = 0.195 \quad (3.12)$$

Therefore, the β_{22} of the impeller was calculated to be $\beta_{22}=11^\circ$. It should be noted that β_{22} corresponds to the ideal velocity triangle. In real flows, the finite number of blades and viscous effects lead to a reduction in c_{u2} (slip) and additional hydraulic losses. Therefore, the blade outlet setting angle β_2 is selected by applying a correction to β_{22} , i.e.,

$$\beta_2 = \beta_{22} + \Delta\beta \quad (3.13)$$

where $\Delta\beta$ is chosen to compensate for slip and loss effects (determined based on empirical experience, Ref. [99]). In this study, β_2 was finalized as $\beta_2=24^\circ$ after considering the above effects and ensuring a balanced compromise between head requirement and loss minimization across all spanwise streamlines. All the above blade angle designs were conducted along five different streamlines in the impeller. The blade WA for all streamlines was initially set to 120° , and the influence of varying WA will be discussed in detail later. The final blade angle distribution is illustrated in Fig. 3.2(b).

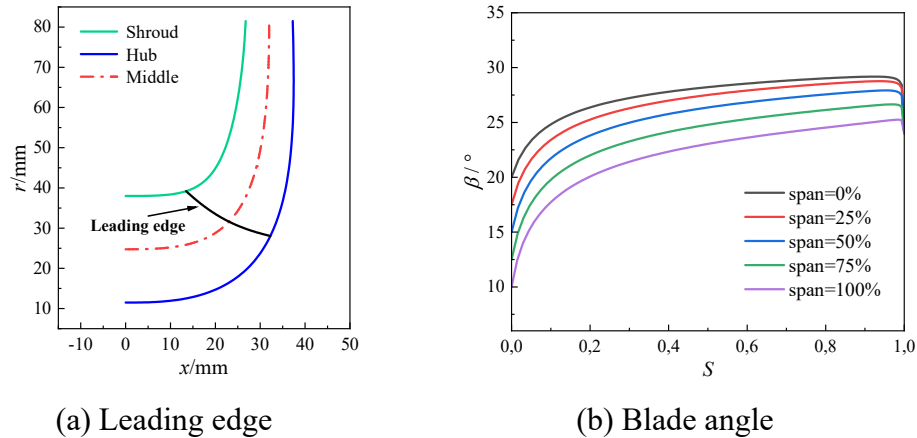
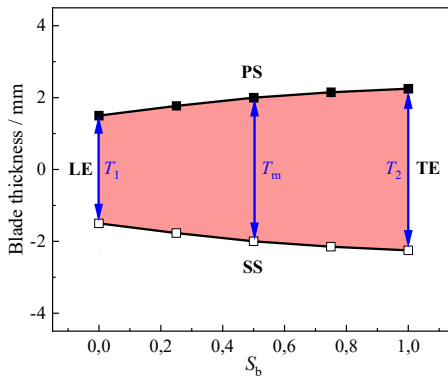


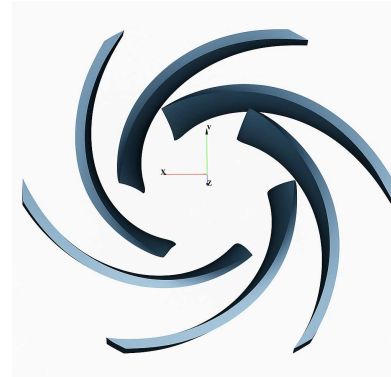
Fig. 3.2. The blade angle design of the centrifugal pump impeller.

The final blade geometry was obtained by symmetrically thickening the designed mean camber surface on both sides. In this study, the blade thickness was determined by comprehensively considering the impeller dimensions, material strength, and manufacturing constraints. Since the impeller was fabricated using photosensitive resin via 3D printing for hydraulic performance testing, the blade thickness design needed to balance the material brittleness, interlayer bonding strength, and hydraulic performance requirements. According to previous design experience [100], the LE thickness T_1 significantly influences the flow behavior at the inlet and is typically set to $T_1=(0.01-0.015) \times D_2=1.5-2.5\text{mm}$. Considering the relatively low mechanical strength of the resin material, $T_1=3.0\text{mm}$ was selected in this study. The midsection thickness T_m , which bears the main mechanical load of the blade, must provide sufficient structural strength and generally satisfies $T_m=(0.02-0.04) \times D_2=3-6\text{mm}$. Here, $T_m=4.0\text{mm}$ was adopted. Typically, the TE thickness T_2 should be minimized to reduce wake losses. However, due to the brittle nature of the resin and the limited interlayer bonding strength inherent in 3D printing, cracks or deformations are more likely to occur in the TE region. To enhance structural strength and printing stability, a gradually increasing thickness profile along the flow direction was employed, making T_2 slightly larger than T_m . In this design, $T_2=4.5\text{mm}$. The blade thickness was smoothly varied along the streamline, with a rounded LE transition for improved flow behavior. The final thickness distribution curve from the LE to the TE is illustrated in Fig. 3.3(a).

When determining the number of blades z , this study aimed to balance two key considerations: minimizing blade blockage and surface friction while ensuring that each flow passage has sufficient length to maintain stable fluid motion and allow adequate energy transfer from the blades to the liquid. Referring to centrifugal pumps with similar n_q , the typical range for z is 5 to 7. The specific influence of z on the performance parameters of the centrifugal pump will be analyzed in detail in the following sections to identify the most suitable blade number. The final three-dimensional geometry of the designed impeller is shown in Fig. 3.3(b), where the number of blades is $z=6$.



(a) Blade thickness



(b) 3D blades

Fig. 3.3. The 3D blades design of the centrifugal pump impeller.

3.1.2 Hydraulic Design of Volute

In this study, the design of the volute was carried out using a quasi-one-dimensional method based on the principles of momentum conservation and the energy equation, combined with the cross-sectional area law to ensure smooth kinetic energy recovery along the flow path. The first step in volute design is to determine the base circle diameter D_3 , which should generally be slightly larger than the impeller outlet diameter D_2 , in order to maintain an appropriate clearance between the volute tongue and the impeller. The size of this clearance is a critical parameter that influences the pump performance, operational stability, and noise characteristics. Typically, $D_3 = (1.03 \sim 1.08) \times D_2$ [99]. Considering the combined effects of noise, vibration, and cavitation, the final value of D_3 in this study was defined as:

$$D_3 = (1.03 \sim 1.08) D_2 = 175 \text{ mm} \quad (3.14)$$

Next, the inlet width of the volute b_3 should be determined. Typically, b_3 must be larger than the impeller outlet width b_2 which includes the thickness of both the shroud and hub and should also provide sufficient clearance to compensate for rotor axial displacement and manufacturing tolerances. Since the centrifugal pump in this study has a relatively low n_q , it is necessary to control the leakage losses through the clearance between the impeller and volute while improving flow compatibility. Therefore, the final value of b_3 in this design was determined as [99]:

$$b_3 = b_2 + (4 \sim 8) = 14.5 \text{ mm} \quad (3.15)$$

In this design, the selected value of b_3 is 2 mm larger than b_2 on each side, which helps reduce the meridional velocity at the volute inlet and thereby mitigates flow impact and the occurrence of secondary flows to some extent.

Next, the cross-sectional shape of the volute must be determined. Common configurations include rectangular, pear-shaped, trapezoidal, and circular cross-sections. In this study, a circular cross-section was adopted. Compared with pear-shaped or rectangular designs, a circular volute provides better flow continuity and lower local energy losses, effectively reducing secondary flow and separation caused by abrupt cross-sectional transitions. Additionally, the circular structure facilitates mesh generation in subsequent numerical simulations, ensuring solution stability and convergence. During the experimental prototype manufacturing phase, the circular flow passage is also easier to mold and finish, which helps maintain high structural accuracy and smoothness of the internal flow channel.

After determining the basic geometric parameters (D_3 and b_3) and adopting a circular cross-section, this study employed a geometry-based method for the parametric hydraulic design of the volute cross-sections. The core concept of this method is to define the geometric characteristic parameters of the volute-specifically, the proportional

relationship between the cross-sectional area A and the curvature radius r_c to determine the volute expansion law. In this study, to balance flow stability and energy recovery, a linear increase of A/r_c from 0 to 11 mm along the circumferential direction of the volute was adopted, as illustrated in Fig. 3.4(a). This linear distribution ensures a smooth circumferential expansion of the volute passage, allowing the fluid velocity to gradually decrease and the pressure to recover uniformly. As a result, it improves both the flow continuity and the energy conversion efficiency in the volute.

Based on the specific A/r_c values at each circumferential position, along with the predetermined b_3 and the geometric constraint of a circular cross-section, the cross-sectional area A at different locations can be calculated, and the corresponding cross-sectional geometry can be determined. The final variation of the volute cross-sectional area A obtained in this study is illustrated in Fig. 3.4(b), while Fig. 3.4(c) shows the geometric configurations of several representative volute cross-sections.

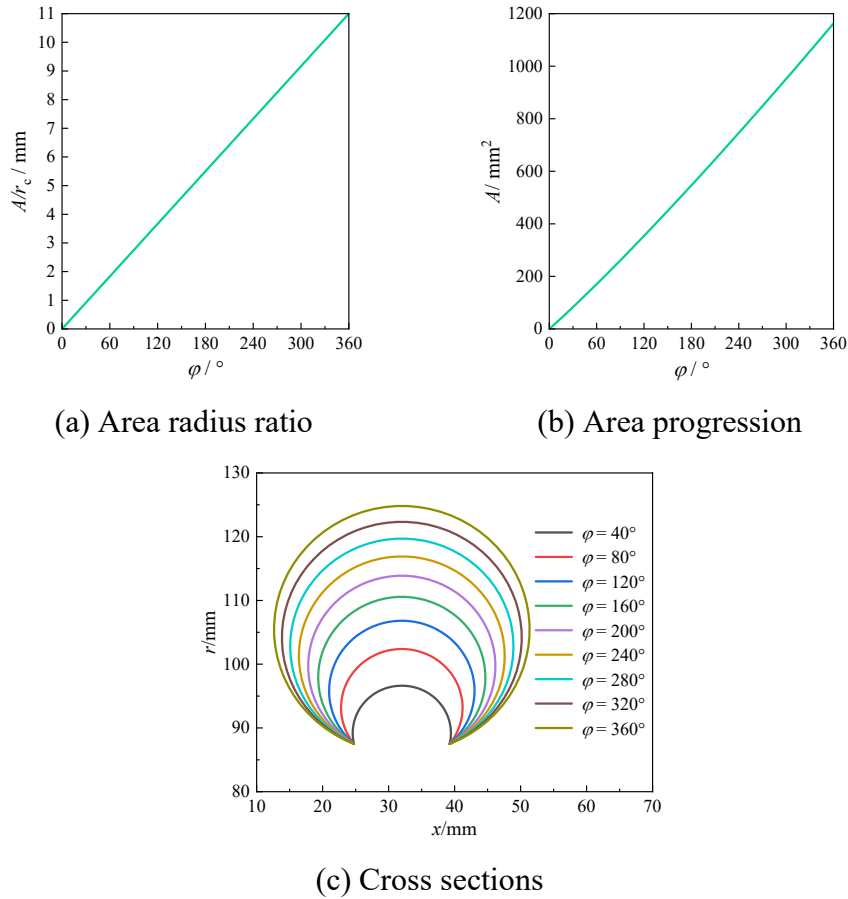


Fig. 3.4. The cross sections design of the volute.

The diffuser section of the volute further reduces the flow velocity, converting a larger portion of the fluid kinetic energy into pressure energy, while also minimizing flow losses in the outlet pipeline. In the design of the diffuser section, the first step is to determine the volute outlet diameter D_4 . Based on the classical velocity coefficient correction

method, this study introduces the volute outlet diameter coefficient k_{D4} , and defines D_4 as follows [99]:

$$D_4 = k_{D4} \sqrt[3]{\frac{Q_{des}}{n}} = 55\text{mm} \quad (3.16)$$

where the typical range of k_{D4} is 3-4, and in this study, it is set to 3.4 [99]. The height of the diffuser section is another important design parameter, as it directly affects the flow uniformity and the overall size of the pump. Under the constraints of maintaining an appropriate diffusion angle and satisfying installation requirements, the height should be kept as small as possible. Considering the limitations of the existing experimental test bench and manufacturing conditions, the final diffuser height was set to 200mm. In this study, a radial design approach was adopted for the volute diffuser, where the A gradually increases along the radial direction, as illustrated in Fig. 3.5(a). The volute outlet features a circular cross-section with an outlet diameter of $D_4=55\text{mm}$, and the outlet direction is perpendicular to the impeller inlet direction. The final spiral contour of the volute is shown in Fig. 3.5(b).

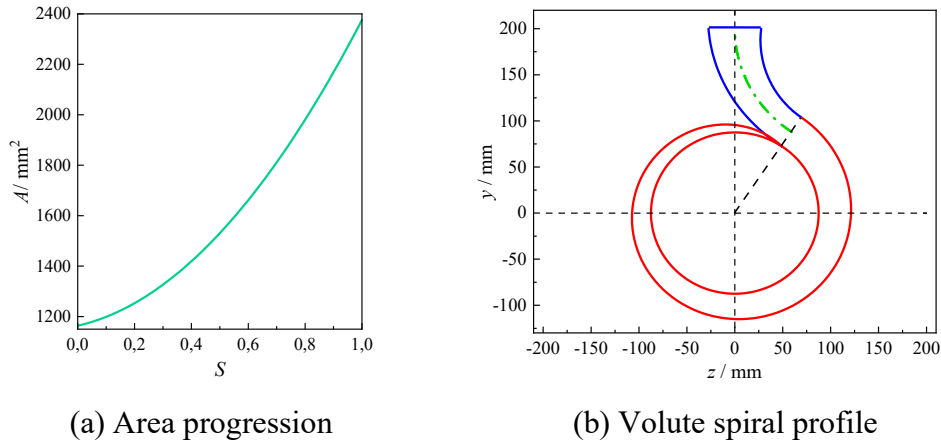


Fig. 3.5. The diffusion section design of the volute.

The volute tongue serves to separate the diffuser section from the spiral flow channel, preventing the radial jet flow from the impeller outlet from directly impinging in the volute and generating vortices. A properly selected tongue setting angle ψ_0 can effectively reduce local turbulence and pressure pulsation, while improving the circumferential uniformity of the outlet pressure distribution in the volute. In general, centrifugal pumps with higher n_q require larger values of ψ_0 . In this study, the tongue setting angle was selected as $\psi_0=20^\circ$, which aligns with empirical design recommendations for low-to-medium n_q centrifugal pumps. The final spiral contour of the volute is shown in Fig. 3.6(a), and its corresponding 3D structure is illustrated in Fig. 3.6(b).

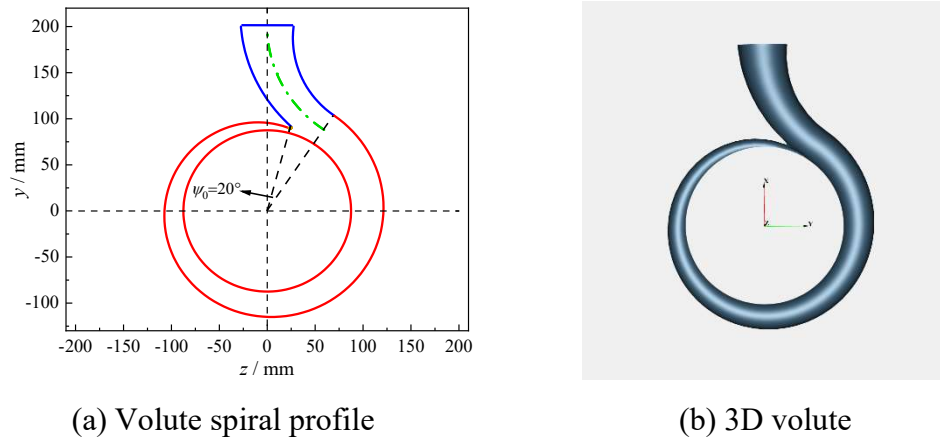


Fig. 3.6. The hydraulic design of the volute.

3.2 Hydraulic Design of Tandem Impeller

Based on the initial design parameters, the Q_{des} of the pump was defined as $30\text{m}^3/\text{h}$, the H_{des} as 14.2m , and the impeller rotational speed n as 2000rpm . Following the design procedure outlined in Section 3.1, the core hydraulic components of this centrifugal pump with the $n_q=25$ were designed at the SAM institute of the RPTU University Kaiserslautern-Landau. The main geometric parameters of the preliminarily designed pump are listed in Table 3.1, and the notations of the key dimensions are illustrated in Fig. 3.7.

Table 3.1. Primary parameters of the centrifugal pump.

Parameter	Symbol	Unit	Value
Impeller inlet diameter	D_j	(mm)	76
Impeller hub diameter	D_h	(mm)	23
Impeller outlet diameter	D_2	(mm)	163
Impeller outlet width	b_2	(mm)	10.5
Volute inlet diameter	D_3	(mm)	175
Volute inlet width	b_3	(mm)	14.5
Volute outlet diameter	D_4	(mm)	55

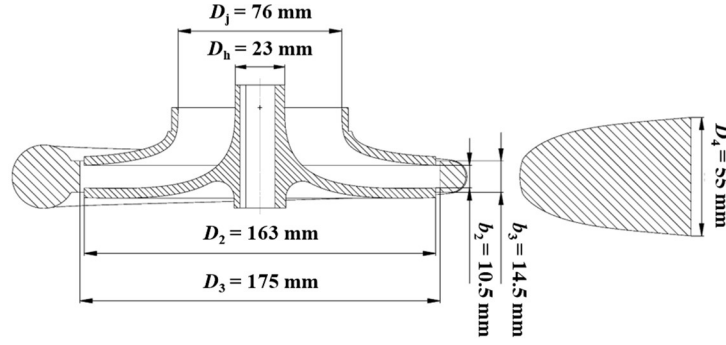


Fig. 3.7. Main dimensions of the centrifugal pump.

The impeller designed in Section 3.1.1 was used as the baseline model in this study and is referred to as the conventional impeller. Based on this reference impeller, the hydraulic design of the tandem impeller was carried out. The most distinctive feature of the tandem impeller is the presence of a mid-span gap in the blade. Therefore, the first step is to determine the gap position. In this study, the gap position was first located and then symmetrically thickened on both sides to achieve the desired gap width. As illustrated in Fig. 3.8, the gap position point P is defined by the intersection of the meridional contours on the shroud surface and hub surface. The parameters b_s and b_h represent the meridional chord lengths from the LE to the TE along the shroud and hub surfaces, respectively. Similarly, l_s and l_h denote the meridional distances from the LE to the gap position point P along the shroud and hub surfaces. The dimensionless ratios l_s/b_s and l_h/b_h were used to define the relative gap position, as expressed in Eq. 3.17. In this study, the gap position parameter P was set to 0.5, meaning that the gap is located at the mid-chord region of the blade. Finally, by connecting the corresponding position points P on the shroud and hub surfaces, the mid-gap line was obtained.

Subsequently, a radial gap size Δr was introduced based on the previously defined mid-gap line. A larger Δr generally results in greater flow losses, thereby reducing the overall impeller performance. In this study, taking into account both mesh generation requirements and manufacturing constraints, the radial gap was set to $\Delta r = 0.5 \text{ mm}$. As illustrated in the enlarged view of Fig. 3.8, the upper and lower gap surfaces were obtained by offsetting the mid-gap line in the radial direction by $\pm \Delta r / 2 = \pm 0.25 \text{ mm}$. This approach ensures a uniform gap width throughout the entire blade span and enables a smooth and continuous transition of the fluid between the PS and SS of the blade.

$$P = \frac{l_s}{b_s} = \frac{l_h}{b_h} = 0.5 \quad (3.17)$$

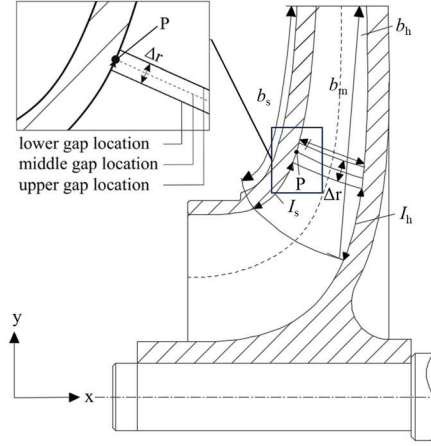


Fig. 3.8. The gap design of the tandem impeller.

By defining the gap position of the blade using the dimensionless parameter P , the junction interface between the front and rear blades of the tandem configuration can be flexibly adjusted without altering the overall meridional contour of the blade. This approach also facilitates subsequent numerical simulations and optimization studies. After determining the gap size and position, geometric fillet smoothing was applied to the TE of the front blade and the LE of the rear blade to improve the flow characteristics in the connection region of the tandem blades. This filleting treatment not only helps to alleviate velocity gradient variations and reduce local turbulence intensity at the tandem blade gap but also enhances the mesh quality in the numerical simulations.

Next, to further optimize the energy transfer characteristics and flow matching of the tandem impeller, a circumferential offset was introduced for the rear blade. The offset was always oriented toward the PS, allowing the fluid from the PS of the front blade to smoothly pass through the gap and attach to the SS of the rear blade. As illustrated in Fig. 3.9, the circumferential offset of the rear blade can be represented by the dimensionless parameter D , which is defined as:

$$D = \frac{\Delta u}{r_m} \geq 0.08 \quad (3.18)$$

In this definition, Δu represents the arc length of the circumferential offset of the rear blade relative to the front blade, and r_m is the radius at the gap position of the blade. When $D=0$, the front and rear blades are perfectly aligned in projection. As D increases, the rear blade gradually shifts in the direction of rotation, thereby modifying the fluid interaction region between the two blades. Preliminary flow analysis indicates that when the offset is too small ($D < 0.08$), the flow in the gap is mainly concentrated on the SS, which limits its participation in the energy conversion process within the rear blade channel, leading to increased local recirculation and losses. Therefore, by adjusting D , the flow pattern in the gap can be effectively controlled to achieve a balance between energy transfer and hydraulic loss. Subsequent chapters will present a comparative analysis of the flow characteristics and pump performance under different D values to determine the optimal

offset configuration.

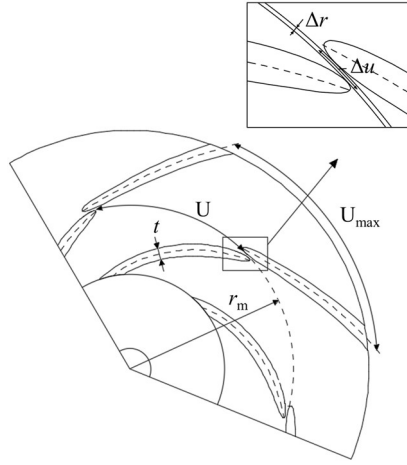


Fig. 3.9. The offset design of the tandem impeller rear blade.

3.3 Optimization of Tandem Impeller Rear Blades

The presence of the gap in the tandem impeller blades inevitably disrupts the continuity of the internal flow pattern and increases hydraulic losses. To improve the flow matching between the front and rear blades, reduce impact losses, and enhance the hydraulic performance of the pump, this section introduces an optimization approach that takes the inlet attack angle of the rear blade β_{21} as the primary optimization variable. As illustrated in Fig. 3.10, the proposed optimization procedure is based on the velocity triangle method, which is used to evaluate and predict the flow angle at the rear blade inlet. During the entire optimization process, only β_{21} is adjusted, while the other geometric angle distributions remain consistent with those of the conventional impeller design.

The design of the β_{21} is based on the velocity triangle method. First, the velocity triangle at the rear blade inlet is constructed using the circumferential velocity u at the mid-gap radius r_m and the outlet velocity components at the TE of the front blade. During the design process, the flow is assumed to be incompressible and steady, while vortex effects are neglected to simplify the calculation. In this study, the average meridional velocity of the flow entering the rear blade, denoted as c_{m21} is estimated. To begin, the representative flow area A_F at the rear blade inlet section must be determined, which can be calculated as follows:

$$A_F = U_D \times b_m \times \left(1 - \frac{t_B}{U_D} \right) \quad (3.19)$$

where b_m represents the channel width at the gap position of the blade, t_B is the blade thickness at the gap, and U_D denotes the local circumferential pitch at the gap radius. The value of U_D can be calculated using the number of blades z and the radius r_m , as expressed by the following equation:

$$U_D = \frac{2\pi \cdot r_m}{Z} \quad (3.20)$$

next, based on the continuity equation, the average c_{m21} through this cross-section can be determined, defined as:

$$c_{m21} = \frac{Q}{Z \cdot A_F} \quad (3.21)$$

Finally, under the assumption of no pre-swirl, the β_{21} can be determined from the velocity triangle relationship shown in Fig. 3.10, as follows:

$$\beta_{21} = \arctan\left(\frac{c_{m21}}{2\pi \cdot r_{21} \cdot n - c_{u21}}\right) \quad (3.22)$$

To further investigate the influence of the β_{21} on the flow characteristics, a series of computational cases with different β_{21} values were established based on the aforementioned method for detailed analysis. The radial distribution of the rear blade angle was designed to vary approximately linearly, while maintaining a consistent outlet angle to ensure overall flow continuity throughout the impeller. By comparing the results from numerical simulations and experimental measurements across different configurations, the impact of varying β_{21} on the pump performance curves will be comprehensively evaluated.

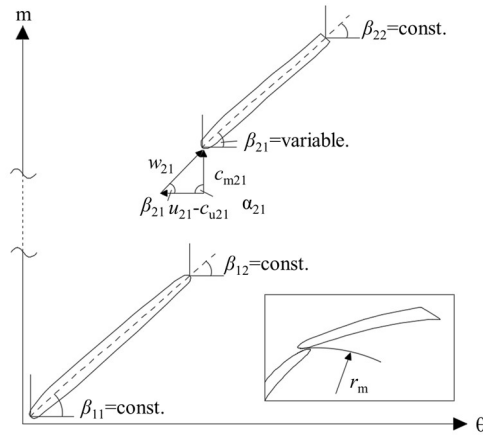


Fig. 3.10. The inlet blade design of the tandem impeller rear blade.

4 Experimental Setup and Test Methods

To accurately measure the performance of the centrifugal pump and verify the influence of the tandem impeller on performance, this research conducted experimental measurements on a centrifugal pump with different impellers. A closed hydraulic test loop for the centrifugal pump is designed and constructed in this chapter, and a test centrifugal pump with $n_q=25$ is designed to be connected to this loop. The convenient design allows for quick impeller replacement, enabling performance measurements of the pump with different impeller configurations. The experimental measurements focus on external characteristic parameters such as pump H , η , and P , while also providing precise validation for numerical simulation results.

4.1 Experimental Loop Design

A closed experimental loop for the centrifugal pump was constructed in the laboratory of the SAM at the RPTU. The 3D structure of the constructed closed experimental loop is shown in Fig. 4.1, with blue arrows indicating the flow direction of water. A more detailed experimental pipeline and instrument layout is shown in Fig. 4.2. This test system enables measurement of centrifugal pump performance parameters under multiple speeds and operating conditions. It primarily consists of a water tank, a centrifugal pump, variable-frequency motor drive system, measurement instruments, and the data collection system. The water tank provides circulating water flow to the system and allows flexible adjustment of liquid level and system pressure through its two attached valves. The system pressure is monitored by a pressure meter connected to the tank. The test pump is mounted on a horizontal plane and driven by an electric motor via a coupling. An frequency converter is connected to the motor to enable flexible speed adjustment. Absolute pressure sensors are installed upstream and downstream of the pump to measure the pressure difference and calculate the head. A torque meter installed between the pump and motor to measure torque and n . Flow is obtained through an electromagnetic flow meter installed on the pressure pipe. An electronic control valve is also installed on the downstream part of the pipe, enabling flexible adjustment of pump operating conditions. The output signals from all sensors and actuators are connected to a computer via a data processing system and integrated using the 2019 version of the LABVIEW program. This program can control the opening of electronic control valve and record measurement data. When replacing the impeller, simply close the throttle and open the drain valve to empty the flow in the test section piping. Next, adjust the expansion joint to remove the inlet pipe. Ultimately, the impeller can be quickly disassembled and reassembled.

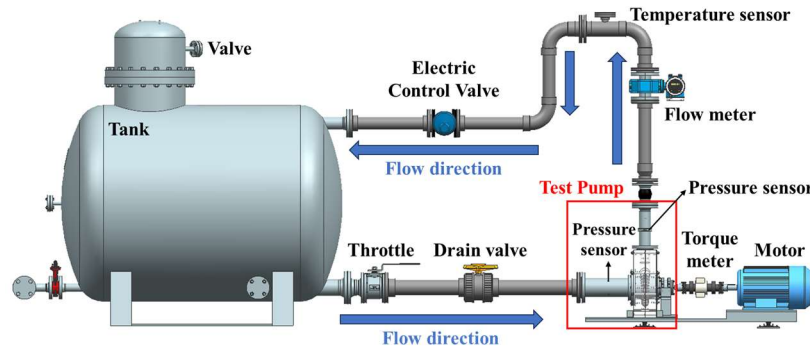


Fig. 4.1. 3D experimental system loop.

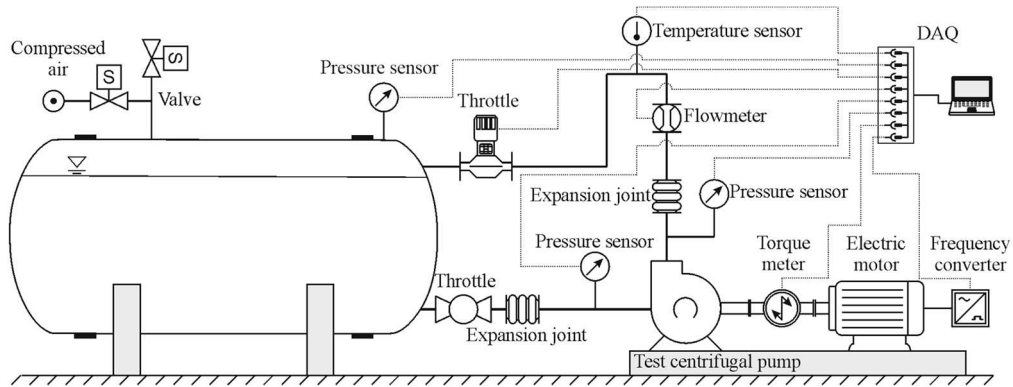


Fig. 4.2. 2D experimental loop and instruments layout diagram.

4.2 Structural Design of the Test Pump

Based on the hydraulic model of the centrifugal pump with $n_q=25$ designed in the previous chapter, for this research a modular-structured centrifugal pump was designed for measurement purposes. This pump accommodates the convenient installation and disassembly of different impellers during experiments while ensuring structural compatibility. The overall structure is shown in Fig. 4.3. The test pump primarily consists of an inlet section, pump section, and outlet section, featuring a compact and stable overall structural layout. The inlet direction remains horizontal, and the outlet direction remains vertical. The inlet and outlet pipes maintain the 90° , facilitating connection to the experimental loop. A centrifugal impeller is installed in the internal cavity of the pump section, and the impeller is connected to the pump shaft in the horizontal direction. The inlet pressure measurement holes of the pump are evenly distributed in four directions along the mid-section of the inlet pipe. The outlet pressure measurement holes are positioned in the mid-section of the outlet pipe, with pressure collection rings welded around the periphery of holes.

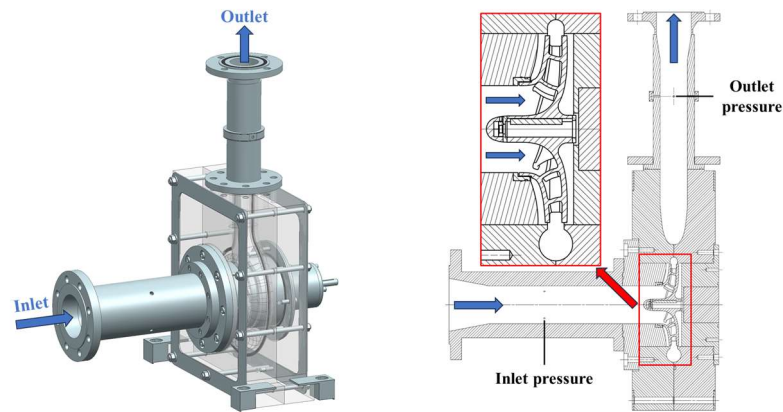


Fig. 4.3. Test pump general structure.

The structural disassembly of the components of the test pump is shown in Fig. 4.4. The inlet section primarily consists of an aluminum inlet pipe and inlet assembly, facilitating overall disassembly, assembly, and positioning. An aluminum sealing ring is fixed on the inlet component, maintaining a slight gap between the sealing ring and the impeller shroud. This design not only ensures no mechanical contact friction during impeller rotation but also allows liquid in the front side room to flow back through the gap to the impeller inlet, which effectively reduces axial force. The inlet pipe, inlet components, and sealing ring feature a modular design. As an integrated assembly, the whole unit can be disassembled horizontally along the pump shaft (as indicated by the red arrow), enabling rapid impeller replacement and structural adjustments. This enhances experimental η and improves the maintainability of the pump unit. The pump section adopts a symmetrical split-case design, consisting of the volute suction side and the volute hub side. A sealing ring is positioned between these two parts to prevent fluid leakage during operation. To enable visual internal flow observation, the pump casing material is selected as PMMA GS transparent acrylic, which offers excellent mechanical strength and optical transmittance. This symmetrical split-case design also significantly simplifies the manufacturing process for the volute flow channel. The outlet section primarily consists of an outlet pipe and flange plate vertically mounted above the pump unit, both made of stainless steel to enhance the overall structural stability and sealing integrity. The substantial weight of stainless steel components help establish more reliable contact pressure between the pump unit and outlet section, which improves sealing performance at the connection and enhances the vibration resistance of the pump during operation. Meanwhile, standardized flange interfaces also facilitate the connection of flow meters and pressure measurement equipment.

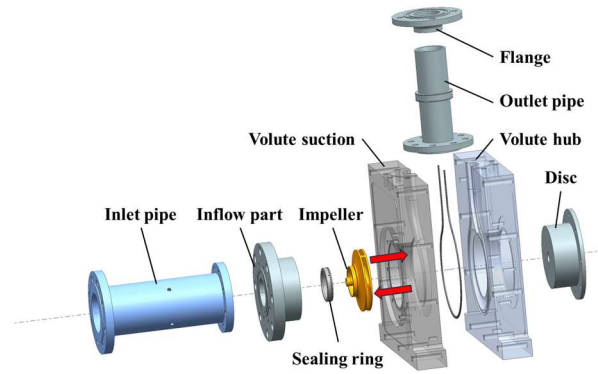


Fig. 4.4. Test pump structural decomposition.

Two test impeller structures are designed in this research, as shown in Fig. 4.5, which are the conventional impeller and the tandem-blade impeller. Both share identical configurations, consisting of three components, namely the shroud, twisted blades, and hub. The blades are positioned between the shroud and hub plates to form a sealed flow channel. All test impellers in this research were exported as STL files based on CAD modeling and manufactured using MJP multi-jet polymer 3D printing technology to ensure geometric accuracy and repeatability. As shown in fig. 4.5, the MJP process offers high resolution and excellent surface finish, enabling accurate reproduction of the 3D contours of twisted blades. This ensures high precision in blade design parameters and clearance size. Additionally, the photopolymer resin material used in MJP printing exhibits excellent size stability, sufficient rigidity, and adequate water resistance. It is suitable for long-term utilization in the low-pressure hydraulic testing environment adopted in this research, which can ensure the structural stability of the impeller during operation and the consistency of experimental data.

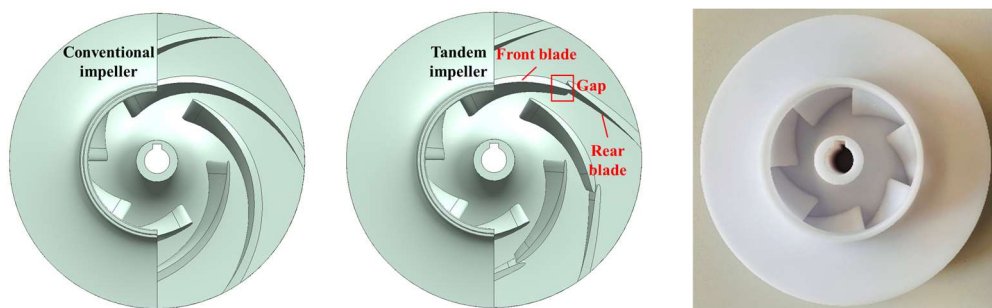


Fig. 4.5. Impeller structure models.

4.3 Measurement Parameters and Instruments

Based on the closed-loop system depicted in Fig. 4.1, multiple parameters including Q , pressure p , n , and T are measured and recorded in real time to comprehensively evaluate the performance indicators of the centrifugal pump. The corresponding instrument types,

technical indicators, and accuracy requirements for each measurement parameter in the test system are listed in Table 4.1. Among these, Q is directly read from an electromagnetic flowmeter manufactured by Endress+Hauser. Its measurement range is 0-180m³/h, with a measurement accuracy of $\pm 0.5\%$, capable of covering the flow rate requirements for all operating points of the test pump in this research. The p measurement employs two types of high-precision pressure sensors, supplied by ABB and Endress+Hauser respectively. Their measurement ranges are -2 to 2MPa and 0 to 1MPa, with measurement accuracies of $\pm 0.04\%$ and $\pm 0.2\%$, respectively. The pump outlet pressure $p_{\text{tot},2}$ and inlet pressure $p_{\text{tot},1}$ measured at the points shown in Fig. 4.3 can be used to calculate the H of the test pump based on Eq. 4.1.

To further evaluate the input energy and efficiency performance of the test pump, an integrated torque-speed sensor manufactured by KTR is integrated in the system, which can output both T and n signals simultaneously. The torque measurement range is -50 to 50N·m with an accuracy of $\pm 0.1\%$. The rotational speed measurement range is 0 to 10,000 r/min with an accuracy of $\pm 0.2\%$. This sensor is directly installed between the motor and the pump shaft. By measuring the actual T and n data, the P of the pump can be calculated based on Eq. 4.2. Finally, the operating η of the test pump can be further calculated based on the combination of the output hydraulic power $P_e = \rho \cdot g \cdot Q \cdot H$, as shown in Eq. 4.3.

$$H = \frac{\Delta p_{\text{tot},12}}{\rho \cdot g} = \frac{p_{\text{tot},2} - p_{\text{tot},1}}{\rho \cdot g} \quad (4.1)$$

$$P = \frac{2 \cdot \pi \cdot n \cdot T}{60} \quad (4.2)$$

$$\eta = \frac{P_e}{P} = \frac{60 \cdot \rho \cdot g \cdot Q \cdot H}{2 \cdot \pi \cdot n \cdot T} \quad (4.3)$$

Table 4.1. Measurement instrument list on the test rig.

Apparatus	Electromagnetic flowmeter	Pressure sensor	Pressure sensor	Torque meter	Speed sensor
Type	Endress+Hauser, E7186F19000	ABB 266MST	Cerabar M PMC41-RE12P1J11 M1	KTR DATAFL EX 16/50	KTR DATAFL EX 16/50
Parameter	Flow rate Q	Pressure p	Pressure p	Torque T	Rotational speed n
Unit	m ³ /h	MPa	MPa	N·m	r/min
Range	0-180	-2-2	0-1	-50-50	0-10000
Accuracy	$\pm 0.5\%$	$\pm 0.04\%$	$\pm 0.2\%$	$\pm 0.1\%$	$\pm 0.2\%$

4.4 Experimental Measurement Methods

Hydraulic performance tests for the centrifugal pump are conducted in the pipe circulation system shown in Fig. 4.1, following the direction of the blue arrows. Before starting the test, the impeller is installed on the shaft in the pump casing. After installing the inlet section, the throttle is opened and the sealing condition of the pipe system is checked. First, connect the power cables from the inverter cabinet to the power distributor and turn on the power for each electronic data collection device. Then, introduce compressed air into the water tank to boost pressure in the whole test pipe system. The pressure in the pipe system is controlled to reach 1.8bar by monitoring the absolute pressure sensor on the water tank. Next, open the vent valves on the pressure sensor connection lines at the inlet and outlet of the test pump to ensure all air has been completely purged from the sensor lines. Then, start the pre-programmed LABVIEW software and open the electronic control valve to over 90%. At this time, start the frequency converter cabinet and set the frequency to above 15Hz and maintain the motor in preheating operation for 15 minutes to protect the frequency converter cabinet from overheating. Subsequently, continue adjusting the frequency to the experimental motor speed specified in the test requirements. Before measurement begins, re-open the vent valves on the sensor lines and verify the absolute pressure of the line operation. During formal measurements, first open the electronic control valve to its maximum position. Subsequently, continuously reduce the valve opening via the LABVIEW program to conduct real-time measurements of experimental data. When the flow rate approaches its minimum value, the opening of the valve is gradually increased again to complete real-time data recording. Ultimately, the measurements allow for the two sets of performance parameter data for the test pump during the valves gradual closing and opening processes, enabling the examination of potential timing effects. Specifically, after each adjustment of the valve opening, wait at least 2 minutes for the test pump to stabilize before starting to record data to eliminate measurement errors caused by unstable pump operation. For each impeller, the above experimental measurements are repeated three times to further eliminate random errors. It is important to note that before replacing the next set of impellers for measurement, it is necessary to wait 30 minutes before restarting the frequency converter to prevent damage caused by overcharging of the capacitor.

4.5 Data Processing Methods and Uncertainty Analysis

The pump performance curve for each impeller is measured at a constant n . For each impeller, two sets of performance curves are measured, corresponding to the conditions of gradually opening and closing the valve, respectively. Each map is measured at least 12 different flow conditions. The target H and η are automatically calculated in the data acquisition program LABVIEW based on Eqs. 4.1 and 4.3. Each sensor is sampled at a frequency of 5kHz within 10 seconds during measurements, and the average value is

calculated. In this experiment, when the actual test n deviates from the design speed n_{sp} , the measured parameters should be converted. When the speed difference is within 20%, the conversion equations for flow rate Q_T , head H_T , and efficiency η_T under the n_{sp} relative to the original measurement data are defined by Eqs. 4.4, 4.5, and 4.6.

$$\frac{Q_T}{Q} = \frac{n_{sp}}{n} \quad (4.4)$$

$$\frac{H_T}{H} = \left(\frac{n_{sp}}{n} \right)^2 \quad (4.5)$$

$$\eta_T = \eta \quad (4.6)$$

Although random errors are avoided to the greatest extent possible during the experiment, the measurement results are still influenced by various factors such as sensor accuracy and the pipe system. A reasonable assessment of the uncertainty in the experimental results can guarantee the reliability of the measurements. Thus, an uncertainty analysis of the experimental measurements is also carried out in this research, with specific steps in reference to the ANSI/ASME PTC 19.1-2019 standard [101, 102]. The uncertainty mainly includes random errors $s_{\bar{X}}$ in the measurement process and systematic errors $b_{\bar{X}}$ inherent to the sensor. Random errors are mainly attributed to factors such as the experimental environment, measurement methods, and readings. The $b_{\bar{X}}$ is mainly caused by the inherent uncertainty of the measurement instruments themselves. These two errors are defined by Eqs. 4.7 and 4.8, respectively.

$$s_{\bar{X}} = \frac{1}{\sqrt{N}} \cdot \sqrt{\sum_{j=1}^N \frac{(X_j - \bar{X})^2}{N-1}} \quad (4.7)$$

$$b_{\bar{X}} = \left[\sum_{m=1}^M (b_{\bar{X}_m})^2 \right]^{0.5} \quad (4.8)$$

The total uncertainty of the experimental measurement at the 95% confidence level is $U_{\bar{X}} = 2u_{\bar{X}}$, where $u_{\bar{X}}$ is defined as:

$$u_{\bar{X}} = \sqrt{(s_{\bar{X}})^2 + (b_{\bar{X}})^2} \quad (4.9)$$

where N denotes the sample count, X_j represents the sample variable value for each independent measurement, M indicates the number of factors contributing to systematic error, and $b_{\bar{X}_m}$ denotes the standard deviation estimate for the M th error source, which is related to the calibration of the sensor itself.

Ten sets of Q , H , and n data are collected within 10 seconds under three different Q and used for uncertainty analysis. The uncertainty parameters calculated according to Eqs. 4.7, 4.8, and 4.9 are listed in Table 4.2. Under 95% confidence level, the maximum error for Q measured in this experiment is $\pm 0.36 \text{ m}^3/\text{h}$, the maximum error for H is $\pm 0.14 \text{ m}$, and

the maximum error for n is ± 8.04 r/min. As a result, the measurements in this experiment exhibit high precision.

Table 4.2. Results of measurement uncertainty calculations.

Notation	Q (m ³ /h)	H (m)	n (r/min)
Average value $\bar{X}$	35.88/29.51/24.11	12.24/14.02/15.15	2007.26/2009.14/ 2011.11
Standard deviation	0.05/0.03/0.05	0.02/0.03/0.07	0.15/0.10/0.18
$s_{\bar{X}}$	0.02/0.01/0.02	0.01/0.01/0.03	0.05/0.04/0.06
$b_{\bar{X}}$	0.18/0.15/0.12	0.06/0.07/0.07	4.01/4.02/4.02
$u_{\bar{X}}$	0.18/0.15/0.12	0.06/0.07/0.07	4.01/4.02/4.02
$U_{\bar{X}}$	0.36/0.30/0.24	0.12/0.14/0.14	8.02/8.04/8.04
$\bar{X} \pm U_{\bar{X}}$	35.88 $\pm$ 0.36/	12.24 $\pm$ 0.12/	2007.26 $\pm$ 8.02/
	29.51 $\pm$ 0.30/	14.02 $\pm$ 0.14/	2009.14 $\pm$ 8.04/
	24.11 $\pm$ 0.24	15.15 $\pm$ 0.14	2011.11 $\pm$ 8.04

5 Numerical Simulation and Validation

Research on the performance and internal flow characteristics of the centrifugal pump would involve comparative analysis of multiple designs. With the rapid development of CFD technology, the reliability of numerical simulations is constantly improving, while simultaneously saving substantial resources and time that would otherwise be consumed by experimental research on each design. Therefore, numerical simulations of the test centrifugal pump are also predicted in this chapter. The physical model and key geometric parameters for numerical simulations are primarily introduced, and the computational process and simulation method validation are described.

5.1 Numerical Calculation Theory

CFD technology is a numerical method that effectively simulates the behavior of fluid flow, widely applied in the research of internal flow characteristics of fluid machinery. The key point is how to apply CFD technology reasonably and efficiently. The essence of CFD is to solve the fundamental governing equations of fluid mechanics, which include the continuity equation, momentum conservation equation, and energy conservation equation by numerical algorithms. Under specified boundary and initial conditions, the flow process in fluid machinery is quantitatively described. Currently, turbulence simulation primarily includes two major categories, Direct Numerical Simulation (DNS) and indirect numerical simulation [103]. Among these, DNS can fully resolve turbulent structures on all scales, with very high simulation accuracy. However, it consumes enormous computational resources, particularly dealing with complex geometries and high Reynolds number flows, which is difficult to meet the economic requirements of practical engineering applications [103, 104]. As a result, it is more appropriate to employ indirect numerical simulation methods for simulating typical complex internal flow in pump equipment. The RANS method is one of the most representative indirect numerical simulation approaches.

5.1.1 Governing Equations

The RANS method obtains closed averaged governing equations by performing temporal or spatial averaging of the instantaneous flow variables, and incorporates turbulence models for closure. This approach significantly reduces computational cost while maintaining acceptable accuracy, and has therefore been widely applied in engineering studies of pump internal flow simulations [105-107]. In the RANS framework, the Navier-Stokes (N-S) equations are subjected to Reynolds averaging, and the resulting Reynolds equations are solved to obtain the mean flow field parameters of turbulence. The density variations are also treated in an averaged sense, while the turbulent transport equations are time-averaged as well. For incompressible flow simulations, the governing

equations in the RANS approach include the conservation of mass and momentum. The conservation of mass, also referred to as the continuity equation, is defined according to the law of mass conservation as follows:

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho u_i)}{\partial x_i} = 0 \quad (5.1)$$

where ρ denotes the fluid density; t represents time; and u_i denotes the velocity tensor.

The momentum conservation equation, also known as the N-S equation, is defined in tensor form by the following equation:

$$\frac{\partial(\rho u_i)}{\partial t} + \frac{\partial(\rho u_i u_j)}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left(\mu \frac{\partial u_i}{\partial x_j} - \rho \overline{u'_i u'_j} \right) + S_i \quad (5.2)$$

where S_i denotes the generalized source term in the momentum conservation equation; μ is the kinetic viscosity; and $-\rho \overline{u'_i u'_j}$ is the Reynolds stress tensor.

5.1.2 Turbulence Model

For the Reynolds-averaged solution method, the velocity fluctuations u'_i cannot be directly solved. To close the momentum equation Eq.5.2, the Reynolds Stress tensor is linked to the velocity gradient based on the Boussinesq assumption, which is expressed as:

$$-\rho \overline{u'_i u'_j} = \mu_t \left(\frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i} \right) - \frac{2}{3} \delta_{ij} \left(\rho k + \mu_t \frac{\partial \bar{u}_k}{\partial x_k} \right) \quad (5.3)$$

where μ_t is the turbulent viscosity, k is the turbulent kinetic energy, and δ_{ij} is the Kronecker delta ($\delta_{ij}=1$ when $i=j$; $\delta_{ij}=0$ when $i \neq j$).

(1) Standard k - ω turbulence model

To further determine the turbulent viscosity μ_t , a variety of eddy-viscosity turbulence models have been developed. Among them, the standard k - ω model is an empirical approach modified based on the transport equations of turbulent kinetic energy and specific dissipation rate. This model is particularly suitable for simulating shear flow propagation at low Reynolds numbers. In the k - ω framework, the turbulent viscosity μ_t is defined using the turbulent kinetic energy k and the specific dissipation rate ω , as expressed by:

$$\mu_t = \rho \frac{k}{\omega} \quad (5.4)$$

Wilcox [108] first proposed the standard k - ω turbulence model and introduced additional terms into the equation for turbulent kinetic energy k to enhance the accuracy of the model. The definition equation for k is:

$$\rho \frac{\partial(k)}{\partial t} + \rho \frac{\partial}{\partial x_j} (u_j k) = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_{k1}} \right) \frac{\partial k}{\partial x_j} \right] + P_k - \beta' \rho k \omega + P_{kb} \quad (5.5)$$

The definition equation for the turbulence frequency ω is:

$$\frac{\partial(\rho\omega)}{\partial t} + \frac{\partial(\rho u_j \omega)}{\partial x_j} = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_{\omega 1}} \right) \frac{\partial \omega}{\partial x_j} \right] + \alpha_1 \frac{\omega}{k} P_k - \beta \rho \omega^2 + P_{\omega b} \quad (5.6)$$

where β' , α_1 , β , σ_{k1} , and $\sigma_{\omega 1}$ are constants with values of 0.09, 5/9, 0.075, 2 and 2, respectively; P_k represents the turbulent kinetic energy production rate due to turbulent viscous forces; P_{kb} and $P_{\omega b}$ represent the turbulent kinetic energy production rates due to buoyancy forces, respectively.

(2) SST k - ω turbulence model

For the aforementioned standard k - ω turbulence model, the inlet value of ω is often sensitive and may significantly affect the computational results. To address this limitation, Menter [109] proposed the Shear Stress Transport (SST) k - ω model in 1994. This model is capable of providing more accurate predictions of the onset and intensity of flow separation under adverse pressure gradients. In addition, cross-diffusion terms are incorporated into the model, which enhances its applicability both in the near-wall region and in the far-field, making it particularly suitable for flows with adverse pressure gradients. Unlike the standard k - ω model, the nonlinear damping functions are not employed in the SST k - ω model. Instead, the original k - ω formulation is retained for the near-wall region, while the k - ε model is applied in the far-field region. The corresponding turbulent viscosity μ_t is defined as:

$$\mu_t = \frac{\rho a_1 k}{\max(a_1 \omega, SF_2)} \quad (5.7)$$

The definition equation for the turbulent kinetic energy k in the SST k - ω turbulence model is:

$$\frac{\partial(\rho k)}{\partial t} + \frac{\partial(\rho \bar{u}_j k)}{\partial x_j} = P_k - \beta' \rho k \omega + \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_{k2}} \right) \frac{\partial k}{\partial x_j} \right] \quad (5.8)$$

The definition equation for the turbulence frequency ω is:

$$\frac{\partial(\rho\omega)}{\partial t} + \frac{\partial(\rho \bar{u}_j \omega)}{\partial x_j} = \alpha_2 S^2 - \beta_2 \rho \omega^2 + \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_{\omega 2}} \right) \frac{\partial \omega}{\partial x_j} \right] + 2(1 - F_1) \rho \frac{1}{\omega \sigma_{\omega 2}} \frac{\partial k}{\partial x_j} \frac{\partial \omega}{\partial x_j} \quad (5.9)$$

where a_1 is set to 0.31; F_1 is the hybrid function that realizes the conversion between the k - ω and k - ε models; S is the strain rate tensor. Concurrently, a constraint is imposed on the vorticity-viscosity equation, defined as:

$$v_t = \frac{\mu_t}{\rho} \quad (5.10)$$

In the above equations, each constant is expressed using a hybrid form of internal and external constants, with the specific equations defined as follows:

$$\varphi = \varphi_1 F_1 + \varphi_2 (1 - F_1) \quad (5.11)$$

where φ_1 is an internal constant; φ_2 is an external constant.

The hybrid function establishes the relationship between the near-wall distance and flow variables, which determines the success of the SST k - ω turbulence model in computational calculations. The definition equations for the hybrid function and closure function are:

$$F_2 = \tanh \left[\left[\max \left(\frac{2\sqrt{k}}{\beta' \omega y}, \frac{500\nu}{y^2 \omega} \right) \right]^2 \right] \quad (5.12)$$

$$P_k = \min \left(\tau_{ij} \frac{\partial u_i}{\partial x_j}, 10\beta' k \omega \right) \quad (5.13)$$

$$F_1 = \tanh \left\{ \left[\min \left[\max \left(\frac{\sqrt{k}}{\beta' \omega y}, \frac{500\nu}{y^2 \omega} \right), \frac{4\sigma_{\omega 2} k}{CD_{k\omega} y^2} \right] \right]^4 \right\} \quad (5.14)$$

$$CD_{k\omega} = \max \left(2\rho\sigma_{\omega 2} \frac{1}{\omega} \frac{\partial k}{\partial x_i} \frac{\partial \omega}{\partial x_i}, 10^{-10} \frac{500\nu}{y^2 \omega} \right) \quad (5.15)$$

where a_1 is 0.31; α_1 is 5/9; α_2 is 0.44; β_1 is 3/40; β_2 is 0.0828; β' is 0.09; σ_{k1} is 0.85; σ_{k2} is 1; $\sigma_{\omega 1}$ is 0.5; $\sigma_{\omega 2}$ is 1/0.856.

(3) RANS/LES Hybrid Model

Due to its high computational efficiency and acceptable accuracy, the RANS turbulence model offers a favorable balance between cost and performance in engineering applications. However, since it is based on Reynolds time-averaging, its capability to resolve the detailed structures of the real turbulence field is limited, making it insufficient for in-depth investigations of local turbulent structures in centrifugal pumps. In capturing the unsteady force characteristics of centrifugal pumps, the RANS method can identify dominant frequency components in the spectrum, but its ability to resolve broadband or non-periodic features remains inadequate [110]. In contrast, LES provides a more accurate representation of complex turbulent structures. Nevertheless, for high-Reynolds-number three-dimensional rotating flows, such as those in centrifugal pumps, LES requires extremely fine grids and small time steps, leading to a significant increase in computational cost. To achieve a balance between accuracy and efficiency, the present study employs a hybrid RANS/LES model for analyzing the flow and excitation characteristics of centrifugal pumps, which ensures sufficient resolution of local flow details while effectively controlling the computational expense.

The baseline model for the RANS/LES hybrid approach is the Detached Eddy Simulation (DES) model [111]. The fundamental concept is to achieve a smooth transition between RANS and LES solutions by comparing the turbulence length scale L_t with the grid spacing $\Delta_{\max}$. Typically, the RANS formulation is applied in the boundary layer regions of the flow field, while LES is employed in the core regions with flow separation. The DES model based on the SST k - ω framework is realized by modifying the turbulent

kinetic energy dissipation term in Eq. 5.8, expressed as $D_k = \beta' \rho k \omega$. The modified dissipation term is given as:

$$D_k = \beta' \rho k \omega F_{DES} \quad (5.16)$$

where the factor F_{DES} is further expressed as:

$$F_{DES} = \max\left(\frac{L_t}{C_{DES} \Delta_{\max}}, 1\right) \quad (5.17)$$

where, $C_{DES} \Delta_{\max}$ is the LES subgrid model filtering scale, $C_{DES}=0.65$, $\Delta_{\max}$ is the locally maximum grid size, L_t is the turbulence length scale, defined as:

$$L_t = \sqrt{k} / \beta' \omega \quad (5.18)$$

In the near-wall region, where $L_t < C_{DES} \Delta_{\max}$ and $F_{DES}=1$, the turbulent kinetic energy dissipation term D_k is identical to that in the SST $k-\omega$ model, i.e., DES is solved using the RANS model. In the turbulent core region far from the wall, where $L_t > C_{DES} \Delta_{\max}$, Eq. 5.16 transforms to:

$$D_k = \frac{\rho k^{\frac{3}{2}}}{C_{DES} \Delta_{\max}} \quad (5.19)$$

At this case, DES employs the LES model for solution.

5.2 Numerical Calculation Model Construction

In this study, a combined approach of numerical simulation and experimental test was employed to conduct a systematic investigation of the centrifugal pump designed in Chapter 3. The numerical simulations were performed on the complete pump structure, with the computational fluid domain constructed strictly according to the pump design specifications presented in Chapter 3, ensuring a high level of consistency with the experimental model described in Chapter 4. The computational model primarily consists of key components such as the impeller, volute, and shroud and hub side rooms. To eliminate boundary interference and ensure computational accuracy, straight pipe sections with a length of five times the impeller diameter D_j were extended at the impeller inlet and the volute outlet, respectively. It is worth noting that in the tandem impeller configuration, the front and rear blade rows were treated as two independent computational domains. This strategy not only facilitates mesh generation and local refinement but also enables the implementation of circumferential offsets between the rear blades and the front blades. The overall numerical model is illustrated in Fig. 5.1.

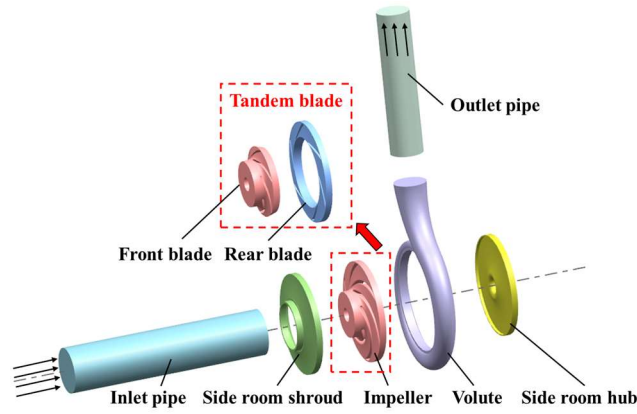


Fig. 5.1. Numerical simulation computational domains.

5.3 Grid Research

The accuracy and reliability of numerical simulation results depend heavily on the quality of the computational grid. The computational model structure in this paper is relatively complex and employs the DES turbulence simulation method, so the accuracy of the computational results is closely related to the grid quality.

5.3.1 Grid Generation

The quality of grid discretization directly affects both the computational efficiency and the accuracy of numerical simulations. Therefore, in this study, a carefully designed meshing strategy was adopted. The entire computational domain was discretized using structured hexahedral grids to ensure a balance between capturing complex geometrical features and maintaining numerical stability in turbulence simulations. The impeller region was meshed with ANSYS TurboGrid, which offers strong adaptability to rotating impeller passages and high accuracy in boundary layer control, making it well suited for generating high-quality meshes of complex impeller surfaces. For the volute, shroud and hub side rooms, as well as the inlet and outlet extension sections, structured block-structured meshes were generated using ANSYS ICEM CFD, providing controllable grid topology and flexibility for local refinement. All near-wall regions of the solid boundaries were refined locally. In particular, multiple boundary-layer mesh layers were deployed along the impeller blade surfaces, the volute tongue, and the volute outlet where pressure gradients are steep, ensuring sufficient resolution of near-wall flow structures. To meet the near-wall resolution requirements of turbulence models (especially the DES model), the first-layer grid spacing in the wall-normal direction was strictly controlled, keeping the average y^+ value below 5, thereby ensuring accurate prediction of wall shear stresses. Meanwhile, the maximum growth rate of the boundary-layer mesh was limited to 1.3 to avoid abrupt variations in the wall-normal direction, which enhances numerical stability. In the tandem impeller configuration, the front and rear blades were treated as two

independent zones while maintaining the same level of mesh refinement. In particular, approximately uniform mesh node density was ensured at the interface between the two blades to improve the continuity and accuracy of flow field data exchange. Furthermore, to guarantee overall mesh quality, the face angles of all elements throughout the computational domain were controlled within the range of 20° - 160° , thereby avoiding numerical errors associated with excessively acute or obtuse angles. The overall mesh structure and the local mesh details in critical regions are illustrated in Fig. 5.2.

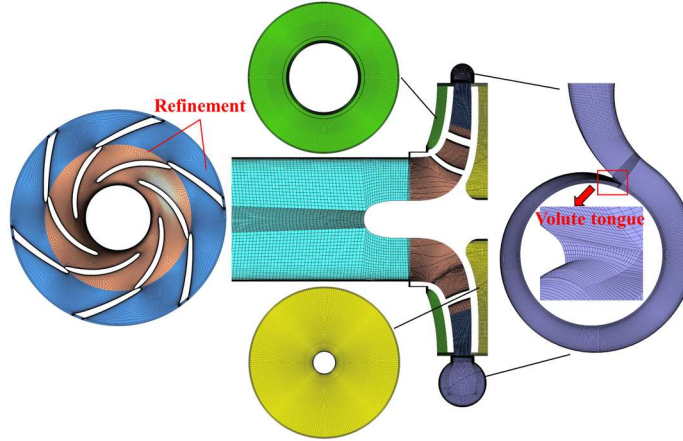


Fig. 5.2. Mesh discretization of the full model.

5.3.2 Grid Independence Research

The accuracy of numerical simulations is influenced by multiple factors, among which mesh quality and quantity are the most critical. While mesh quality was ensured during the discretization process, mesh quantity must be validated through grid independence verification. Therefore, three sets of meshes with different densities were established in this study. Considering that the impeller and volute are the core energy conversion components of the centrifugal pump, mesh refinement was only applied to these two parts. The details of the mesh schemes are listed in Table 5.1. The Grid Convergence Index (GCI) method was employed to perform the grid independence analysis, with the pump H and η under three operating conditions selected as evaluation indicators. Following the procedure described in Ref. [112, 113], the approximate relative error e_a^{32} between the medium and coarse meshes, the extrapolated relative error e_{ext}^{32} , and the fine-grid convergence index GCI_{medium}^{32} were calculated.

First, establish three sets of grids with varying densities based on the grid scale parameter h , which are noted as Scheme 1 (Fine), Scheme 2 (Medium), and Scheme 3 (Coarse). These must satisfy the grid refinement ratios $r_{21}=h_2/h_1$ and $r_{32}=h_3/h_2$, both of which must exceed 1.3. The grid scale parameter h is defined as follows:

$$h = \left[\frac{1}{N} \sum_{i=1}^N (\Delta V_i) \right]^{\frac{1}{3}} \quad (5.20)$$

where, ΔV_i is the volume of the i -th mesh cell, and N is the total number of mesh cells in the computational domain.

Then, calculate the GCI level based on the following equations:

$$p = \frac{1}{\ln(r_{21})} \left| \ln |\varepsilon_{32} / \varepsilon_{21}| + q(p) \right| \quad (5.21)$$

$$q(p) = \ln \left(\frac{r_{21}^p - s}{r_{32}^p - s} \right) \quad (5.22)$$

$$s = 1 \cdot \text{sgn}(\varepsilon_{32} / \varepsilon_{21}) \quad (5.23)$$

where, $\varepsilon_{21} = \phi_2 - \phi_1$, $\varepsilon_{32} = \phi_3 - \phi_2$, ϕ_k represents the computational results of the flow parameters based on the k -th set of grids.

Finally, the error estimation between different grid schemes can be calculated based on Eqs. 5.24-5.27, with the extrapolated value defined as:

$$\phi_{\text{ext}}^{32} = (r_{32}^p \phi_2 - \phi_3) / (r_{32}^p - 1) \quad (5.24)$$

The approximate relative error is defined as:

$$e_a^{32} = \left| \frac{\phi_3 - \phi_2}{\phi_2} \right| \quad (5.25)$$

The relative error of the extrapolated value is defined as:

$$e_{\text{ext}}^{32} = \left| \frac{\phi_{\text{ext}}^{32} - \phi_1}{\phi_{\text{ext}}^{32}} \right| \quad (5.26)$$

The fine-grid convergence index is defined as:

$$\text{GCI}_{\text{medium}}^{32} = \frac{1.25 e_a^{32}}{r_{32}^p - 1} \quad (5.27)$$

Table 5.2 presents the values of H and η obtained from steady simulations under three operating conditions ($0.8Q_{\text{des}}$, $1.0Q_{\text{des}}$, and $1.2Q_{\text{des}}$), together with the corresponding GCI parameters for the three mesh schemes. By comparing the results of different mesh schemes, it can be clearly observed that as the mesh density increases, the numerical solutions become more stable and the errors decrease significantly, demonstrating a favorable trend of grid convergence. Specifically, under the overload condition ($1.2Q_{\text{des}}$), the maximum $\text{GCI}_{\text{medium}}^{32}$ for η is 1.21%, while under the design and part-load operating conditions, both H and η exhibit $\text{GCI}_{\text{medium}}^{32}$ values below 1%. This indicates that the numerical results exhibit satisfactory grid independence across all operating conditions.

Therefore, the medium mesh scheme is considered to have reached the asymptotic convergence region and meets the criterion for grid convergence determination [112].

Table 5.1. Mesh independence research scheme information. (Unit: million)

Mesh scheme	Inlet pipe	Impeller	Volute	Side rooms	Outlet pipe	Total
Fine 1	1.279	5.708	4.554	1.111	0.658	13.310
Medium 2	1.279	3.109	2.461	1.111	0.658	8.618
Coarse 3	1.279	0.799	1.036	1.111	0.658	4.884

Table 5.2. Mesh discretization uncertainty analysis.

Q/Q_{des}	Parameter	Fine 1	Medium 2	Coarse 3	e_a^{32}	e_{ext}^{32}	GCI_{medium}^{32}
0.8	H (m)	15.55	15.70	15.66	0.268%	0.029%	0.037%
	η (%)	76.27	76.41	76.67	0.346%	0.280%	0.349%
1.0	H (m)	14.61	14.64	14.63	0.101%	0.031%	0.039%
	η (%)	79.87	79.94	79.67	0.341%	0.108%	0.135%
1.2	H (m)	12.90	12.94	12.97	0.200%	0.088%	0.110%
	η (%)	78.79	78.90	78.77	0.159%	0.958%	1.210%

Fig. 5.3 illustrates the pressure load distributions at the mid-span of the impeller blades under different mesh schemes. It can be observed that the influence of mesh density on the pressure load distribution is quite limited, with only slight differences appearing near the LE and TE of the blades. As the mesh density increases, the pressure load distributions of the fine and medium mesh schemes become nearly identical. Therefore, based on the results of grid discretization error analysis and pressure load distributions, and by comprehensively considering both computational accuracy and efficiency, the medium mesh (Scheme 2) was ultimately selected for further numerical simulations of the centrifugal pump. The total number of elements in this scheme is 8.62 million. Fig. 5.4 presents the y^+ distribution on the impeller blades under Scheme 2. It can be seen that the y^+ values are slightly higher in some regions near the LE, while relatively low values are maintained on both the PS and SS at the mid-span of the blades. Overall, the average y^+ values on both the PS and SS remain below 5, which satisfies the computational requirements of the SST $k-\omega$ and DES models.

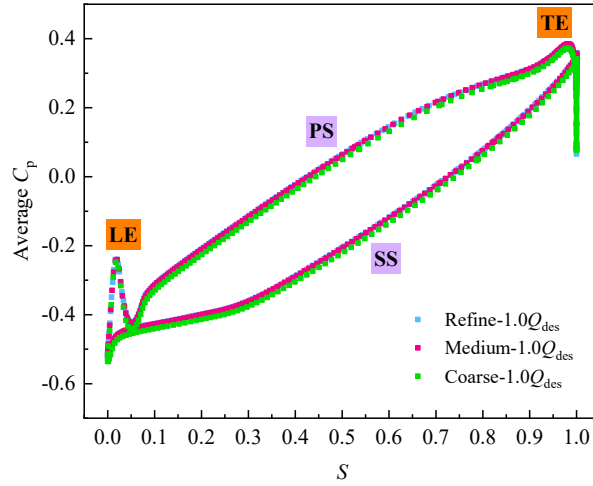


Fig. 5.3. Pressure distribution at midspan of impeller blades under different mesh schemes.

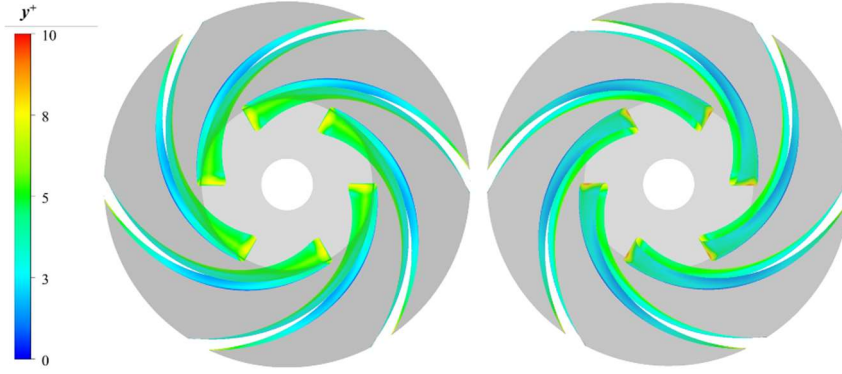


Fig. 5.4. Average y^+ value distribution on impeller blades.

5.4 Numerical Calculation Settings

In this study, both steady and unsteady numerical simulations of the centrifugal pump were performed under different conditions to comprehensively capture its performance characteristics and transient response behavior. All simulations were conducted on the high-performance computing cluster Elwetritsch at Rheinland-Pfälzische Technische Universität Kaiserslautern-Landau (RPTU), which is part of the Alliance for High Performance Rhineland-Palatinate (AHRP). The numerical simulations were carried out using the software ANSYS CFX 2022R2, with the governing equations discretized by the finite volume method (FVM). To ensure comparability across different cases, all simulations were performed with consistent interface treatments, turbulence model selections, and wall function settings. This section provides a detailed description of the boundary conditions imposed in the computational domain, as well as the validation method for the selected time step.

5.4.1 Boundary Conditions

In this study, steady simulations were first performed to obtain the preliminary distribution of the flow field. The results of the steady simulations were then used as the initial field for subsequent unsteady simulations, thereby improving the convergence efficiency and stability of the transient calculations. During the solution of the governing equations, the diffusion terms were discretized using a central-difference scheme, while the convection and dissipation terms were discretized with a second-order upwind scheme, ensuring a balance between computational accuracy and numerical stability. For the steady simulations, the SST $k-\omega$ turbulence model described in Section 5.1.2 was employed, which is well suited for handling boundary layer flows and flow separation. The unsteady simulations adopted the DES model to more accurately capture the transient disturbance characteristics and excitation responses of the unsteady flow in the pump.

The specific boundary conditions are illustrated in Fig. 5.5. At the inlet, a mass flow boundary condition was applied with an additional 5% turbulence intensity to mimic the unstable disturbances under realistic conditions. At the outlet, a pressure boundary condition was imposed, with the pressure fixed at standard atmospheric pressure (1atm). In the steady simulations, the interfaces between the impeller and the inlet pipe, shroud and hub side rooms, and volute were treated using the Frozen Rotor model, which simplifies the relative motion between rotating and stationary components. By contrast, in the unsteady simulations, these interfaces were defined as Transient Rotor-Stator Interfaces, allowing periodic data exchange between the rotating and stationary domains and thereby reproducing the transient flow process more realistically.

For the tandem impeller configuration, since the front and rear impellers operate at the same n , they can be regarded as relatively stationary with respect to each other. Consequently, the interface between them was simplified as a stationary-domain connection, which avoids unnecessary mesh sliding treatments. Furthermore, the time advancement in the unsteady solver was carried out using a second-order Backward Euler scheme for temporal discretization, while the spatial terms were discretized using the high-resolution scheme to minimize numerical dissipation while maintaining numerical stability. The total simulated time for the unsteady computation was 0.36s, corresponding to 12 complete revolutions of the impeller. The pump H and η were monitored during the simulation, and the solution was considered converged once a stable periodic behavior was observed, or when the root-mean-square (RMS) residual dropped below 1×10^{-5} . To ensure the representativeness and stability of the data, the results used for flow pattern analysis were obtained as the time-averaged data over the last two impeller revolutions in the unsteady simulations.

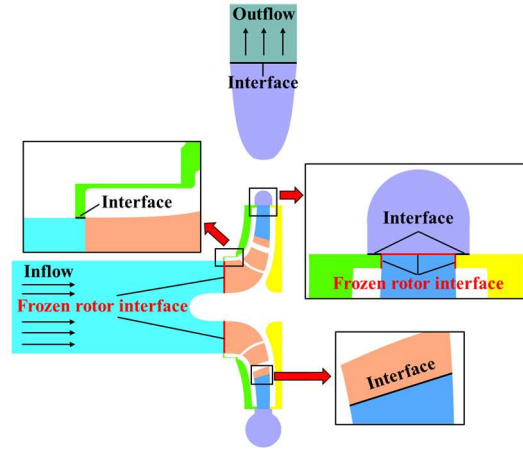


Fig. 5.5. Boundary condition settings.

5.4.2 Time Step Verification

In unsteady simulations, the choice of time step is also a critical factor that can potentially influence the simulation accuracy. Therefore, three different time steps were selected in this study to investigate their effects on the simulation results. The three time-steps were $\Delta t_1 = 2.5 \times 10^{-4} \text{s}$, $\Delta t_2 = 1.67 \times 10^{-4} \text{s}$, and $\Delta t_3 = 8.33 \times 10^{-5} \text{s}$, which correspond to the time required for the impeller to rotate by 3° , 2° , and 1° , respectively. Fig. 5.6 shows the monitored values of H and η during the 6th to 8th revolutions of the impeller under the design condition for the different time steps. As shown in the figure, both H and η exhibit distinct periodic fluctuations at all three time-steps, with the curves almost overlapping. A closer inspection reveals slight deviations at curve inflection points: the η curve at Δt_2 shows minor discrepancies, and the H curve at Δt_1 also deviates slightly. Consequently, Δt_3 was ultimately selected as the time step for unsteady simulations, corresponding to 360 calculation steps per impeller revolution. Under Δt_3 , the Courant number in most flow regions remains below 1, thereby satisfying the stability requirements for unsteady simulations.

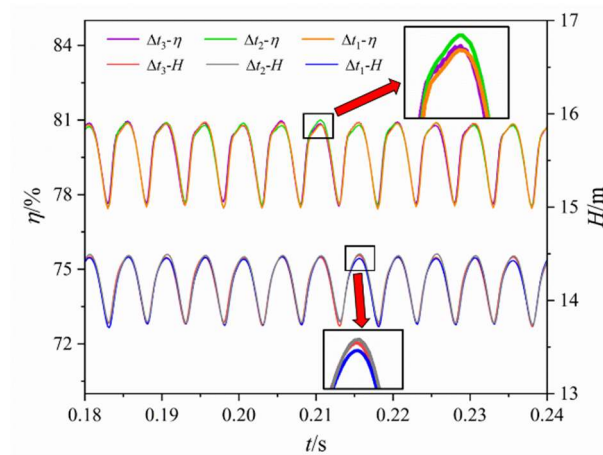


Fig. 5.6. Time step validation.

5.5 Numerical Simulation Accuracy Verification

To validate the accuracy of the numerical simulation method adopted in this chapter, the external characteristic parameters of the centrifugal pump obtained from simulations were compared with the corresponding experimental measurements. The performance comparison results for the conventional impeller are shown in Fig. 5.7, while those for the tandem impeller are presented in Fig. 5.8. As the Q decreases, the H of both impellers gradually increases, but a decline is observed under extremely low flow rates. This phenomenon is attributed to severe flow instabilities, such as backflow, pre-rotation, and flow separation, which cause a sharp reduction in the energy conversion capability of the pump. Under such conditions, the pump η also drops rapidly to a relatively low value. For both impeller types, the pump η reaches its maximum near the Q_{des} and maintains a relatively wide high- η region. However, η decreases at both high and low flow rates, with a notably faster decline under small flow rates. The P monotonically increases with Q . The figures also present the variation curves of H , η , and P obtained from unsteady simulations, where the data were averaged over the last two revolutions of the impeller. The overall trends of the CFD results show excellent agreement with the experimental data, with relative deviations of approximately 3% for H and about 5% for η and P . These deviations are indicated by error bars in the figures. From the error distribution, it can be observed that most simulation results fall within the experimental uncertainty range. Compared with experimental values, the simulations tend to slightly overpredict H and η while underpredicting P . The primary reasons are that non-fluid losses, such as wall roughness, bearing friction, and mechanical seal effects were not considered in the simulations. Moreover, idealized geometry and fluid properties were employed in the numerical model, whereas the experiments inherently involve assembly tolerances and property variations, leading to minor discrepancies in performance. In summary, the numerical results demonstrate good consistency with experimental data, thereby verifying the accuracy and reliability of the simulation approach employed in this research for predicting centrifugal pump performance.

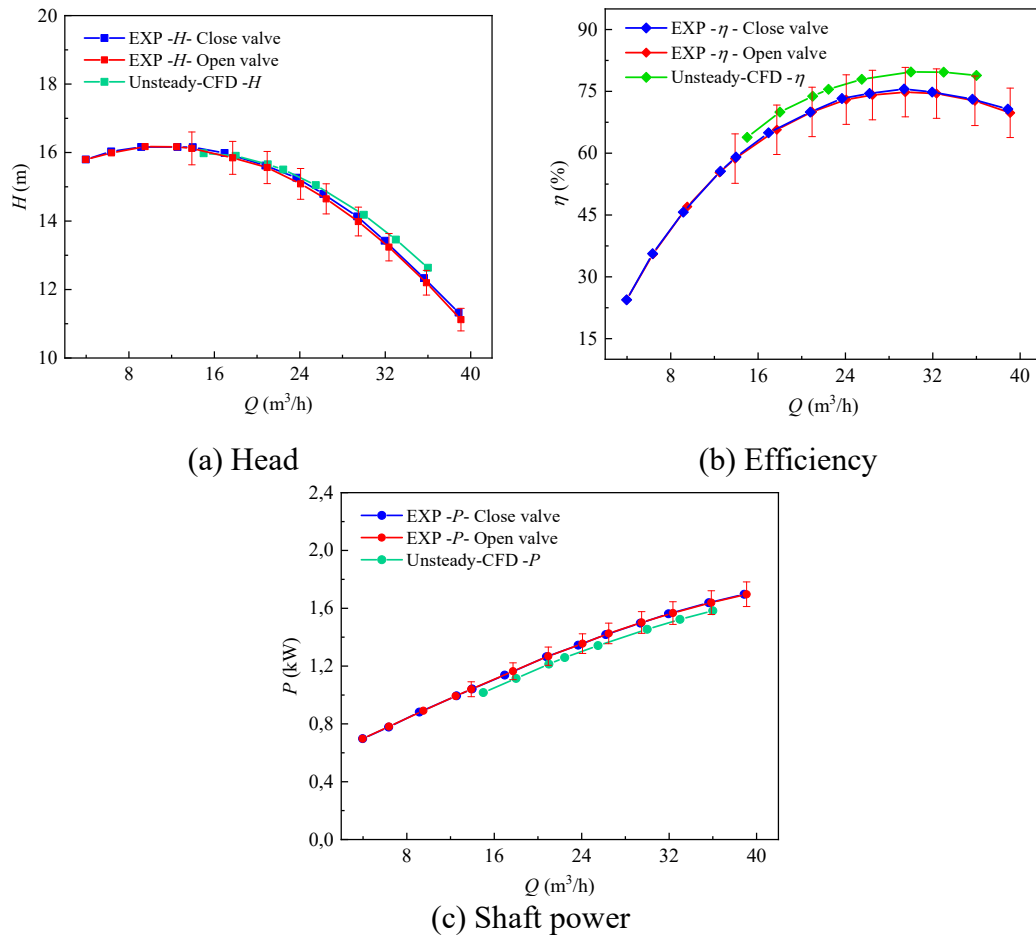


Fig. 5.7. Comparison of experimental and simulation performance curves of the conventional impeller.

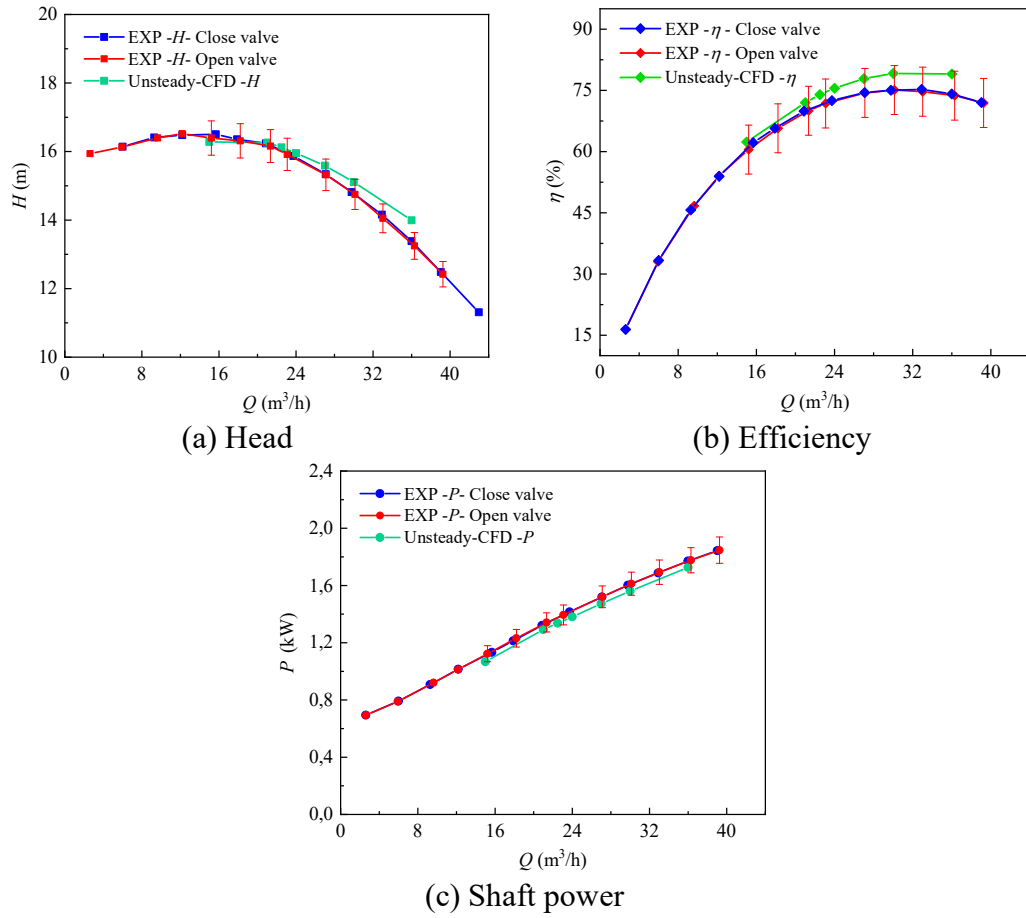


Fig. 5.8. Comparison of experimental and simulation performance curves of the tandem impeller.

6 Comparison Analysis of Centrifugal Pump Performance

This chapter presents a systematic analysis of the impact of different impeller design configurations on the performance of the centrifugal pump. By combining experimental measurements with numerical simulations, a comparative evaluation is conducted between conventional impellers and tandem impellers in terms of hydraulic performance. Furthermore, the underlying mechanisms through which key geometric parameters influence pump performance are investigated. The analysis includes the effects of blade number and WA on the performance of conventional impellers, a sensitivity study of D and β_{21} for tandem impellers, as well as the adaptability of tandem impellers with varying blade numbers.

6.1 Influence of Impeller Design Parameters on Performance

This section focuses on the influence of design parameters of conventional impellers on the hydraulic performance of the pump, with particular attention to two key geometric variables, blade number and blade WA. The effects of these parameters on both the external characteristics and the internal flow behavior are systematically investigated. By constructing impeller models with different parameter settings and conducting comparative measurements and simulations, the regulatory role of these variables on the overall pump performance under various operating conditions is revealed. Finally, based on the research findings, an optimized design scheme for the conventional impeller with superior overall performance is identified.

6.1.1 Influence of Blade Number on Performance

This section first compares the influence of different blade numbers on the performance parameters of the centrifugal pump. In this study, five impeller schemes with different blade numbers were designed, containing 3, 4, 5, 6, and 7 blades, respectively. The geometries are illustrated in Fig. 6.1. All schemes were designed with the same blade WA=120°, while other design parameters were kept identical except for the blade number. Both experimental measurements and numerical simulations were conducted for the five impeller schemes, where the experiments included the processes of gradual valve closing and opening. The numerical simulations were performed using the steady method described in Chapter 5, and all reported parameters were obtained by averaging the results over the last 200 iterations.

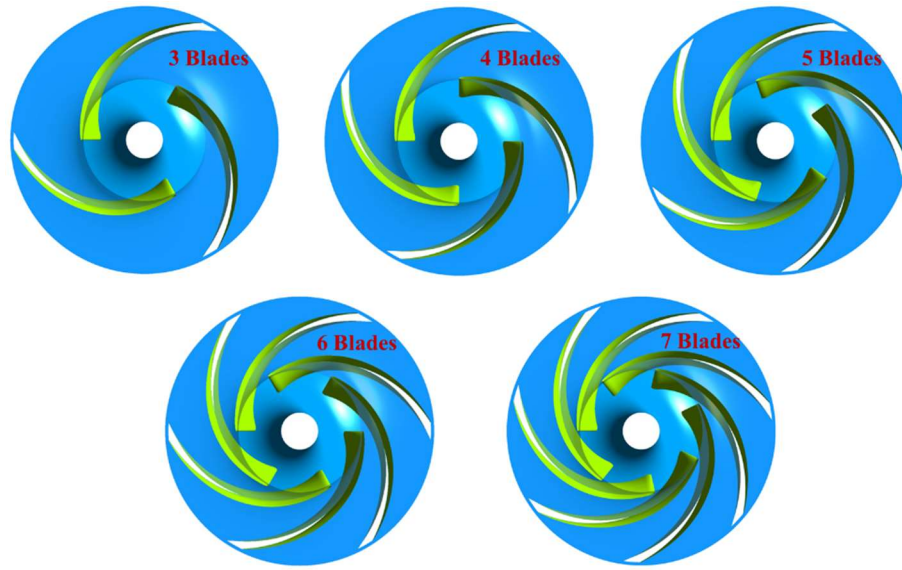


Fig. 6.1. Geometry of impellers with different blade numbers.

Fig. 6.2 illustrates the variations of H with Q for impellers with different blade numbers, obtained from both experimental measurements (EXP) and CFD. The EXP results indicate that the H curves measured during the valve closing and opening processes remain almost identical, showing no evident hysteresis effect. Overall, as the Q decreases, the H gradually increases, and the growth trend tends to level off when approaching the minimum Q . From the comparison among different blade number schemes, it can be observed that increasing the number of blades helps to enhance the pump H under all operating conditions. In particular, the H increases significantly when the blade number rises from 3 to 5. However, when the blade number is further increased to 7, the H curve becomes nearly identical to that of the 6-blade impeller, indicating a saturation effect. This suggests that the sensitivity of H to blade number gradually diminishes once a certain threshold is reached. The CFD results shown in Fig. 6.2(b) exhibit a high level of consistency with the EXP data, further validating the accuracy and reliability of the steady simulation method adopted in this study. As the blade number increases, more blades can perform work on the fluid, thereby producing a higher H . Nevertheless, an increased blade number also narrows the flow passages, making the fluid more prone to local flow separation and energy losses. Consequently, an excessive number of blades may not only fail to continuously improve the H but could also induce flow instabilities, thereby compromising the overall hydraulic performance.

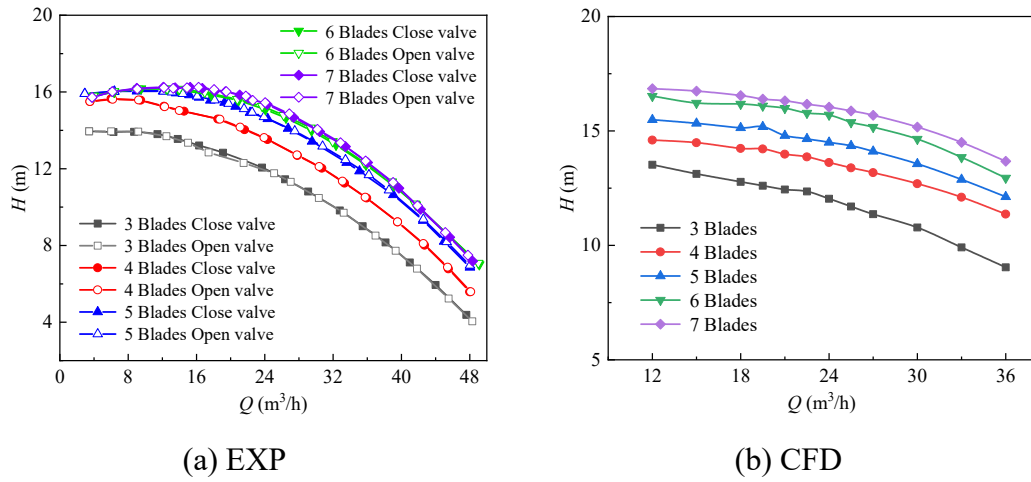


Fig. 6.2. Comparison of head curves for impellers with different blade numbers.

The variations of pump η with Q obtained from EXP and CFD are presented in Fig. 6.3. The EXP results indicate that the η curves measured during valve closing and opening are nearly identical, confirming that no evident hysteresis occurs in the η measurement process. As the Q decreases, the pump η first gradually increases, reaching its maximum near the Q_{des} , and then rapidly declines with further reduction in Q . Notably, once the Q drops below the Q_{des} , the η decreases at a much faster rate. Among the five impeller schemes with different blade numbers, the η differences are negligible when the Q is below 16 m³/h. However, when the Q exceeds 16 m³/h, the 3-blade impeller exhibits the lowest η , whereas the 6-blade impeller demonstrates the highest η . The η of the 4-blade impeller decreases more sharply in the high-flow region, while the η of the 5-blade and 7-blade impellers are relatively close. The CFD results shown in Fig. 6.3(b) exhibit a similar trend to the EXP data. Due to the excessively large flow passage area of the 3-blade impeller, strong flow separation is easily induced, leading to significant energy losses and consequently lower η . The 4-blade and 5-blade impellers, having relatively larger flow passage areas, show their peak η biased toward overload conditions. By contrast, the 6-blade impeller maintains a wide high-efficiency range around the Q_{des} , indicating better operational stability and adaptability, with overall efficiency performance superior to the other designs.

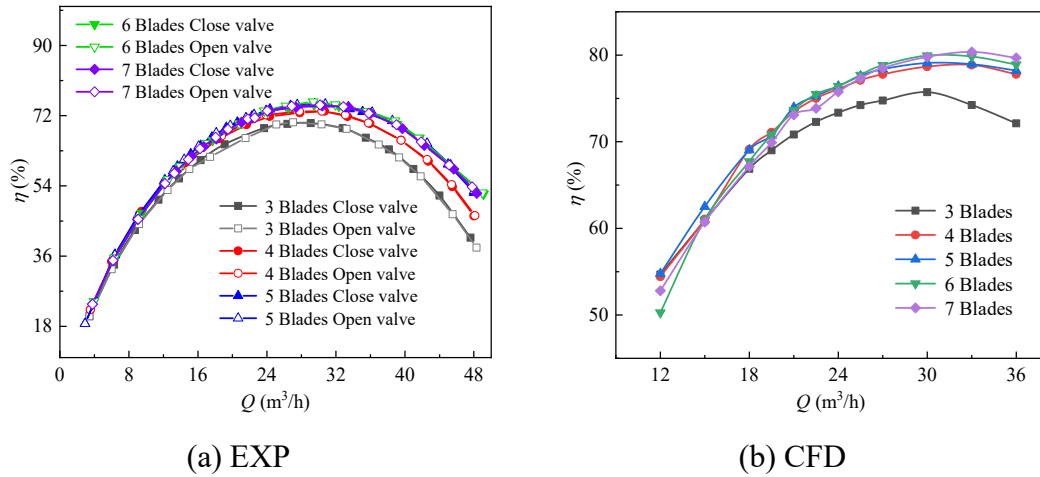


Fig. 6.3. Comparison of efficiency curves for impellers with different blade numbers.

Fig. 6.4 illustrates the variation of P with Q for impellers with different blade numbers. As shown in Fig. 6.4(a), during both valve opening and closing processes, the P curves of all impeller schemes exhibit no significant hysteresis. The P increases gradually with increasing Q , and a higher blade number corresponds to a higher P level. In the part-load region, the differences in P among the impellers are relatively small, whereas in the overload region, these differences become more pronounced. The 7-blade impeller consistently exhibits the highest P , while the 3-blade impeller has the lowest, indicating that an increase in blade number leads to greater energy consumption by the pump. The CFD results in Fig. 6.4(b) similarly show that P increases almost linearly with Q , with higher blade numbers corresponding to higher P . This occurs because a greater number of blades impose more interaction on the fluid during its passage, thereby increasing the total work imparted by the pump. However, this rise in P does not necessarily translate into proportional improvements in η . As indicated by the results in Figs. 6.2 and 6.3, the 6-blade impeller demonstrates excellent performance in both η and H , although it also requires relatively high P , thereby achieving a more balanced energy efficiency. In contrast, while the 7-blade impeller incurs significantly higher P , its H performance does not show a corresponding improvement, resulting in no notable gain in η .

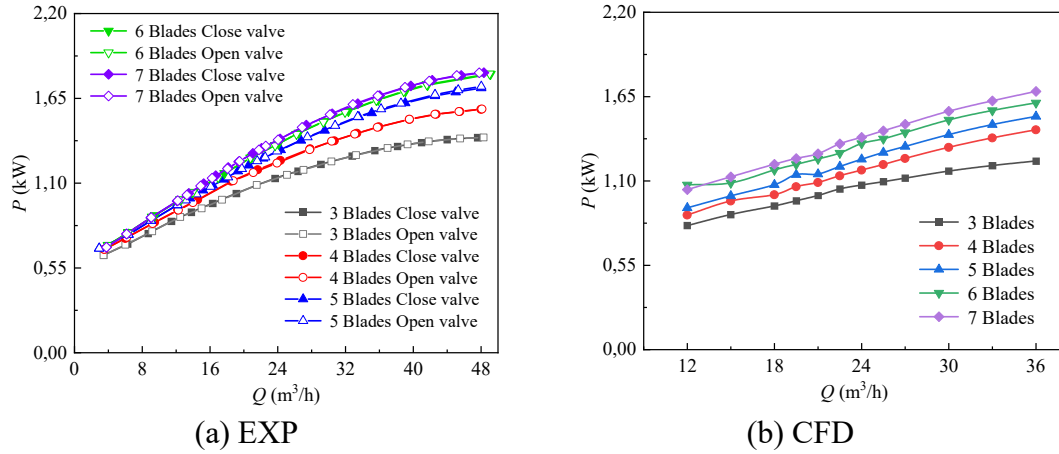


Fig. 6.4. Comparison of shaft power curves for impellers with different blade numbers.

6.1.2 Influence of Blade Wrap Angle on Performance

This section focuses on the influence of blade WA on the performance parameters of centrifugal pumps. Based on the six-blade impeller scheme, which exhibited superior performance in the previous section, six impeller designs with different WA were developed, with a 30° increment between consecutive designs, namely $WA=90^\circ$, 120° , 150° , 180° , 210° , and 240° . The geometries are illustrated in Fig. 6.5. Steady simulations were conducted for all six impeller designs, and physical models of the impellers with $WA=90^\circ$, 120° , and 150° were fabricated and experimentally tested. Similarly, all parameters reported from the numerical calculations represent the averages over the last 200 iterations.

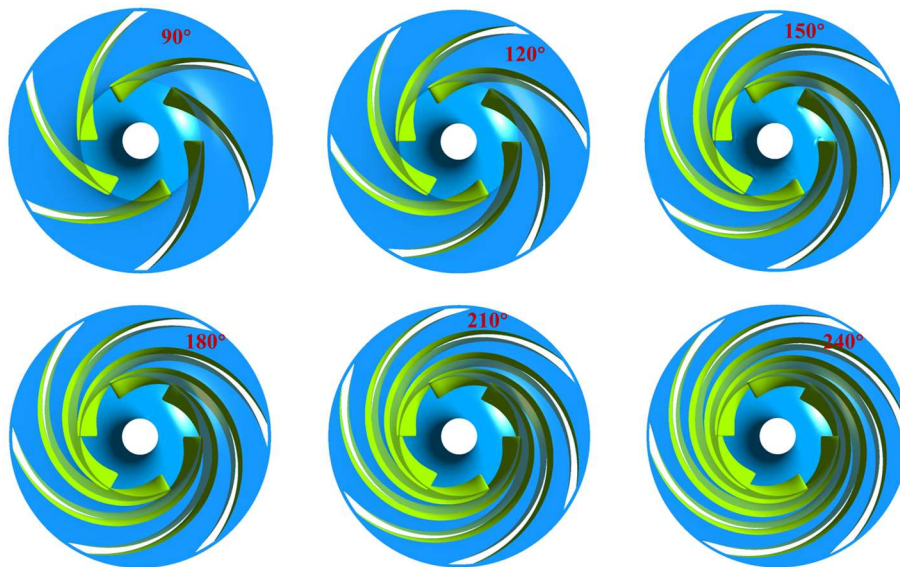


Fig. 6.5. Geometry of impellers with different blade wrap angles.

The H curves under different WA obtained from EXP and CFD are shown in Fig. 6.6. The

EXP results indicate that, for all WA configurations, the H gradually increases as the Q decreases. Notably, for $WA=150^\circ$, a slight hysteresis between the H curves during valve opening and closing is observed, which may be associated with flow delay in the impeller or system inertia. Both EXP and CFD results show that, as WA increases, the overall pump H tends to decrease, particularly in the overload region. For instance, at a Q of $36\text{ m}^3/\text{h}$, the impeller with $WA=90^\circ$ exhibits the highest H , while that with $WA=240^\circ$ has the lowest, resulting in the maximum H difference between the two. This indicates that impellers with larger WA experience more severe flow losses and exit kinetic energy dissipation. Furthermore, although impellers with smaller WA achieve higher H , their H variation with Q is less sensitive, exhibiting a flatter curve. In contrast, impellers with larger WA show lower H but a rapid increase in H as the Q decreases. Therefore, it can be inferred that with further increases in Q , the H of large WA impellers will decrease more sharply, leading to more pronounced H differences among impellers with different WA.

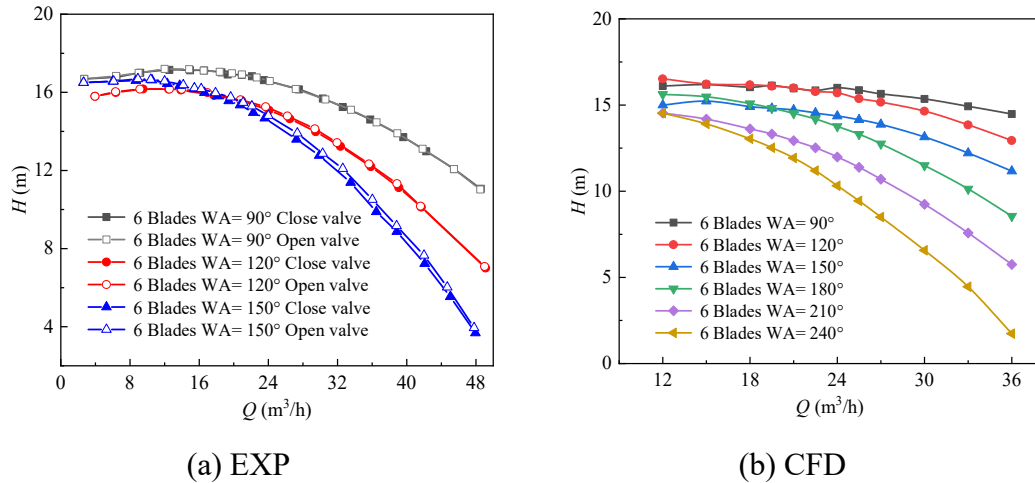


Fig. 6.6. Comparison of head curves for impellers with different blade wrap angles.

Fig. 6.7 presents the η curves under different WA obtained from both EXP and CFD. As shown in Figs. 6.7(a) and 6.7(b), WA still has a significant effect on the η of the centrifugal pump. Both EXP and CFD results indicate that the η first increases and then decreases with Q , exhibiting a typical single-peak distribution. Moreover, as WA increases, the best efficiency point gradually shifts from higher to lower flow rates. At the same Q , impellers with smaller WA (90° and 120°) generally achieve higher η , whereas impellers with larger WA (210° and 240°) show a noticeable decrease in η . This reduction is particularly pronounced under overload conditions. In addition, larger WA leads to lower peak η and a narrower high-efficiency range. Similar to the trends observed in Fig. 6.6, the η differences among different WA impellers are mainly reflected in the high-flow regime. Impellers with smaller WA not only provide higher H but also superior η , whereas large-WA impellers experience severe η decay at higher flow rates due to increased energy

losses, indicating that their internal flow deteriorates more rapidly under off-design conditions, causing losses to rise sharply.

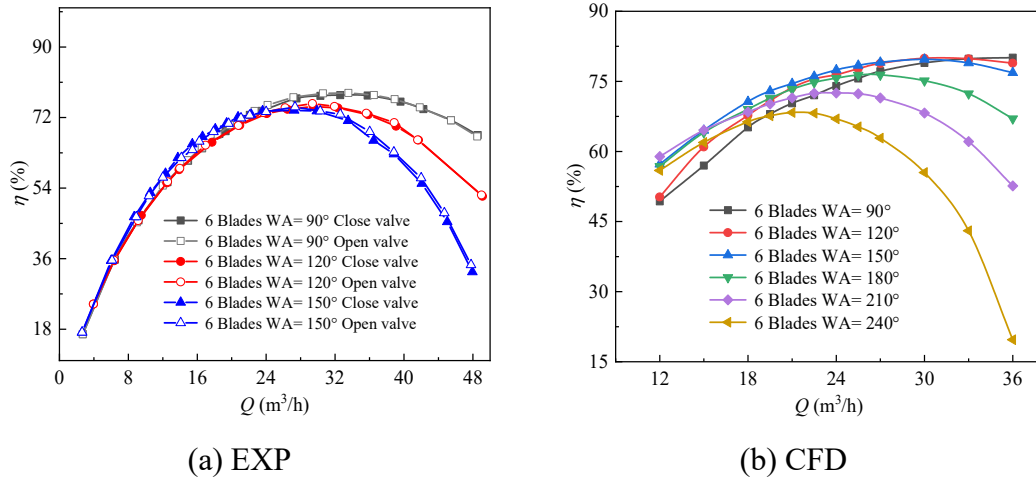


Fig. 6.7. Comparison of efficiency curves for impellers with different blade wrap angles.

Fig. 6.8 illustrates the variation of P for impellers with different WA under various Q . Both EXP and CFD results indicate that P generally increases with Q , but the rate of increase differs significantly among the different WA. EXP data show that the impeller with WA=90° exhibits the highest P under the overload condition, reaching nearly 2.4 kW, whereas the WA=150° impeller has the lowest P , with a maximum of approximately 1.6 kW. Furthermore, the P of large WA impellers tends to increase more slowly with Q and even shows a decreasing trend under overload conditions. CFD results further confirm this pattern, the larger the blade WA, the lower the P . In particular, when WA exceeds 180°, the P curves in the overload region become nearly flat or slightly decrease, clearly lower than those of small WA impellers. Considering the H and η distributions shown in Figs. 6.6 and 6.7, it can be seen that large WA, while effective in reducing P , sacrifice H and η , especially under medium to high flow conditions where pump performance significantly deteriorates. In contrast, small WA impellers, despite higher P consumption, achieve higher H and better η , redemonstrating a more balanced overall performance. Therefore, from the perspective of both energy efficiency and performance, the WA in the range of 90°-120° is preliminarily recommended to achieve favorable performance parameters and a more reasonable power input match.

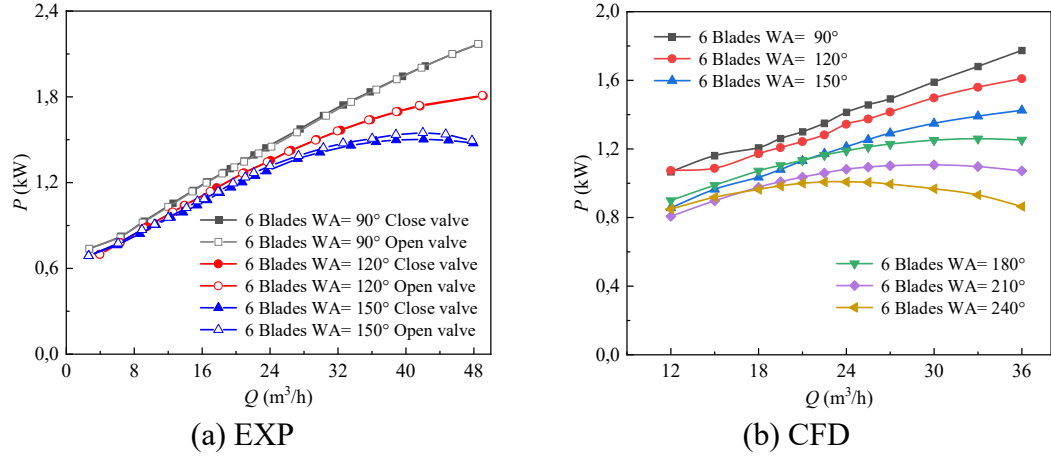


Fig. 6.8. Comparison of shaft power for impellers with different blade wrap angles.

6.1.3 Comparison of Flow Patterns with Different Conventional

Impellers

To further investigate the underlying reasons for the influence of blade number and WA on the performance of the centrifugal pump, this section conducts a comparative analysis of the internal flow patterns of impellers with different blade numbers and WA under various Q . The analysis mainly focuses on three representative operating conditions, overload condition ($1.2Q_{des}$), design condition ($1.0Q_{des}$), and part-load condition ($0.8Q_{des}$). First, the relative velocity and streamline distributions at the mid-span section (span=0.5) of impellers with different blade numbers are compared, as shown in Fig. 6.9. For the 3-blade impeller, the flow channel area in the impeller is relatively large, and the streamlines at the impeller inlet are clearly misaligned with the blade inlet angle, resulting in pronounced flow separation near the PS of the LE and the formation of a large low-velocity region. As the Q decreases, the inlet flow mismatch becomes more severe, leading to more intense flow separation and an enlarged low-velocity region. With an increasing number of blades, the streamlines distribution along the PS and SS of the impeller gradually becomes more uniform, and the extent of low-velocity regions diminishes. In particular, when the blade number increases to 5 or more, the streamlines distribution under overload and design conditions becomes increasingly uniform and stable, with flow separation near the PS of the LE being significantly suppressed. However, under part-load conditions, obvious low-velocity regions and flow separation persist in impellers with all blade numbers, indicating that increasing the blade number does not markedly improve the internal flow patterns under part-load conditions. Consequently, the performance parameter differences observed in Figs. 6.2 and 6.3 are relatively small under part-load conditions but become pronounced under overload conditions.

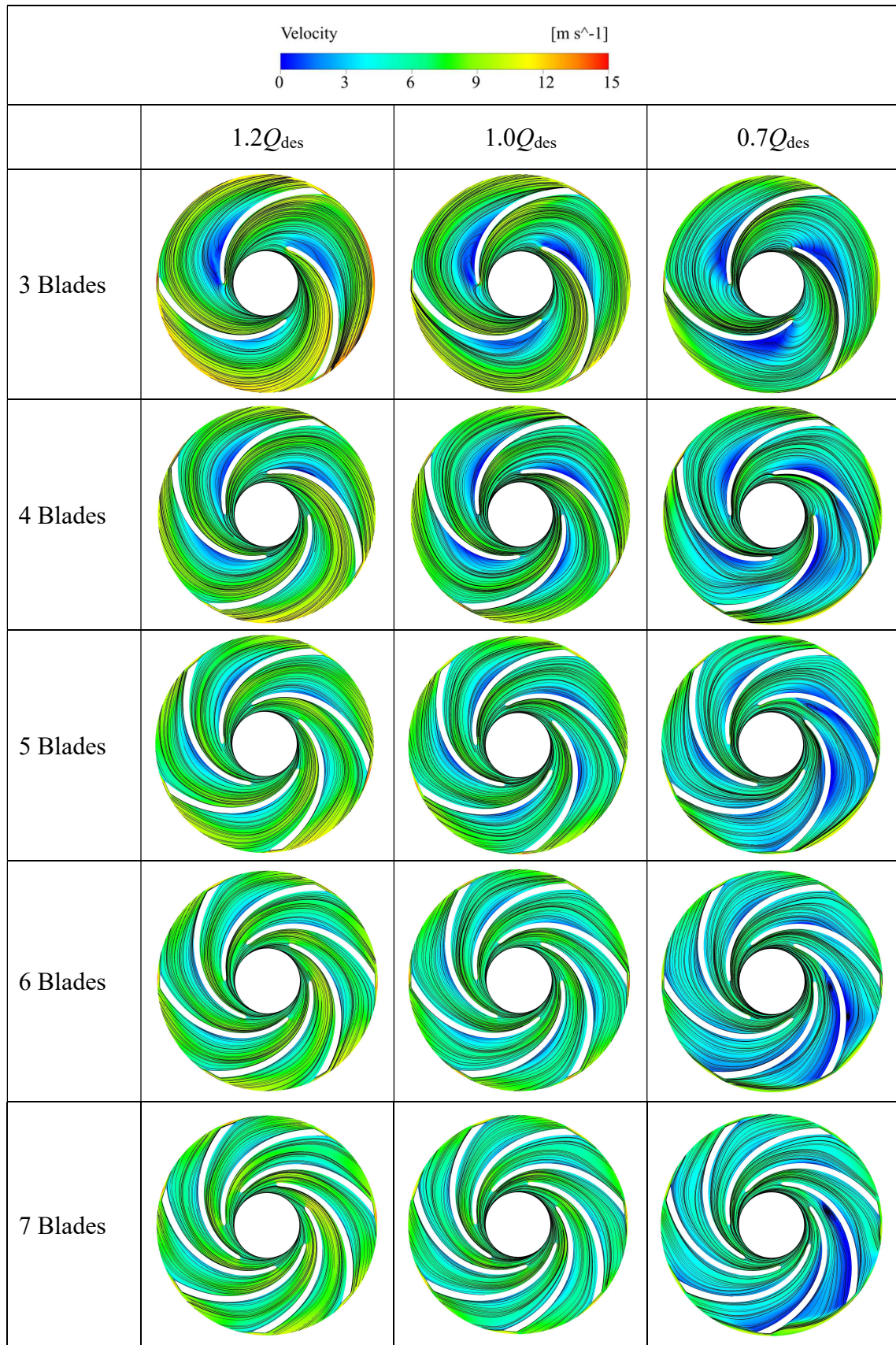


Fig. 6.9. Comparison of velocity streamlines for impellers with different blade numbers.

Based on numerical simulations, the total pressure variable p_{tot} can be used to characterize the flow losses in the rotating domain of the impeller passages [81, 114]. Fig. 6.10 illustrates the distribution of the dimensionless total pressure ratio Ω at the mid-span section (span=0.5) for impellers with different blade numbers. The definitions of Ω and p_{tot} are given in Eqs. 6.1 and 6.2, respectively.

$$\Omega = \frac{p_{tot}}{p_{tot,1}} \quad (6.1)$$

$$p_{tot} = p + \frac{1}{2} \cdot \rho \cdot w^2 - \frac{1}{2} \cdot \rho \cdot u^2 \quad (6.2)$$

The comparison of Ω distributions for different blade numbers was further carried out under the three representative flow conditions. At the impeller inlet, p_{tot} equals $p_{tot,1}$. In the absence of flow losses in the impeller, Ω would remain constant at 1, which means Ω is equivalent to the total pressure in a non-rotating system. For impellers with fewer blades (3 and 4 blades), large low- Ω regions appear, mainly concentrated near the SS of the blade TE and at the impeller outlet. This indicates significant flow losses in these regions, primarily caused by flow separation and secondary flow phenomena. As the blade number increases to 5 or more, the low- Ω regions near the TE diminish, suggesting that flow separation is effectively suppressed and flow losses are reduced accordingly. It is noteworthy that when the blade number increases from 3 to 5, Ω improves substantially, whereas the improvement from 5 to 7 blades is no longer significant. This nonlinear trend suggests the existence of an optimal blade number, approximately 5 to 6 blades. In addition, under part-load conditions, the low- Ω regions are considerably larger than those under overload conditions, indicating greater flow losses. Combined with the observations in Fig. 6.9, it can be concluded that increasing the blade number helps suppress flow instabilities such as separation in the impeller and reduces flow losses, with the improvement being more pronounced under design and overload conditions. However, an excessive number of blades reduces the impeller flow passage area, which not only fails to further improve the internal flow pattern but may even deteriorate it. Therefore, considering both flow stability and manufacturing cost, impellers with 5 or 6 blades are recommended as the better choice.

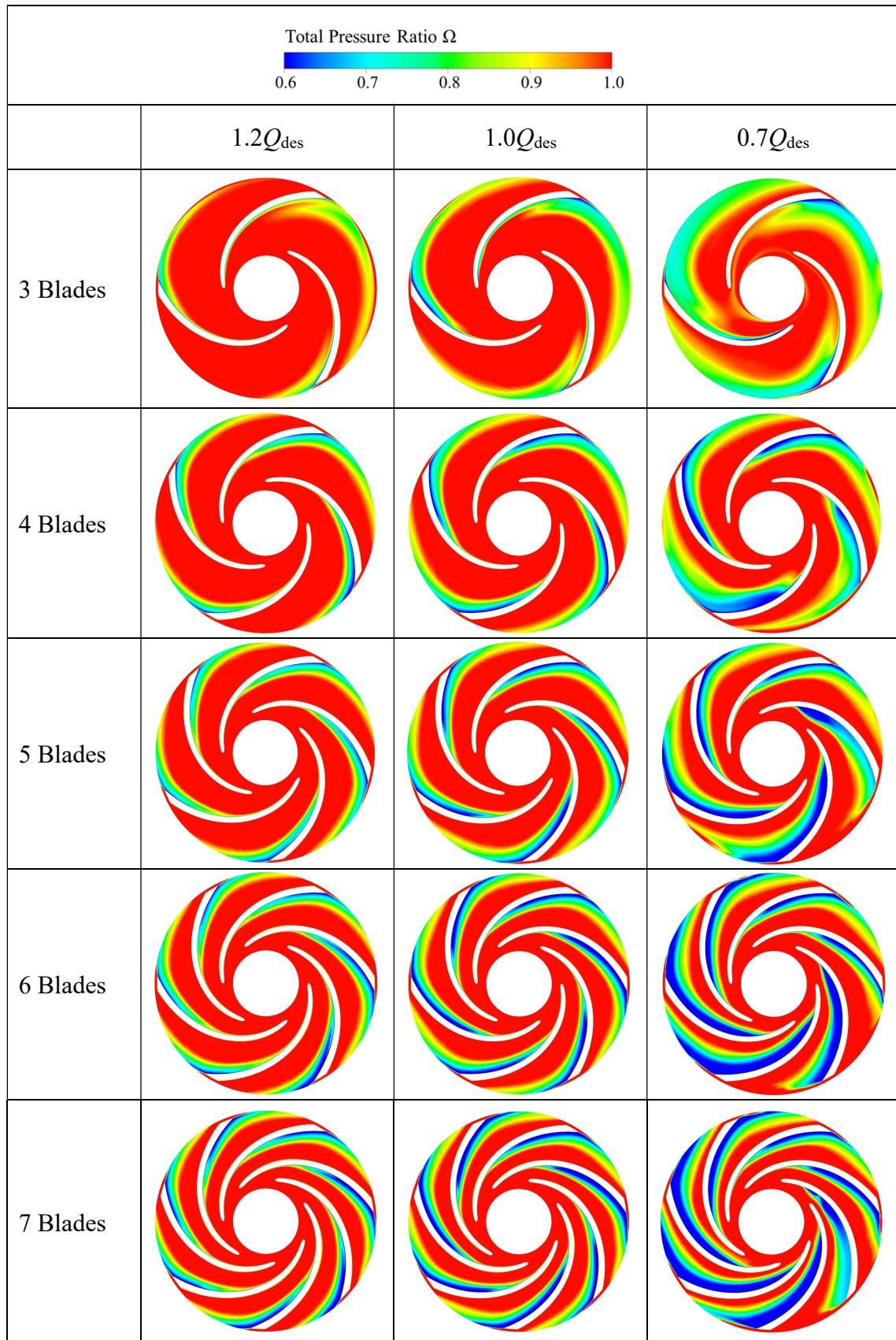
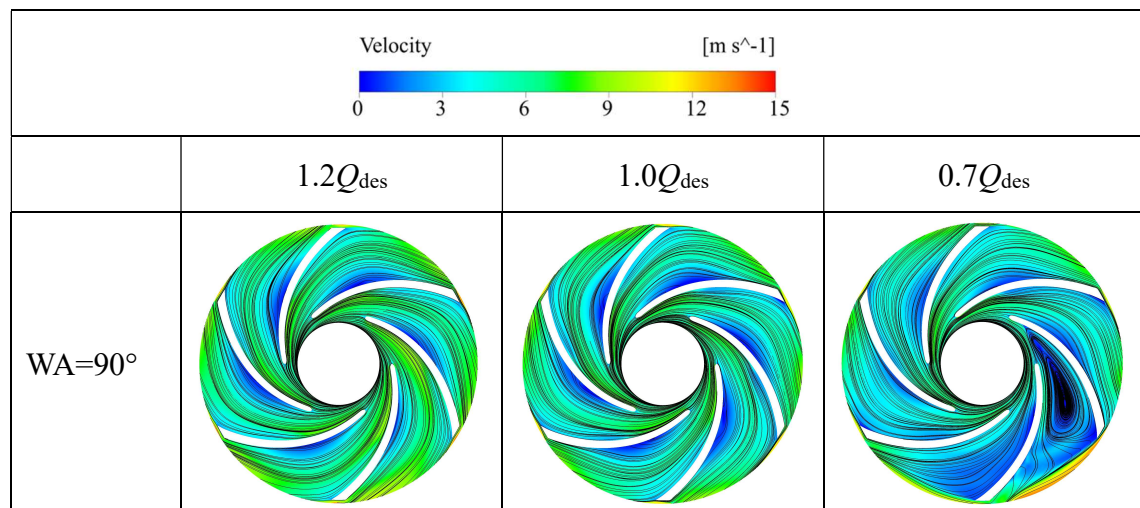


Fig. 6.10. Comparison of total pressure ratio for impellers with different blade numbers.

Subsequently, Fig. 6.11 compares the relative velocity and streamline distributions at the mid-span section (span=0.5) of impellers with different WA under the three representative operating conditions. Based on the conclusions from Figs. 6.9 and 6.10, the analysis focuses on the influence of WA variations on the internal flow patterns of the 6-blade impeller, which demonstrated superior performance. When $WA=90^\circ$, the streamlines distribution in the impeller is generally favorable under overload and design flow conditions, with only slight streamlines bending and the formation of a low-velocity region near the mid-PS of the blade. However, under part-load conditions, the streamlines bend severely and the extent of the low-velocity region increases, with large-scale vortices even forming in some passages, leading to blockage. As WA increases, the flow passage area decreases, resulting in higher flow velocities. Nevertheless, when WA exceeds 180° , the passages become excessively narrow, causing distinct high-velocity regions inside the impeller, particularly under overload conditions, which inevitably leads to a sharp increase in frictional losses between the fluid and the blades. In addition, with increasing WA, the streamlines distribution under part-load conditions improves significantly and the velocity distribution becomes more uniform, which explains the continuous shift of the best efficiency point toward part-load conditions observed in Fig. 6.7. Overall, when $WA=120^\circ$, the flow velocity in the impeller is moderate, the streamlines are uniformly distributed without evident low-velocity regions, and flow separation under part-load conditions is alleviated. Therefore, the impeller with $WA=120^\circ$ achieves higher H and η , making it the preferred impeller configuration in this study.



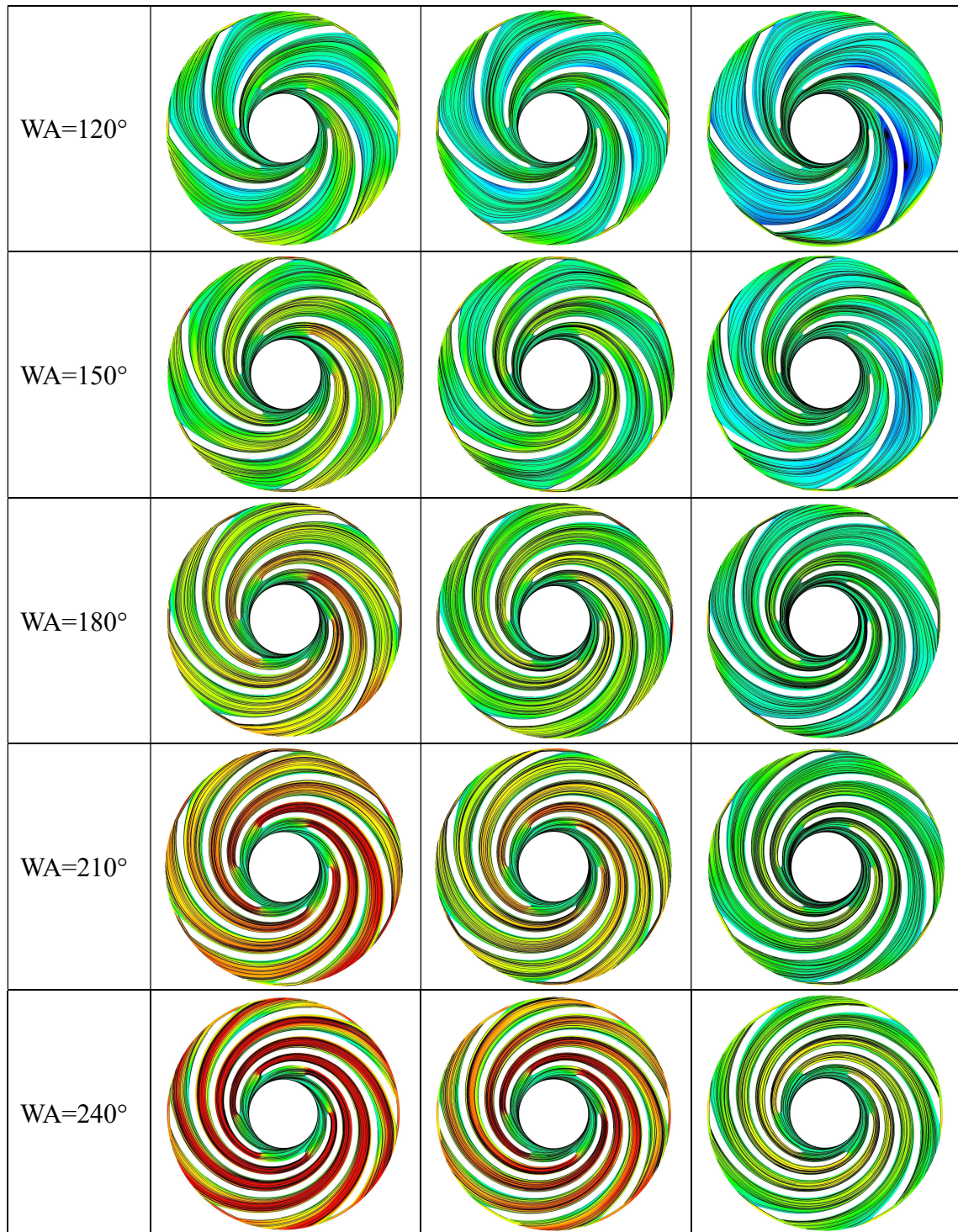
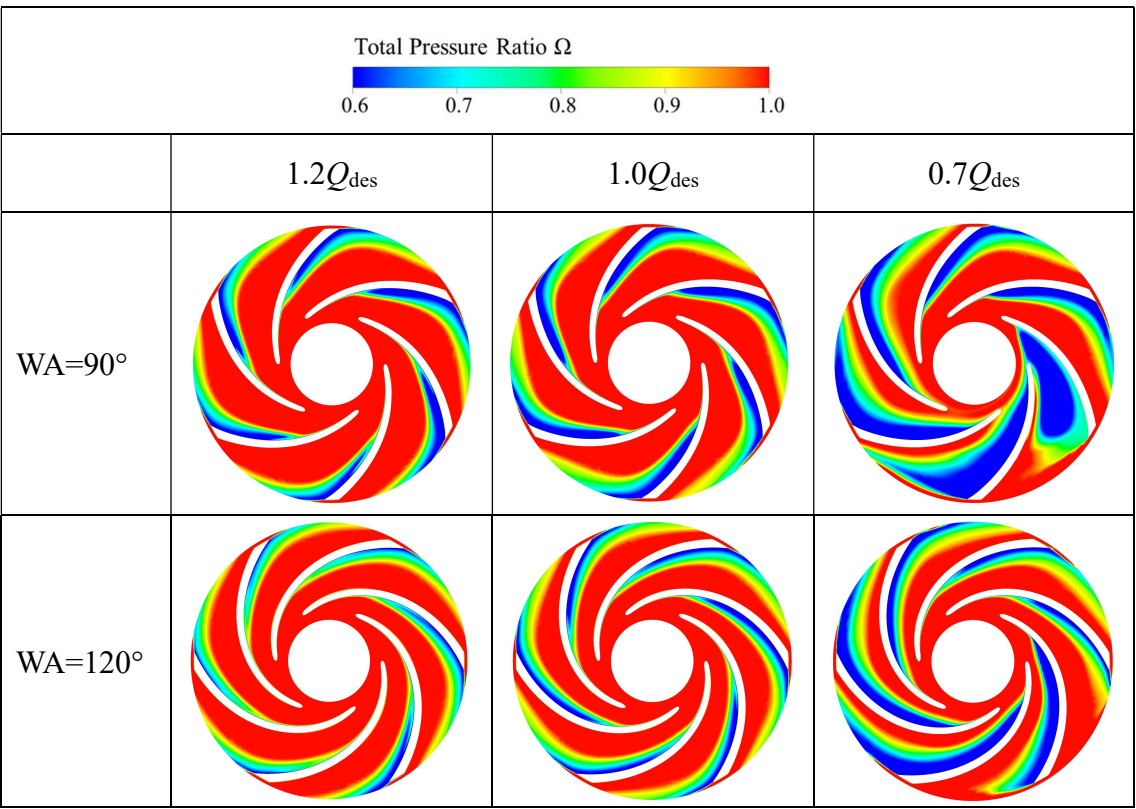


Fig. 6.11. Comparison of velocity streamlines for impellers with different blade wrap angles.

A further comparison of the flow losses in impellers of different WA was conducted, with the distributions of the total pressure ratio Ω shown in Fig. 6.12. For all WA, the regions of high flow loss are primarily concentrated near the SS at the mid-to-rear portions of the blades and extend toward the impeller outlet. At smaller WA, the flow losses under part-

load conditions are significantly greater than those under overload conditions. In particular, for $WA=90^\circ$, the low- Ω region almost covers the entire blade passage, which corresponds to the vortex region observed in Fig. 6.11. As the WA increases, the total pressure ratio Ω under overload and design conditions first increases and then decreases. When $WA=120^\circ$, the low- Ω region in the impeller is minimized, and the Ω in most of the main flow passages is nearly 1. Under part-load conditions, Ω increases with WA ; however, due to the reduced passage area, low- Ω regions still occupy most of the flow channels. Especially at $WA=240^\circ$, the Ω in the passages remains at a relatively low level, mainly due to the excessively long flow path and aggravated frictional losses caused by the large WA . These results indicate that the design of blade WA requires a balance between flow control and loss suppression, as excessively small or large WA are both unfavorable for optimizing impeller performance.

By comprehensively comparing the flow patterns in Fig. 6.11 and the loss distributions in Fig. 6.12, it is evident that the impeller with $WA=120^\circ$ exhibits more stable flow characteristics and lower flow losses under all operating conditions. Therefore, based on the flow pattern analysis in this section, the impeller with 6-blades and $WA=120^\circ$ is selected as the conventional impeller with superior overall performance, which will serve as the baseline for the subsequent redesign involving tandem blades.



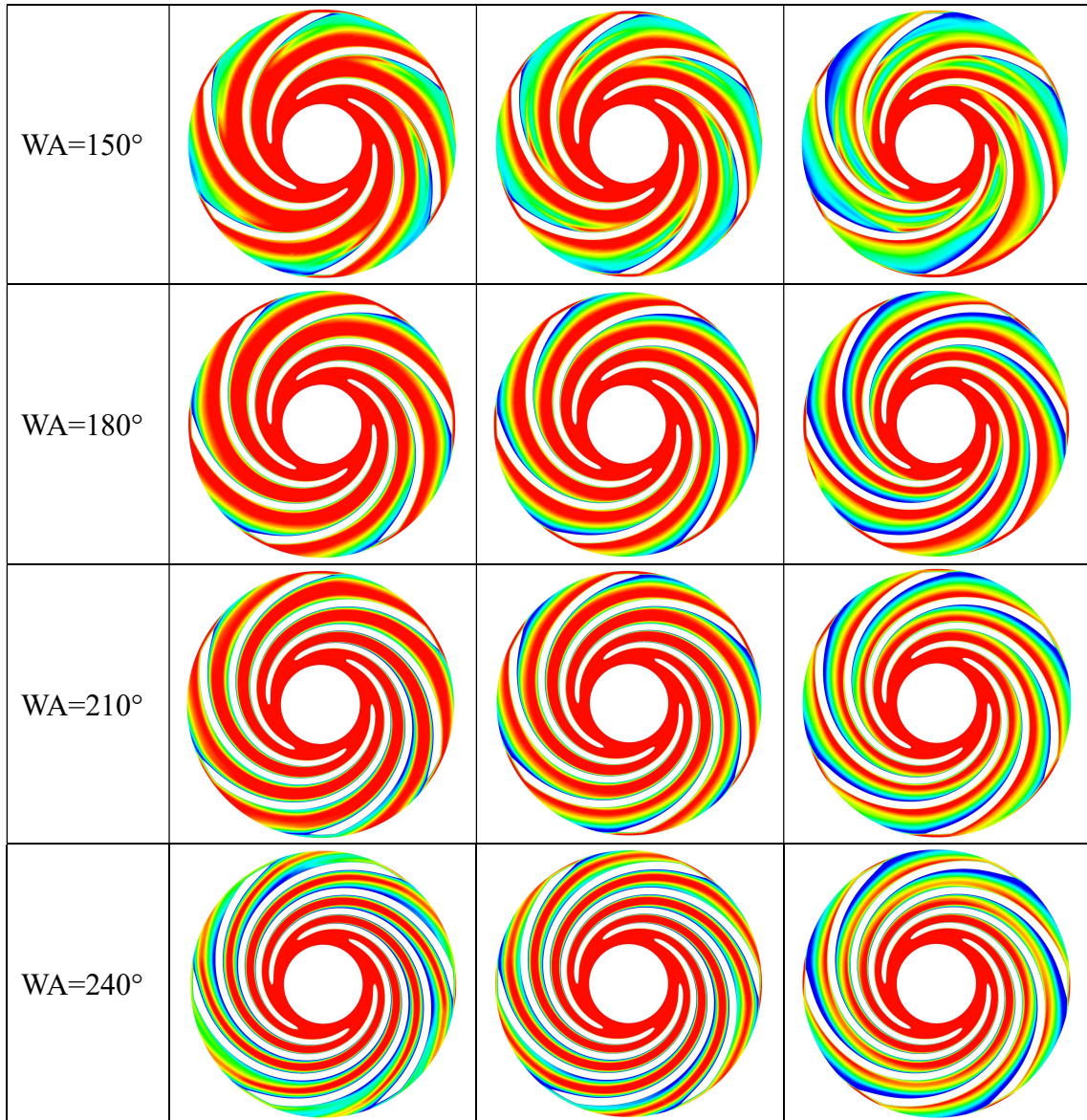


Fig. 6.12. Comparison of total pressure ratio for impellers with different blade wrap angles.

6.2 Performance Analysis of Tandem Impeller

To systematically analyze the hydraulic performance of tandem impellers, this section investigates the effects of several key structural parameters on performance indicators such as H , η , and P . The analysis primarily focuses on two major geometric factors, the D and the β_{21} . On this basis, the tandem impeller with superior performance will be identified and further compared with the conventional impeller to comprehensively evaluate the performance advantages and engineering application potential of tandem blade configurations.

6.2.1 Influence of Axial Offset on Performance

Based on numerical simulation results, this section presents a comparative analysis of the influence of rear-blade offset angle on the performance parameters of tandem impellers. Four different circumferential relative offsets D were considered, namely $D=0.08$, 0.13 , 0.18 , and 0.23 , corresponding to rear-blade circumferential rotation angles of $\Delta\varphi=4.58^\circ$, 7.45° , 10.31° , and 13.18° , respectively. The blade distributions of the four tandem impellers are illustrated in Fig. 6.13.

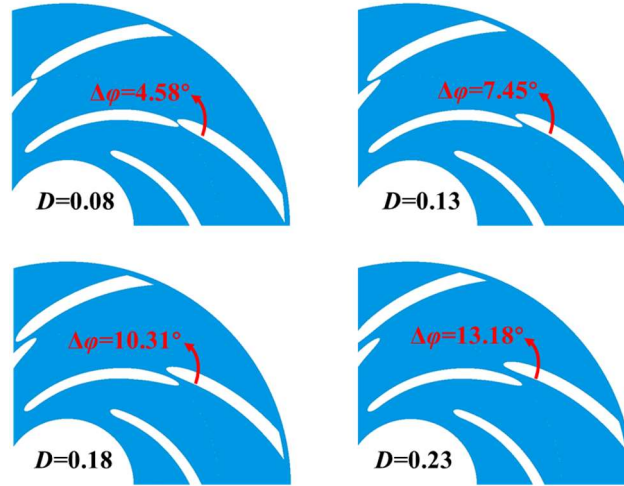


Fig. 6.13. Tandem impellers with different axial offset.

The performance parameter comparisons of the centrifugal pump with four tandem impellers, obtained from steady simulations, are presented in Fig. 6.14. All statistical values represent the averages over the last 200 iterations of the simulations. Fig. 6.14(a) shows that the influence of D on H becomes more pronounced as the Q decreases, while it is negligible under overload conditions. A larger D corresponds to a smaller slope of the H curve, indicating a slower increase in H as the Q decreases. Therefore, under part-load conditions, smaller D yield higher H . However, excessively small D tend to cause unstable fluctuations in the H curve under part-load conditions, as observed for $D=0.08$. D also exerts a noticeable influence on η , as illustrated in Fig. 6.14(b). When the D increases beyond 0.13 , the η under all operating conditions decreases significantly, particularly under part-load conditions. This reduction is primarily attributed to the decrease in H , which results in lower P_e . In addition, when the D is excessively large, η curves exhibit unstable fluctuations under part-load conditions due to the influence of H variations. The P distributions shown in Fig. 6.14(c) indicate that D has only a minor effect on P characteristics across different operating conditions. For all four tandem impellers, P increases with Q , with only slight differences observed at very low flow rates.

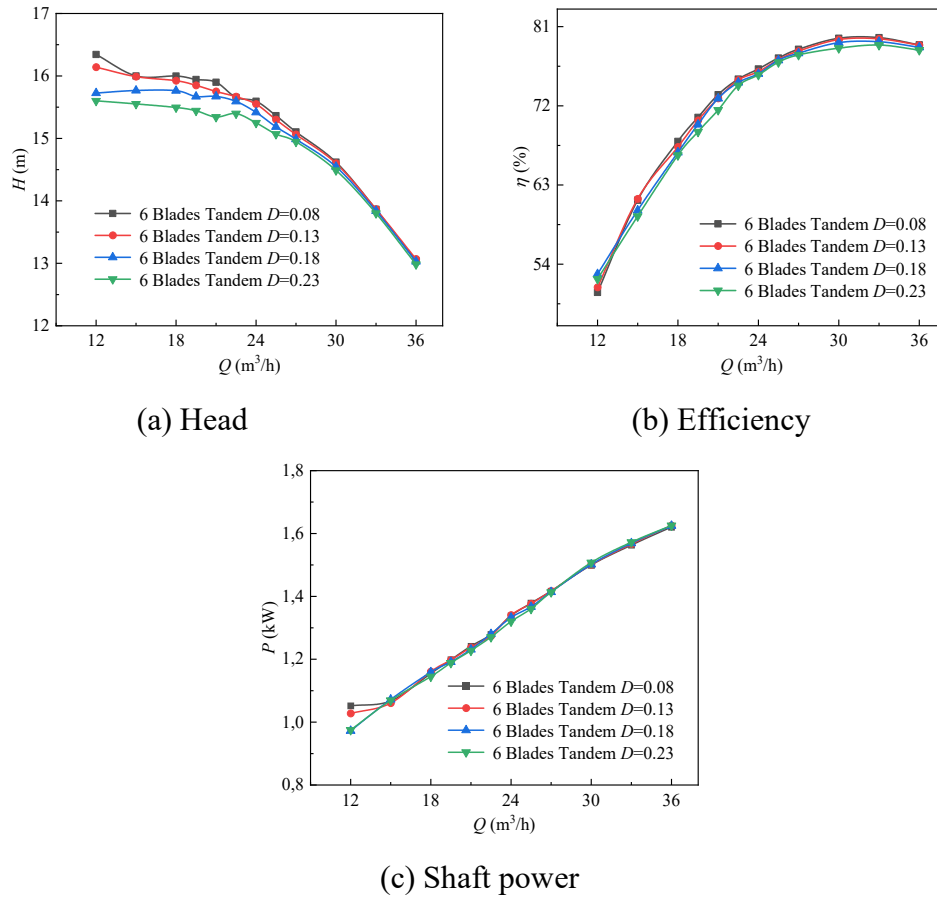


Fig. 6.14. Comparison of performance curves for tandem impellers with different axial offset.

The main conclusion drawn from the above analysis is that both excessively large and excessively small circumferential offsets D have adverse effects on the external characteristics of the centrifugal pump. As shown in Fig. 6.14, when $D=0.13$, the pump exhibits relatively high H and η , and the performance parameters vary stably with Q without noticeable fluctuations. Therefore, in this study, the tandem impeller adopts a fixed rear-blade circumferential offset of $D=0.13$, corresponding to $\Delta\varphi=7.45^\circ$.

6.2.2 Influence of Rear Blade Angle on Performance

Due to the existence of the mid-passage gap in the tandem impeller, the inlet setting angle of the rear blade becomes a critical parameter influencing the flow compatibility in the impeller. In this section, the effect of the β_{21} on the pump performance is investigated through a comparative analysis based on both experimental measurements and numerical simulations. Four different cases (Case 1-Case 4) were designed and analyzed using steady numerical simulations. The blade angle settings at Span=0.5 for each case are listed in Table 6.1, including the outlet setting angle of the front blade β_{12} , the β_{21} , and the angle difference $\Delta\beta=\beta_{21}-\beta_{12}$. The rear blade profiles for the four cases after angle modification are shown in Fig. 6.15. Furthermore, based on the configurations of Case 1 and Case 3,

impeller models were fabricated and tested experimentally.

Table 6.1. Blade angle setting information at Span=0.5

Case scheme	β_{12} (°)	β_{21} (°)	$\Delta\beta$ (°)
Case 1	26.8	26.8	0.0
Case 2	26.8	16.8	-10.0
Case 3	26.8	10.8	-16.0
Case 4	26.8	6.8	-20.0

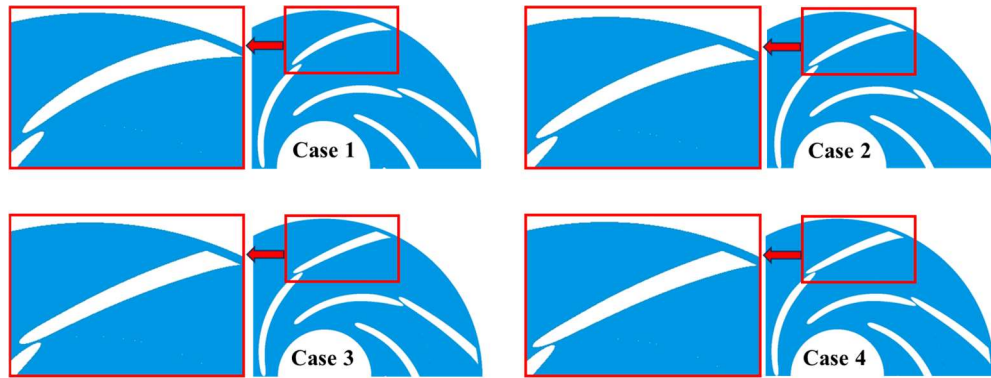


Fig. 6.15. Tandem impellers with different rear blade angle.

As shown in Fig. 6.16, both the experimental measurements and numerical simulations demonstrate that the H gradually increases with the reduction of β_{21} under all operating conditions. The improvement in H is particularly evident when comparing Case 1 with Case 3. However, the H in Case 4 shows no significant difference compared with Case 3, indicating that the enhancement effect of reducing β_{21} has an upper limit. In addition, decreasing β_{21} helps to mitigate the unstable fluctuations of the H curve under part-load conditions.

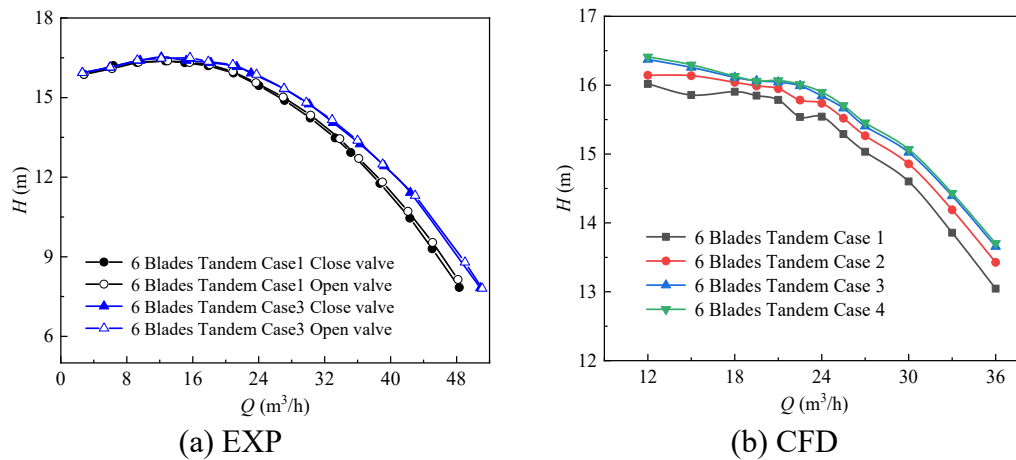


Fig. 6.16. Comparison of head for tandem impellers with different rear blade angle.

The η curves presented in Fig. 6.17 indicate that variations in β_{21} exert only a limited influence on the pump η . The EXP results show that the η of Case 3 is slightly lower than that of Case 1 near the design condition, while it becomes marginally higher under overload conditions. The CFD results reveal that, when the Q is below 18 m³/h, the η of Case 2 is slightly reduced compared with Case 1, whereas the η of Case 3 and Case 4 remain almost identical to that of Case 2. These observations suggest that reducing β_{21} may lead to a slight η decrease under certain flow conditions. However, this reduction does not intensify with further decreases in β_{21} .

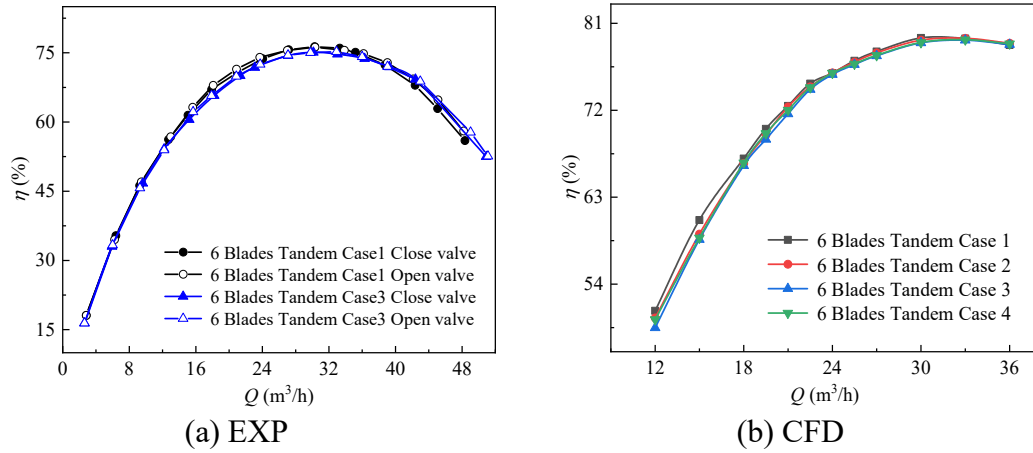


Fig. 6.17. Comparison of efficiency for tandem impellers with different rear blade angle.

The P curves obtained from both EXP and CFD results, as shown in Fig. 6.18, indicate that P gradually increases across all operating conditions as β_{21} decreases. With increasing Q , the P of Case 3 becomes progressively higher than that of Case 1, and the difference grows with further Q increase. However, the CFD results reveal that when β_{21} is reduced to the value of Case 3, the P no longer continues to increase with further reductions in β_{21} . The influence of β_{21} on P exhibits a trend highly consistent with the H variation shown in Fig. 6.16. Therefore, although a smaller β_{21} enables the fluid to gain more energy and consequently enhances the pump H , it also requires greater P input, which ultimately results in the overall η remaining nearly unchanged.

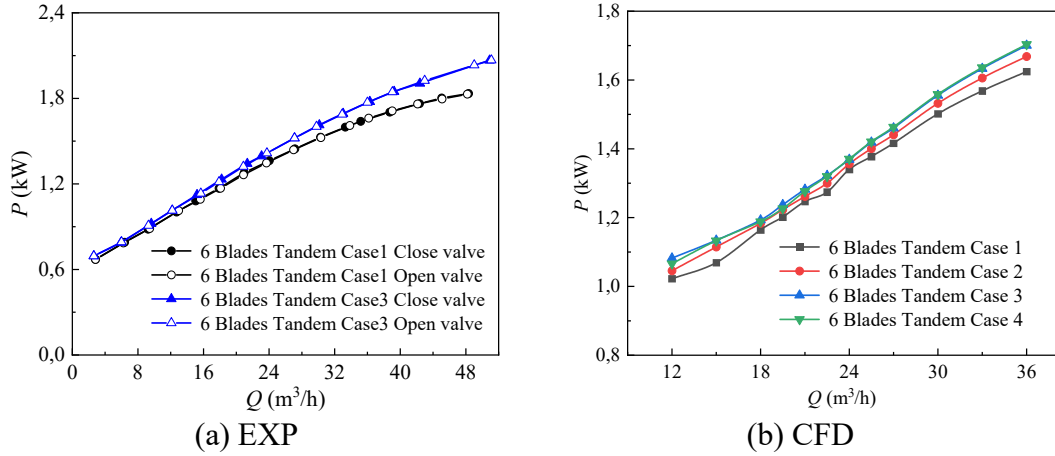


Fig. 6.18. Comparison of shaft power for tandem impellers with different rear blade angle.

The main conclusion drawn from the comparative analysis in this section is that reducing β_{21} can effectively enhance the pump H under all operating conditions. Although a decrease in β_{21} leads to an increase in P , the corresponding improvement in H compensates for the adverse impact of higher P on η . As a result, the reduction of β_{21} does not exert a significant influence on the pump η . However, the enhancement of H by continuously decreasing β_{21} is limited, as an excessively small β_{21} no longer improves the H and instead causes a decline in η under part-load conditions. Therefore, considering the distribution characteristics of all performance parameters, Case 3 is identified as the optimal configuration for the tandem impeller in this study.

6.2.3 Performance Comparison between Tandem Impeller and Conventional Impeller

To more comprehensively demonstrate the performance advantages of the tandem impeller design in centrifugal pumps, this section conducts a comparative analysis between the selected 6-blade conventional impeller and the tandem impeller Case 3, based on experimental measurements and unsteady numerical simulations results. The experimentally measured performance curves include both shut-valve and open-valve operating conditions, while the unsteady simulation results are obtained by averaging the data over the last two revolutions of the impeller to account for the inherent periodic flow fluctuations during pump operation.

Fig. 6.19 compares the H curves of the conventional impeller and the tandem impeller. The experimental results in Fig. 6.19(a) indicate that neither impeller exhibits hysteresis in the H curve. As the Q decreases, the rising trend of H gradually weakens and even shows a tendency to decline under very low flow conditions. Across the entire operating range, the tandem impeller consistently achieves a significantly higher H than the conventional impeller. Moreover, the H enhancement provided by the tandem impeller

increases with the Q , reaching a maximum head rise of $\Delta H_{\max}=1.7\text{m}$ at approximately $48\text{m}^3/\text{h}$. The simulation results shown in Fig. 6.19(b) exhibit the same phenomena and trends, further confirming the high reliability of the numerical predictions. In the simulations, the maximum head rise of the tandem impeller is $\Delta H_{\max}=1.2\text{m}$ at a Q of $36\text{m}^3/\text{h}$, which is very close to the experimentally measured head rise of $\Delta H=1.1\text{m}$ at the same Q . Therefore, the tandem impeller can significantly improve the H across a wide range of operating conditions.

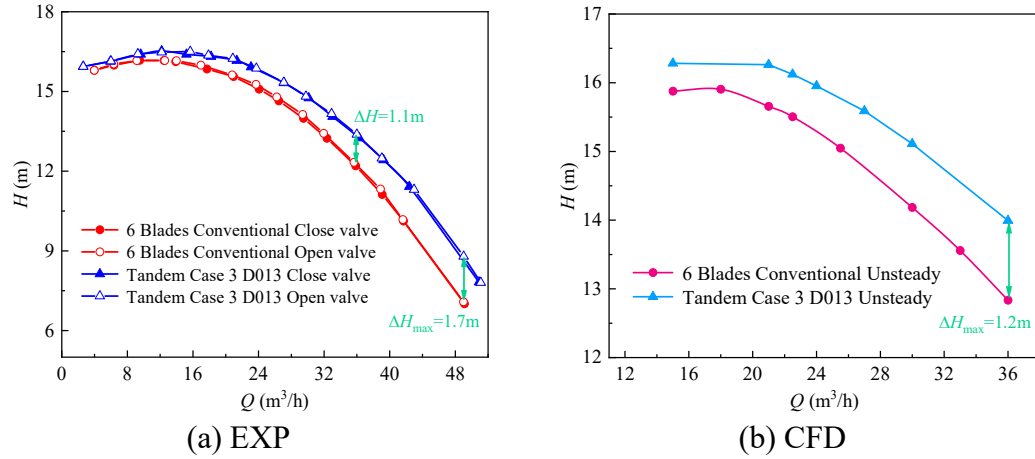


Fig. 6.19. Comparison of head curves between conventional impeller and tandem impeller.

The η characteristics of the two impellers are compared in Fig. 6.20. EXP results indicate that the pump η of the tandem impeller is nearly identical to that of the conventional impeller under the Q_{des} and part-load conditions. Both impellers achieve their maximum η in the vicinity of the Q_{des} . At higher flow rates, particularly when the discharge exceeds $36\text{m}^3/\text{h}$, the tandem impeller exhibits a distinctly higher η than the conventional impeller, suggesting that it provides a broader operating range with sustained high η . The CFD results shown in Fig. 6.20(b) reveal a slight reduction in the tandem impeller η , especially under part-load conditions. However, with increasing Q , the discrepancy between the two impellers gradually diminishes and becomes negligible at around $Q=36\text{m}^3/\text{h}$. The lower η of the tandem impeller under part-load conditions may be attributed to the SST $k-\omega$ model tendency to overpredict unsteady flow phenomena in the pump. Nevertheless, under overload conditions, the tandem impeller demonstrates improved η and even surpasses the conventional impeller, which is in good agreement with the EXP results.

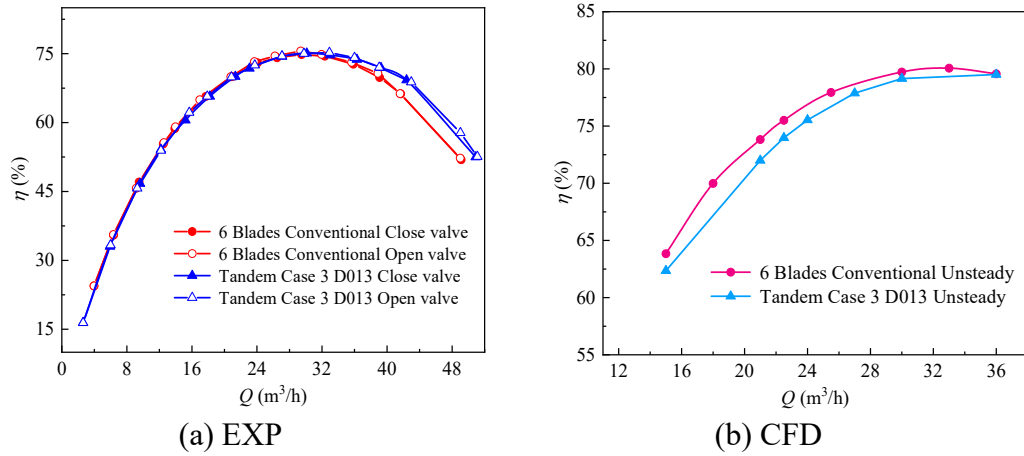


Fig. 6.20. Comparison of efficiency curves between conventional impeller and tandem impeller.

The P distributions shown in Fig. 6.21 indicate that the tandem impeller exhibits a noticeably higher P than the conventional impeller in both EXP and CFD. As the Q increases, the P of both impellers rises gradually, with the tandem impeller showing a faster rate of increase. Consequently, the P of the tandem impeller is significantly higher than that of the conventional impeller under overload conditions. Although the fluid in the tandem impeller absorbs more energy, the increase in rotational resistance, hydraulic friction, and unsteady flow structures results in a corresponding rise in P , which partially offsets the positive effect of H enhancement on the η .

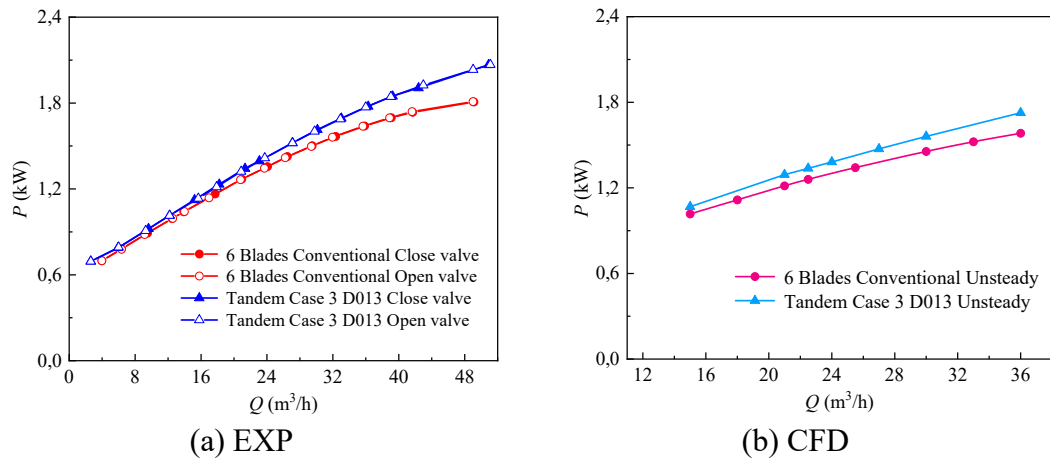


Fig. 6.21. Comparison of shaft power curves between conventional impeller and tandem impeller.

The comparison of pump performance between the tandem and conventional impellers indicates that, by selecting appropriate rear-blade inlet angle β_{21} and circumferential offset D , the tandem impeller can effectively increase the pump H by more than 1m. Although the P of the tandem impeller is slightly higher, the H enhancement compensates for the potential adverse impact on η , allowing the pump to maintain a similar η level to that of

the conventional impeller.

6.3 Adaptability Analysis of Tandem Impeller with Different Blade Numbers

The results presented in Section 6.2 have demonstrated that a properly designed tandem impeller can effectively enhance the performance of the centrifugal pump. To further evaluate the applicability of tandem impellers, this section conducts an adaptability analysis with different blade numbers, systematically assessing the influence of tandem impellers on pump performance under various blade counts. The comparative performance analyses in this section are all based on experimental measurements. In these tests, the conventional impellers were designed with a $WA=120^\circ$, while the tandem impellers adopted a fixed circumferential offset of $D=0.13$, and the blade angles were configured according to Case 3 in Table 6.1. The comparative analysis between the 6-blade conventional and tandem impellers has already been presented in Section 6.2.3, therefore, this section focuses on the experimental results of tandem impellers with blade numbers of 3, 4, 5, and 7. The blade profiles of tandem impellers with different blade numbers at the mid-span (span=0.5) are shown in Fig. 6.22.

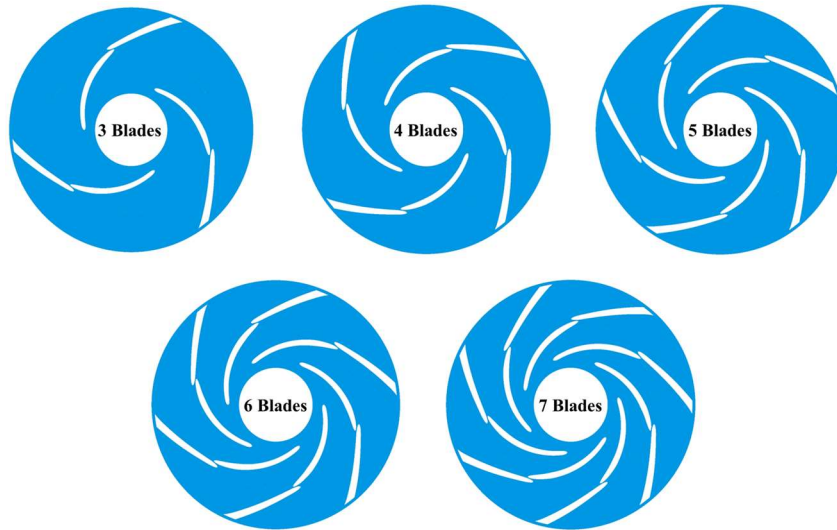


Fig. 6.22. Geometry of tandem impellers with different blade numbers.

6.3.1 Comparison of Head

The pump H characteristics of conventional impellers and their corresponding tandem impellers with different blade numbers were compared based on experimental measurements, as shown in Fig. 6.23. For all blade numbers, the tandem impellers consistently exhibited higher pump H than the conventional impellers. The H improvement provided by the tandem impellers increased progressively with the Q , while the difference between the two types of impellers remained negligible when the Q was

below $16\text{m}^3/\text{h}$. For each blade number, the maximum H increment $\Delta H_{\max}$ occurred at a Q of approximately $48\text{m}^3/\text{h}$. Specifically, $\Delta H_{\max}=1.2\text{m}$ for the 3-blade impeller, $\Delta H_{\max}=1.4\text{m}$ for the 4-blade impeller, and $\Delta H_{\max}=1.5\text{m}$ for the 5-blade impeller. As shown previously in Fig. 6.19, the 6-blade impeller achieved $\Delta H_{\max}=1.7\text{m}$, while the 7-blade impeller further increased to $\Delta H_{\max}=2.0\text{m}$. These results indicate that the enhancement effect of tandem impellers on pump H becomes more pronounced as the blade number increases.

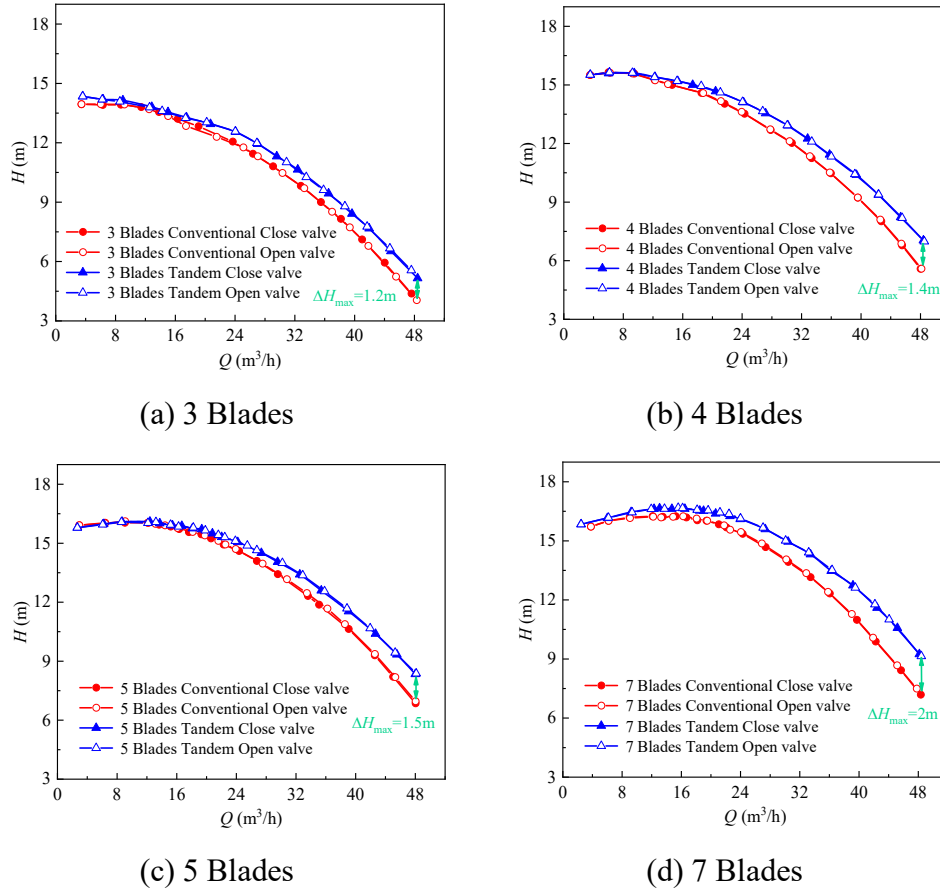


Fig. 6.23. Comparison of head curves between conventional and tandem impellers with different blade numbers.

6.3.2 Comparison of Efficiency

The η curves of conventional and tandem impellers with different blade numbers, obtained from experimental measurements, are compared in Fig. 6.24. For the 3-blade impeller, the η of the tandem configuration was nearly identical to that of the conventional impeller at Q below $36\text{m}^3/\text{h}$. When the Q exceeded $36\text{m}^3/\text{h}$, the tandem impeller exhibited higher η , and the improvement became more pronounced as the Q increased. For the 4-blade impeller, the tandem configuration also showed almost identical η to the conventional impeller across most operating conditions. However, when the Q exceeded

$32\text{m}^3/\text{h}$, the tandem impeller achieved higher η , and the improvement was greater compared with the 3-blade case, extending to a wider range of overload conditions. The η curve of the 5-blade tandem impeller exhibited a similar trend to the 3 and 4 blade cases, with η improvement mainly observed under overload conditions. In contrast, when the blade number increased to 7, the tandem impeller showed reduced η relative to the conventional impeller under design and part-load conditions. Nevertheless, under overload conditions, the tandem impeller still achieved higher η , though the critical Q at which the improvement occurred was shifted to approximately $40\text{m}^3/\text{h}$.

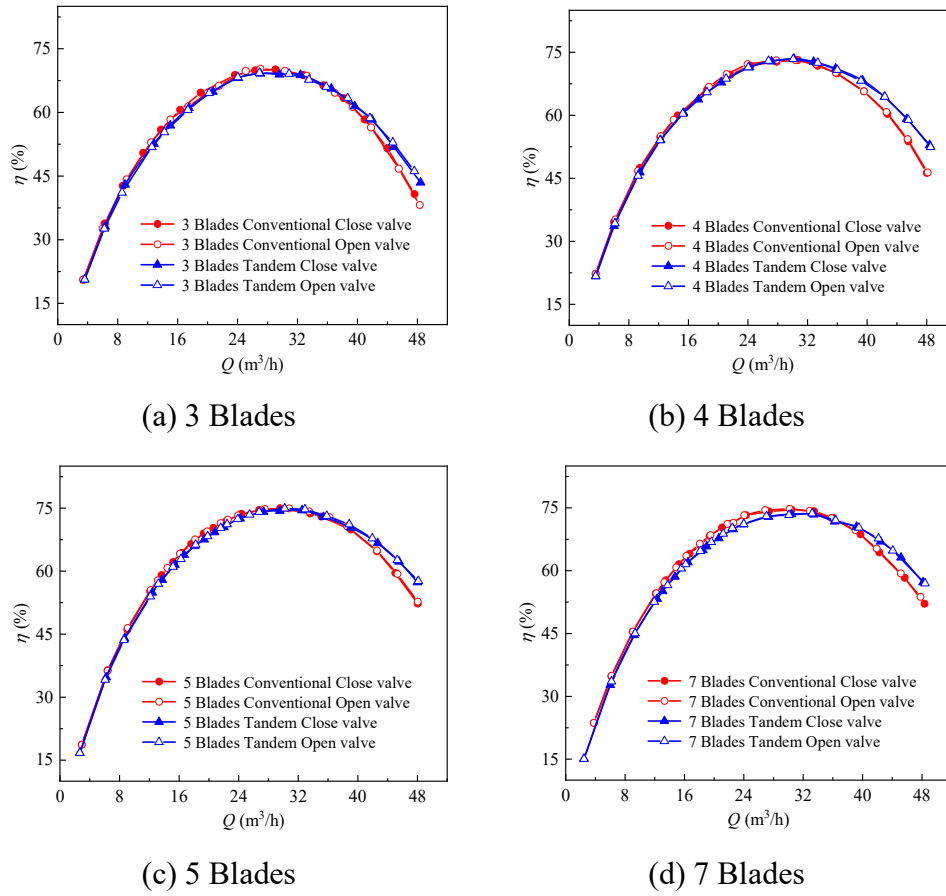


Fig. 6.24. Comparison of efficiency curves between conventional and tandem impellers with different blade numbers.

6.3.3 Comparison of Shaft Power

Fig. 6.25 compares the P distributions of conventional and tandem impellers with different blade numbers, based on experimental measurements. For all blade numbers, the P of the tandem impellers increased more rapidly with Q than that of the conventional impellers. Under part-load conditions, the P of the two types of impellers was nearly identical. However, as the Q increased, the tandem impellers exhibited progressively higher P , and the difference became more pronounced under overload conditions. In

particular, when the blade number increased to 7, the P of the tandem impeller was significantly higher than that of the conventional impeller. This can be attributed to the larger hydraulic friction losses and intensified unsteady flow structures caused by the higher blade count, which substantially increased the torque acting on the blades. This phenomenon also explains the reduction in η observed for the 7-blade tandem impeller in Fig. 6.24(d).

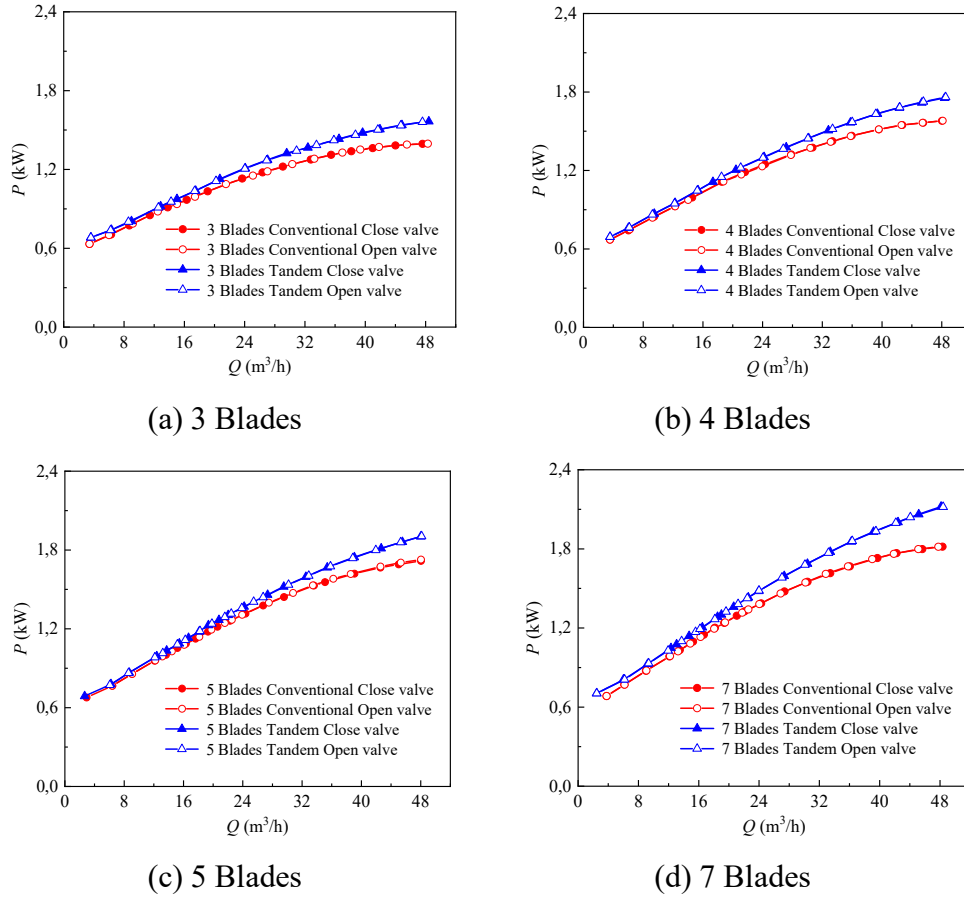


Fig. 6.25. Comparison of shaft power curves between conventional and tandem impellers with different blade numbers.

6.3.4 Performance Improve Evaluation of Tandem Impeller

Based on the performance test results of impellers with different blade numbers, this section further provides a quantitative evaluation of the performance improvement achieved by the tandem impeller. To this end, the relative head increment percentage, ΔH_{re} , is introduced and defined by Eq. (6.3). ΔH_{re} reflects the magnitude of H enhancement of the tandem impeller relative to the conventional impeller under the same operating conditions.

$$\Delta H_{re} = \frac{H_{\text{tandem}} - H_{\text{conv}}}{H_{\text{conv}}} \times 100\% \quad (6.3)$$

where H_{tandem} denotes the experimentally measured head of the tandem impeller, and H_{conv} represents the experimentally measured head of the conventional impeller.

Fig. 6.26 presents the relative head increment percentage ΔH_{re} of tandem impellers with five different blade numbers compared to their conventional counterparts under three typical operating conditions ($1.2Q_{\text{des}}$, $1.0Q_{\text{des}}$, and $0.7Q_{\text{des}}$). For all blade numbers, ΔH_{re} consistently increased with Q , indicating that the H improvement of tandem impellers becomes more pronounced under overload conditions. Under the part-load condition ($0.7Q_{\text{des}}$), the 3-blade tandem impeller exhibited the highest H improvement, with ΔH_{re} exceeding 5%, whereas the 5-blade case showed the lowest improvement, less than 2%. Under the design condition ($1.0Q_{\text{des}}$), the 7-blade tandem impeller achieved the highest enhancement, followed by the 6-blade impeller, both with ΔH_{re} exceeding 6%. Under the overload condition ($1.2Q_{\text{des}}$), the ΔH_{re} of the 6-blade impeller approached 10%, while the 7-blade case also exceeded 9%, whereas the 5-blade impeller consistently exhibited the lowest increment. These findings demonstrate that the H improvement effect of tandem impellers is more significant under overload condition, which is consistent with the conclusions drawn in Section 6.3.1. Moreover, the influence of blade number on H improvement shows a certain dependency on operating conditions. Overall, the 6-blade tandem impeller provides superior H enhancement across different flow conditions.

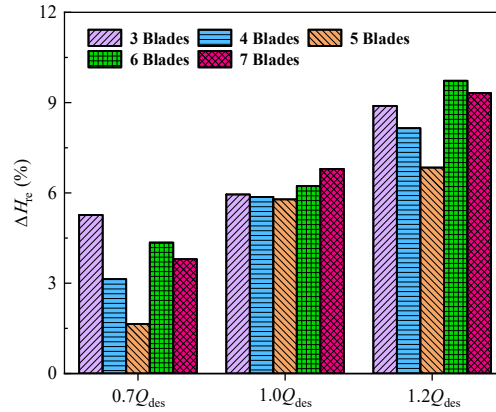


Fig. 6.26. Comparison of ΔH_{re} with different blade numbers.

To further evaluate the energy consumption differences among various impeller configurations during H enhancement, this section introduces the unit energy consumption index ψ_H , which is defined by Eq. (6.4). ψ_H represents the P required per unit H and thus provides a direct measure of the energy cost-effectiveness of the pump.

$$\psi_H = \frac{P}{H} \quad (6.4)$$

Fig. 6.27 compares the values of ψ_H between the conventional and tandem impellers for different blade numbers, based on experimental measurements under 4 representative operating conditions: part-load condition ($0.7Q_{\text{des}}$), design condition ($1.0Q_{\text{des}}$), large flow condition ($1.2Q_{\text{des}}$), and overload condition ($1.6Q_{\text{des}}$). With increasing Q , ψ_H of both

impellers gradually rises, indicating that higher P input is required to achieve H improvement under overload conditions, resulting in poorer energy performance of the pump. Under $0.7Q_{\text{des}}$ and $1.0Q_{\text{des}}$, the ψ_H values of the tandem impeller are nearly identical to those of the conventional impeller. Under $1.2Q_{\text{des}}$, the tandem impeller exhibits slightly lower ψ_H , and when the Q further increases to $1.6Q_{\text{des}}$, the ψ_H of the tandem impeller becomes significantly lower. This trend indicates that the tandem impeller progressively demonstrates superior energy consumption characteristics with increasing Q , which is consistent with the η enhancement observed in Fig. 6.24. A comparison across different blade numbers shows that the 3-blade impellers consistently exhibit the highest ψ_H values under all conditions, with the difference becoming more pronounced as Q increases. As the blade number increases, ψ_H gradually decreases, with the 6-blade tandem impeller consistently presenting the lowest ψ_H across all operating conditions. These results suggest that the tandem impeller achieves comparable energy consumption performance to the conventional impeller under design and part-load conditions, while offering a clear advantage in cost-effectiveness under overload conditions. Moreover, an appropriate blade number configuration (6 blades) can further enhance the performance benefits of the tandem impeller.

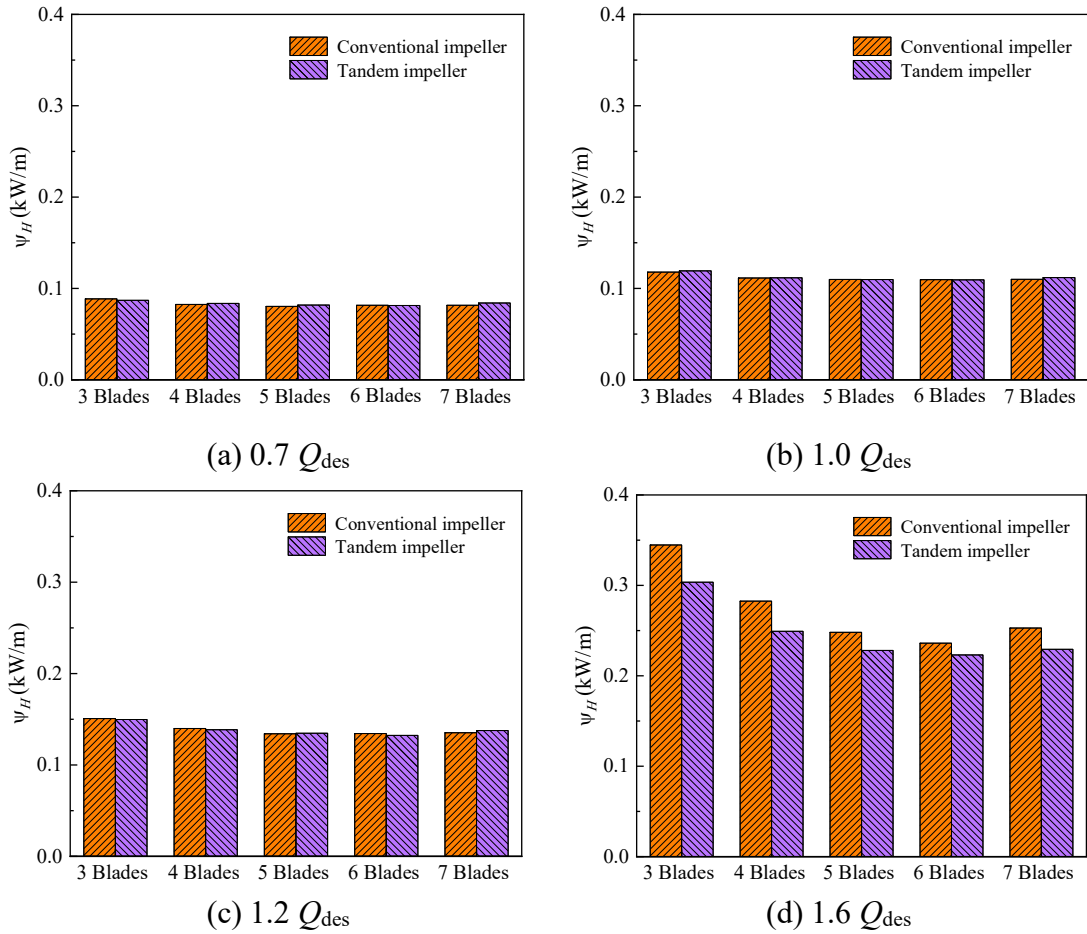


Fig. 6.27. Comparison of ψ_H with different blade numbers.

7 Comparison Analysis of Internal Flow Characteristics of the Centrifugal Pump

To further elucidate the internal flow mechanisms underlying the performance differences between the conventional impeller and the tandem impeller, this chapter presents a comparative analysis of the internal flow characteristics of the centrifugal pump based on unsteady numerical simulations. The analysis focuses on several key aspects, including the mainstream flow patterns, vortex evolution, and unsteady excitation forces, to systematically evaluate the internal flow behaviors under the two impeller configurations. The investigation encompasses the distribution of pressure and velocity fields, contour lines of the potential rothalpy, vortex development features, and unsteady excitation forces. Through this comprehensive analysis, the underlying mechanisms by which the tandem impeller enhances the hydraulic performance of the centrifugal pump are revealed.

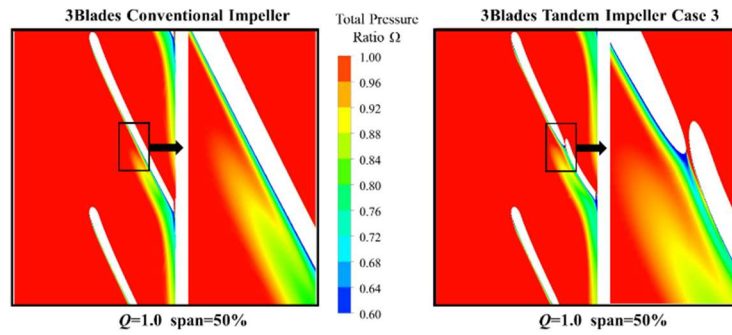
7.1 Comparative Analysis of Mainstream Flow Patterns

The internal mainstream flow patterns of centrifugal pumps equipped with the two impeller configurations are first compared and analyzed in this section, focusing on representative operating conditions under different blade numbers. The analysis examines the spatial distribution and uniformity of key flow parameters, including pressure, velocity, and potential rothalpy. By comparing the tandem impeller with the conventional impeller, the influence mechanisms of structural modifications on the internal flow patterns of critical components are further clarified, thereby elucidating the underlying causes of the observed variations in the centrifugal pump performance.

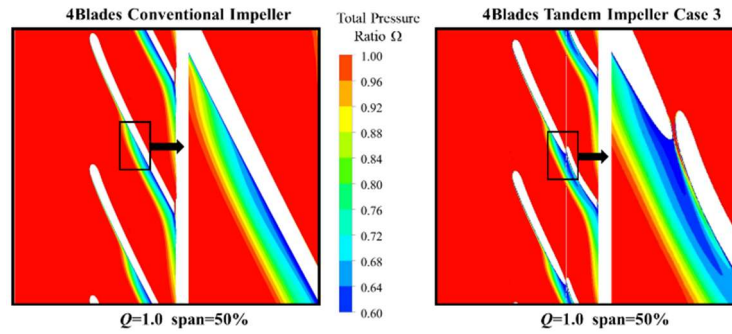
7.1.1 Pressure and Velocity Distribution

The blade-to-blade cross sections of the impeller provide a more intuitive means of comparing the influence of conventional and tandem blade configurations on the internal flow patterns in the impeller passages. Fig. 7.1 illustrates the distributions of the total pressure ratio Ω at the blade-to-blade cross section located at span=50% under the design condition for both impellers with different blade numbers. For all impellers, Ω at the inlet remains equal to 1, corresponding to the total pressure in a non-rotating system, indicating that no flow loss occurs at the impeller inlet. Regions of low Ω values first appear on the SS around the middle of the blades and progressively extend toward the TE. Comparing the two impellers with the same blade number, the tandem impeller exhibits a broader pressure gradient zone along the SS and at the outlet, as well as a more pronounced low- Ω region in the gap. This phenomenon becomes increasingly evident as the blade number rises and the flow area decreases, indicating that for impellers with more blades, the tandem impeller tends to experience reduced flow stability near the SS of the rear blade

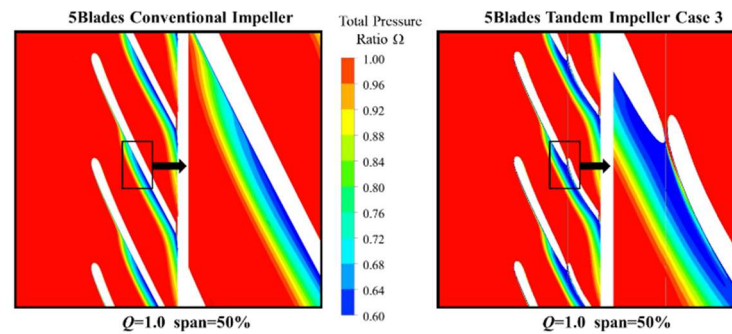
and at the blade TE, potentially resulting in higher flow losses. When comparing across different blade numbers, it can be observed that with increasing blade count, the conventional impeller gradually develops a stable low- Ω region along the mid-to-rear section of the SS. However, strong local pressure gradients persist near the outlet. In contrast, the tandem impeller exhibits slight flow separation in the gap, which becomes more evident with increasing blade number. Detailed views further reveal that as the blade number increases, a newly formed boundary layer develops along the rear blade of the tandem impeller.



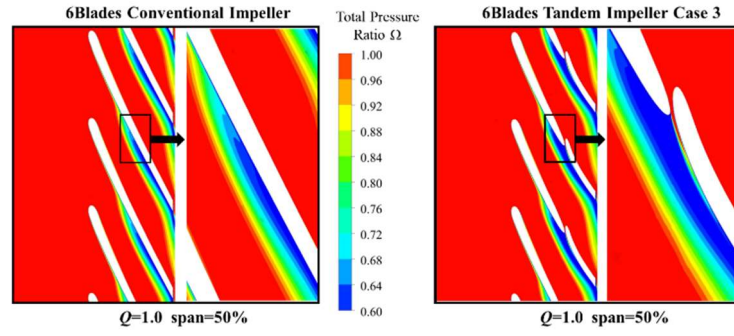
(a) 3 Blades



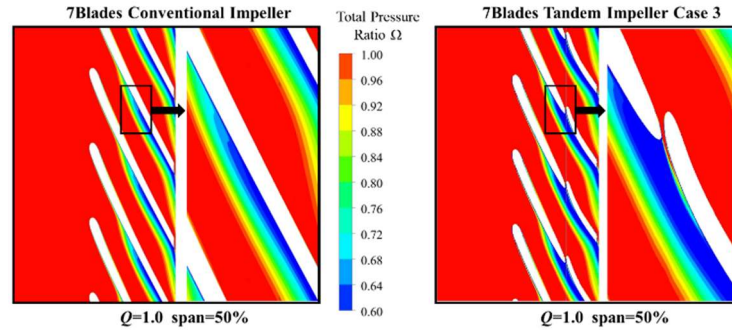
(b) 4 Blades



(c) 5 Blades



(d) 6 Blades

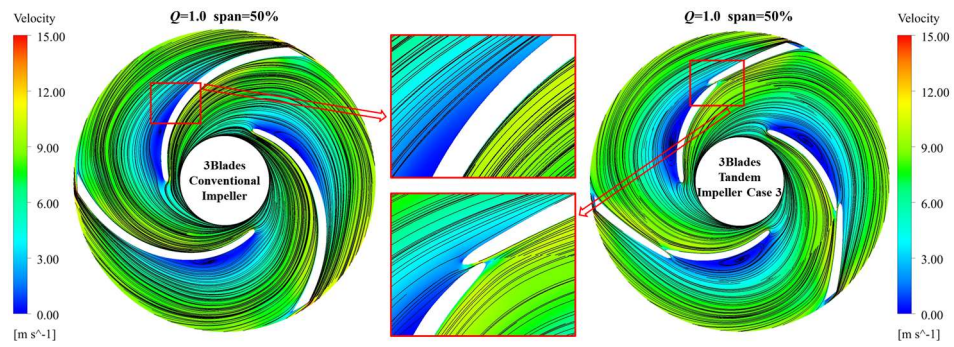


(e) 7 Blades

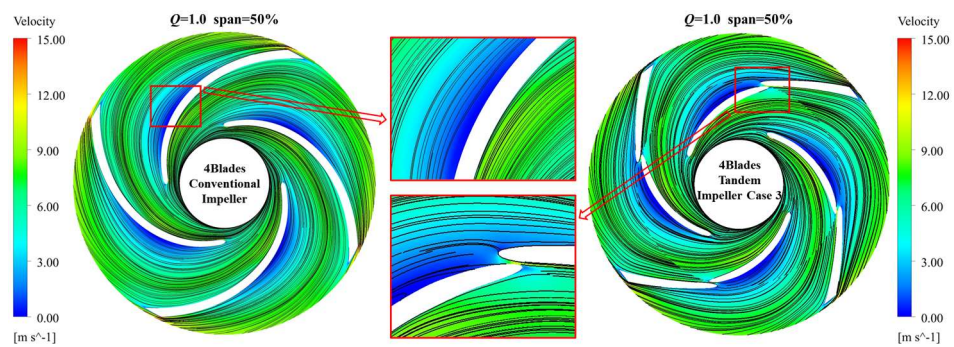
Fig. 7.1. Comparison of total pressure ratio of conventional and tandem impellers with different blade numbers.

This study further compares the velocity and streamlines distributions of conventional and tandem impellers with different blade numbers under the design condition. Fig. 7.2 presents the relative velocity contours and streamlines at the mid-span (span=50%) cross section for both impellers. When the blade number is small, distinct low-velocity regions can be observed near the blade LE and along the PS for both impellers. These regions result from severe flow separation caused by the mismatch between the inlet flow direction and the blade setting angle. As the number of blades increases, this phenomenon gradually diminishes. In the tandem impeller, the gap is located near the TE of the low-velocity region, effectively acting as a flow barrier that suppresses the further expansion of the low-velocity region when the blade number is small. Detailed observations reveal that part of the flow can travel from the PS of the front blade to the SS of the rear blade. However, slight flow separation and localized low-velocity regions still occur along the SS. Moreover, the rear blade of the tandem impeller, with a reduced inlet setting angle β_{21} , achieves better alignment with the incoming flow. When the blade number exceeds 5, small-scale low-velocity regions persist around the gap, but the streamlines distribution along the SS of the rear blade becomes more uniform compared with that of the conventional impeller. Particularly for the 6-blade and 7-blade impellers, the tandem impeller exhibits a more uniform velocity field and denser streamlines distribution near

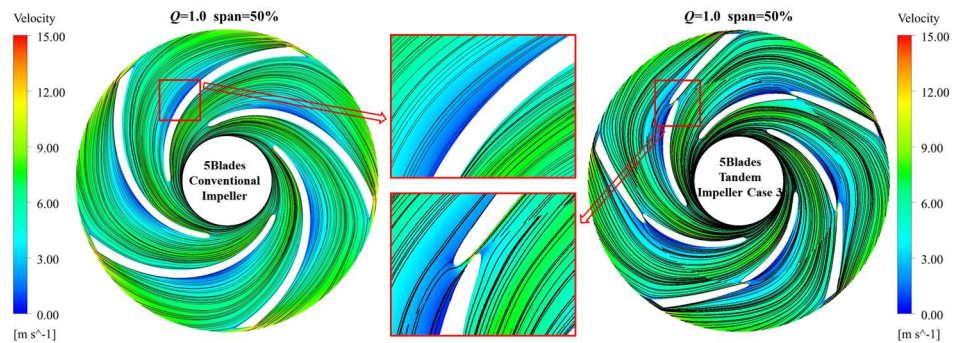
the rear part of the flow passage and at the impeller outlet.



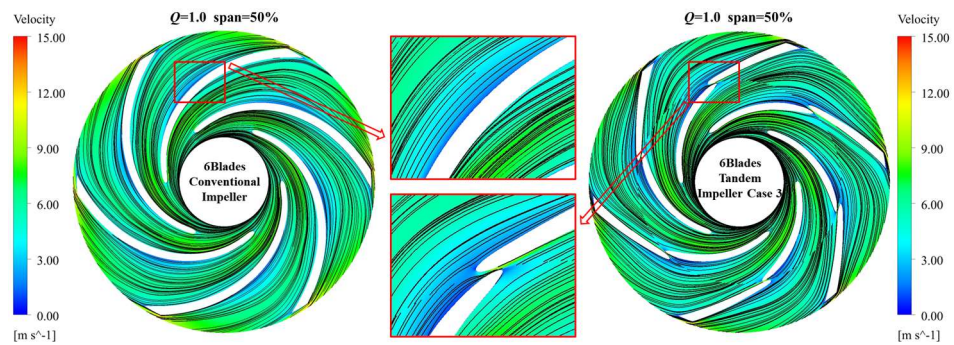
(a) 3 Blades



(b) 4 Blades



(c) 5 Blades



(d) 6 Blades

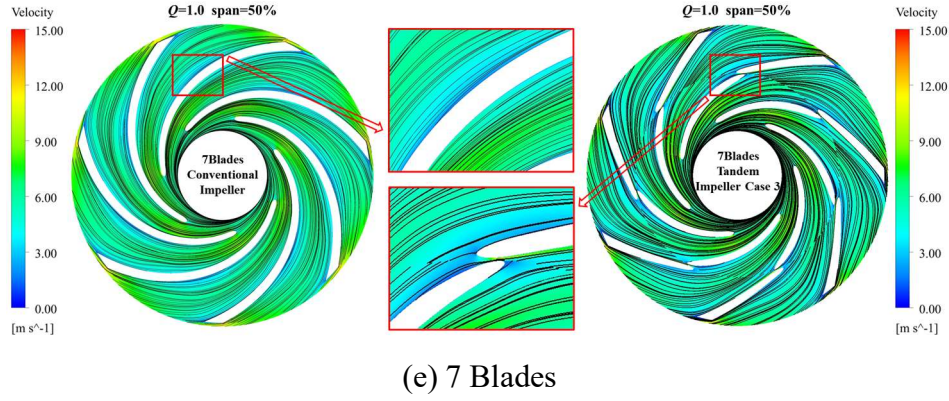


Fig. 7.2. Comparison of velocity streamlines of conventional and tandem impellers with different blade numbers.

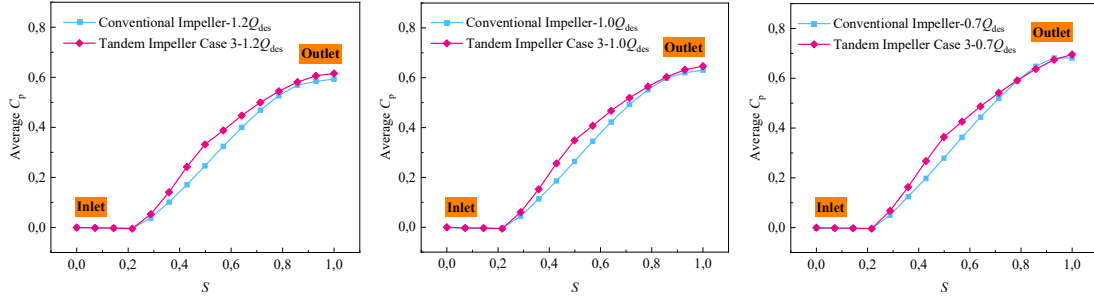
The most significant advantage of the tandem impeller lies in its ability to enhance the H of the centrifugal pump while maintaining the same level of η . To elucidate the mechanism behind this performance improvement, a quantitative analysis was conducted. First, the pressure variation from the impeller inlet to the outlet was evaluated for both impellers. The averaged pressure coefficient, denoted as C_p , was introduced to characterize this variation. In this study, C_p is defined as follows:

$$C_p = (P_{ave} - P_{ref}) / 0.5 \rho u_2^2 \quad (7.1)$$

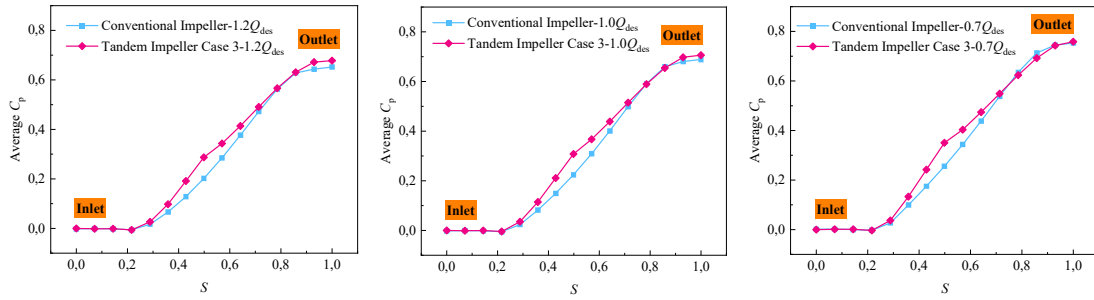
where P_{ave} denotes the average pressure over the final two revolutions of the unsteady simulation, P_{ref} represents the reference pressure defined as the average pressure at the impeller inlet, and u_2 is the circumferential velocity at the impeller outlet.

Fig. 7.3 compares the C_p distribution curves along the streamlines direction from the inlet to the outlet of two impellers with different blade numbers under three typical flow rates ($1.2Q_{des}$, $1.0Q_{des}$, and $0.7Q_{des}$). The horizontal coordinate S represents the dimensionless distance, where $S=0$ denotes the impeller inlet and $S=1$ denotes the impeller outlet. For both impeller types, the C_p value at the inlet region ($S=0-0.2$) remains constant across all blade numbers and operating conditions, consistent with the phenomenon observed in Fig. 7.1. From the LE of the blade ($S=0.25$), the work done by the blade on the fluid begins, and C_p also starts to rise rapidly. Moreover, the smaller the Q , the steeper the increase in C_p . For the same blade numbers, the C_p values in the middle section ($S=0.4-0.8$) of the tandem impeller are significantly higher than those of the conventional impeller. Moreover, as Q increases and blade numbers grow, more blades exhibit higher C_p values. Furthermore, under various operating conditions for different blade numbers, the increase in C_p for the tandem impeller always peaks at the gap position ($S=0.5$), with this phenomenon becoming more pronounced under overload conditions with more blade numbers. The above observations indicate that, across all blade numbers and operating conditions, the gap region between the tandem blades effectively enhances the energy

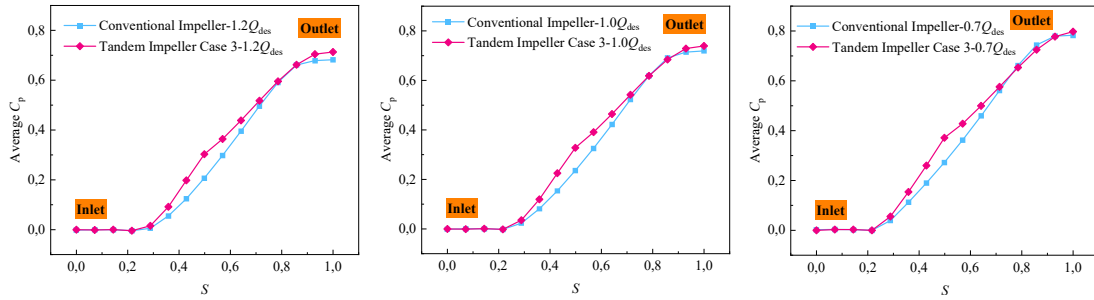
conversion capability of impellers. This strengthening effect becomes more pronounced with increasing blade number and flow rate. This is also the primary reason why the tandem impeller in Fig. 6.23 can increase the H of the centrifugal pump across multiple operating conditions with different blade numbers compared to the conventional impeller. Moreover, at higher blade numbers and larger flow rates, the C_p values in the gap region of the tandem impeller increase more significantly, corresponding to the $\Delta H_{\max}$ observed under overload conditions, as illustrated in Fig. 6.23(d).



(a) 3 Blades



(b) 4 Blades



(c) 5 Blades

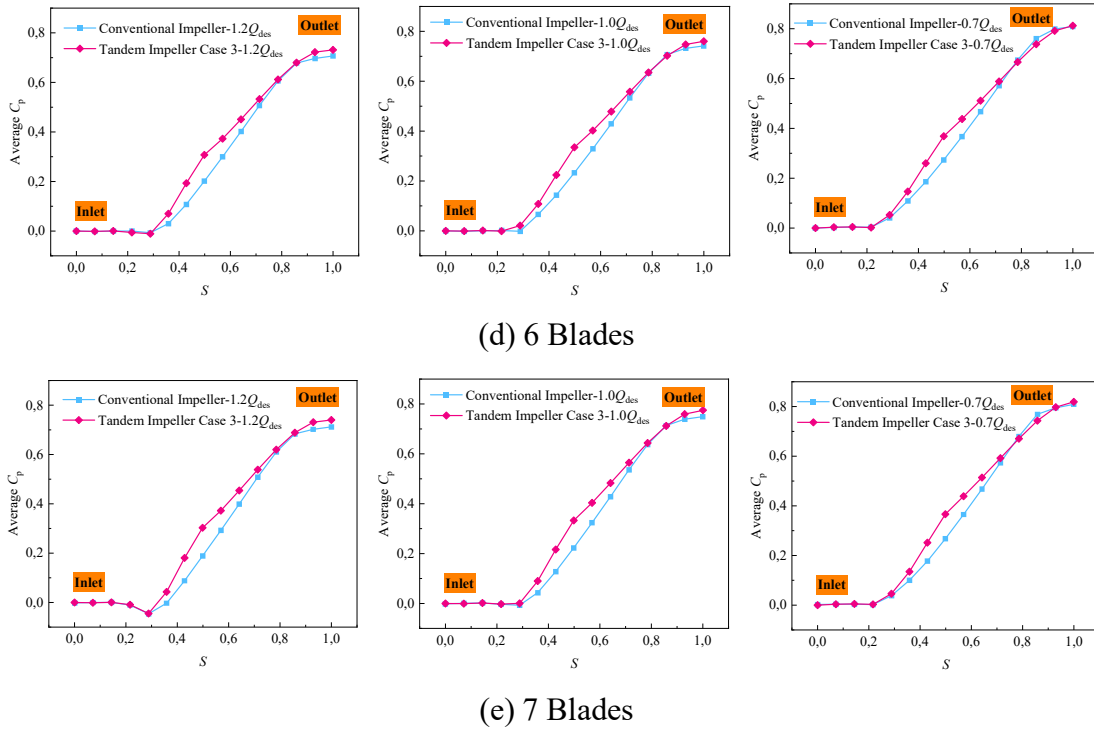
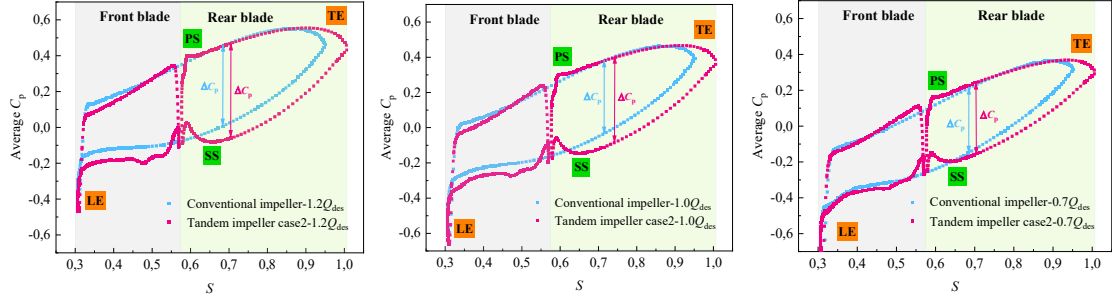


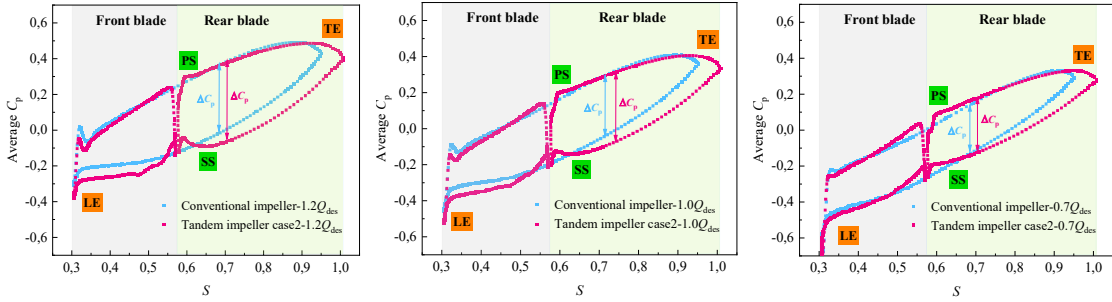
Fig. 7.3. Comparison of pressure distribution of conventional and tandem impellers with different blade numbers.

The pressure load distribution on the blade surface is a key indicator of the impeller energy conversion capability. In this study, the average pressure load on the blade surfaces of both impeller types was analyzed under three operating conditions and for five different blade numbers. Fig. 7.4 presents the variation of the averaged pressure coefficient C_p along the blade surface, from the LE to the TE, at the mid-span position (span=50%) for both impellers. The pressure difference between the PS and SS, denoted as ΔC_p , directly reflects the impeller work capacity. As the number of blades decreases and the Q increases, the ΔC_p value on each blade rises accordingly. This indicates that fewer blades result in a higher work-load per blade, and higher Q also demand greater work from each blade. Across all five blade numbers and three operating conditions, the most notable differences in C_p between the tandem and conventional impellers are observed on the rear blade surface region of the tandem impeller ($S=0.55-1$). The tandem impeller employs a modified rear blade, which exhibits a larger ΔC_p value on the rear blade surface compared with the conventional impeller, indicating a larger enclosed area under the C_p curve. Under the same blade number, the tandem impeller shows a more significant increase in ΔC_p under $1.2Q_{des}$, with both the front and rear blades contributing to higher ΔC_p values. Across different blade numbers, the ΔC_p value on each rear blade of the tandem impeller remains higher than that of the conventional impeller. The cumulative effect of multiple blades with higher ΔC_p leads to a more substantial increase in the overall pump H . Moreover, the tandem impeller achieves a more rational pressure load distribution

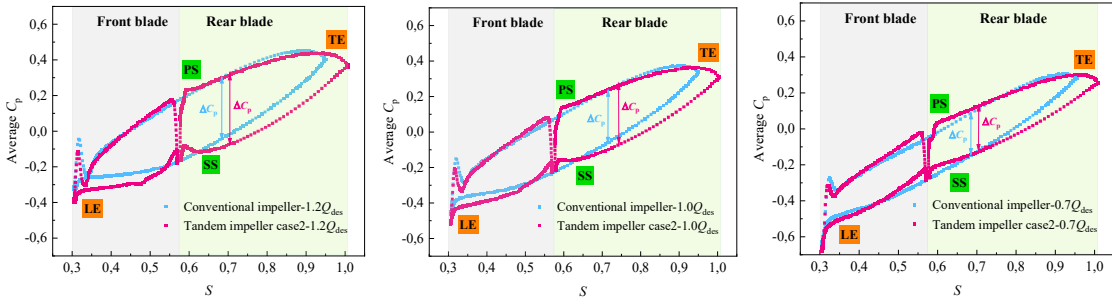
through a two-stage loading process, which helps suppress flow separation that typically occurs in the mid-blade high-load region of the conventional impeller. In summary, the enhanced work capability of the modified rear blade and its more stable and efficient energy conversion are the direct reasons why the tandem impeller can increase the pump H . This improvement in pressure load characteristics becomes more pronounced under overload conditions, and the cumulative effect with an increasing number of blades further contributes to a greater enhancement in pump H .



(a) 3 Blades



(b) 4 Blades



(c) 5 Blades

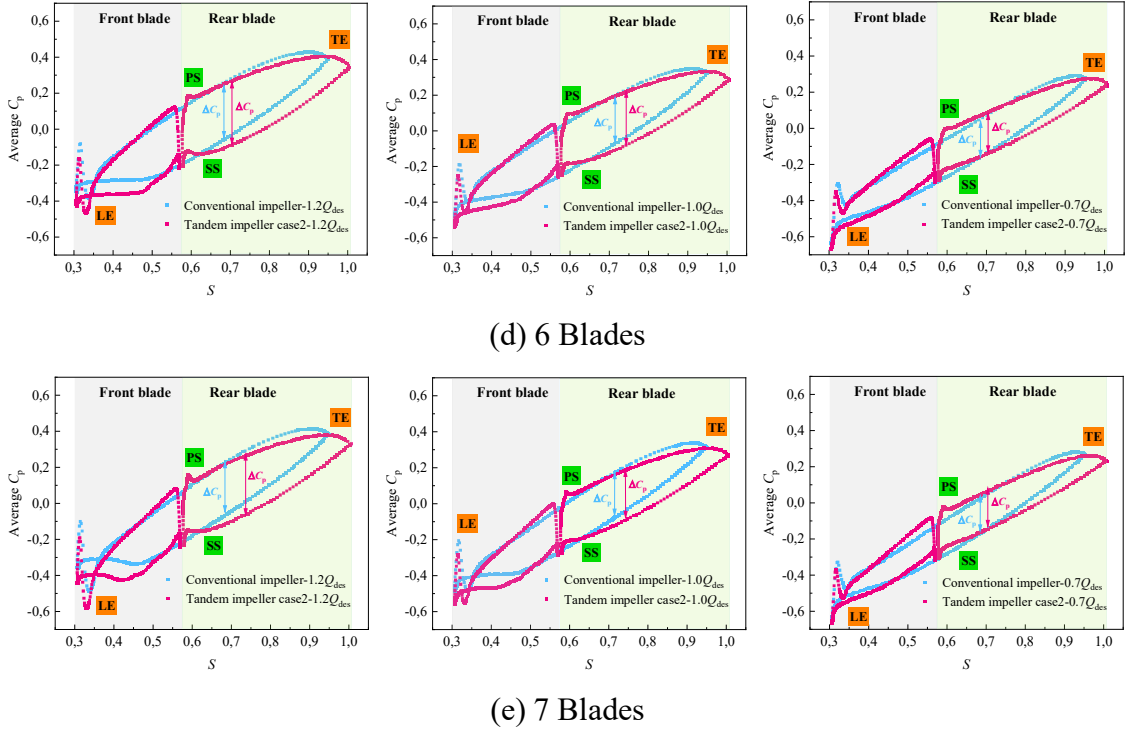


Fig. 7.4. Comparison of blade surface load distribution of conventional and tandem impellers with different blade numbers.

The reason for the H increase in the tandem impeller can also be explained by the Euler equation, which is the fundamental relationship for energy conversion in incompressible Newtonian fluids. It can be defined as:

$$g \cdot H_{th} = u_{22} \cdot c_{u,22} - u_{11} \cdot c_{u,11} \quad (7.2)$$

For the spinless inflow in this study, it is possible to simplify Eq. 7.2 to Eq. 7.3:

$$g \cdot H_{th} = u_{22} \cdot c_{u,22} \quad (7.3)$$

The conventional impeller and tandem impeller in this study share identical rotational speeds and impeller diameters, meaning the circumferential velocity u_{22} at the impeller outlet is the same. Therefore, the H increase in the tandem impeller can only be attributed to the $c_{u,22}$. The ratio of $c_{u,22}$ to u_{22} at the outlet of two impellers is calculated for four operating conditions with five blade numbers in Fig. 7.5. The values of $c_{u,22}$ in the figure represent the mass flow-averaged at the impeller outlet, obtained from the average of the last two rotations of the unsteady simulations.

The $c_{u,22}/u_{22}$ values for both impeller types increase with the blade numbers, with the tandem impeller exhibiting a greater increase. The $c_{u,22}/u_{22}$ values for both impellers decrease with increasing Q , and the tandem impeller shows a more gradual change trend. Under all conditions with five impeller blades, the $c_{u,22}/u_{22}$ value of the tandem impeller consistently exceeds that of the conventional impeller. For the same blade number, the

$c_{u,22}/u_{22}$ value of the tandem impeller is significantly higher than that of the conventional impeller under overload conditions. For different blade numbers, as the number of blades increases, the $c_{u,22}/u_{22}$ value of the tandem impeller also becomes progressively higher than that of the conventional impeller. As a result, based on Eq. 7.3, the H of the tandem impeller would be higher than that of the conventional impeller across all operating conditions for any blade numbers, with the greatest difference under overload conditions where more blades are employed.

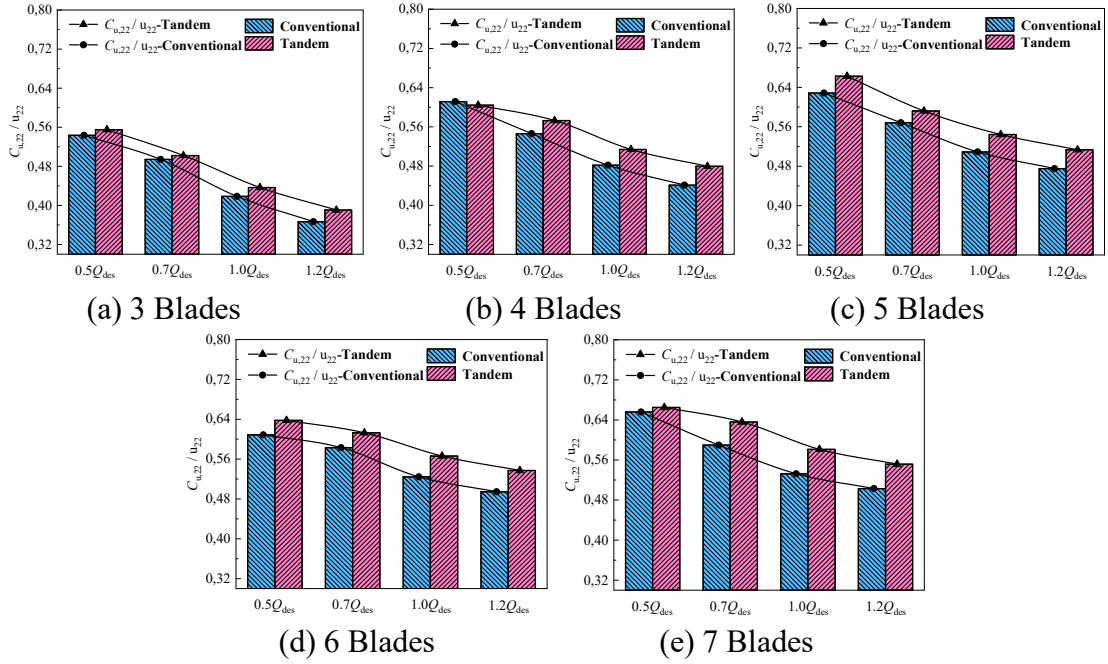
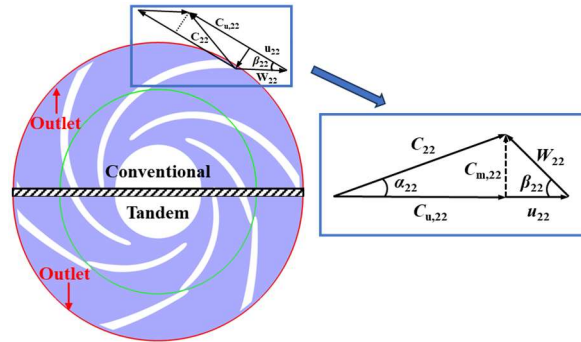


Fig. 7.5. Comparison of $c_{u,22}/u_{22}$ distribution of conventional and tandem impellers with different blade numbers.

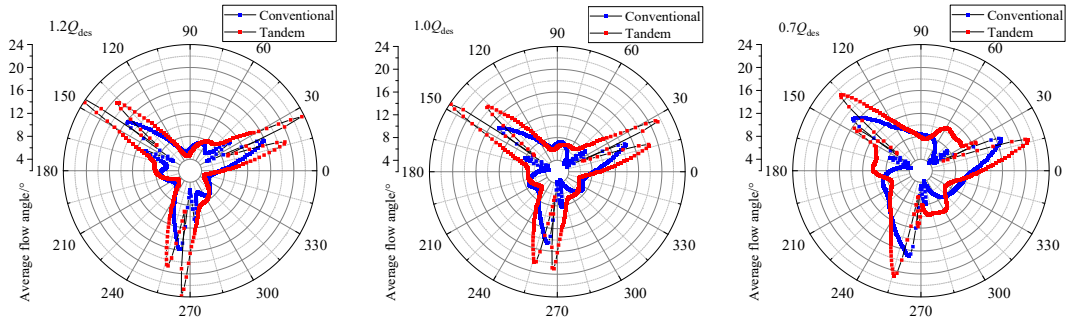
To further observe the flow uniformity at the impeller outlet and analyze the reasons for the differences in $c_{u,22}$, this study statistically analyzed the circumferential distribution of the average relative flow angle β_{22} at span=50% of the impeller outlet. As shown in Fig. 7.6(a), the fluid velocity relationship at the outlet of both impellers can be represented by the velocity triangle. The absolute flow angle α_{22} characterizes the angle between the absolute velocity c_{22} and the circumferential velocity $c_{u,22}$, while the relative flow angle β_{22} reflects the angle between the fluid relative velocity w_{22} and the flow direction at the blade outlet edge. Ideally, β_{22} should approach the outlet angle of the blade as closely as possible. However, due to the effects of secondary flow, jet wakes, and non-uniform flow, β_{22} often deviates from the design value. Therefore, investigating the distribution of β_{22} can help identify discrepancies between the actual flow pattern and the ideal pattern. Furthermore, the variation in β_{22} is closely related to that of $c_{u,22}$, with larger β_{22} values typically yielding greater $c_{u,22}$ values, thereby increasing the H of the pump.

Fig. 7.6(b)-(f) compares the circumferential distribution of β_{22} at the impeller outlet for

the two impeller configurations under different blade numbers, where all β_{22} values represent the average over the final two revolutions of the impeller. Both impellers exhibit distinct periodic fluctuations of β_{22} along the circumferential direction, with the number of periods corresponding exactly to the blade numbers. The peaks and troughs of β_{22} are closely associated with unsteady flow phenomena such as jet-wake interactions and flow separation at the impeller outlet. The comparison between the two impellers shows that, for the same number of blades, the tandem impeller exhibits higher β_{22} peaks and troughs than the conventional impeller, and this difference increases with Q . Across all three operating conditions, the overall β_{22} distribution range of the tandem impeller is 5° - 10° higher than that of the conventional impeller, while also exhibiting improved circumferential uniformity. Under varying blade numbers, the tandem impeller consistently reduces the low-angle regions observed in the conventional impeller, with the degree of improvement becoming more pronounced as the blade count increases, indicating suppression of flow separation near the blade TE. As the number of blades increases, the overall β_{22} values of the tandem impeller become significantly higher than those of the conventional impeller, particularly in the trough regions. Consequently, with the increase and homogenization of β_{22} at the tandem impeller outlet, the $c_{u,22}$ is substantially enhanced. Ultimately, the tandem impeller achieves higher energy conversion efficiency and improved flow stability per unit mass at the impeller outlet. These factors constitute the main reasons for the observed head improvement, especially under overload conditions and with a larger number of blades.



(a) Position



(b) 3 Blades

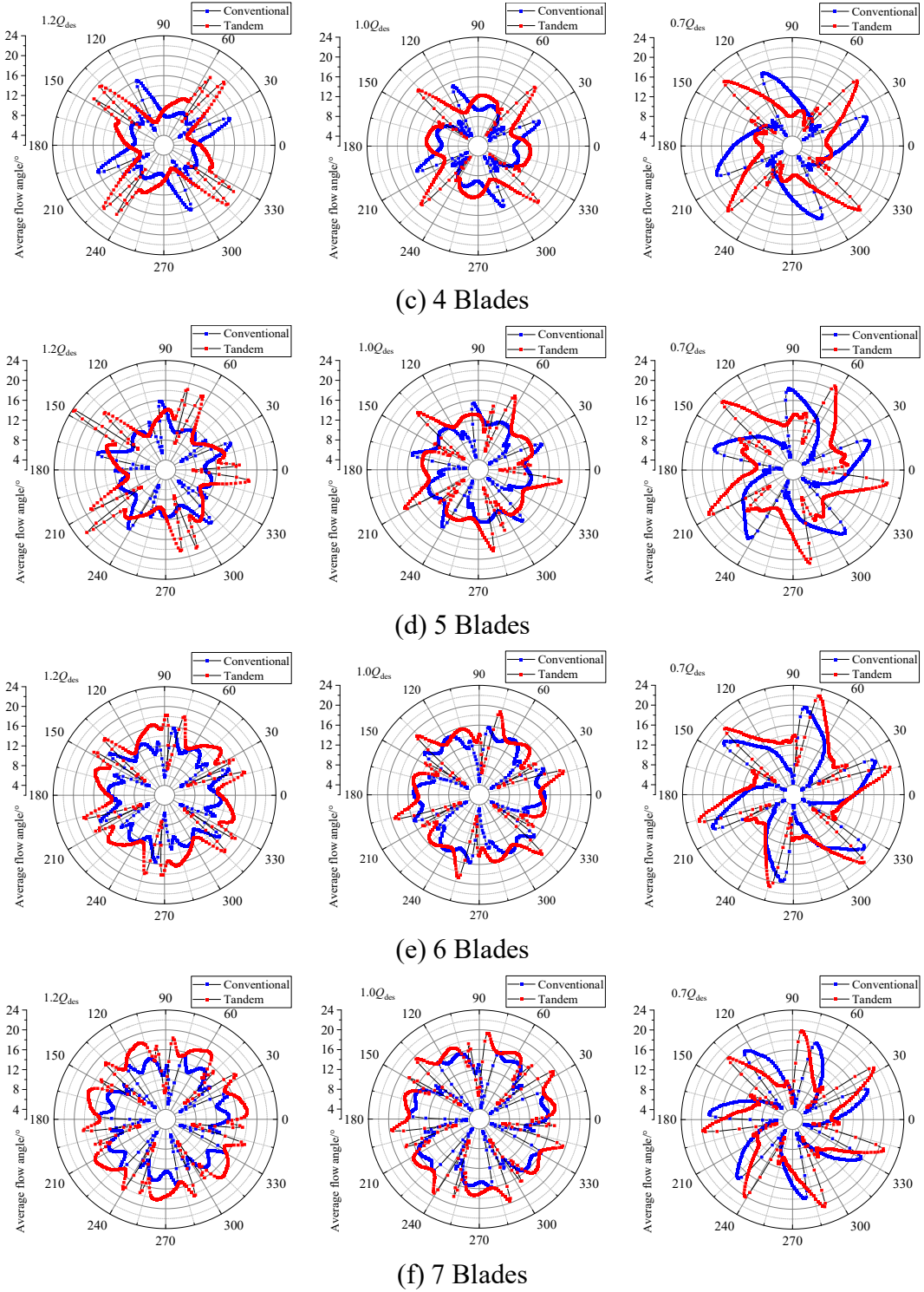
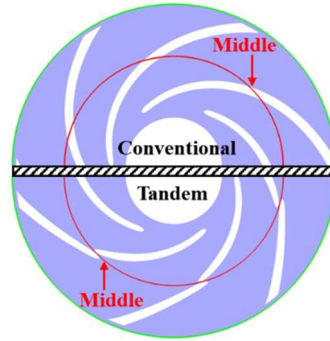


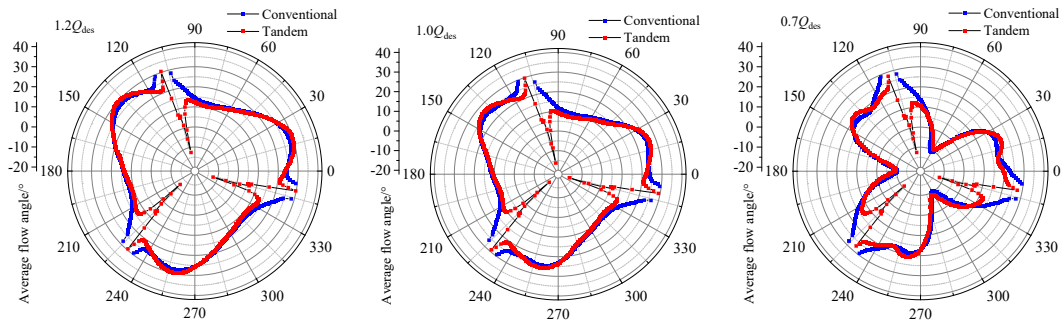
Fig. 7.6. Comparison of flow angle of conventional and tandem impellers outlet with different blade numbers.

The most distinctive structural feature of the tandem impeller compared with the conventional impeller is the gap located in the middle of the impeller. Therefore, Fig. 7.7 also presents the circumferential distribution of β_{22} at the gap position, with Figs. 7.7(b)-(f) showing the β_{22} distributions under three representative operating conditions for five

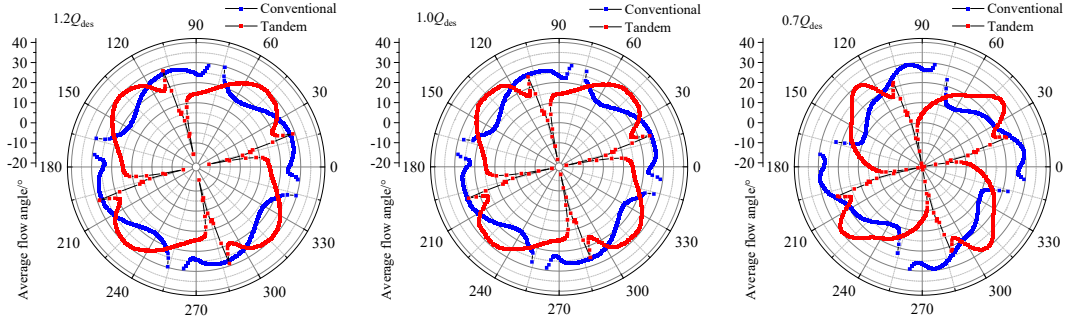
different blade numbers. Across all blade numbers, the most notable difference is that the β_{22} value at the gap of the tandem impeller drops sharply, and in some cases even becomes negative, compared with the conventional impeller. This indicates that the flow in the gap region is highly unstable, and reverse flow may even occur. Such behavior is closely related to the flow separation near the TE of the front blade and the LE of the rear blade. For both impellers, β_{22} fluctuations are more pronounced under part-load conditions than under overload conditions, with the tandem impeller exhibiting lower β_{22} values in the gap region. This suggests that flow deterioration in the tandem impeller gap is most severe under part-load conditions. As the blade number and Q increase, the β_{22} distributions in the flow passages of both impellers become progressively more stable, reaching their most uniform state when the blade number is 6. A further comparison reveals that in the conventional impeller, large velocity gradients in the boundary layers on the blade PS and SS lead to rapid increases in β_{22} . In contrast, the tandem impeller exhibits stronger β_{22} fluctuations under part-load conditions, with steep rises and drops indicating intensified unsteady behavior in the flow passages. Therefore, the unstable flow in the tandem impeller gap may cause additional hydraulic losses and increase the torque acting on the blades, potentially offsetting the positive impact of the H improvement on η .



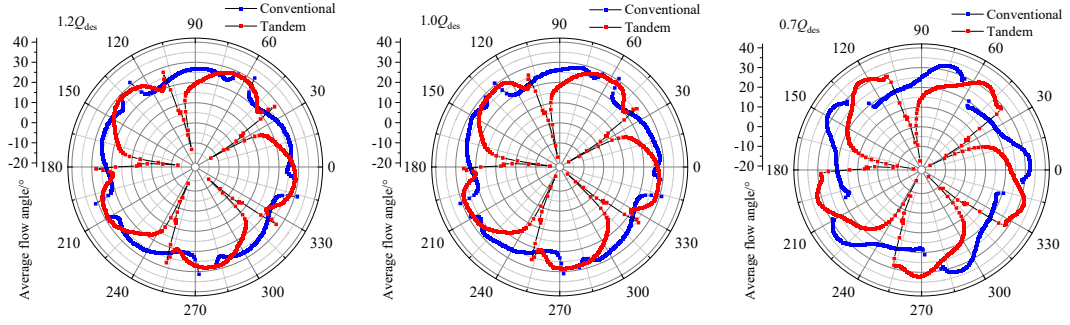
(a) Position



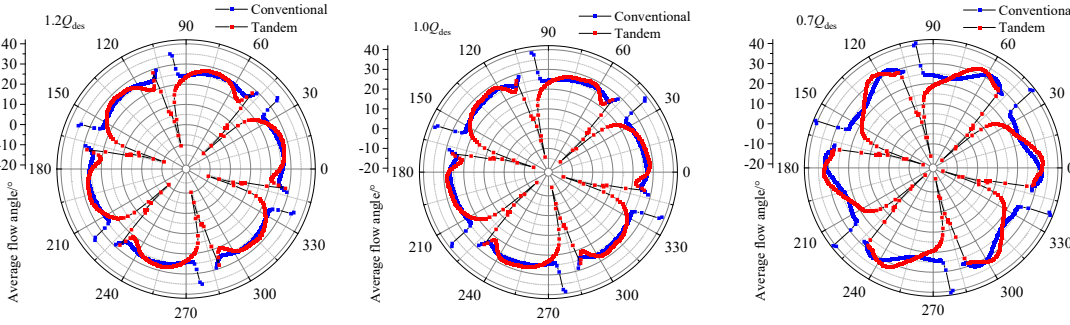
(b) 3 Blades



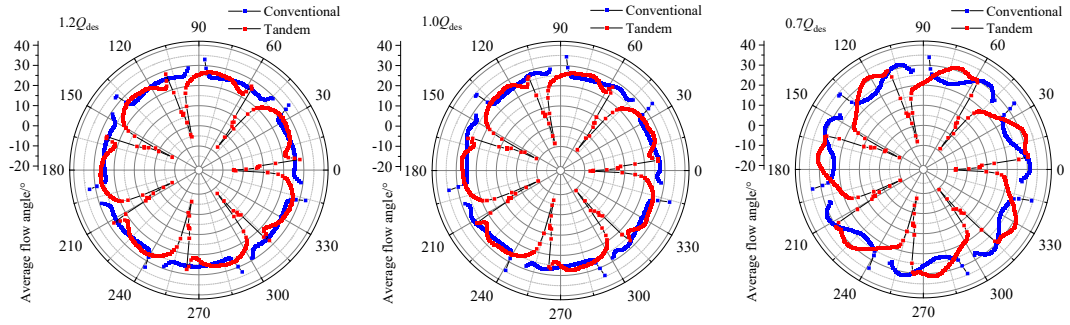
(c) 4 Blades



(d) 5 Blades



(e) 6 Blades



(f) 7 Blades

Fig. 7.7. Comparison of flow angle of conventional and tandem impellers middle with different blade numbers.

7.1.2 Potential Rothalpy Distribution

The quantitative analysis in the previous section reveals that the flow patterns in both conventional and tandem impellers are unstable, particularly when the number of blades

is low and flow rates are small. To further compare and analyze flow stability in the impeller, this study introduces the concept of two-force competition in the impeller. Under the basic assumptions of steady-state, inviscid flow, and zero blade thickness, the momentum equation for relative motion in the impeller is:

$$\begin{cases} \mathbf{V}_r \cdot \nabla \mathbf{V}_r = -\nabla I_p - 2\boldsymbol{\omega}_i \times \mathbf{V}_r \\ I_p = \frac{p}{\rho} - \frac{\boldsymbol{\omega}_i^2 r_a^2}{2} \end{cases} \quad (7.4)$$

where I_p is referred to as the potential rothalpy in this research [115, 116]. The Poisson equation for relative motion in the impeller is obtained by taking the divergence of this equation:

$$\Delta I_p - 2\boldsymbol{\omega}_i \cdot \nabla \times \mathbf{V}_r = 2Q_r \quad (7.5)$$

where Q_r denotes the second invariant of the relative velocity gradient tensor $\nabla \mathbf{V}_r$ in the impeller domain. Rigid vorticity reasonably characterizes vortex motion. When the rigid rotational motion represented by rigid vorticity significantly dominates the shear motion represented by deformable vorticity, vortex motion becomes predominant, i.e., the relative vortex intensity Ω_R exceeds 0.5, signaling the generation of local secondary vortices. Specifically, an equivalent relationship between Q_r and Ω_R is:

$$Q_r = \lambda_{ci}^2 \cdot \left[1 - 3 \left(\frac{\lambda_{cr}}{\lambda_{ci}} \right)^2 \right] > 0 \Leftrightarrow \Omega_R = \frac{(\boldsymbol{\omega} \cdot \mathbf{r})^2}{2 \cdot [(\boldsymbol{\omega} \cdot \mathbf{r})^2 - 2\lambda_{ci}^2 + 2\lambda_{cr}^2 + \lambda_r^2]} > 0.5 \quad (7.6)$$

The equivalent relationship in the above equation indicates that secondary vortices in the impeller passages are driven jointly by the potential rothalpy of the rotor and the Coriolis force F_c . Since F_c is always vertical to the relative velocity $\mathbf{V}_r$ and does not perform work, under actual conditions, the potential rothalpy gradient $-\nabla I_p$ (PRG) becomes the sole driving force for actively inducing secondary flow.

According to Newton's second law, force is the primary reason for the change in the state of motion of an object. As shown in Fig. 7.8, under ideal flow conditions (red line), when the $F_c = -2\boldsymbol{\omega}_i \times \mathbf{V}_r$ opposes and exceeds the PRG, the fluid can flow along the geometric contour of the blade. Under ideal conditions, the PRG and F_c are always vertical to the relative velocity $\mathbf{V}_r$. Therefore, this study assesses flow stability by evaluating the angle between the PRG and $\mathbf{V}_r$ in both impellers, as well as the magnitude of the PRG itself. A closer angle of 90 degrees between the PRG and $\mathbf{V}_r$ indicates more stable flow.

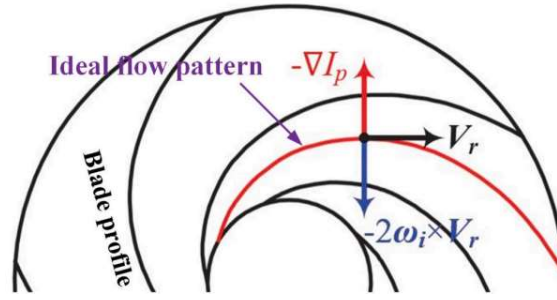
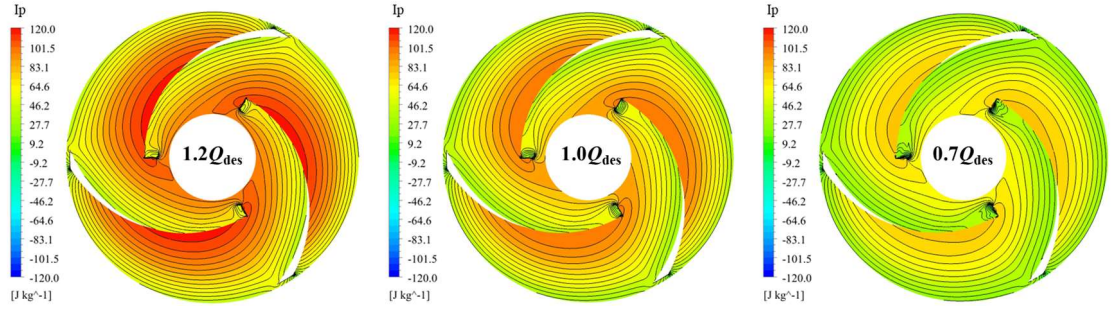
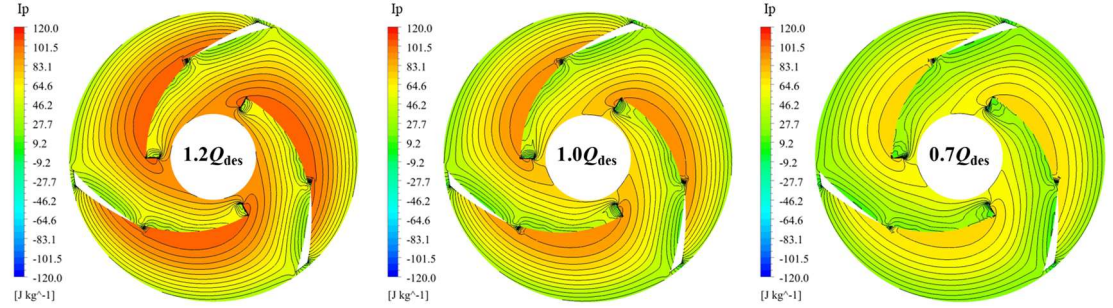


Fig. 7.8. Two force actions under ideal flow pattern in the impeller.

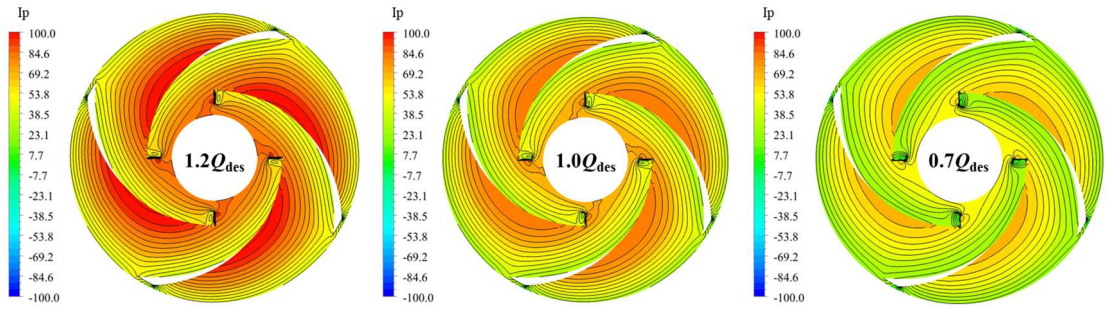
The comparison of the iso-contour distributions of the I_p for the two impeller configurations at the mid-span (span=50%) and on the blade surfaces under five different blade numbers is illustrated in Fig. 7.9. Under ideal and stable flow conditions, the I_p iso-contours are expected to align with the streamlines. Both impellers exhibit higher I_p values at larger flow rates and with fewer blades. In both configurations, a wide region of positive I_p appears on the PS near the mid-span, and the I_p contour lines deviate from the blade surface. High I_p values indicate the presence of PRG directed from the PS to SS along the S1 surface, which induces secondary flows toward the SS, forming axial separation vortices. This high- I_p phenomenon is more pronounced in the conventional impeller, indicating that the tandem impeller effectively suppresses mid-passage flow separation. Near the impeller outlet, the difference in I_p values between the PS and SS gradually decreases for both impellers. Consequently, the increasingly dominant centrifugal force F_c causes the flow to deflect from the SS toward the PS, leading to velocity slip. For the same blade number, the I_p values in the tandem impeller passage are consistently lower than those in the conventional impeller under various operating conditions, and the reduction becomes more significant under overload conditions. As the number of blades increases, the tandem impeller exhibits more uniform and smaller I_p distributions across the passage, which favors a more gradual energy conversion and thereby contributes to a higher H generation. However, for all blade numbers, closed-loop I_p contours appear near the LE of both the front and rear blades in the tandem impeller, indicating enhanced local flow instability. This feature becomes particularly evident under part-load conditions with fewer blades, where the closed I_p regions mark the onset of spanwise separation vortices and backflow, leading to increased flow losses in the tandem impeller passage.



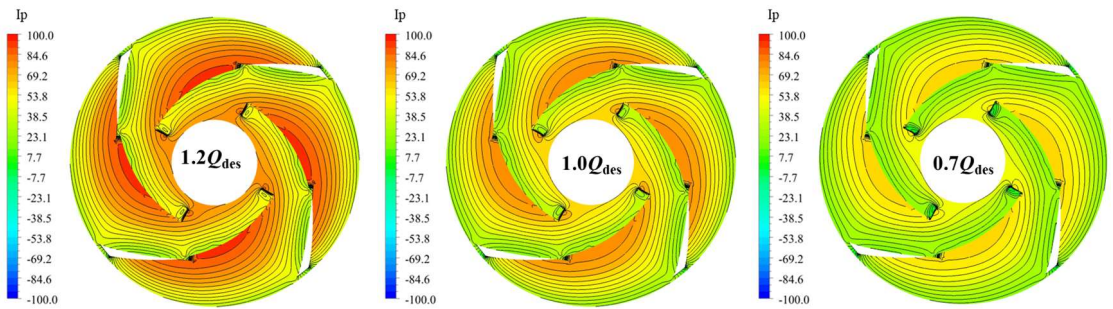
(a) Conventional impeller with 3 blades



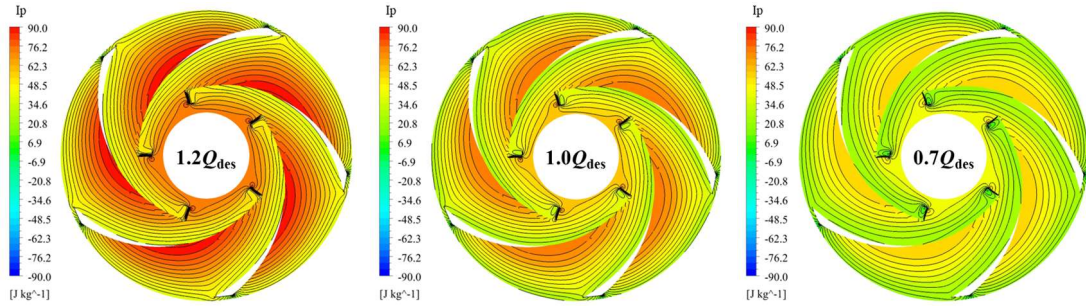
(b) Tandem impeller with 3 blades



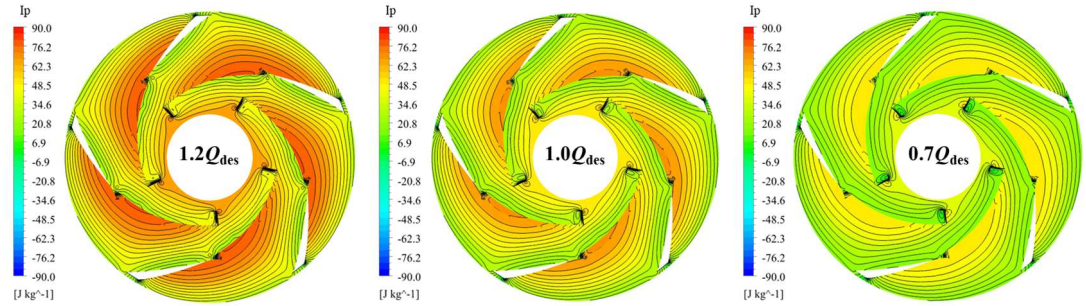
(c) Conventional impeller with 4 blades



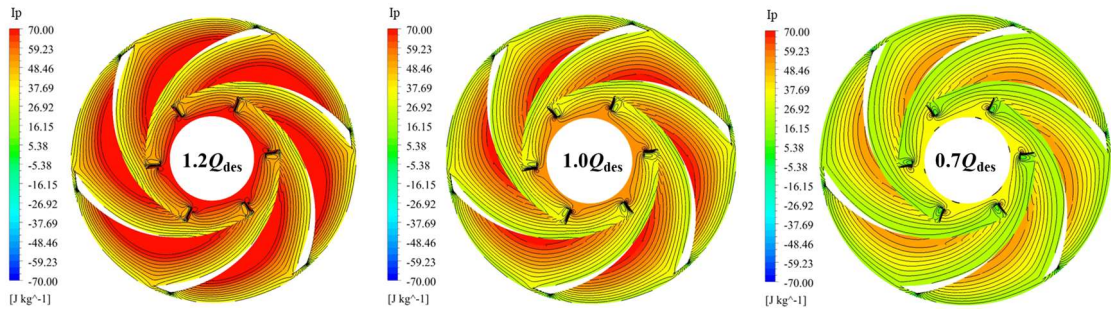
(d) Tandem impeller with 4 blades



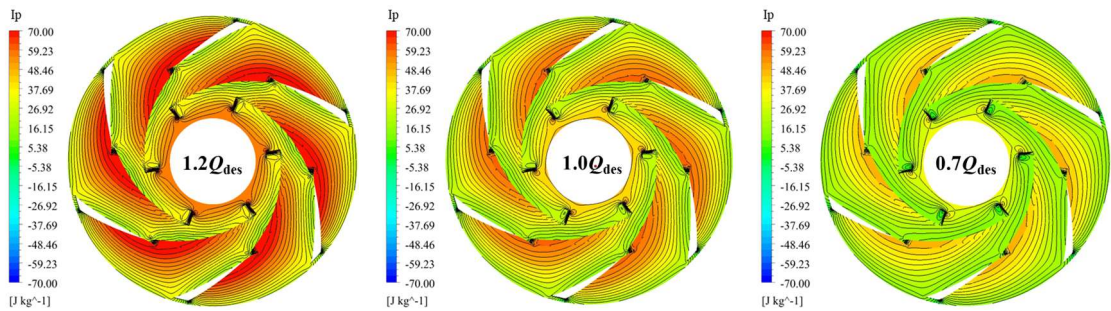
(e) Conventional impeller with 5 blades



(f) Tandem impeller with 5 blades



(g) Conventional impeller with 6 blades



(h) Tandem impeller with 6 blades

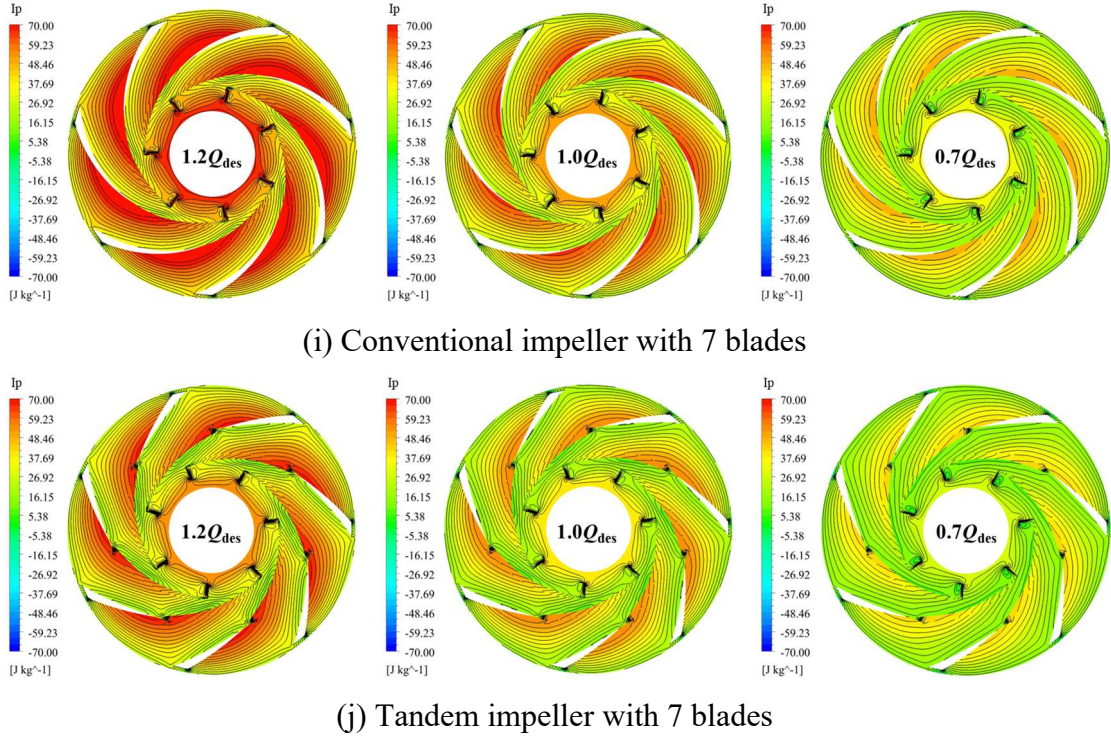
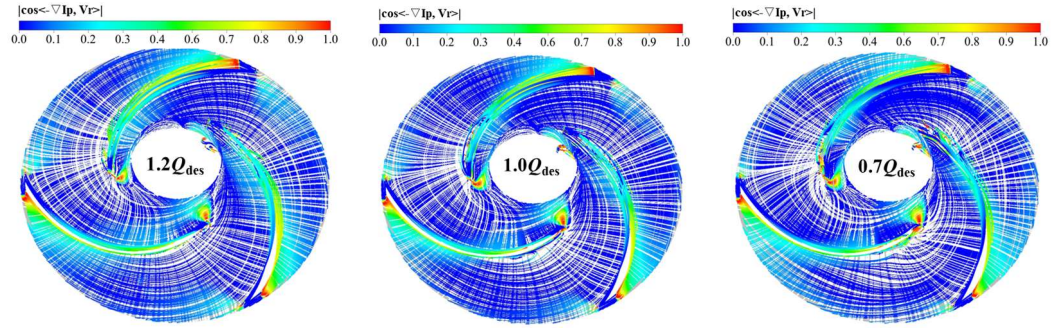


Fig. 7.9. Comparison of potential rothalpy of conventional and tandem impellers with different blade numbers.

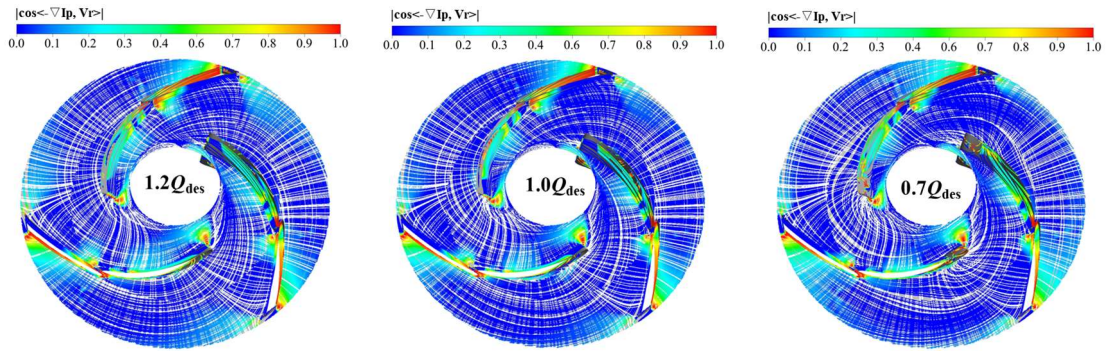
The distribution of relative velocity streamlines and PRG contour lines on the two impellers at span=50% cross-section and blades with five blade numbers is shown in Fig. 7.10. The color of the relative velocity streamlines in the figure represents the cosine of the angle between the PRG vector and the relative velocity vector: $|\cos\langle -\nabla I_p, V_r \rangle|$. When the cosine value approaches 0, the angle between the two vectors tends toward 90 degrees, indicating flow approaching an ideal steady state. Conversely, when the cosine value approaches 1, it indicates that the PRG and relative velocity vectors are nearly collinear, signifying extremely high flow instability.

In both impeller configurations, the cosine values in the core flow region at the mid-passage remain close to 0, indicating that the flow in this region is relatively stable. However, near the PS of the LE and TE, the orthogonality between the two local velocity vectors decreases significantly. This effect is especially pronounced under part-load conditions and with fewer blades, where the cosine values approach 1, signifying distinct flow separation and vortex formation, characteristic of flow instability. For the same blade number, the tandem impeller exhibits noticeably higher cosine values, approaching 1, at the PS of the front-blade TE, the SS of the rear-blade LE, and the PS of the rear blade, compared with the conventional impeller. As the blade number increases, the cosine values at multiple locations, particularly in the gap and near the TE, become significantly higher for the tandem impeller than for the conventional design, especially under overload conditions. These results indicate that the flow instability in the tandem impeller

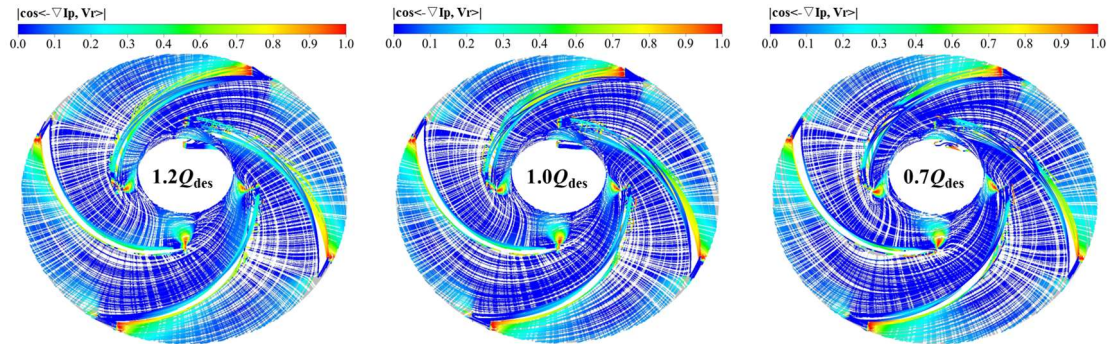
intensifies markedly at the gap and on the PS of the rear blade. Such unsteady flow patterns lead to greater torque generation on both the blades as well as the front and rear shrouds. Moreover, as the number of blades increases, the cumulative torque in the tandem impeller becomes substantially larger than that in the conventional impeller. Ultimately, the continuously increasing torque in the tandem configuration offsets the positive effect of H enhancement on η .



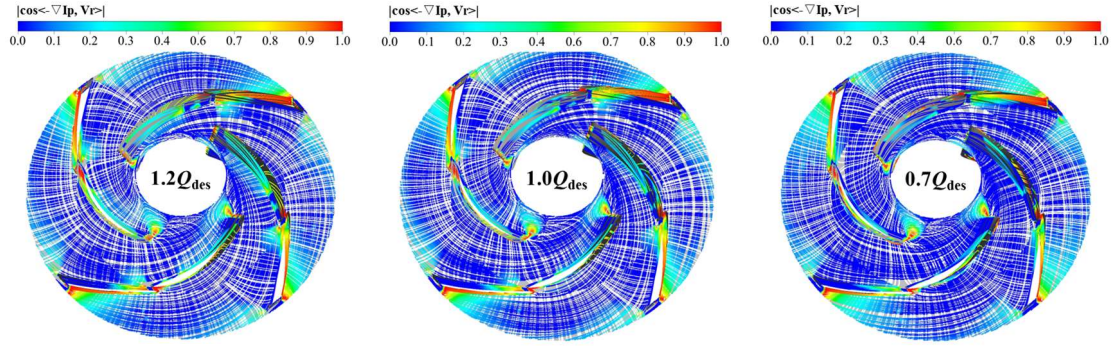
(a) Conventional impeller with 3 blades



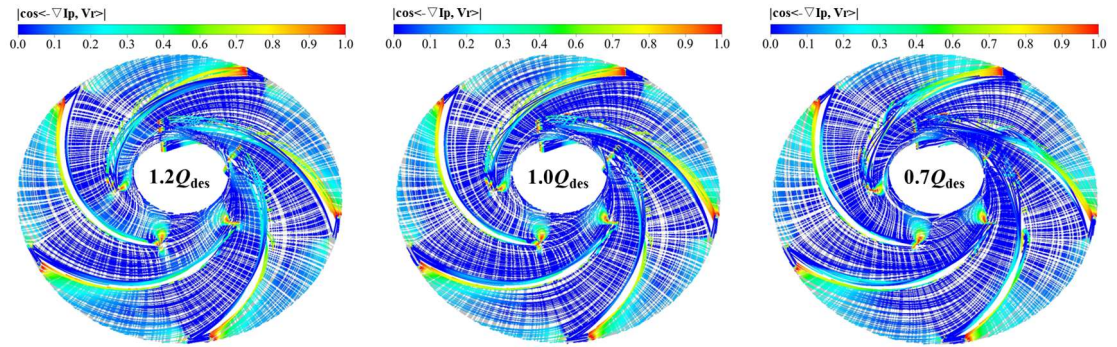
(b) Tandem impeller with 3 blades



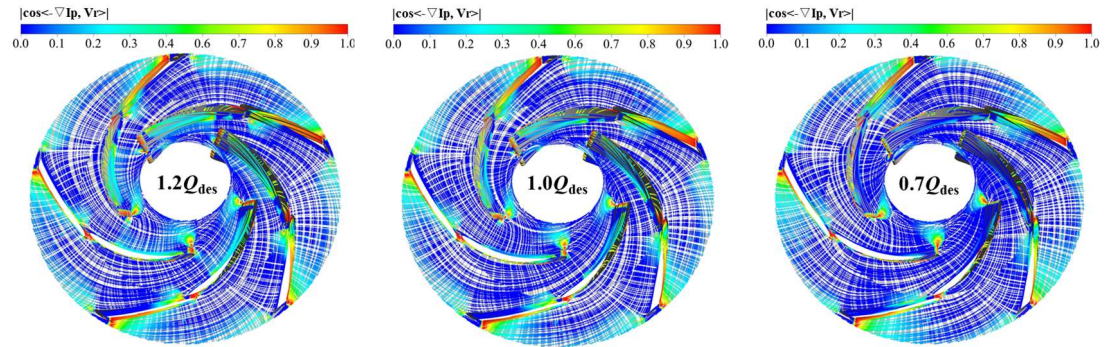
(c) Conventional impeller with 4 blades



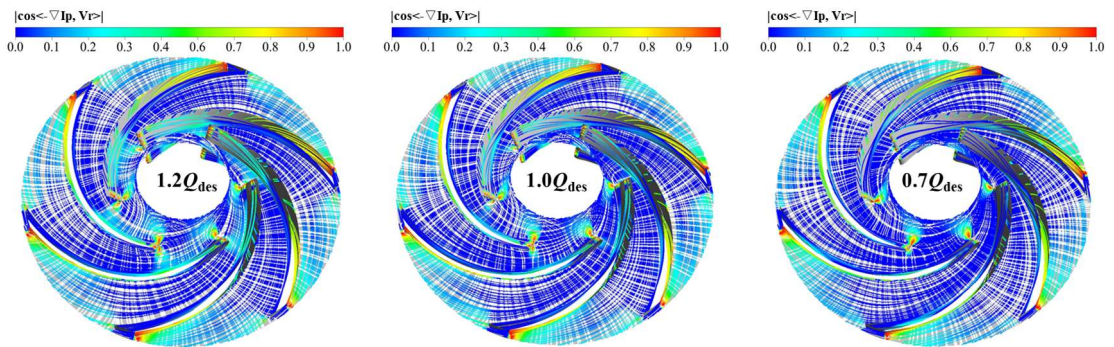
(d) Tandem impeller with 4 blades



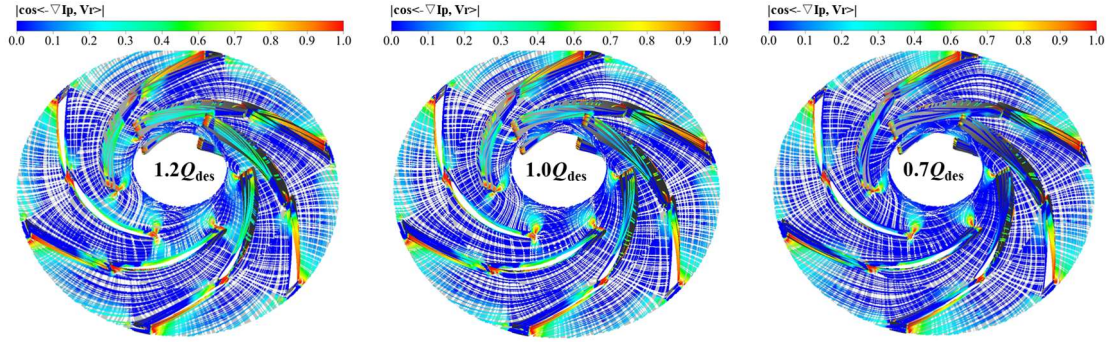
(e) Conventional impeller with 5 blades



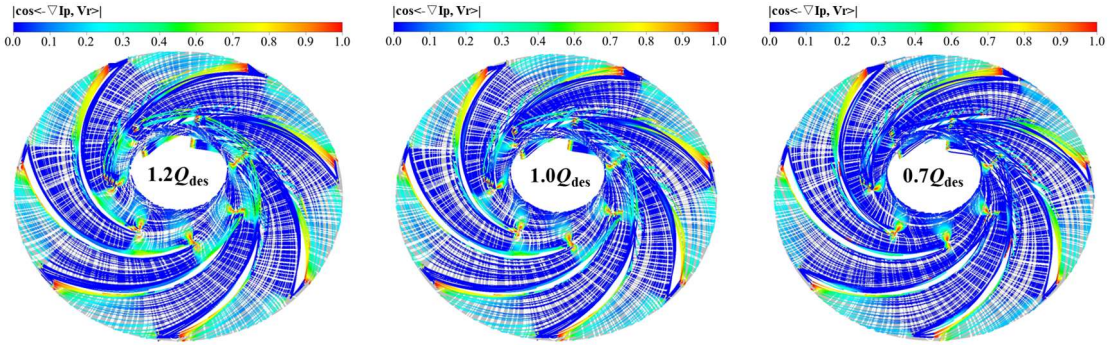
(f) Tandem impeller with 5 blades



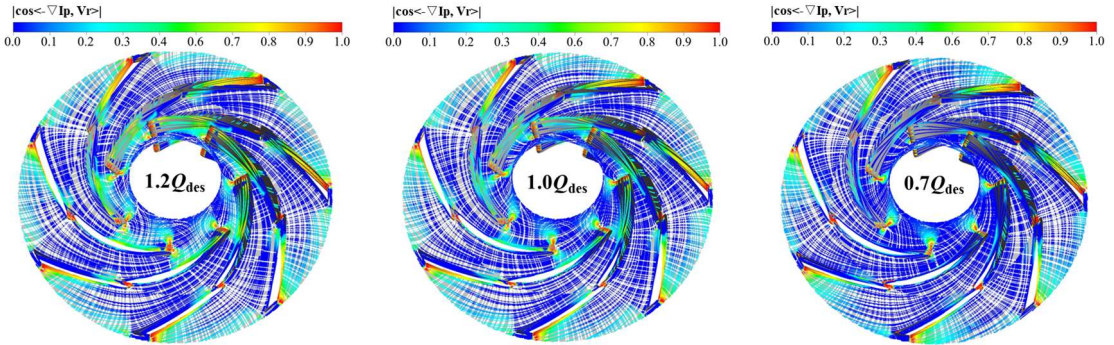
(g) Conventional impeller with 6 blades



(h) Tandem impeller with 6 blades



(i) Conventional impeller with 7 blades

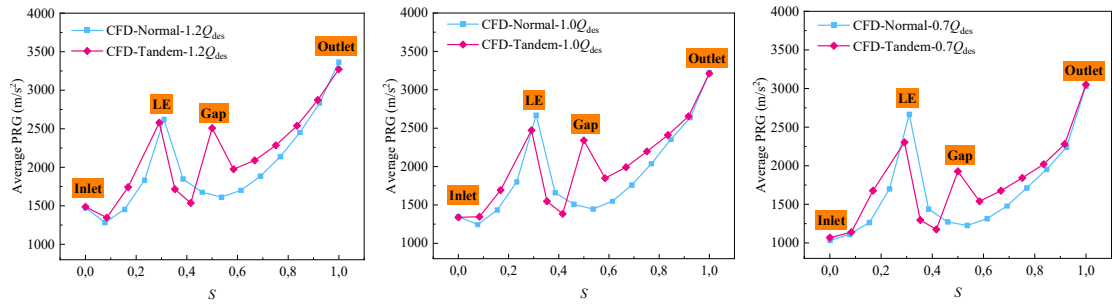


(j) Tandem impeller with 7 blades

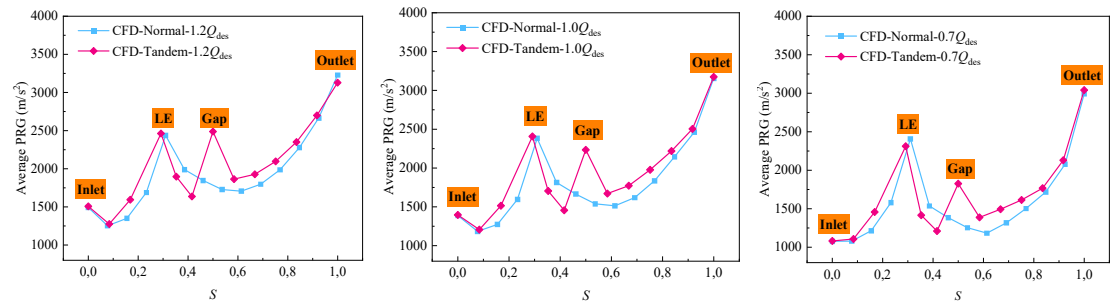
Fig. 7.10. Comparison of PRG distribution of conventional and tandem impellers with different blade numbers (white lines-PRG contour lines, and the dyed lines-relative velocity streamlines).

The PRG itself is also a crucial indicator for assessing flow stability. To provide a more intuitive comparison of the PRG between the two impellers, a quantitative analysis of the average PRG from the inlet to the outlet was conducted. The average PRG distribution curves for both impellers with 5 blade numbers are shown in Fig. 7.11. For all blade numbers, the PRG of the conventional impeller exhibits an overall increasing trend from the impeller inlet to the outlet, with two distinct peaks appearing near the LE and TE of the blades. These peaks correspond closely to the regions of flow instability observed in Fig. 7.10. As both the blade number and the Q increase, the overall PRG values for both

impellers rise accordingly. For a given blade number, the distribution trend of PRG in the tandem impeller is generally similar to that of the conventional impeller, and their magnitudes are comparable. However, a notable difference arises in the mid-gap region between the front and rear blades of the tandem impeller, where an additional distinct PRG peak appears. This PRG peak in the gap increases significantly with rising Q . Across different blade numbers, the PRG values near the blade LE increase progressively as the number of blades grows for both impellers. Moreover, the PRG in the gap region of the tandem impeller becomes increasingly higher than that of the conventional impeller with more blades, especially under overload conditions, where the peak difference can reach up to 50%. Combining the observations from Figs. 7.10 and 7.11, it is evident that the adverse pressure gradients near the LE and TE, as well as in the gap of the tandem impeller, induce highly unstable flow patterns. The sharp PRG rise in the gap region is identified as the primary cause of increased torque acting on the blades. Although the tandem impeller achieves greater H enhancement under overload conditions and with more blades, the accompanying increase in torque offsets the positive impact of H improvement on the η .



(a) 3 Blades



(b) 4 Blades

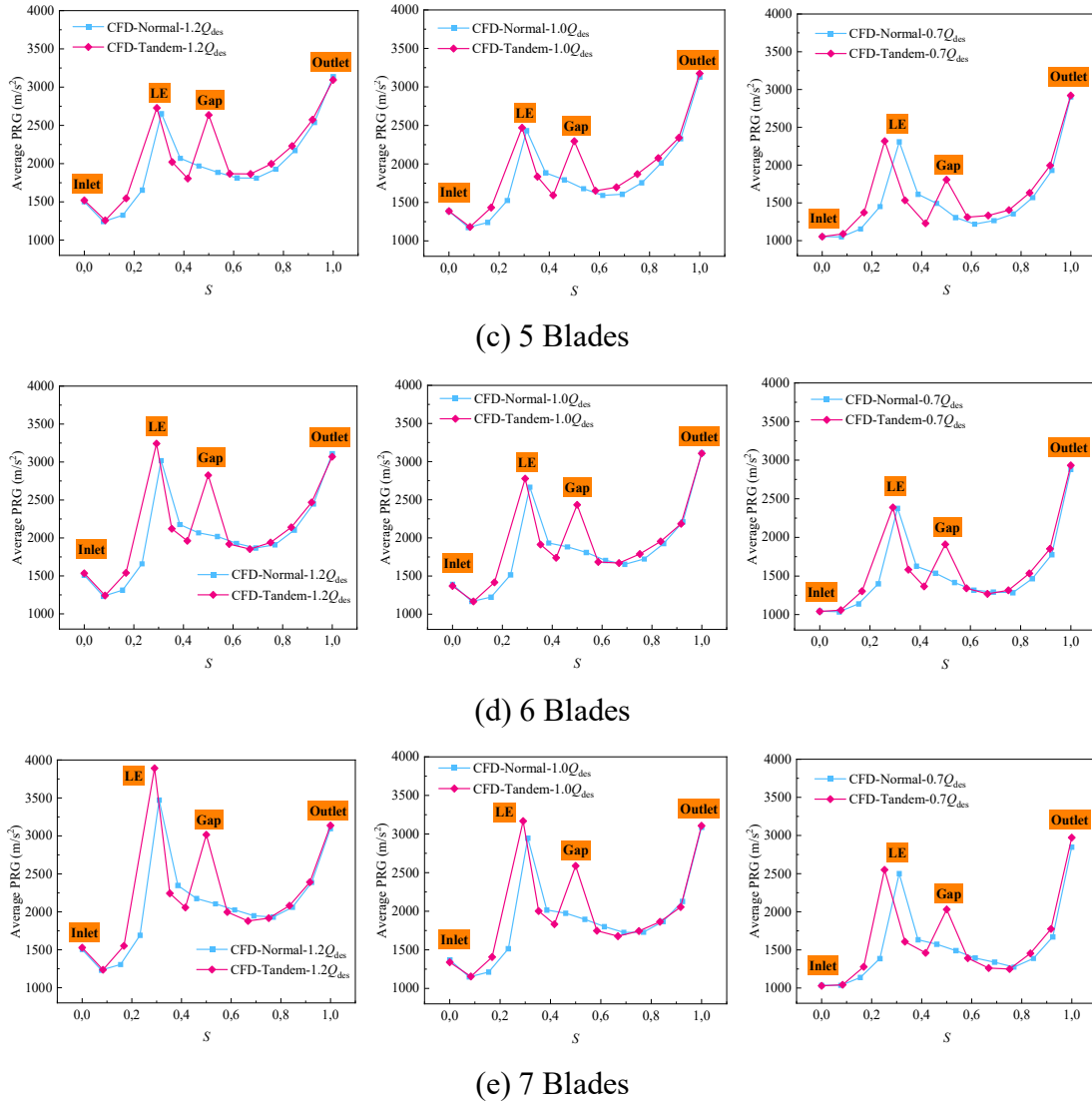


Fig. 7.11. Comparison of average PRG value from conventional and tandem impellers inlet to outlet with different blade numbers.

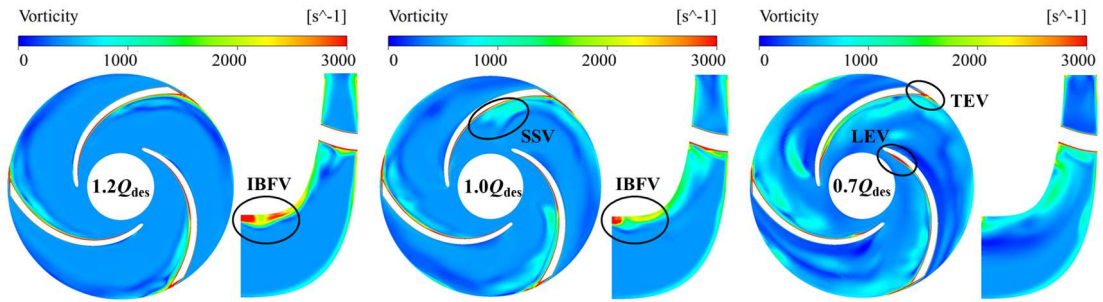
7.2 Comparison Analysis of Vortex Characteristics

This section compares the internal vortex characteristics of centrifugal pumps with two different impeller structures, focusing on the effect of blade number variation on vortex behavior. By employing vortex distribution analysis, 3D vortex structure identification methods, and the relative vorticity transport equation as introduced in Section 2.4.2, this study systematically reveals the vortex evolution patterns within the flow channels of the two impeller models. Further investigation explores the mechanism by which impeller structural modifications influence vortex-induced energy losses, thereby providing a physical basis for explaining the potential reasons behind performance differences between conventional and tandem impellers.

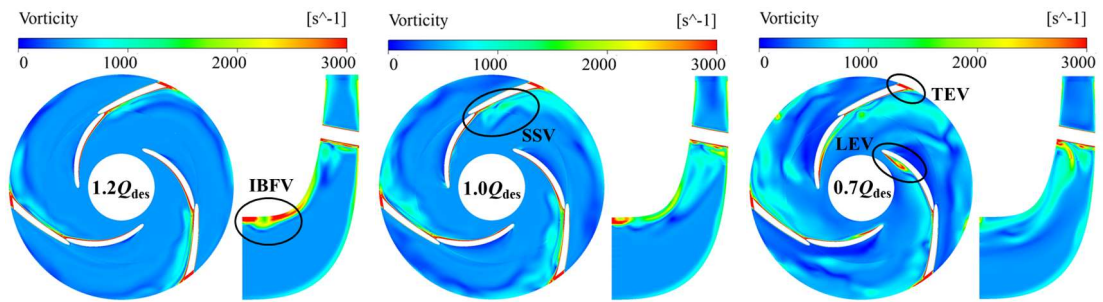
7.2.1 Vorticity Distribution and Intensity

Fig. 7.12 displays the vorticity distribution (Eq.2.9) contour plots of conventional and tandem impellers in the yz -plane and xy -plane for different blade numbers. The primary vortex structures in centrifugal pump impellers invariably generate high vorticity. The major vortices include: the IBFV formed on the shroud of the impeller inlet where the backflow from the shroud chamber converges with the main flow, the LEV caused by mismatch between the inlet flow angle and the blade settling angle, the separation vortex caused by flow separation on the SS of the mid-blade (SSV), and the separation vortex generated by wake separation at the blade TE (TEV). For all blade numbers, the IBFV intensity increases with the Q , which can be attributed to the intensified recirculation occurring in the front chamber of the impeller. As the blade number increases, the influence region of the IBFV also expands, gradually extending toward the blade LE, while the impeller structural modification exerts relatively minor influence on the IBFV characteristics. The LEV is predominantly observed in impellers with fewer blades and gradually vanishes when the blade number exceeds 5. The LEV originates on the SS of the blade LE, and its strength increases progressively as the Q decreases. Similarly, the presence and development of the LEV are only marginally affected by changes in the impeller structure. For different blade numbers, the SSV always originates from the mid-span region of the blade SS and subsequently diffuses downstream along the blade contour toward the impeller outlet. Among all the vortex structures, the SSV exhibits the largest spatial extent, and its influence region further expands as the blade number and Q decrease. The SSV is observed in every flow passage of the impeller. Therefore, its overall impact on the impeller becomes more pronounced with an increasing number of blades. In the tandem impeller, the gap is located slightly downstream of the SSV initiation zone. High-pressure fluid from the PS of the front blade can pass through the gap and merge into the SS of the rear blade, which makes the vorticity distribution of the rear-blade SSV more complex and unstable. As the number of blades increases, a significant vorticity amplification phenomenon can be observed in the gap region of the tandem impeller, particularly under overload conditions. The SSV observed near the rear blades of the tandem impeller exhibits not only greater intensity but also a broader region of influence, which consequently leads to increased vortex-induced flow losses and higher blade torque. For all blade-number configurations, the TEV is consistently observed to originate at the blade TE and extend toward the impeller outlet in the direction opposite to the impeller rotation. Although the TEV generally possesses relatively high intensity, its influence range is significantly smaller than that of the SSV. As the number of blades increases, the interference effect of the TEV on the internal flow pattern of the impeller becomes progressively more pronounced. In the tandem impeller, the TEV demonstrates higher intensity and a wider impact region compared with the conventional impeller, and this characteristic becomes increasingly evident with a greater number of blades. This

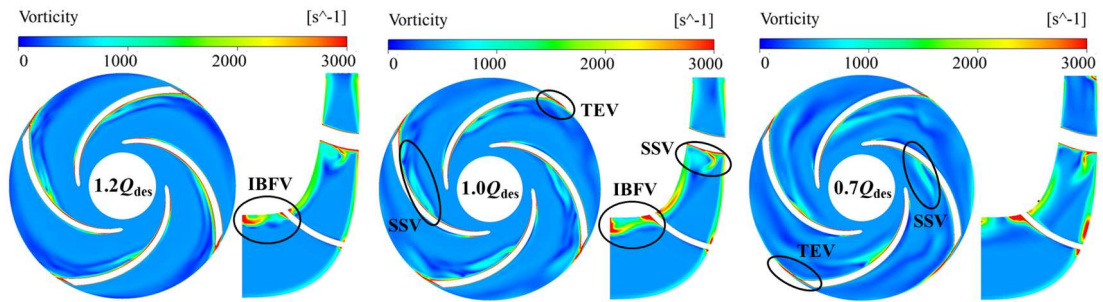
behavior is likely closely related to the strong SSV present in the rear-blade region, which may induce or amplify the formation of the TEV.



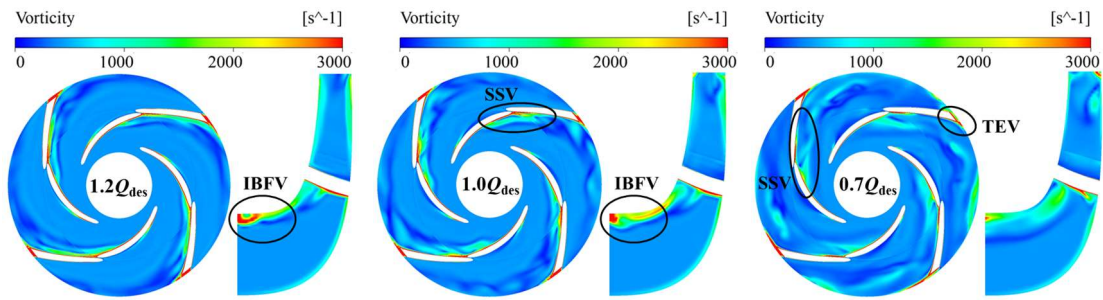
(a) Conventional impeller with 3 blades



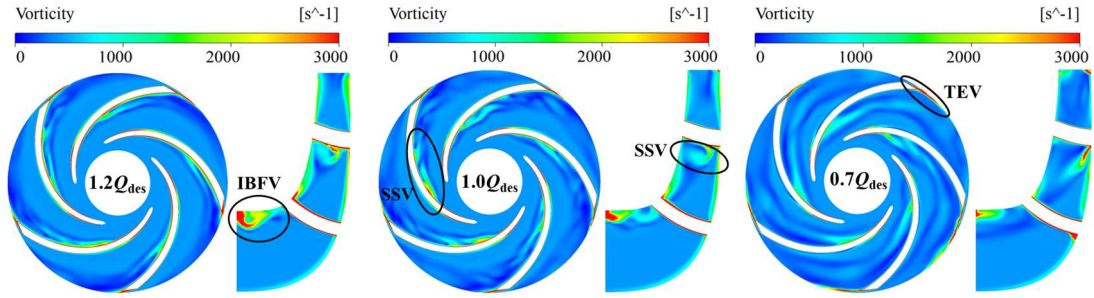
(b) Tandem impeller with 3 blades



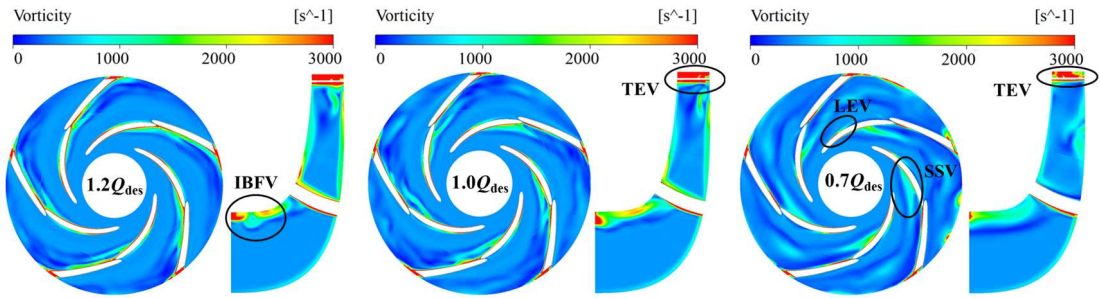
(c) Conventional impeller with 4 blades



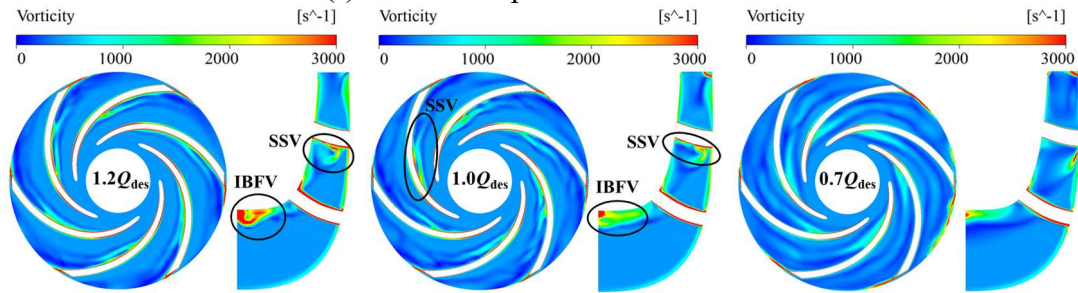
(d) Tandem impeller with 4 blades



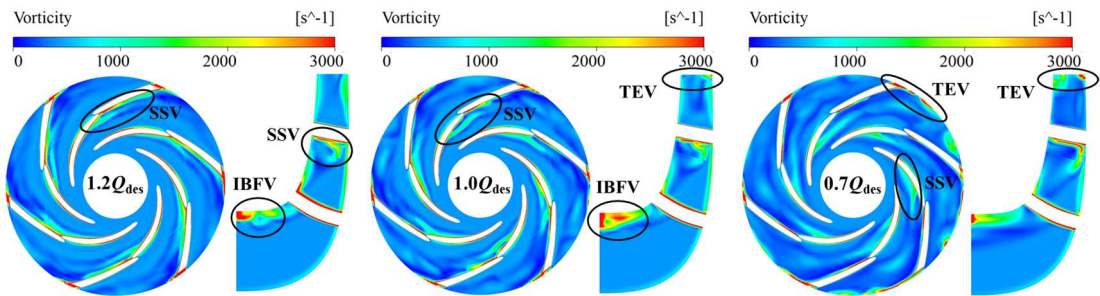
(e) Conventional impeller with 5 blades



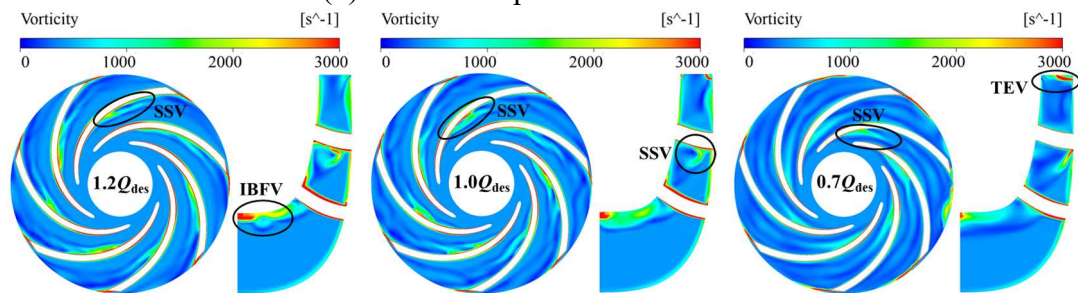
(f) Tandem impeller with 5 blades



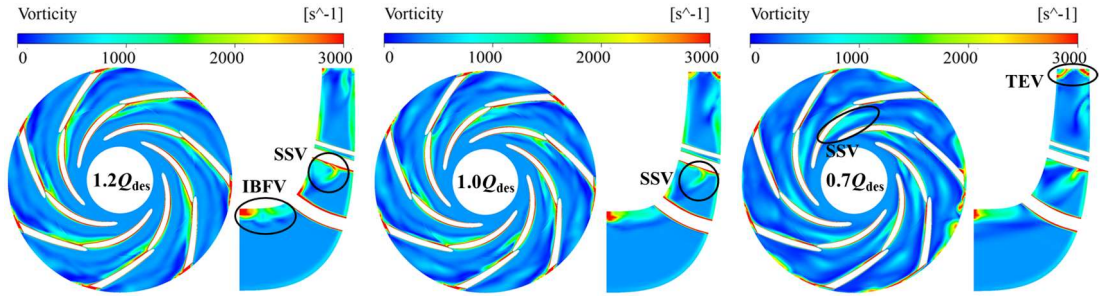
(g) Conventional impeller with 6 blades



(h) Tandem impeller with 6 blades



(i) Conventional impeller with 7 blades



(j) Tandem impeller with 7 blades

Fig. 7.12. Comparison of vorticity in conventional and tandem impellers with different blade numbers.

To comparatively investigate the unsteady vorticity distribution characteristics in the impeller, Fig. 7.13 presents the vorticity contours on the blade-to-blade section of the two six-bladed impellers at different time instants under the design condition. The time interval between successive frames in Figs. 7.13(a) and (b) is $1/4T_n$, where $T_n=0.03s$ represents one full rotation period of the impeller. For both impellers, the high-vorticity LEV observed near the blade LE gradually weakens over time and then reappears with increased strength after approximately T_n . As shown in Fig. 7.13(a), the vorticity intensity of the SSV in the conventional impeller alternates among adjacent flow passages. As indicated by the black arrows, the SSV in a given passage originates from the mid-blade region and continuously develops toward the blade TE over time, exhibiting a distinct periodic behavior with a period of approximately $1/2T_n$. In contrast, Fig. 7.13(b) shows that in the tandem impeller, the high-vorticity region of the SSV periodically shifts from the TE of the front blade to the SS of the rear blade, with a shorter oscillation period of approximately $1/4T_n$. This indicates that the unsteady behavior of the SSV in tandem impellers is more complex, and that the presence of gap accelerates the unstable evolution of vortex structures. The continuous evolution of the SSV and its interaction with downstream TEV are likely to be primary sources of flow-induced excitation in impellers.

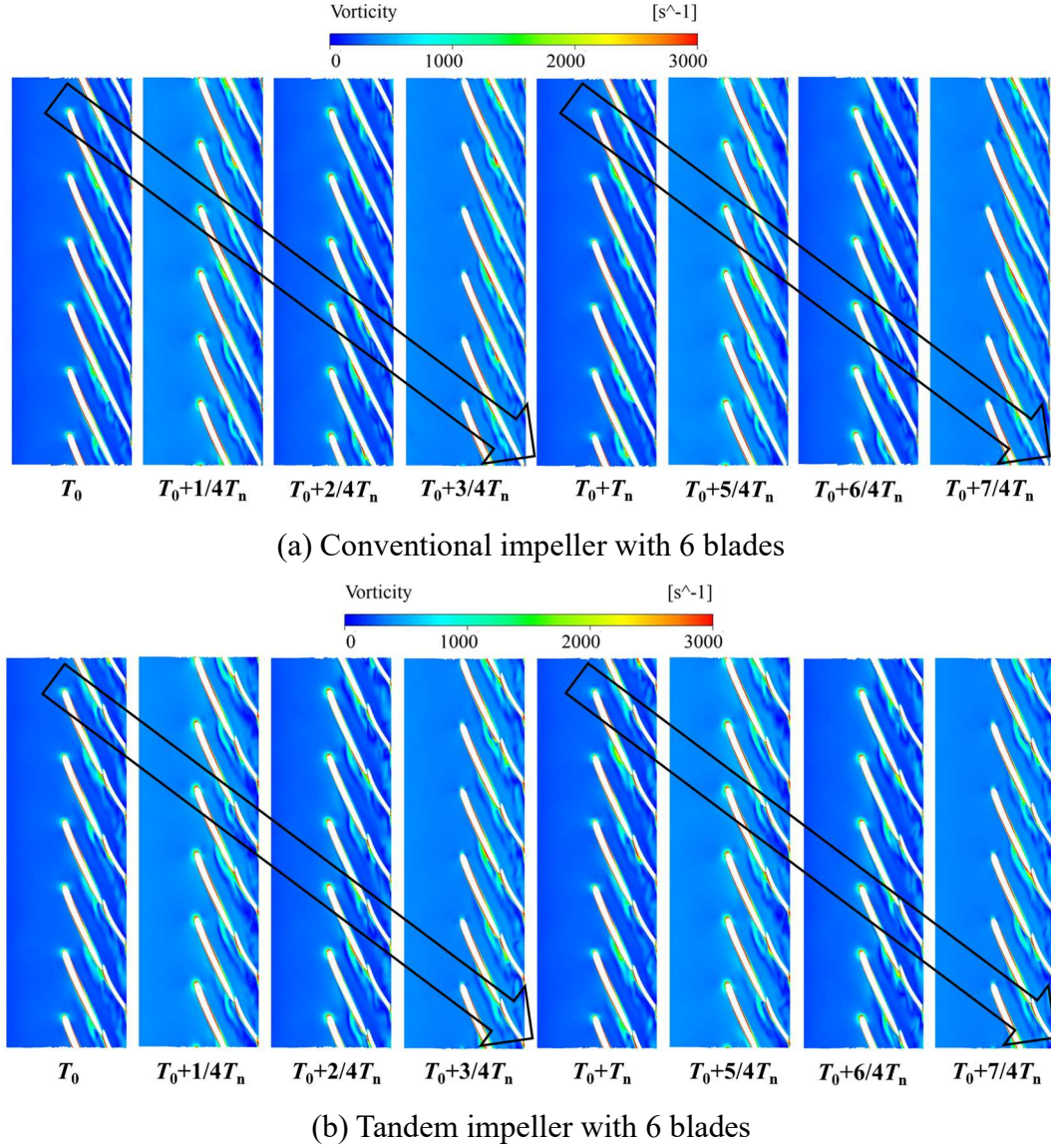


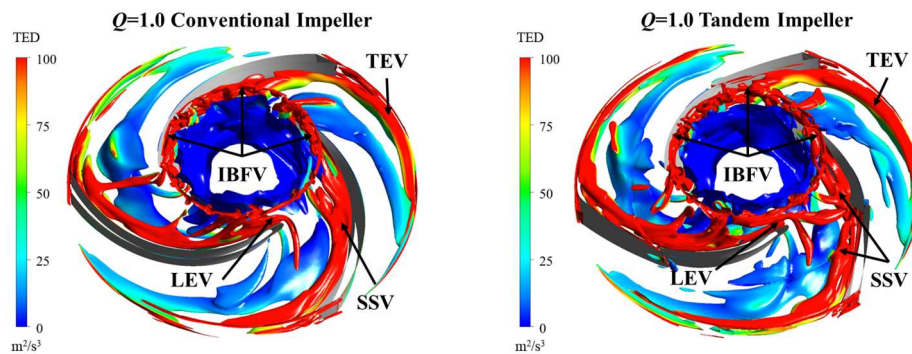
Fig. 7.13. Transient of vorticity in conventional and tandem impellers under different moments.

7.2.2 Vortex Evolution and Transport Characteristics

As revealed in the previous section, both impellers are prone to inducing unstable flow structures in their flow passages. Therefore, a detailed analysis of the vortex structures in the impeller channels is essential for understanding the evolution of flow patterns and the mechanisms underlying performance variations.

Based on the Ω vortex identification method (Eq.2.11), the vortex distribution patterns in the conventional impeller and the tandem impeller of the centrifugal pump under different blade numbers were investigated. This analysis aims to further elucidate the variations in vortex characteristics in the blade passages following modifications to the impeller geometry. The vortex surfaces were colored according to the turbulent energy dissipation

(TED) to indicate the magnitude of flow losses induced by the vortical structures. The results are presented in Fig. 7.14. The Ω method successfully identifies the typical high-intensity vortex structures in the impeller, including the IBFV, LEV, SSV, and TEV, which are consistent with the results shown in Fig. 7.12. For all blade numbers, the IBFV consistently appears near the impeller inlet in the form of an annular strip. The IBFV maintains an extremely high TED and is only weakly affected by variations in blade number or impeller geometry. The LEV with high TED can be clearly observed when the impeller has three blades. It originates near the blade LE and extends as a narrow band toward the midsection of the blade's SS. In both impeller configurations, the LEV maintains a high TED and connects with the SSV located in the middle of the blade, forming an extensive region of intense vortical structures on the SS. However, as the blade number increases, the LEV gradually disappears because the incoming flow becomes progressively aligned with the blade inlet setting angle. For all blade numbers, the SSV appears as a strip-like structure distributed along the middle SS of the blades. In particular, for the 3-blade case, the SSV exhibits the widest coverage and the highest TED, indicating the pronounced effects of flow separation and strong shear interactions. Under the same conditions and for the same blade number, the tandem impeller consistently exhibits a broader distribution of the SSV. The SSV in the tandem impeller forms a two-segment strip along the blade SS, with the segments separated at the gap. Notably, the SSV on the rear blades shows higher TED and a wider coverage area, and in cases with fewer blades, it even connects with the TEV at the blade TE to form an extensive vortical band. The TEV exhibits the highest TED and the largest rearward extension when the impeller has three blades, largely due to the induced effects of the high-intensity SSV. As the blade number increases to 4 or more, the TEV area significantly decreases and its intensity diminishes. For the tandem impeller, the TEV area is slightly larger than that of the conventional impeller when the blade number is large. However, with increasing blade number, the coupling effect of the SSV on the TEV gradually weakens.



(a) 3 Blades

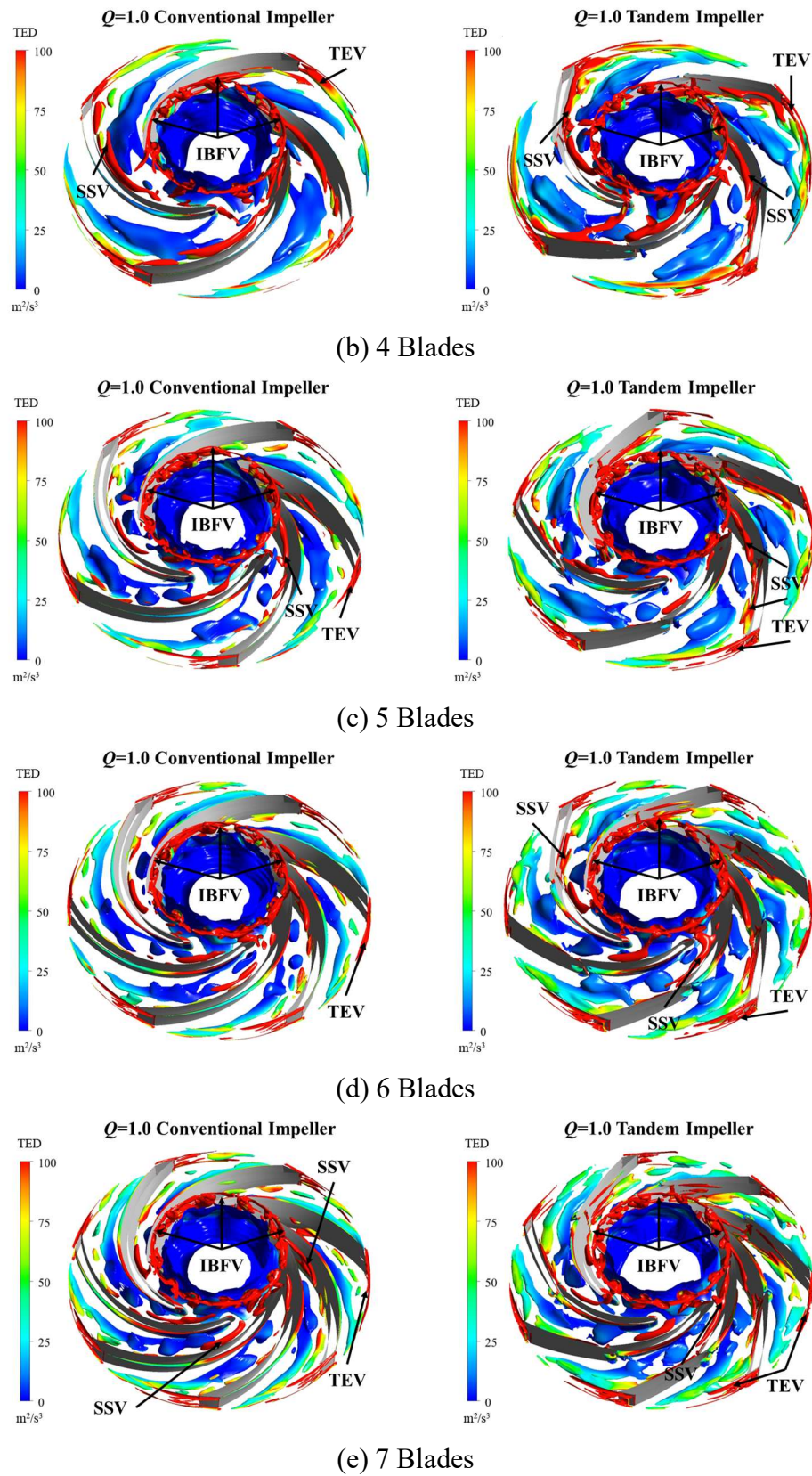
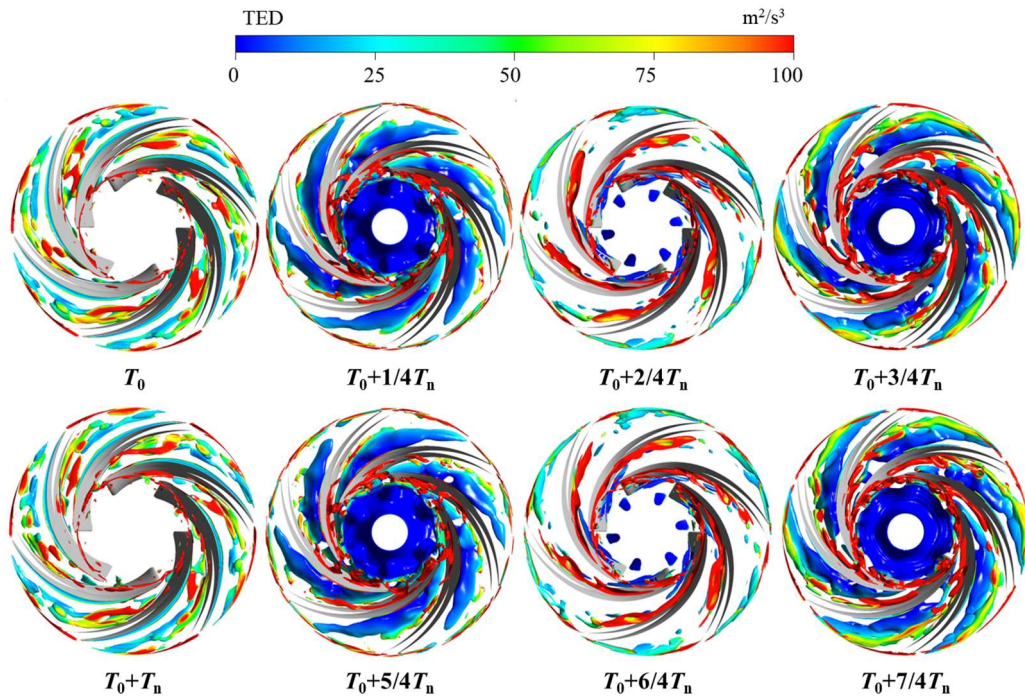


Fig. 7.14. Comparison of omega vortex identification in conventional and tandem impellers with different blade numbers.

This study also investigated the unsteady evolution characteristics of vortex structures in the two impellers. Fig. 7.15 illustrates the temporal variation of Ω vortex structures in the 6-blade impellers under the design condition. In Fig. 7.15(a), the IBFV exhibits a periodic increase and decrease over time, with a period of approximately $1/4 T_n$. The SSV initially originates at the mid-section of the blades with very high TED. After $1/4 T_n$, the SSV develops toward the blade exit, with its coverage area significantly increasing, although its TED decreases notably. After another $1/4 T_n$, the SSV reappears at the blade mid-section with high TED. Therefore, in the conventional impeller, the SSV exhibits a periodic evolution with a temporal cycle of $1/2 T_n$. The TEV, in contrast, maintains a high-TED strip along the impeller exit at all times. Fig. 7.15(b) illustrates that the unsteady evolution characteristics of the IBFV in the tandem impeller are generally consistent with those in the conventional impeller. The SSV in the mid-region of the tandem impeller still undergoes a periodic cycle of initiation, development, and dissipation over time, with a period of approximately $1/2 T_n$. However, when the SSV in the tandem impeller expands toward the impeller exit, high-TED SSV can still be observed on the SS of the blades, particularly on the SS of the rear blades. Furthermore, the SSV in the tandem impeller exhibits a more fragmented and heterogeneous structure, especially at the moment it reaches the impeller exit. As a result, the SSV in the tandem impeller produces greater flow losses and is more prone to inducing severe excitation characteristics at multiple instances.



(a) Conventional impeller with 6 blades

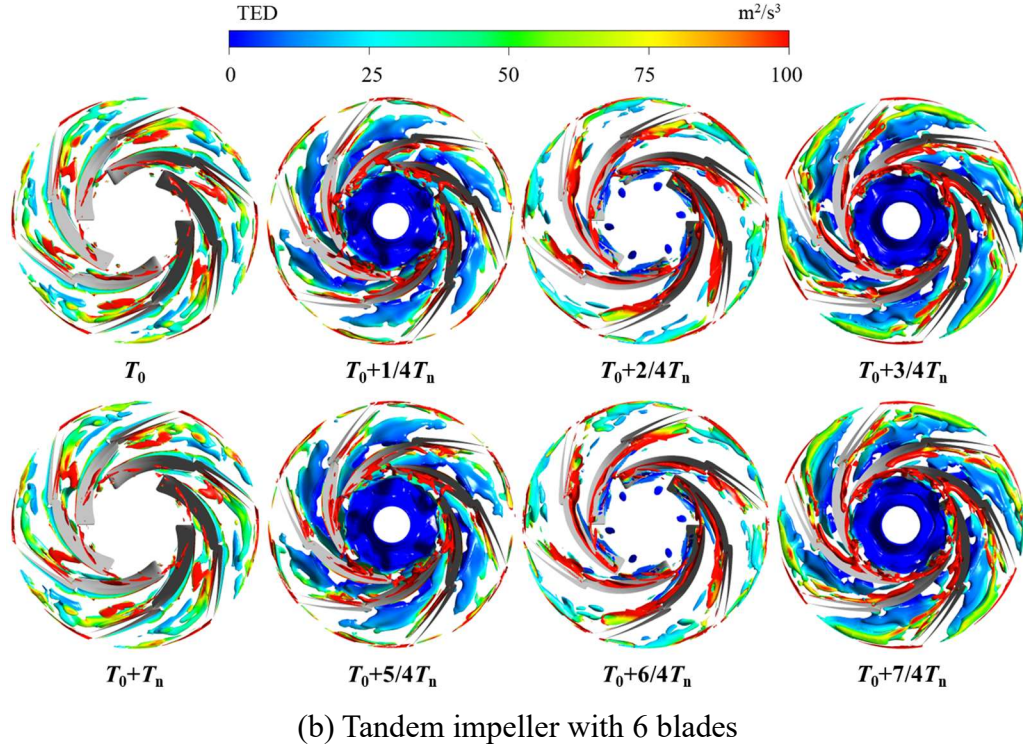
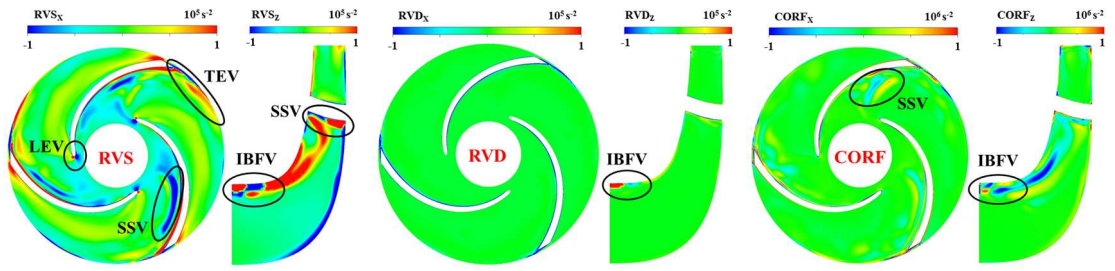


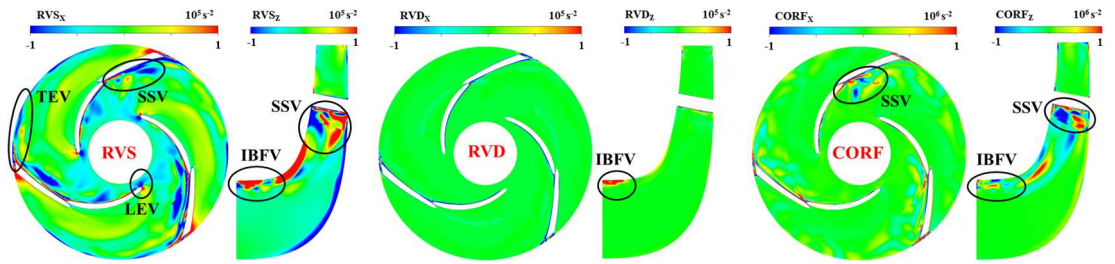
Fig. 7.15. Transient of omega vortex identification in conventional and tandem impellers under different moments.

This study focuses on analyzing the first three terms of the relative vorticity transport equation (Eq.2.16) in both the yz -plane and xy -plane of the impeller region to characterize the vortex evolution behavior. As shown in Fig. 7.16, the spatial distribution of high-vorticity regions in both impellers corresponds closely to the regions of strong intensity in the various vorticity transport terms. For all blade numbers, the RVS term exhibits alternating positive and negative regions at the location of the IBFV, a feature observed in both impeller types. The RVD term shows only a localized region of very high intensity at the IBFV, while the CORF term also displays positive and negative distributions there, though its magnitude and extent are significantly smaller than those of the RVS. This indicates that the stretching and deformation of the rotating fluid predominantly govern the development of the IBFV, while the CORF induced by impeller rotation has a secondary but noticeable influence. Across different blade numbers, only the RVS exhibits localized extremum values at the LEV. In both impellers, the RVS shows a distinct maximum on the PS and a minimum on the SS near the LE of the blades. However, this pattern diminishes progressively as the blade number increases. Therefore, under configurations with fewer blades, the development of the LEV is primarily driven by the stretching and deformation of the rotating fluid. For all blade numbers, distinct RVS extrema can be observed in the y - z and x - y planes near the midspan region of the blades, corresponding precisely to the SSV locations shown in Figs. 7.12 and 7.14. As the number of blades increases, the magnitude and spatial extent of the RVS extrema at the SSV

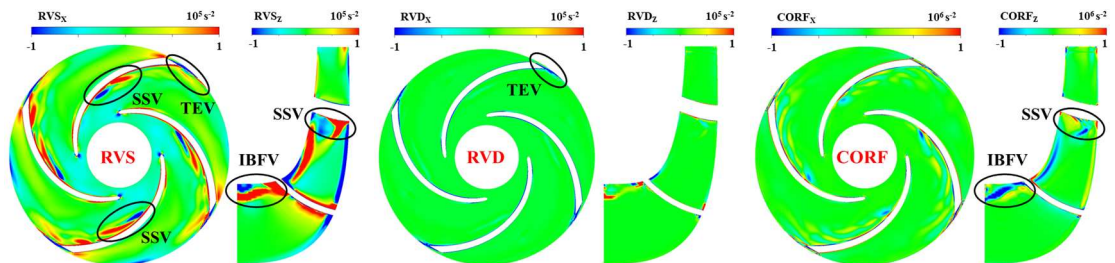
regions become more pronounced. Under identical blade counts, the RVS extrema in the mid-blade region of the tandem impeller are larger and cover a wider area than those in the conventional impeller, particularly on the SS of the rear blade. Localized CORF extrema are also observed near the SSV, especially in the gap of the tandem impeller, where they exhibit alternating positive and negative distributions. This indicates that the intensified SSV in the tandem impeller is primarily driven by the enhanced stretching and deformation of the rotating fluid, and secondarily by the rotational inertia effects of the gap flow. At the TEV region, extrema of RVS, RVD, and CORF are observed for all blade numbers. Among them, the RVS exhibits the widest spatial coverage, whereas RVD and CORF show smaller but comparable localized extrema. As the blade number increases, the spatial extent of all three terms extrema near the TEV gradually decreases. Under the same blade number, the differences in the magnitudes of these three terms between the conventional and tandem impellers are minor. Therefore, the generation and evolution of the TEV are primarily attributed to the stretching and deformation of the rotating flow field.



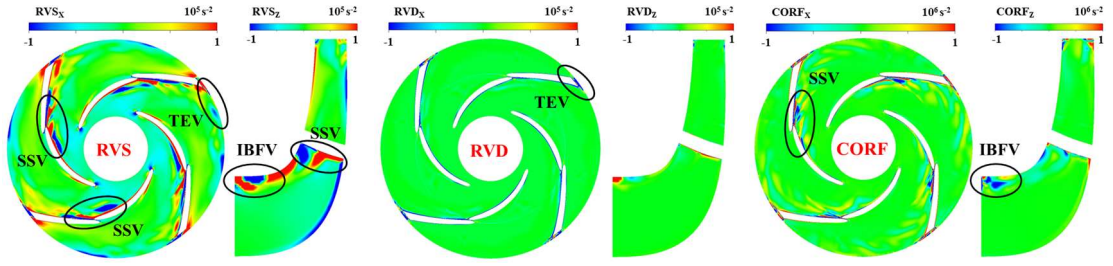
(a) Conventional impeller with 3 blades under $1.0Q_{des}$



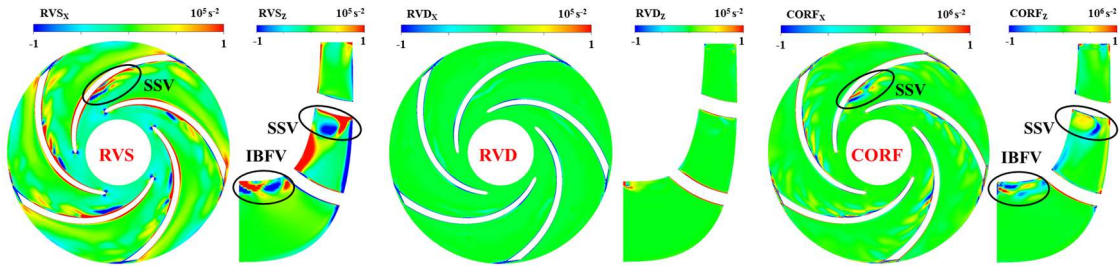
(b) Tandem impeller with 3 blades under $1.0Q_{des}$



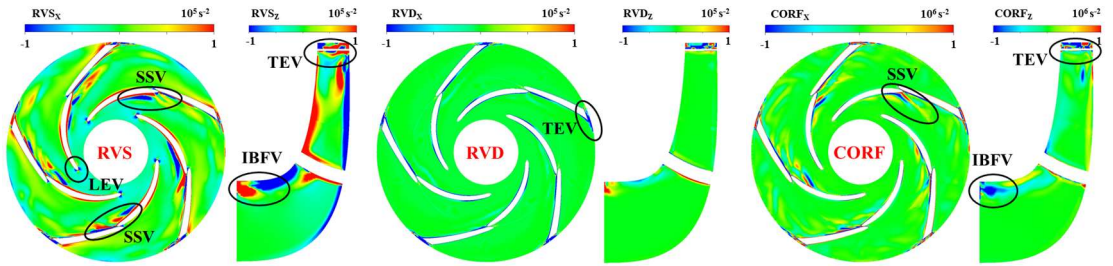
(c) Conventional impeller with 4 blades under $1.0Q_{des}$



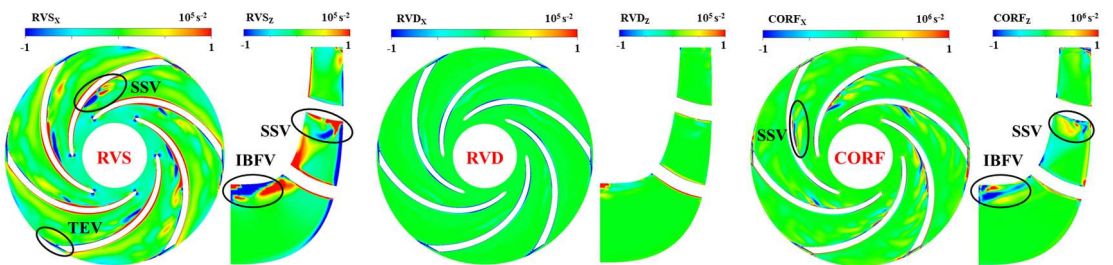
(d) Tandem impeller with 4 blades under $1.0Q_{des}$



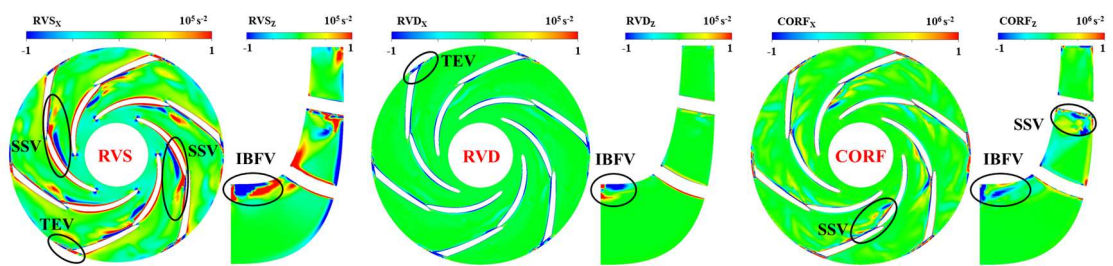
(e) Conventional impeller with 5 blades under $1.0Q_{des}$



(f) Tandem impeller with 5 blades under $1.0Q_{des}$



(g) Conventional impeller with 6 blades under $1.0Q_{des}$



(h) Tandem impeller with 6 blades under $1.0Q_{des}$

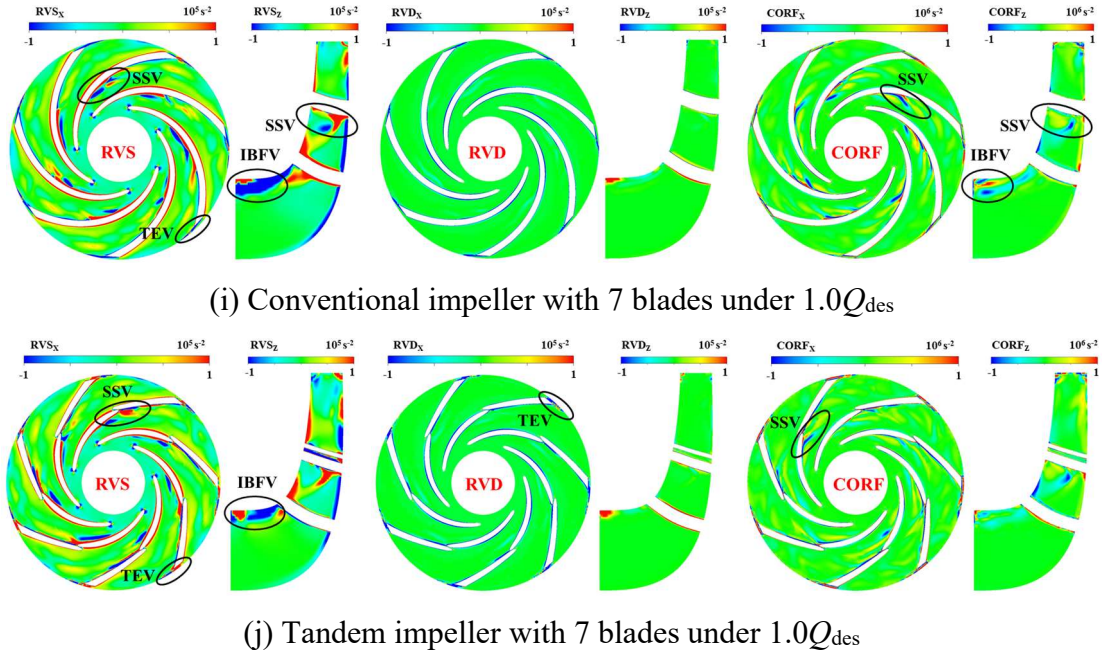


Fig. 7.16. Comparison of vortex transport terms in conventional and tandem impellers with different blade numbers.

The above analysis indicates that the evolution of the core vortex structures in both impellers is primarily governed by the RVS and CORF. Therefore, to further investigate the unsteady transport characteristics of vortex evolution, Figs. 7.17 and 7.18 present the instantaneous variations of the RVS and CORF, respectively, for the two 6-blade impellers under the design condition.

Fig. 7.17 shows that the RVS at the LEV initially exhibits a minimum value, then gradually increases to a maximum, and subsequently decreases back to a minimum, completing one full impeller rotation period T_n . This indicates that the evolution of the LEV is primarily driven by the alternating positive and negative stretching and deformation of the rotating fluid within a single impeller cycle. Fig. 7.17(a) illustrates that, in the conventional impeller, the RVS at the SSV displays an alternating high-low distribution across different passages. As indicated by the black arrows, the high-RVS regions in the SSV periodically migrate from the mid-span toward the impeller exit, with a transfer period of approximately $1/2T_n$. At the TEV, the RVS also exhibits periodic maxima and minima over time, with a period of $1/2T_n$. Fig. 7.17(b) shows that in the tandem impeller, the RVS at the SSV exhibits opposite trends at the TE of the front blade and the SS of the rear blade, with maxima and minima alternating. Over time, the RVS of SSV evolution still follows a $1/2T_n$ periodicity, but the SS of the rear blade consistently exhibits distinct RVS maxima or minima. The RVS evolution at the TEV in the tandem impeller is almost identical to that of the conventional impeller.

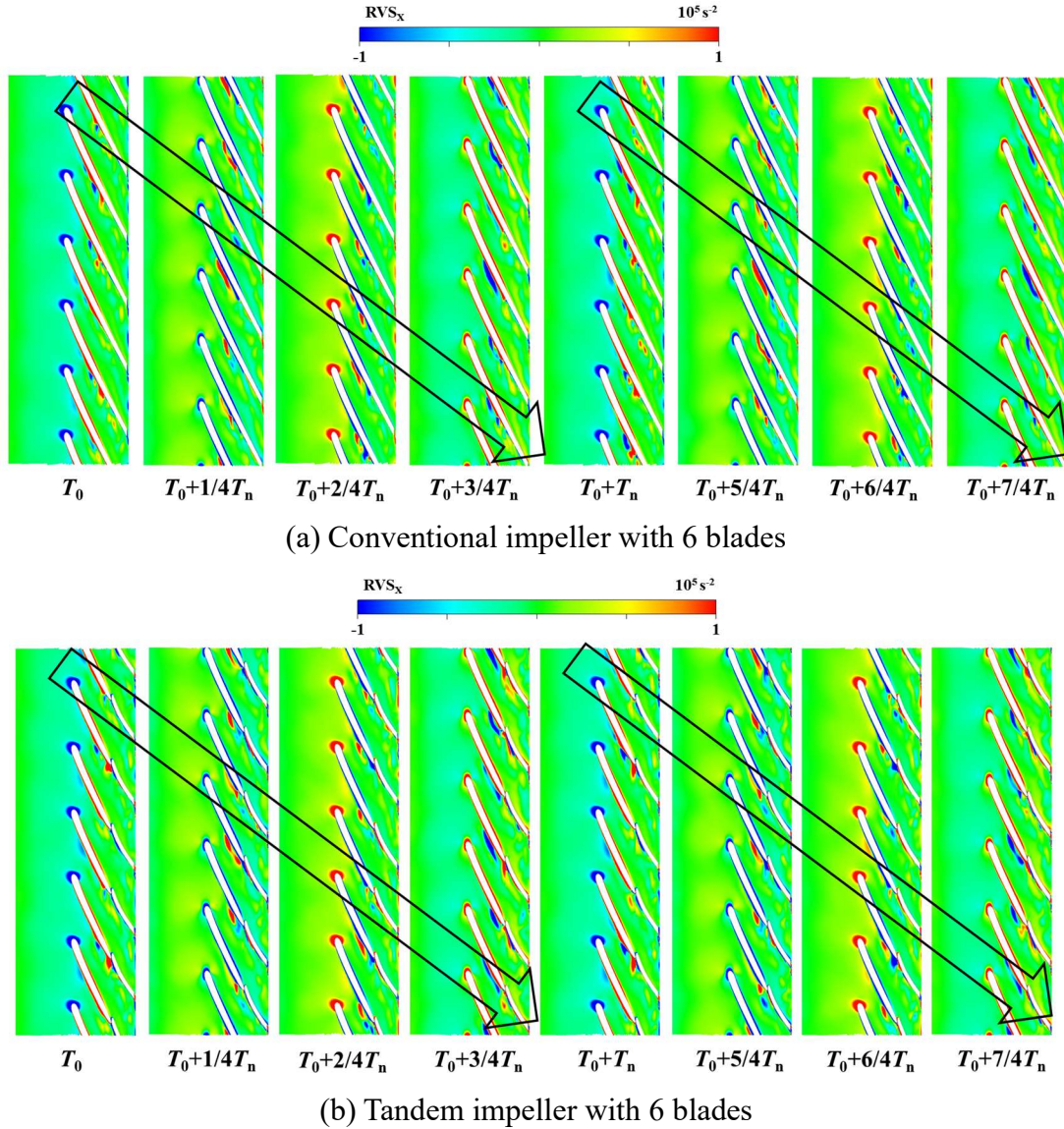
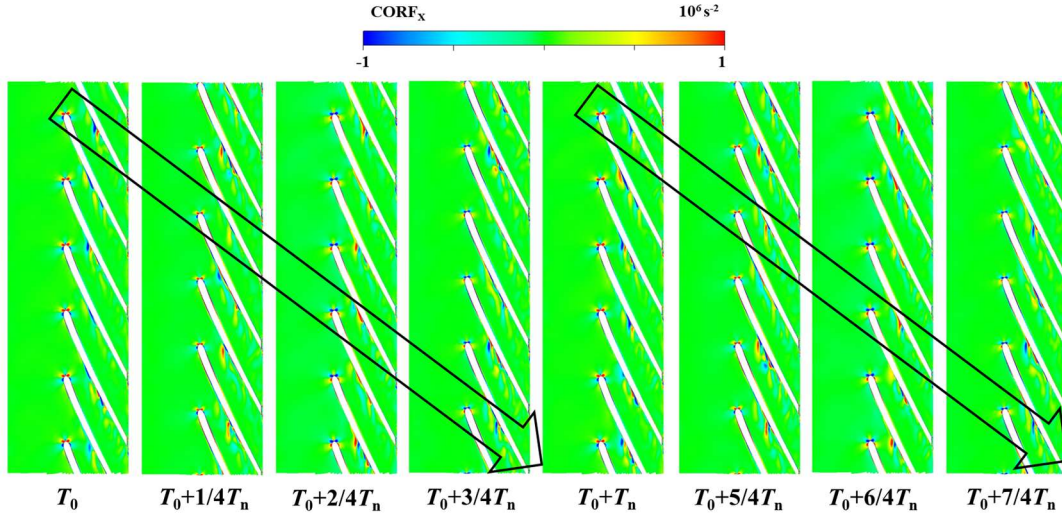


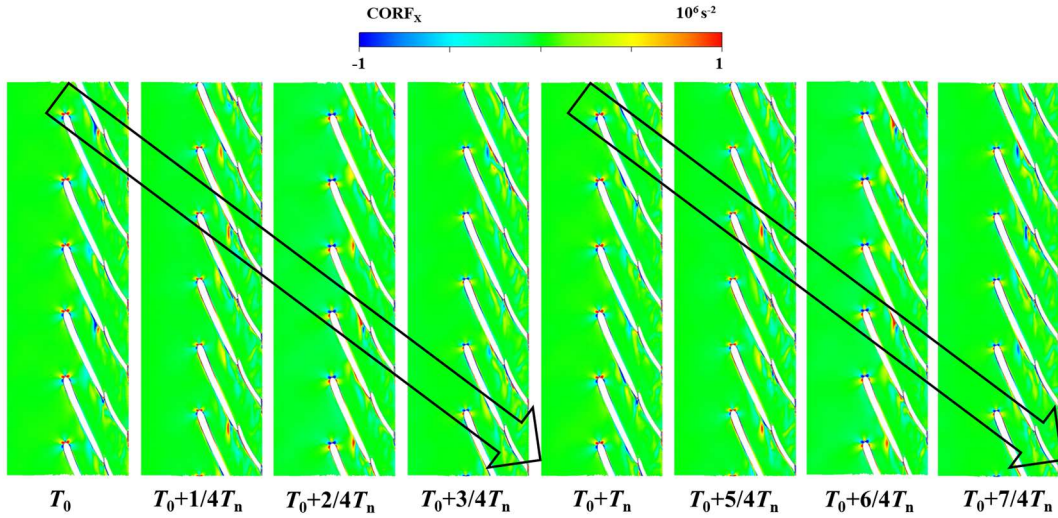
Fig. 7.17. Transient of RVS term in conventional and tandem impellers under different moments.

As shown in Fig. 7.18(a), the CORF at the LEV of the conventional impeller exhibits an alternating pattern of maximum and minimum values, with only slight variations over time. A similar alternating distribution of CORF maxima and minima is also observed at the SSV region. Within the black arrows, the CORF in the same flow passage consistently presents alternating positive and negative values along the mid-span SS of the blade, extending downstream, but its temporal variation remains relatively weak. In Fig. 7.18(b), the tandem impeller shows a distinct alternating pattern of CORF maxima and minima near the TE of the front blade, whereas no significant CORF variations are observed along the SS of the rear blade. As time progresses and the SSV develops downstream, the CORF magnitude at the front blade TE noticeably decreases. By combining the observations from Figs. 7.17 and 7.18, it can be concluded that the generation of the SSV is closely

related to the Coriolis force effects represented by the CORF, while its downstream evolution is primarily driven by fluid stretching and deformation represented by the RVS. Under the persistently strong influence of RVS, the SSV in the tandem impeller develops more rapidly and exhibits a more fragmented and complex vortex structure.



(a) Conventional impeller with 6 blades



(b) Tandem impeller with 6 blades

Fig. 7.18. Transient of CORF term in conventional and tandem impellers under different moments.

7.3 Comparison Analysis of Unsteady Excitation Force Characteristics

This section presents a comparative analysis of the unsteady excitation characteristics between the conventional impeller and the tandem impeller, focusing primarily on two aspects, the unsteady pressure pulsation behavior in the main flow components and the

unsteady excitation forces acting on the impeller. For the pressure pulsation characteristics, several monitoring points were arranged at key locations in the impeller and volute. Based on CFD results, a comprehensive comparison was conducted in the time domain, frequency domain, and time-frequency domain to examine the unsteady pressure evolution. For the unsteady excitation force characteristics, both the axial and radial forces acting on the impeller were analyzed in terms of their time-domain variations, frequency spectra, and vector distributions. Overall, this section further investigates the mechanisms by which changes in the internal flow pattern of the impeller influence the unsteady excitation characteristics of the centrifugal pump.

7.3.1 Pressure Pulsation Characteristics

To facilitate the analysis of pressure pulsation characteristics at various locations during the unsteady simulations, a series of monitoring points were arranged in the conventional impeller, tandem impeller, and volute, as illustrated in Fig. 7.19. Taking the 6-blade conventional and tandem impellers as examples, three groups of monitoring points were placed in a single impeller passage at the 0.5 span position, near the PS, SS, and mid-passage region. For the PS and SS, all monitoring points were positioned 2mm away from the blade surface to avoid interference from the boundary layer mesh. On the PS, five monitoring points were evenly distributed from the LE to the TE, denoted as PS_{c1}-PS_{c5} in the conventional impeller and PS_{t1}-PS_{t5} in the tandem impeller. Similarly, on the SS, five points were evenly placed from the LE to TE, denoted as SS_{c1}-SS_{c5} for the conventional impeller and SS_{t1}-SS_{t5} for the tandem impeller. In the mid-passage region, five monitoring points were arranged uniformly from the inlet to the outlet, denoted as m_{c1}-m_{c5} for the conventional impeller and m_{t1}-m_{t5} for the tandem impeller. In addition, two extra monitoring points were installed in the gap region of the tandem impeller, g_{t1} located on the PS near the TE of the front blade, and g_{t2} located downstream of g_{t1} on the SS of the LE of the rear blade. Both were placed 2mm away from the blade surface, as shown in the enlarged view of Fig. 7.19(b). Within the volute, six pressure pulsation monitoring points V₁-V₆ were positioned at the mid-span along the central line of the flow passage, from the small cross-sectional area region to the volute outlet. Among them, V₅ was located close to the tongue region, as indicated in Fig. 7.19.

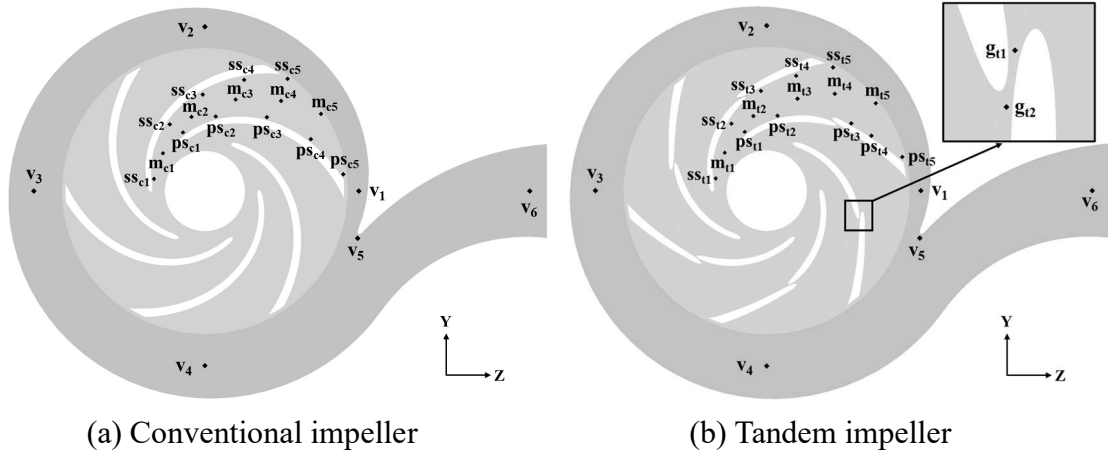


Fig. 7.19. Distribution of monitoring points.

In this study, a quantitative comparison of the pressure pulsation intensity at different monitoring positions was first conducted. The pressure pulsation intensity was characterized by the standard deviation of the pressure pulsation signal, denoted as A_P , which was calculated based on the pressure fluctuations recorded during the 8th to 12th revolutions of the impeller. The parameter A_P is defined by Eq.7.7, where a larger value of A_P indicates a higher intensity of pressure pulsation.

$$A_P = \sqrt{\sum_{i=1}^N (P(i) - \bar{P})^2 / (N - 1)} \quad (7.7)$$

Fig. 7.20 presents the statistical distribution of A_P values at various monitoring positions for the conventional 6-blade impeller under three representative conditions. Figs. 7.20(a)-(d) correspond to the blade SS, mid-passage region, blade PS, and volute interior, respectively. For the SS, PS, and mid-passage, three typical positions, inlet, middle, and outlet were selected for comparison. As shown in Figs. 7.20(a)-(c), the A_P values increase gradually from the LE to the TE of the blade, with a more pronounced rise at $0.7Q_{des}$. The smallest A_P values occur under the design condition $1.0Q_{des}$, while the largest appear at $0.7Q_{des}$. A comparison of different positions in the impeller indicates that the PS exhibits the highest A_P values, followed by the mid-passage, whereas the SS shows relatively lower A_P values. At $0.7Q_{des}$, the increase in A_P from the LE to TE is much steeper on the PS than on the SS. Fig. 7.20(d) shows that the pressure pulsation in the volute is significantly higher on the small cross-section side than on the large one, and the difference becomes more evident under overload conditions. The pressure pulsation intensity near the volute outlet is noticeably weaker than that in the volute passage, owing to its greater distance from the impeller domain. As the Q increases, the overall pressure pulsation intensity in the volute passage also increases. In summary, the pressure pulsation intensity peaks near the blade TE, and it becomes even more pronounced under part-load conditions. The PS of the blade exhibits stronger pressure pulsations than the SS, while in the volute, the small cross-section region experiences significantly higher

pulsation levels under overload conditions.

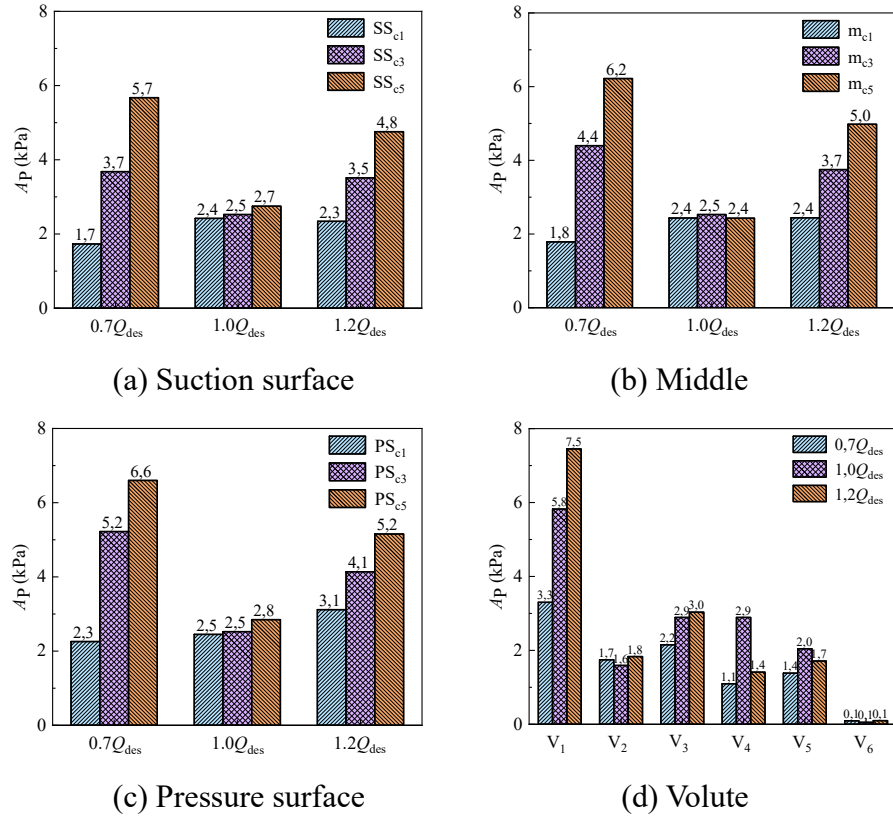


Fig. 7.20. Comparison of pressure pulsation intensity in the pump with 6 blades conventional impeller under different conditions.

Fig. 7.21 compares the A_p values of the conventional impeller and the tandem impeller with 6 blades under different conditions at various blade positions. As shown in Fig. 7.21(a), both impellers exhibit the lowest A_p values under the design condition and the highest under the part-load condition, particularly at the mid-chord and TE regions of the SS. Under both design and part-load conditions, the A_p values of the tandem impeller are consistently higher than those of the conventional impeller, with the difference being more pronounced near the blade TE. In contrast, under the overload condition, the tandem impeller shows lower A_p values than the conventional impeller, again with a larger difference at the TE. Similar trends can be observed in the mid-passage region and along the PS, as illustrated in Figs. 7.21(b) and 7.21(c). Notably, under part-load conditions and on the PS of the blade, the tandem impeller exhibits a significantly higher A_p than the conventional one, indicating stronger unsteady pressure fluctuations. Conversely, under overload conditions, especially in the mid-passage region, the A_p of the tandem impeller becomes considerably lower than that of the conventional impeller.

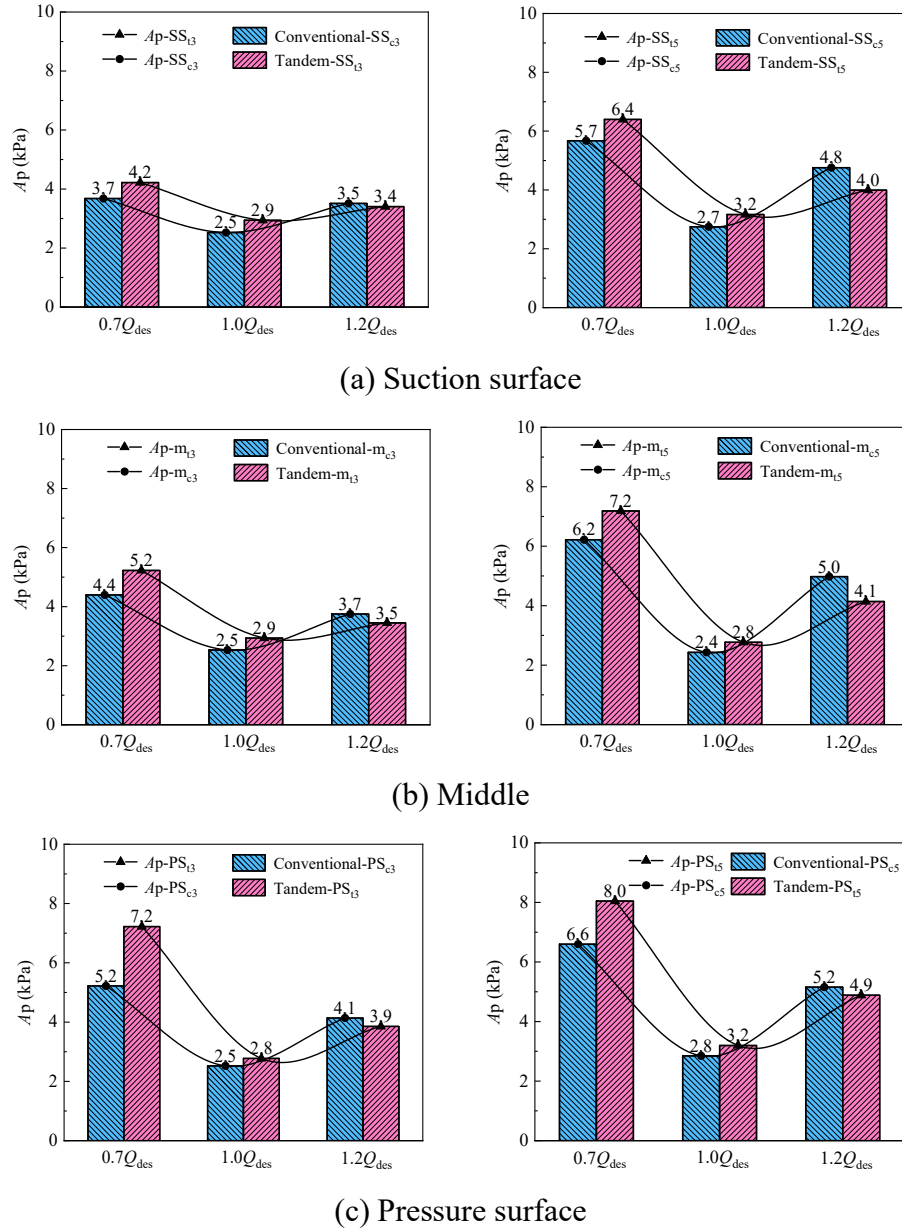


Fig. 7.21. Comparison of pressure pulsation intensity in the pump with 6 blades conventional and tandem impellers under different conditions.

Fig. 7.22 compares the A_P values in the volute for the 6-blades conventional impeller and the tandem impeller under different operating conditions. For both impellers, the A_P value at V_1 increases with Q , showing a similar growth trend and comparable magnitude. At V_2 , the A_P remains relatively stable across all flow conditions, and the difference between the two impellers is negligible. The A_P at V_3 also increases slightly with the rise in Q , but the rate of increase is modest. Notably, the tandem impeller consistently exhibits a higher A_P than the conventional impeller at this location. At V_4 and V_5 , the A_P variations with Q are relatively small. The tandem impeller shows a higher A_P at V_4 under the overload condition, while at V_5 , it exceeds the conventional impeller under the part-load condition.

At V_6 , both impellers maintain consistently low A_P values, with variations that can be considered negligible. In summary, the pressure pulsation intensity on the small cross-sectional side of the volute increases noticeably under the overload condition. However, the overall influence of the tandem impeller on the pressure pulsation intensity at different positions in the volute remains relatively limited, indicating that the modification primarily affects the impeller domain rather than the volute domain.

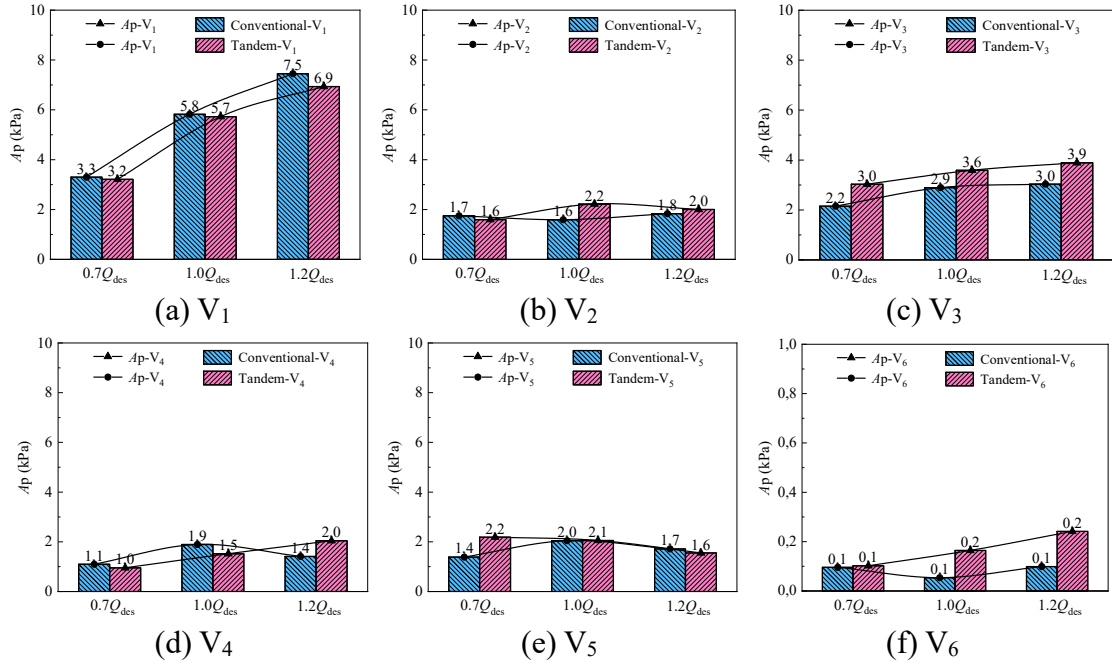


Fig. 7.22. Comparison of pressure pulsation intensity in the volute under different conditions.

To further compare the pressure pulsation intensity between the conventional impeller and the tandem impeller with different blade numbers, Fig. 7.23 presents the variation of A_P values at several key positions in the impellers under the design condition. At the mid-span and TE regions of the three representative blade sections, the A_P values for both impellers decrease progressively with the increase in blade number, and the rate of decrease gradually slows down. Moreover, the A_P of the tandem impeller are consistently higher than those of the conventional impeller across all blade numbers and positions, with the difference being more pronounced at lower blade counts. A positional comparison further indicates that the pressure pulsation on the SS of the tandem impeller is significantly stronger than that of the conventional one. In particular, the A_P at SS_{i3} and m_{i3} in the tandem impeller exceed those at SS_{i5} and m_{i5} by a larger margin compared with the conventional impeller. These observations suggest that the tandem impeller exhibits substantially stronger pressure pulsations when the number of blades is small, especially in the mid-region of the SS. Combined with the findings in Fig. 7.14, it can be inferred that the intensified evolution of the SSV induced by the mid-gap structure in the tandem

impeller is the primary cause of the significantly enhanced pressure pulsation intensity.

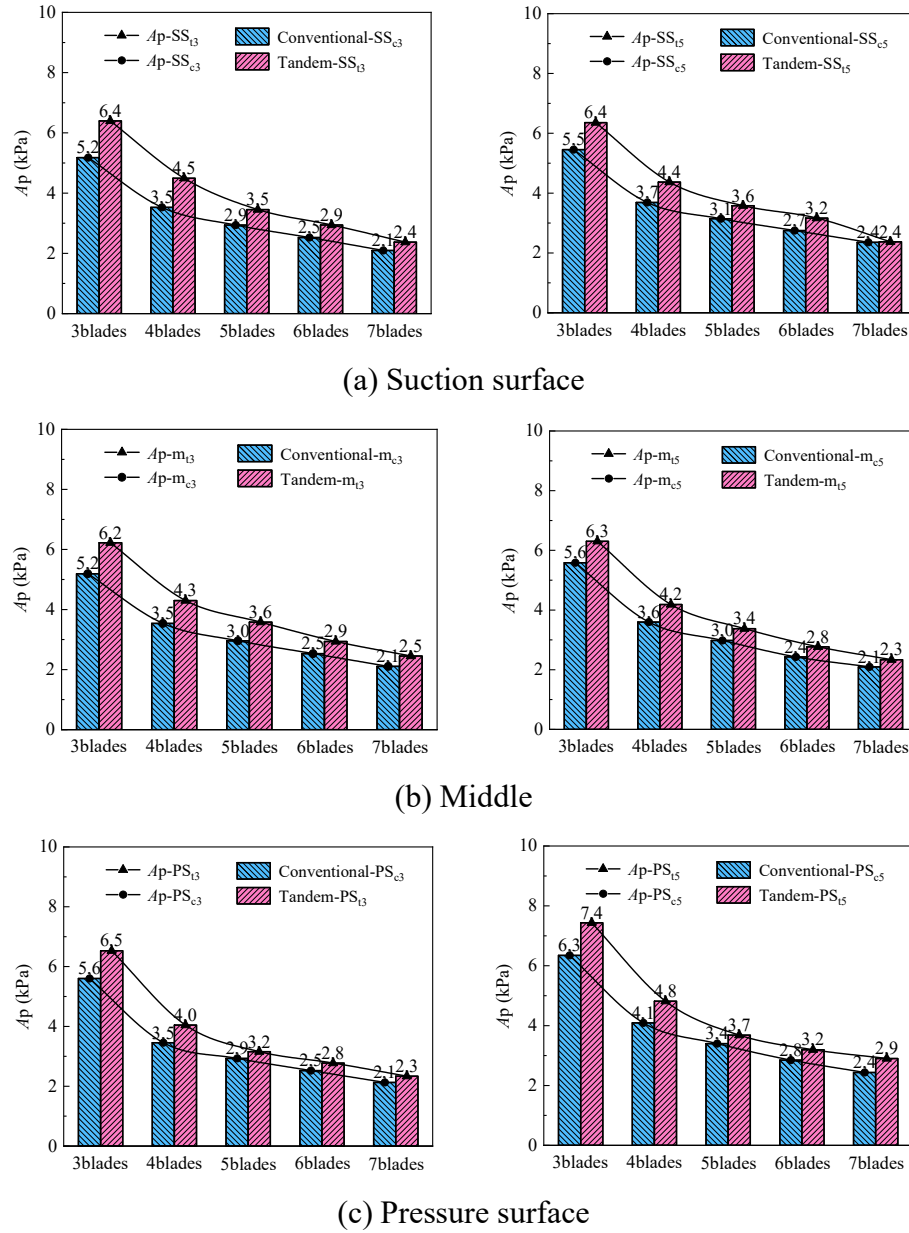


Fig. 7.23. Comparison of pressure pulsation intensity in the pump with conventional and tandem impellers under different blade numbers ($1.0Q_{des}$).

Fig. 7.24 presents the variation of A_P values in the volute for the conventional and tandem impellers with different blade numbers under the design condition. At monitoring points V_1 - V_4 , the A_P values in the volute decrease with increasing blade number for both impellers, with the most pronounced and rapid decline observed at V_1 and V_4 . When the blade number ranges from 3 to 5, the A_P values in the volute of the tandem impeller are significantly higher than those of the conventional impeller. However, when the blade number exceeds 6, the difference between the two impellers diminishes. In addition, at the small cross-sectional side of the volute (V_1 and V_2), the difference in A_P values

between the tandem and conventional impellers is more prominent than that observed on the large cross-sectional side. At the tongue region (V_5), both impellers exhibit nearly identical A_P values, with minimal influence from blade number variations. Therefore, it can be concluded that the tandem impeller primarily enhances the pressure pulsation intensity on the small cross-section side of the volute, particularly when the number of blades is small. Combined with the results shown in Fig. 7.23, this indicates that the variation in pressure pulsation intensity in the volute is mainly governed by the pressure fluctuations at the impeller outlet.

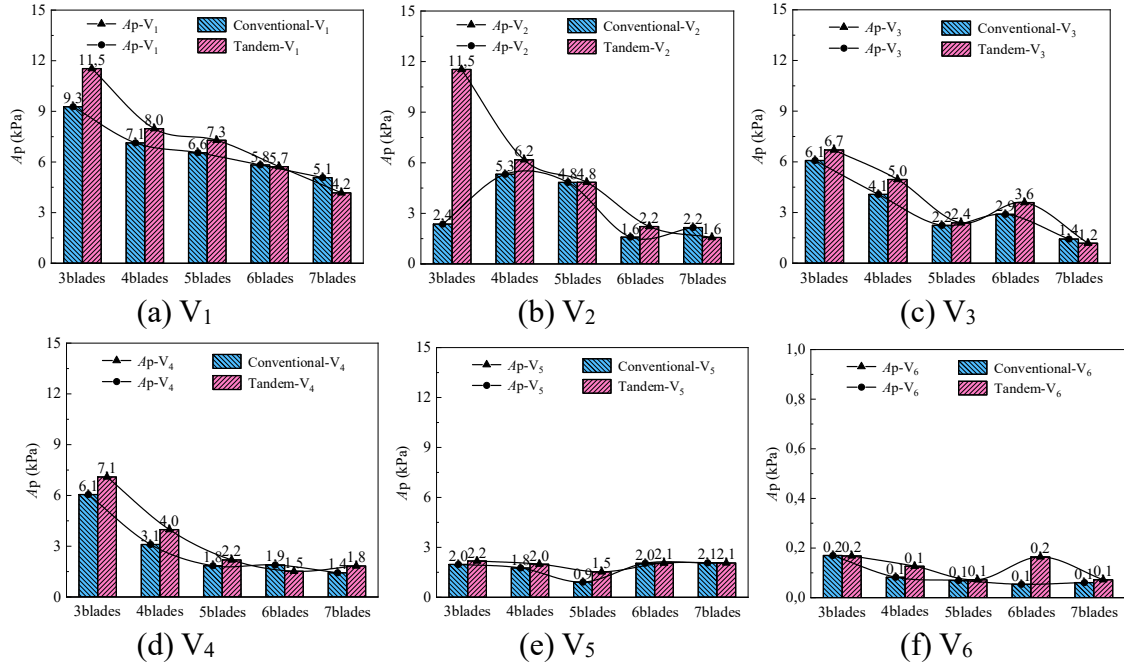


Fig. 7.24. Comparison of pressure pulsation intensity in the volute with conventional and tandem impellers under different blade numbers ($1.0Q_{des}$).

This study also conducted a separate analysis of the pressure pulsation intensity in the gap of the tandem impeller. Fig. 7.25 presents the variation of A_P values at g_{t1} and g_{t2} , whose locations are shown in Fig. 7.19(b), under three representative flow conditions as a function of blade number. Under both overload and design conditions, the A_P values at g_{t1} and g_{t2} decrease with increasing blade number, with the rate of decrease gradually slowing down. Under the part-load condition, the A_P values initially decrease with increasing blade number, but remain nearly constant once the blade number exceeds 5. Under the overload condition, the A_P at g_{t1} is slightly higher than that at g_{t2} when the blade number is small, while the difference narrows and eventually disappears as the blade number increases. Under the design condition, the difference between g_{t1} and g_{t2} remains small. However, under the part-load condition, the A_P value at g_{t1} consistently exceeds that at g_{t2} , and the difference grows larger with increasing blade number. Combined with the results shown in Fig. 7.23, it can be observed that the pressure pulsation intensity in

the gap of the tandem impeller is significantly higher than that at the mid-blade region of the conventional impeller. Therefore, the unsteady flow patterns induced by the gap further enhance the pressure pulsations, particularly under the part-load condition, where a larger pressure pulsation amplitude difference occurs between the inlet and outlet of the gap. As the Q decreases and the blade number increases, the strong pressure pulsations in the tandem impeller gap may exert a detrimental influence on the operational stability of the centrifugal pump.

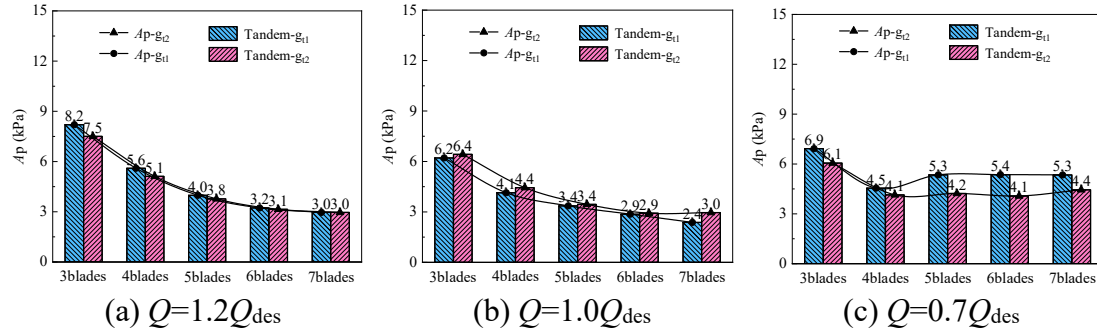


Fig. 7.25. Comparison of pressure pulsation intensity in the gap of tandem impellers under different blade numbers.

A more in-depth comparative analysis of the pressure pulsation characteristics in the two impellers was conducted in this study, based on the time-domain pressure fluctuation signals recorded during the 8th to 12th revolutions of the impeller. In the time-domain comparison, the raw pressure signals were dimensionless processed, and a dimensionless pressure pulsation coefficient, denoted as C_{PP} , was introduced. The coefficient C_{PP} is defined by Eq.7.8 as follows:

$$C_{PP} = \frac{P - \bar{P}}{0.5\rho u_2^2} \quad (7.8)$$

where P represents the transient pressure; $\bar{P}$ is the average pressure within 8-12 revolutions.

Fig. 7.26 presents a time-domain comparison of the pressure fluctuation signals on the SS and PS of the blades for both 6-bladed impellers under three representative conditions. As shown in Figs. 7.26(a) and (b), under the design condition, the pressure fluctuations on the SS of both impellers exhibit stable and periodic behavior, with 24 distinct peaks and troughs observed within four impeller revolutions. Under overload and part-load conditions, however, the amplitude of the pressure fluctuations increases significantly and the periodicity weakens, particularly near the blade TE and under the part-load condition, where the amplitude rise is more pronounced. A comparison between the two impellers indicates that the pressure fluctuation amplitude in the tandem impeller is consistently higher than that in the conventional impeller, with the difference being especially evident near the TE and under the part-load condition. Similar trends can be observed on the PS

of the blades, as shown in Figs. 7.26(c) and (d). When deviating from the design condition, both impellers exhibit larger and more irregular pressure fluctuations on the PS compared to the SS. Moreover, the enhancement of fluctuation amplitude induced by the tandem impeller is more pronounced on the PS. These observations are consistent with the distribution pattern of pressure pulsation intensity presented in Fig. 7.21, confirming that the tandem impeller amplifies unsteady pressure fluctuations, particularly under off-design conditions.

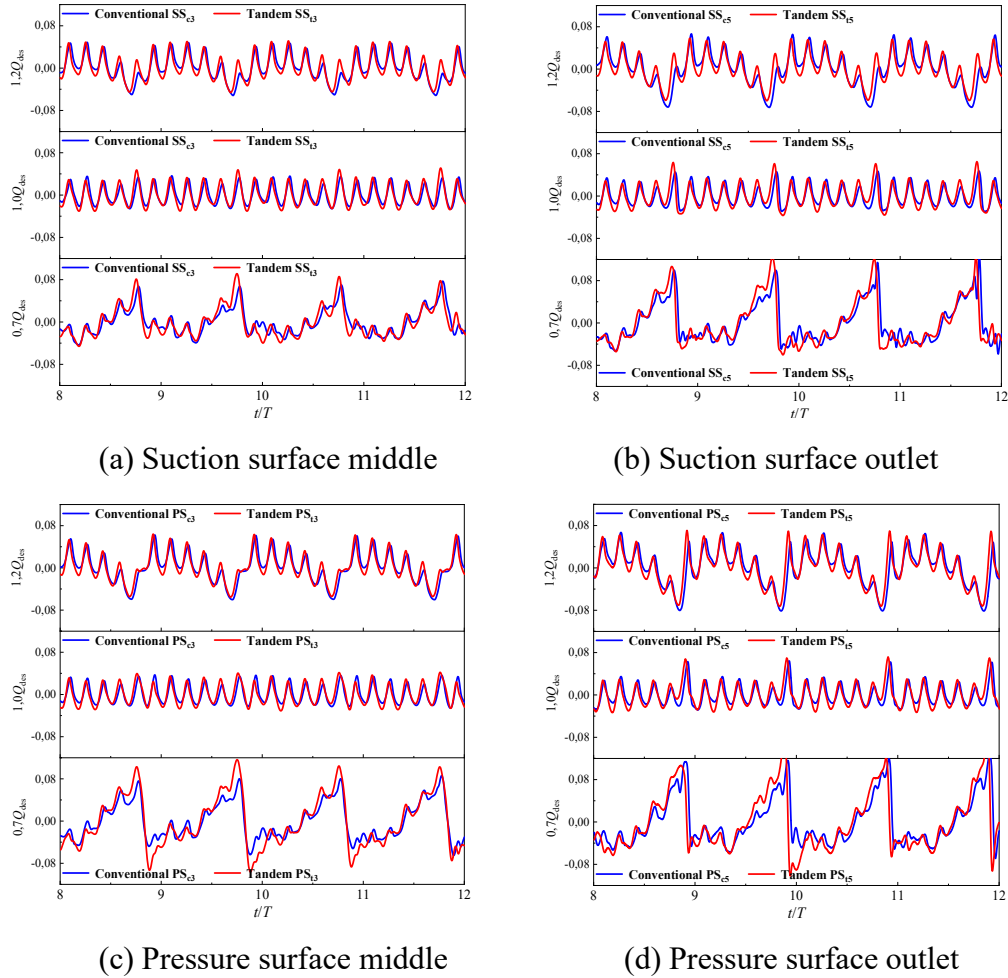


Fig. 7.26. Comparison of pressure pulsation time domain signals in the pump with 6 blades conventional and tandem impellers under different conditions.

The influence of blade number on the time-domain pressure pulsation signals of the two impellers was further investigated. Fig. 7.27 illustrates the time-domain signals at the mid-span positions on both the SS and PS under the design condition for five different blade numbers. On both sides of the blades, the pressure fluctuations exhibit a distinctly periodic behavior under the design condition, with the number of pressure fluctuation cycles corresponding directly to the blade number. Moreover, the amplitude of the pressure fluctuations increases progressively as the blade number decreases, with this

trend being more pronounced on the PS. A comparison between the two impeller configurations reveals that the tandem impeller consistently exhibits a slightly higher fluctuation amplitude than the conventional impeller, with a more noticeable increase on the SS. As the blade number decreases, the amplitude difference between the tandem and conventional impellers becomes more significant, while the periodicity of the pressure fluctuation remains unchanged. These findings are consistent with the pressure pulsation intensity distributions shown in Fig. 7.23, indicating that although the gap in the tandem impeller induces stronger pressure pulsations when the blade number is reduced, it does not disrupt the inherent periodicity of the pressure fluctuations.

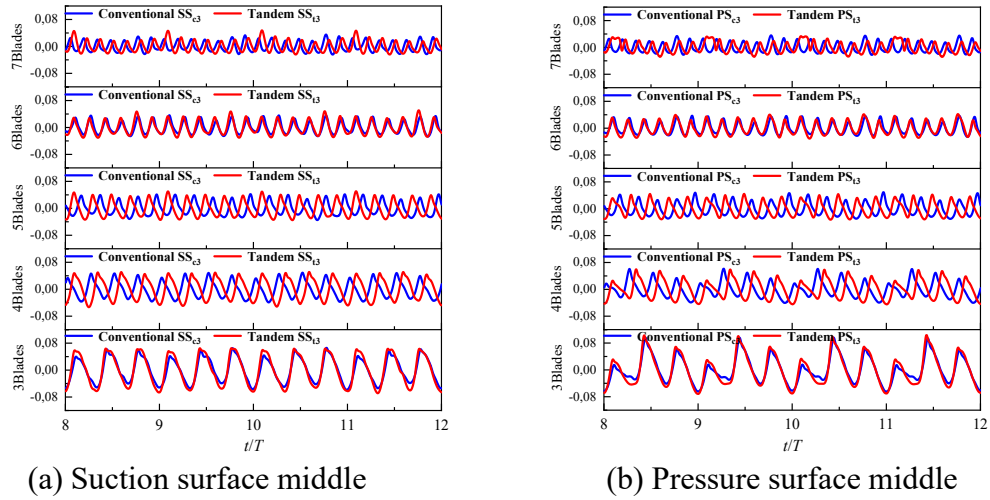


Fig. 7.27. Comparison of pressure pulsation time domain signals in the pump with conventional and tandem impellers under different blade numbers ($1.0Q_{des}$).

The influence of blade number on the time-domain pressure pulsation signals within the volute for both impeller configurations was also investigated. Fig. 7.28 presents the time-domain pressure signals at monitoring points V_1 - V_4 under the design condition for five different blade numbers. At all monitoring points, both impellers exhibit distinctly periodic pressure fluctuations, with the number of fluctuation cycles per impeller revolution corresponding precisely to the blade number. The pressure fluctuation amplitude in the volute increases consistently as the blade number decreases, particularly at V_1 and V_2 , which are located near the volute smaller cross-sectional area. When the blade number is small, the tandem impeller shows a notably larger fluctuation amplitude than the conventional impeller, with the difference being most significant at V_1 and V_2 . As the blade number exceeds 5, the amplitude difference between the two impellers becomes negligible. These observations, in conjunction with the trends shown in Fig. 7.24, indicate that the pressure pulsation amplitude in the volute is higher for impellers with fewer blades and greater near the smaller cross-sectional region of the volute. The tandem impeller primarily influences the pressure pulsations in this smaller cross-section region but does not alter the inherent periodicity of the volute pressure pulsations.

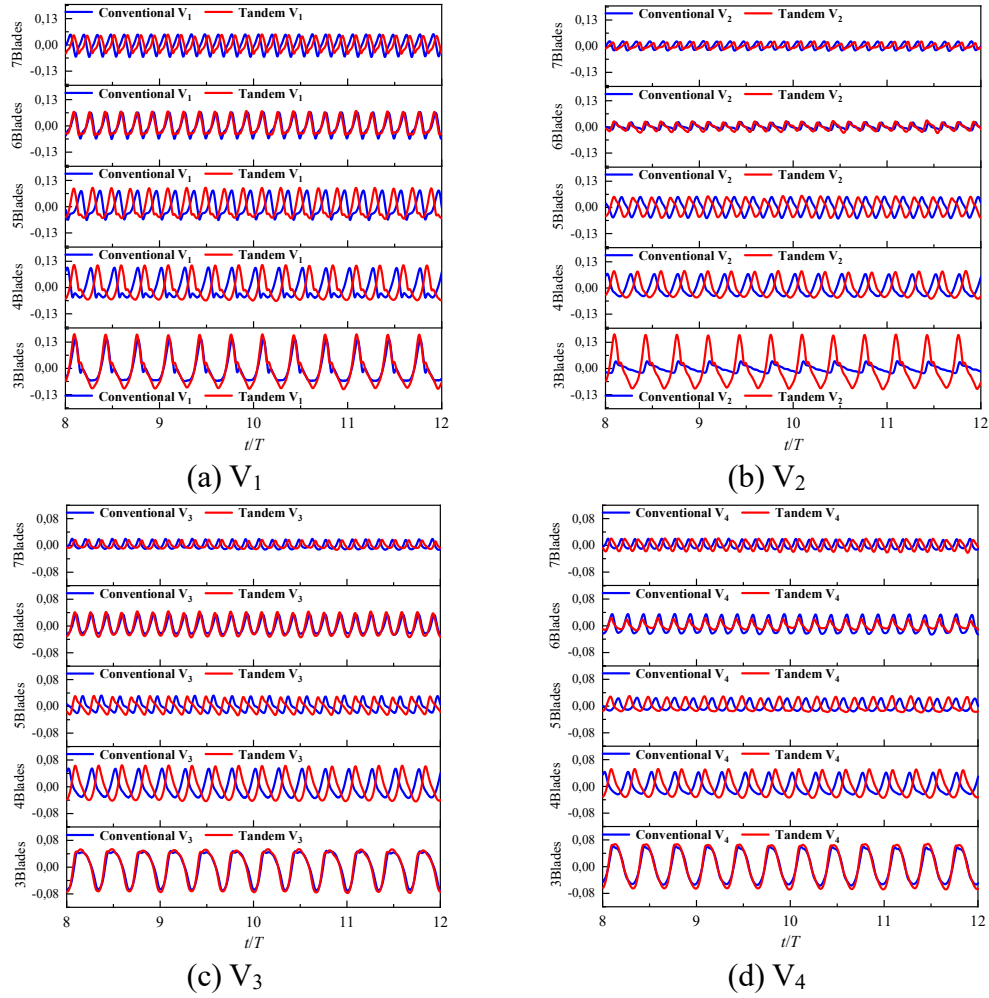


Fig. 7.28. Comparison of pressure pulsation time domain signals in the volute with conventional and tandem impellers under different blade numbers ($1.0Q_{des}$).

Fig. 7.29 investigates the influence of blade number on the time-domain pressure pulsation behavior in the gap of the tandem impeller. The temporal evolution of pressure at monitoring points g_{t1} and g_{t2} is presented for five blade numbers under three representative flow conditions, as shown in Figs. 7.29(a)-(c). Under the design condition, both g_{t1} and g_{t2} exhibit distinctly periodic pressure oscillations for all blade numbers, with the amplitude of pressure fluctuations increasing progressively as the blade number decreases. Under off-design conditions, the periodicity of the pressure fluctuations weakens, particularly under part-load conditions with larger blade numbers, where a noticeable shift in oscillation frequency occurs. Under the design and overload conditions, the amplitude difference between g_{t1} and g_{t2} remains small, showing only minor variations when the blade number is low. However, under the part-load condition, the pressure fluctuation amplitude at g_{t1} becomes significantly higher than that at g_{t2} , and when the blade number exceeds 5, the time-domain signal gradually becomes irregular and disordered. These results indicate that as the blade number increases, the high-intensity unsteady flow structures in the tandem impeller gap can alter the intrinsic pressure

pulsation frequency, particularly under off-design conditions.

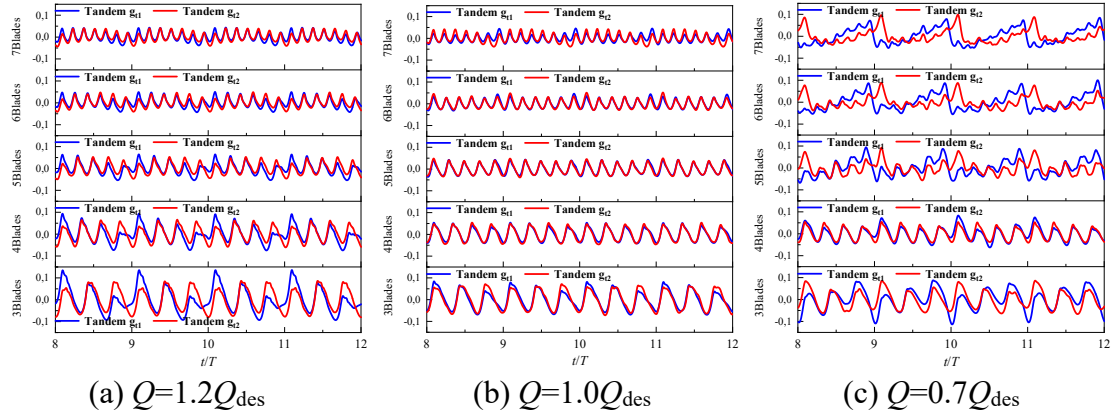


Fig. 7.29. Comparison of pressure pulsation time domain signals in the gap of tandem impellers under different blade numbers.

The time-domain analysis of pressure pulsation signals presented above provides an intuitive understanding of their temporal variation and instantaneous characteristics. However, the periodic components and energy distribution features contained in the pressure pulsations have not yet been fully elucidated. Therefore, in this study, the FFT method was employed to further identify the contribution of different frequency components in the pulsation signals and to clarify their correspondence with internal flow patterns in the pump. The frequency-domain analysis of pressure pulsations based on the FFT approach enables a deeper understanding of the dominant frequency characteristics and the potential excitation sources in the two impeller configurations, thereby providing valuable insight into the underlying mechanisms of unsteady hydraulic excitation in centrifugal pumps.

Fig. 7.30 presents the frequency-domain spectra of pressure pulsations on the PS and SS of both 6-bladed impellers under three representative conditions. As shown in Figs. 7.30(a) and (b), on the SS under the design condition, the dominant frequency consistently corresponds to 6 times the shaft frequency (f_n), i.e., $f_{BPF}=6f_n$. Under overload conditions, however, the amplitude of f_n increases significantly, particularly for the conventional impeller, where it even surpasses $6f_n$, thus becoming the new dominant frequency. Under part-load conditions, the amplitude of f_n again increases markedly and exceeds that of $6f_n$, while the amplitudes of its low-order harmonics also rise noticeably. Moreover, for the tandem impeller, both f_n and $6f_n$ amplitudes are consistently higher than those of the conventional impeller. A similar phenomenon can be observed on the PS in Figs. 7.30(c) and (d). Under off-design conditions, the $6f_n$ amplitude decreases significantly, whereas the amplitudes of f_n and its lower harmonics increase sharply. Under part-load conditions, multiple low-order harmonics of f_n exhibit pronounced high amplitudes. Although both impeller types share identical excitation frequencies, the tandem impeller consistently exhibits higher amplitudes at the same frequency compared with the conventional

impeller. In summary, under off-design conditions, the unstable flow structures on both the PS and SS of the impeller lead to a shift in the dominant pressure pulsation frequency, from the $6f_n$ to lower-order f_n , reflecting the increased influence of large-scale unsteady flow phenomena.

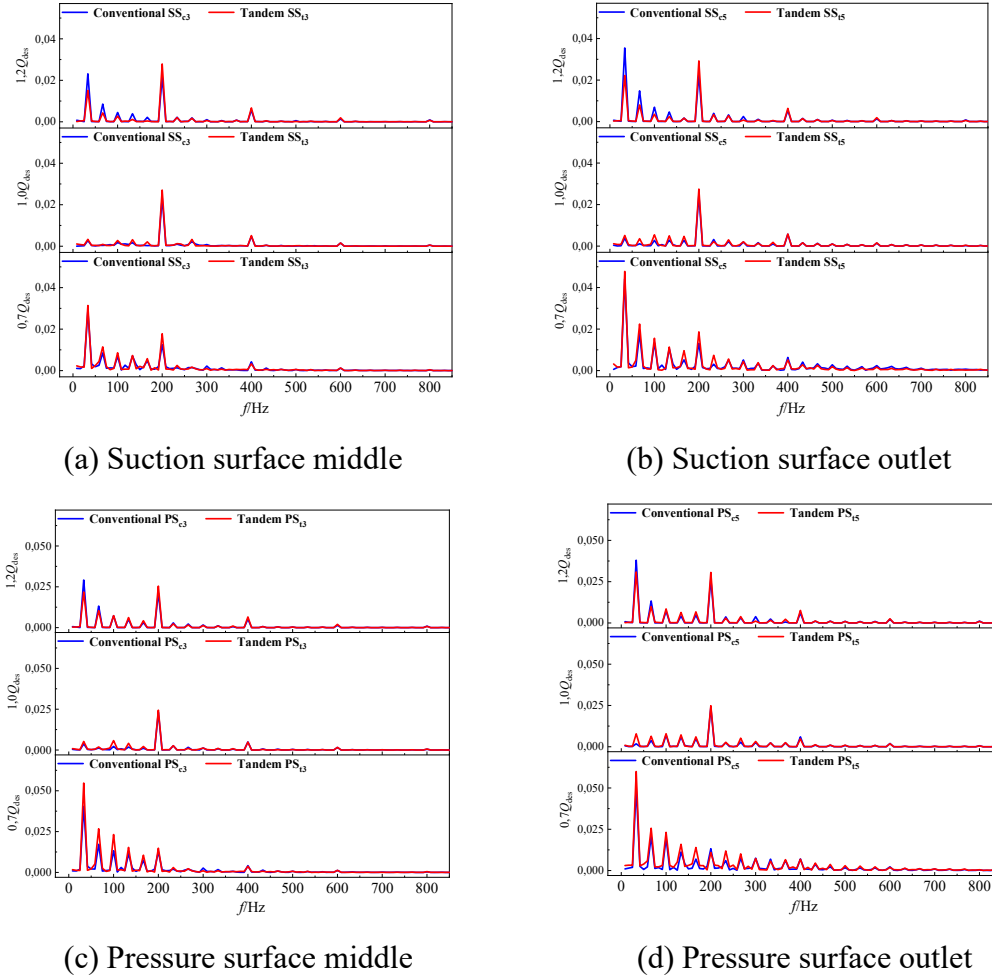


Fig. 7.30. Comparison of pressure pulsation frequency domain signals in the pump with 6 blades conventional and tandem impellers under different conditions.

Figs. 7.31(a)-(e) present the frequency-domain spectra of pressure pulsations on the SS of the blades under the design condition for five different blade numbers. For all blade numbers, the dominant frequency component of the pressure pulsations in both impellers corresponds to the f_{BPF} . The amplitude of the f_{BPF} gradually decreases with increasing blade number in both cases. A comparison between the two impeller configurations shows that, although the dominant frequencies are identical, the tandem impeller consistently exhibits higher amplitudes than the conventional impeller at both the f_{BPF} and its harmonics. Moreover, as the number of blades decreases, the amplitude difference between the tandem and conventional impellers becomes more pronounced, indicating that the tandem impeller experiences stronger periodic pressure fluctuations under lower

blade counts.

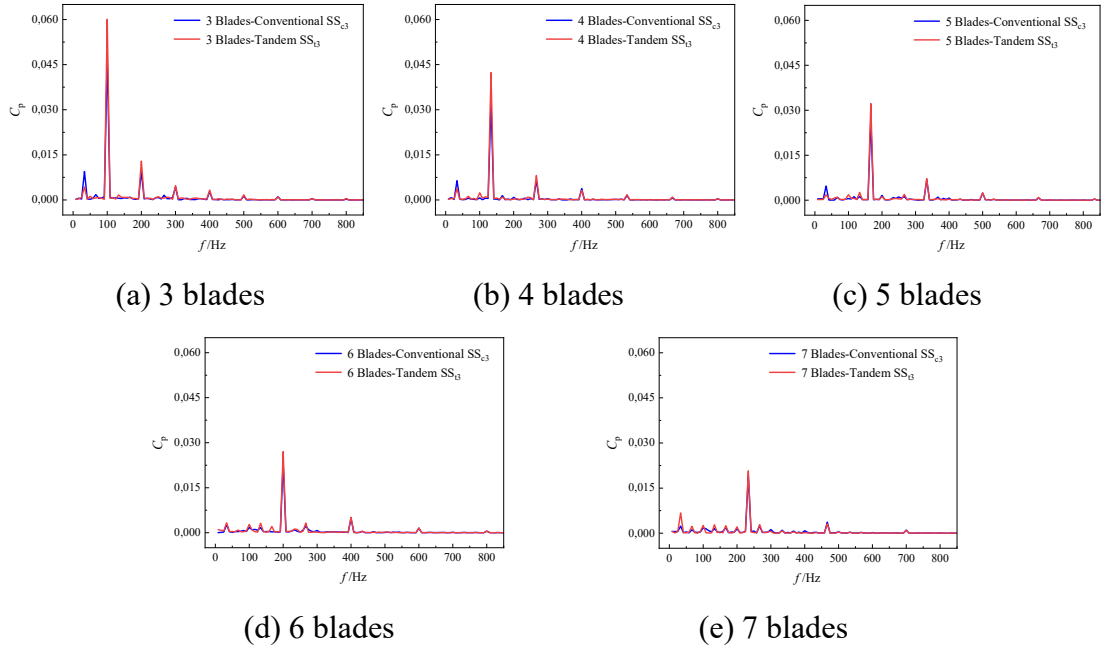
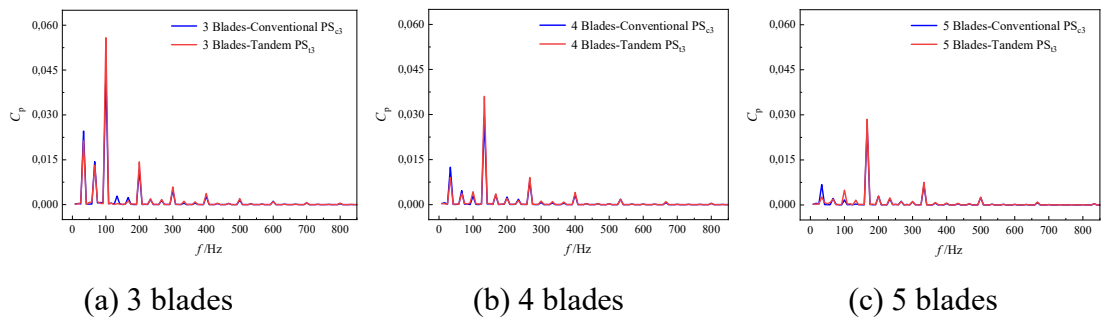


Fig. 7.31. Comparison of pressure pulsation frequency domain signals in the middle of SS with conventional and tandem impellers under different blade numbers ($1.0Q_{des}$).

Fig. 7.32 illustrates the frequency-domain spectra of pressure pulsations at the mid-span position on the PS of both impellers under varying blade numbers. For all tested blade counts, the dominant frequency on the PS remains the f_{BPF} , and its amplitude decreases progressively as the number of blades increases. However, compared with the SS, several high-amplitude low-frequency components are observed on the PS of both impellers. These components correspond primarily to the f_n and its low-order harmonics, and their amplitudes increase as the blade number decreases. A comparison of the two impellers reveals that when the blade number is below 6, the tandem impeller exhibits a higher amplitude at the f_{BPF} than the conventional impeller, with the difference becoming more pronounced at lower blade counts. Conversely, in the shaft-frequency range and its lower harmonics, the conventional impeller shows higher amplitudes than the tandem impeller.



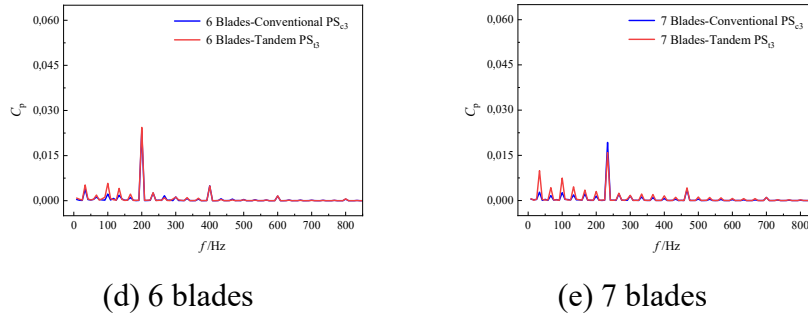
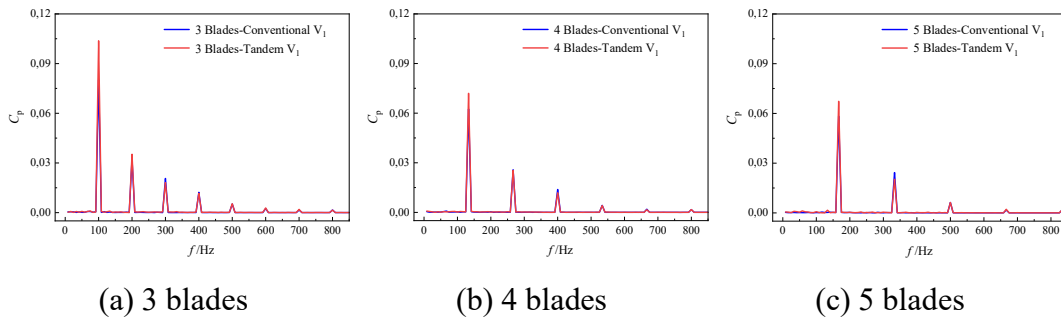


Fig. 7.32. Comparison of pressure pulsation frequency domain signals in the middle of PS with conventional and tandem impellers under different blade numbers ($1.0Q_{\text{des}}$).

Since both the blade number and blade configuration exert a significant influence on the pressure pulsation intensity and time-domain behavior at the small cross-section side of the volute, Fig. 7.33 presents the frequency-domain spectra at monitoring point V_1 under the design condition for different blade numbers. For all blade numbers, the dominant frequency of the pressure pulsation at point V_1 corresponds to the f_{BPF} , and its harmonics also exhibit pronounced high-amplitude peaks. Compared with the signals in the impeller domain, the amplitude of the dominant frequency at V_1 is considerably larger, indicating stronger pulsation in the volute. As the number of blades increases, the amplitudes of both the f_{BPF} and its harmonics decrease progressively, and the number of observable high-amplitude frequency components also diminishes. When the blade number is fewer than 6, the tandem impeller consistently exhibits higher amplitudes at the dominant frequency than the conventional impeller, with the difference becoming more significant as the blade count decreases. These results indicate that the pressure pulsations in the volute are generally stronger than those in the impeller passage, but the frequency-domain characteristics follow similar trends.



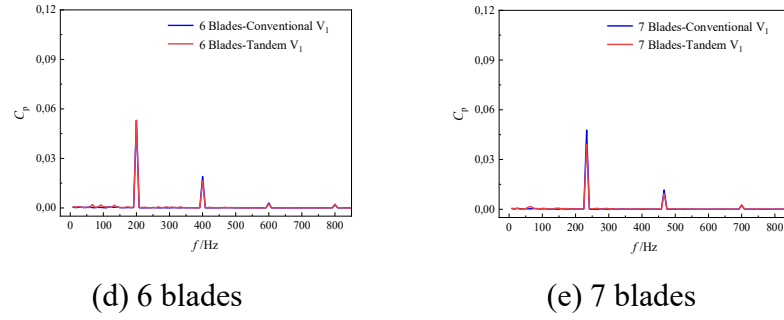


Fig. 7.33. Comparison of pressure pulsation frequency domain signals in the V_1 with conventional and tandem impellers under different blade numbers ($1.0Q_{des}$).

The results presented in Fig. 7.34 indicate that the frequency-domain characteristics at point V_2 exhibit trends similar to those observed at V_1 . For all five blade numbers, the dominant frequency of pressure pulsation at V_2 remains the f_{BPF} for both impeller configurations. However, the amplitudes of the f_{BPF} and its harmonics at V_2 are lower than those observed at V_1 . Moreover, the decreasing trend of these amplitudes with increasing blade number is more pronounced at V_2 than at V_1 . Taken together, the observations from Figs. 7.33 and 7.34 demonstrate that neither blade number variation nor the application of the tandem impeller alters the dominant frequency characteristics in the volute. Nonetheless, as the blade number decreases and the tandem impeller is employed, the amplitudes of the characteristic frequencies are significantly amplified.

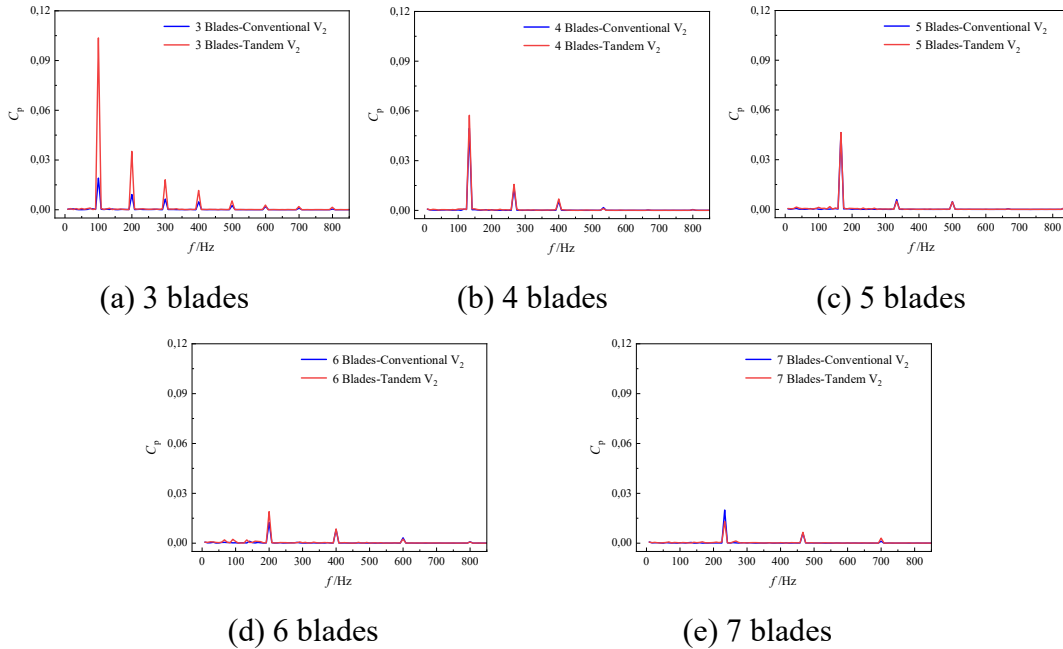
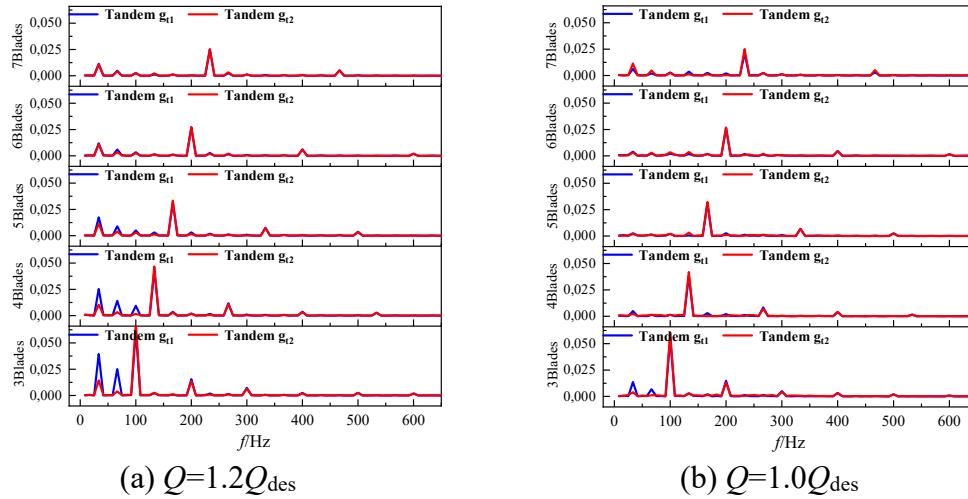


Fig. 7.34. Comparison of pressure pulsation frequency domain signals in the V_2 with conventional and tandem impellers under different blade numbers ($1.0Q_{des}$).

The frequency-domain signals in the gaps of tandem impellers with different blade numbers are shown in Fig. 7.35, where Figs. 7.35(a)-(c) correspond to the three typical flow conditions. Under the design condition, the dominant frequency of the pressure pulsation at both g_{t1} and g_{t2} is the f_{BPF} for all blade numbers. The amplitude of the f_{BPF} increases as the blade number decreases, and the dominant frequency amplitude at g_{t2} is always higher than that at g_{t1} . Under the overload condition, the dominant frequency of pressure pulsation at both g_{t1} and g_{t2} remains the f_{BPF} . However, as the blade number decreases, the amplitudes of the f_n and its low-order harmonics increase significantly, especially at g_{t1} . Although g_{t2} always shows higher amplitudes at the f_{BPF} , the amplitudes of the f_n and its low-order harmonics at g_{t1} are higher than those at g_{t2} . Under the part-load condition, when the blade number exceeds 5, the amplitudes of the f_n at both g_{t1} and g_{t2} become higher than those of the f_{BPF} , making the f_n the dominant frequency, which is more pronounced at g_{t1} . As the blade number decreases, the amplitude of the f_n gradually decreases, while that of the f_{BPF} increases, indicating that the f_{BPF} gradually becomes dominant. Under off-design conditions, the amplitudes of the characteristic frequencies at g_{t1} and g_{t2} exhibit significant differences, which become larger as the blade number decreases. This phenomenon results from the complex pressure pulsation characteristics caused by the unstable flow pattern in the gap, which may adversely affect the operational stability of the centrifugal pump.



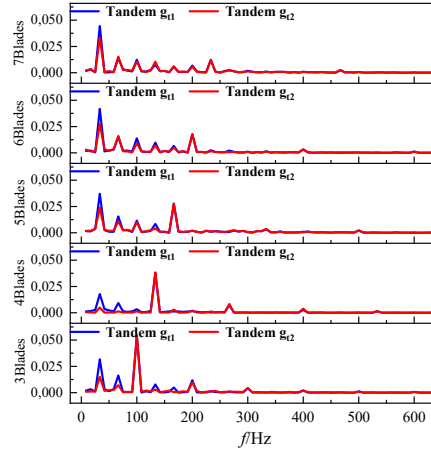
(c) $Q=0.7Q_{des}$

Fig. 7.35. Comparison of pressure pulsation frequency domain signals in the gap with tandem impellers under different blade numbers.

The time-domain and frequency-domain analyses of the pressure pulsation signals in the centrifugal pump indicate that the application of the tandem impeller affects the pressure pulsation characteristics in certain regions of the impeller and volute, with the degree of influence varying according to the blade number. However, time-domain and frequency-domain analyses alone are insufficient to simultaneously capture the localized evolution characteristics of the signal in both time and frequency. To further investigate the unstable characteristics of the pressure pulsation signals and to more accurately reveal the temporal distribution of different frequency components, this study employs the Continuous Wavelet Transform (CWT) for a joint time-frequency analysis of the pressure pulsations [117, 118]. The CWT analysis is performed using the Morlet wavelet implemented in MATLAB, based on the pressure data obtained during the 8th to 12th revolutions of the impeller.

The time-frequency characteristics (CWT results) of the mid-span region on the SS of both impellers under three representative flow conditions are shown in Fig. 7.36. Under the design condition, a distinct high amplitude $f_{BPF}=6f_n$ can be clearly observed in the conventional impeller, and its amplitude remains stable over time. In contrast, for the tandem impeller, the amplitude of the $6f_n$ signal is significantly reduced, while the f_n and several low-order harmonics become clearly distinguishable, as highlighted by the red box. Over time, the f_n component maintains a high amplitude, whereas the amplitudes of its low-order harmonics gradually stabilize only after the ninth revolution. Under the overload condition, both impellers exhibit strong amplitudes at $6f_n$ and f_n . These frequencies remain temporally stable, although the bandwidth of f_n is noticeably narrower than that of $6f_n$. The tandem impeller shows a higher amplitude at $6f_n$ compared with the conventional impeller, while the conventional impeller exhibits higher amplitudes at the low-order harmonics of f_n . Under the part-load condition, the amplitude of the $6f_n$ signal

in both impellers decreases significantly, and its temporal stability deteriorates. As highlighted in the red box, the amplitudes of f_n and its low-order harmonics increase markedly and fluctuate considerably over time. Therefore, under the part-load condition, the pressure pulsation time-frequency characteristics in the mid-span region of the SS become the most complex, the dominant frequency shifts from the f_{BPF} to the f_n , accompanied by multiple high-amplitude characteristic frequencies exhibiting unsteady temporal variations.

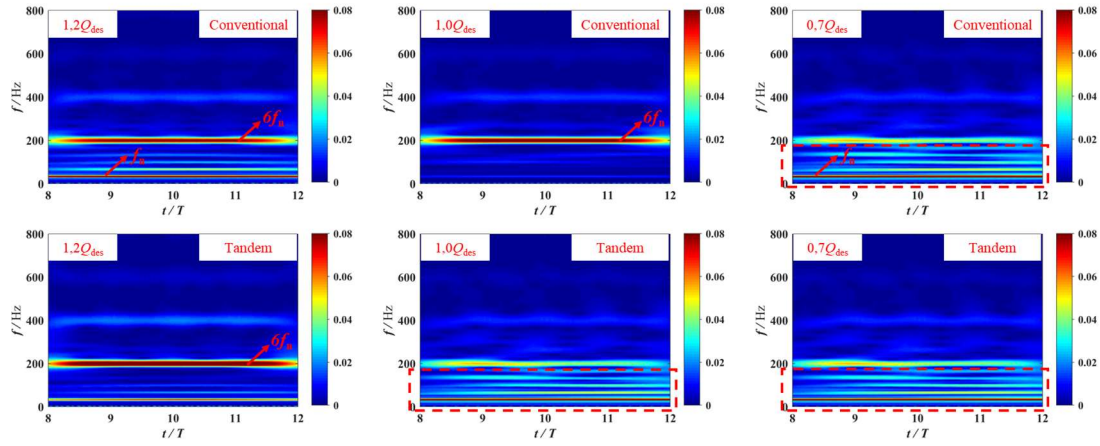


Fig. 7.36. Time-frequency characteristics of pressure pulsations in the middle of SS with conventional and tandem impellers under different flow conditions.

The CWT time-frequency characteristics at the mid-span region on the PS of both impellers are shown in Fig. 7.37. Under the design condition, both impellers exhibit a stable, high amplitude f_{BPF} that remains consistent over time. However, in the tandem impeller, low-frequency components begin to appear with noticeable temporal fluctuations. Under the overload condition, in addition to the strong $6f_n$ component, the amplitudes of f_n and its low-order harmonics increase significantly, and exhibit pronounced unsteady variations over time, as highlighted by the red box. Compared with the SS, the amplitudes of the low-order harmonics of f_n on the PS are higher, and their temporal fluctuations are also more intense. Under the part-load condition, both impellers display multiple high-amplitude low-frequency components that fluctuate irregularly over time, while the dominant frequency shifts to f_n , as indicated by the red box. Notably, in the tandem impeller, the low-frequency components have larger amplitudes and exhibit stronger temporal fluctuations. Overall, the time-frequency signals on the PS are clearly stronger and more complex than those on the SS, particularly under off-design conditions, where the tandem impeller shows multiple unstable low-frequency features associated with unsteady flow behavior.

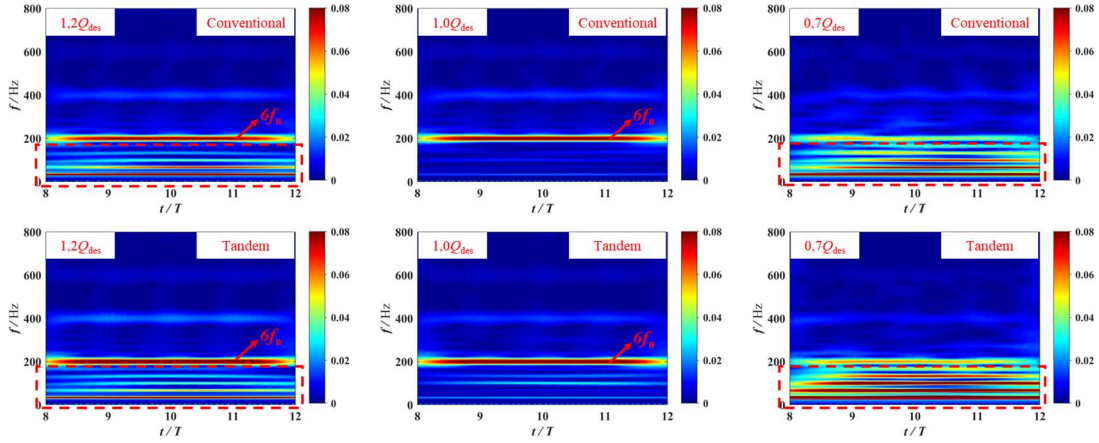


Fig. 7.37. Time-frequency characteristics of pressure pulsations in the middle of PS with conventional and tandem impellers under different flow conditions.

To compare the time-frequency characteristics of both impellers under different blade numbers, the CWT results at the mid-span region on the SS under the design condition are presented in Fig. 7.38. For both impellers, a persistent high amplitude f_{BPF} signal can be observed across all blade numbers. In the conventional impeller, as the blade number decreases, the amplitudes of the shaft frequency f_n , f_{BPF} , and their harmonics all increase slightly, but their temporal fluctuations remain small and stable. In contrast, for the tandem impeller, the amplitudes of the f_{BPF} and its harmonics increase more rapidly with decreasing number of blades, and their temporal stability deteriorates more significantly. Under the same blade number, the tandem impeller consistently exhibits higher amplitudes at the f_{BPF} and its harmonics compared with the conventional impeller, and the time-dependent fluctuation of characteristic frequencies is also noticeably stronger.

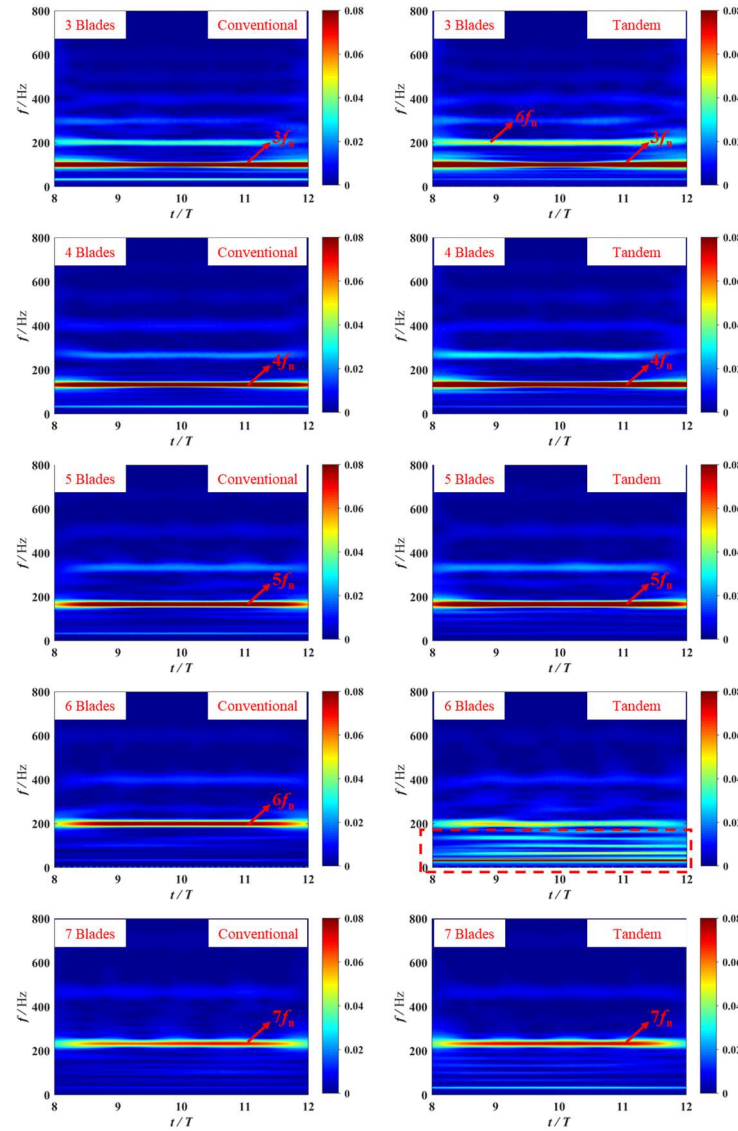


Fig. 7.38. Time-frequency characteristics of pressure pulsations in the middle of SS with conventional and tandem impellers under different blade numbers ($1.0Q_{des}$).

The CWT results at the mid-span region on the PS under the design condition are presented in Fig. 7.39. For all blade numbers, both impellers exhibit a distinct high amplitude f_{BPF} feature that remains clearly visible throughout the entire time period. As the blade number decreases, the amplitude of the f_{BPF} gradually increases. When the blade number is fewer than 5, distinct f_n and low-order harmonics begin to appear in the low-frequency range of the conventional impeller. As the blade number further decreases, the amplitudes of these low-frequency components increase, and their temporal fluctuations become more pronounced, as indicated by the red box. Under the same blade number, the tandem impeller shows more high-amplitude frequency components in the low-frequency range, and their amplitudes fluctuate more irregularly over time, a clear contrast to the behavior on the SS. With decreasing blade number, the low-frequency components in the

tandem impeller become increasingly complex and unsteady, with stronger temporal variability. Therefore, the time-frequency characteristics on the PS are more complex than those on the SS, especially in the tandem impeller with fewer blades, where the low-frequency amplitudes exhibit intensified temporal instability.

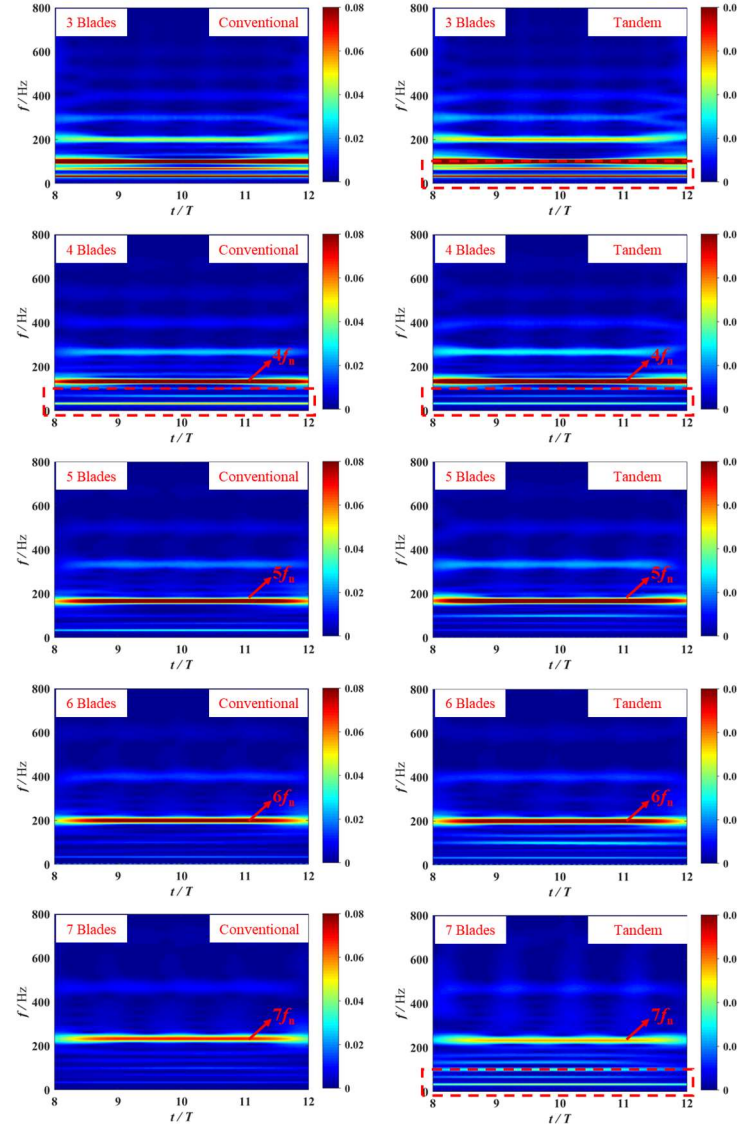


Fig. 7.39. Time-frequency characteristics of pressure pulsations in the middle of PS with conventional and tandem impellers under different blade numbers ($1.0Q_{des}$).

The CWT results at point V_1 under the design condition, illustrating the time-frequency characteristics of pressure pulsations in the volute for different blade numbers, are shown in Fig. 7.40. For all blade numbers, both impellers exhibit clearly identifiable f_{BPF} and its harmonics, with amplitudes significantly higher than those observed in the impeller passages. As the blade number decreases, a greater number of high amplitude f_{BPF} and harmonic components can be observed. In the conventional impeller, when the blade

number is small, the temporal stability of the f_{BPF} and its harmonics decreases compared with cases with more blades. Under the same blade number, the tandem impeller shows poorer temporal stability of the f_{BPF} and its harmonics than the conventional impeller, and this instability becomes more pronounced as the blade number decreases. With 3 blades, the tandem impeller even exhibits interference between multiple characteristic frequencies, indicating a strong coupling of unsteady components. This phenomenon is closely related to the high-intensity unsteady flow patterns occurring near the impeller outlet region.

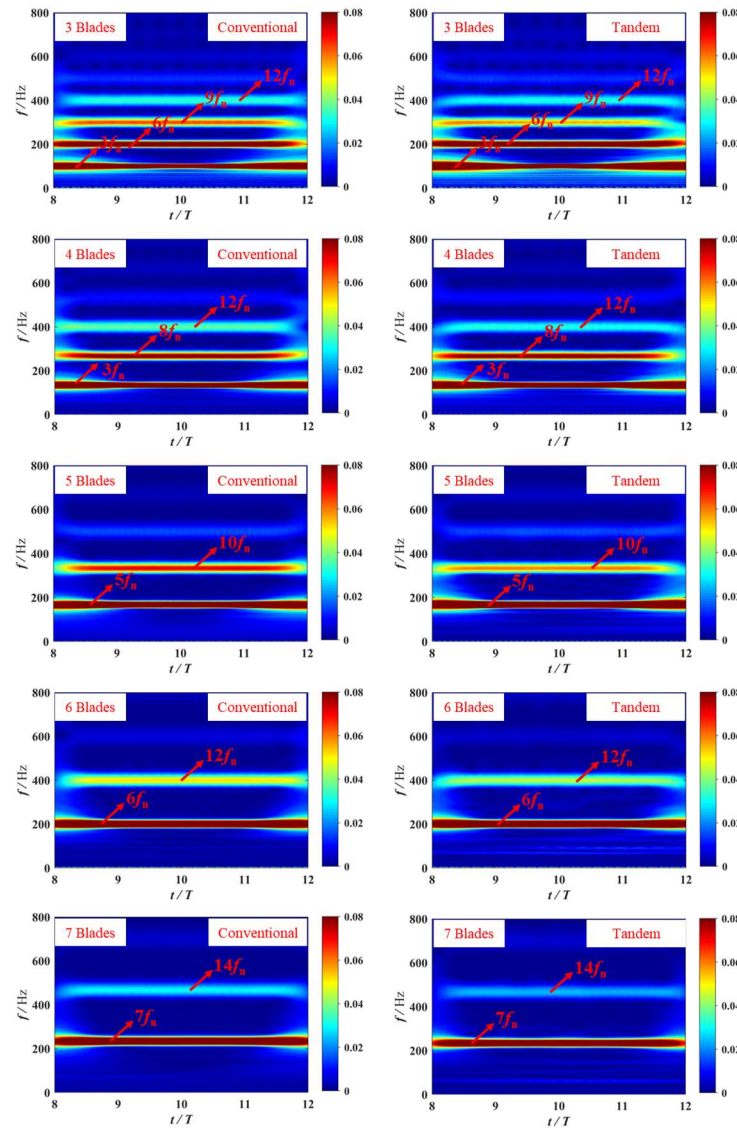


Fig. 7.40. Time-frequency characteristics of pressure pulsations in the V_1 with conventional and tandem impellers under different blade numbers ($1.0Q_{des}$).

The CWT results at point V_2 under the design condition are presented in Fig. 7.41. The time-frequency characteristics of pressure pulsations at V_2 are similar to those observed

at V_1 . However, the amplitudes of the f_{BPF} and its harmonics at V_2 are lower throughout the entire time period compared with those at V_1 . For the same blade number, the tandem impeller exhibits greater temporal instability in its characteristic frequencies than the conventional impeller. In particular, when the blade number is 3, the amplitudes of the f_{BPF} and its harmonics in the tandem impeller are significantly higher and exhibit larger fluctuations than those in the conventional impeller. Furthermore, frequency interference phenomena can be clearly observed. Combining the results from Figs. 7.40 and 7.41, it can be concluded that, influenced by the internal flow pattern of the impeller, the tandem impeller shows worse time-frequency stability of pressure pulsations in the volute. This effect becomes especially pronounced with fewer blade numbers, where the amplitudes of the characteristic frequencies increase and their temporal fluctuations become more irregular.

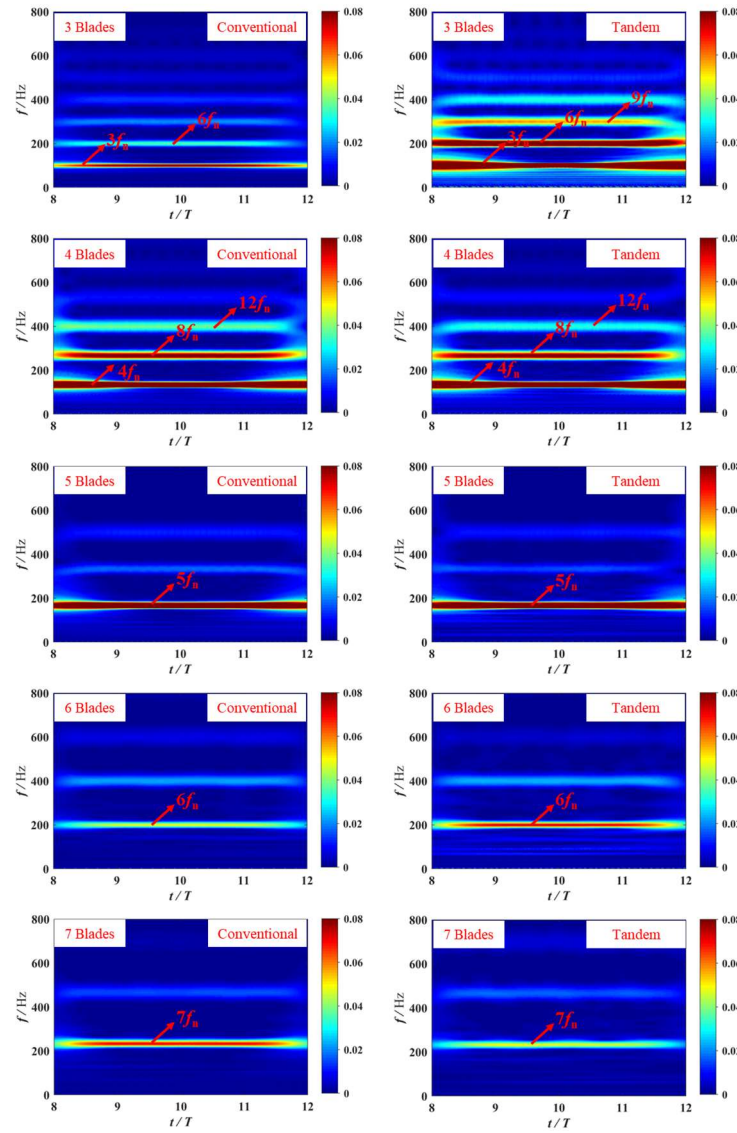


Fig. 7.41. Time-frequency characteristics of pressure pulsations in the V_2 with conventional and tandem impellers under different blade numbers ($1.0Q_{des}$).

The time-frequency characteristics of pressure pulsations in the gap, which is the most distinctive structural feature of the tandem impeller compared with the conventional one, were also investigated. Fig. 7.42 presents the CWT results at monitoring points g_{t1} and g_{t2} of the 6-blade tandem impeller under three representative flow conditions. Under the design condition, the f_{BPF} signals at both g_{t1} and g_{t2} can be clearly identified and remain highly stable over time. At g_{t2} , the f_n and its low-order harmonics can be intermittently observed in the low-frequency range, though their temporal variation is relatively unstable. Under the overload condition, the amplitudes of the f_{BPF} signals at g_{t1} and g_{t2} increase while maintaining temporal stability throughout the cycle. In addition, both g_{t1} and g_{t2} exhibit f_n and partial harmonic components in the low-frequency range, though their low-order harmonics still display mild temporal fluctuations. In contrast, under the part-load condition, the amplitudes of the f_{BPF} signals at g_{t1} and g_{t2} decrease significantly, accompanied by stronger temporal fluctuations. Multiple distinct high amplitude low-frequency components can be observed, and these low-frequency signals exhibit strong temporal instability, as highlighted by the red boxes. At g_{t2} , interference phenomena among low-frequency components during temporal evolution are clear. Therefore, under off-design conditions, the time-frequency characteristics of pressure pulsations in the gap deteriorate markedly, particularly under the part-load condition, where unstable flow patterns in the gap lead to a weakening of the f_{BPF} signal and mutual interference among low-frequency components.

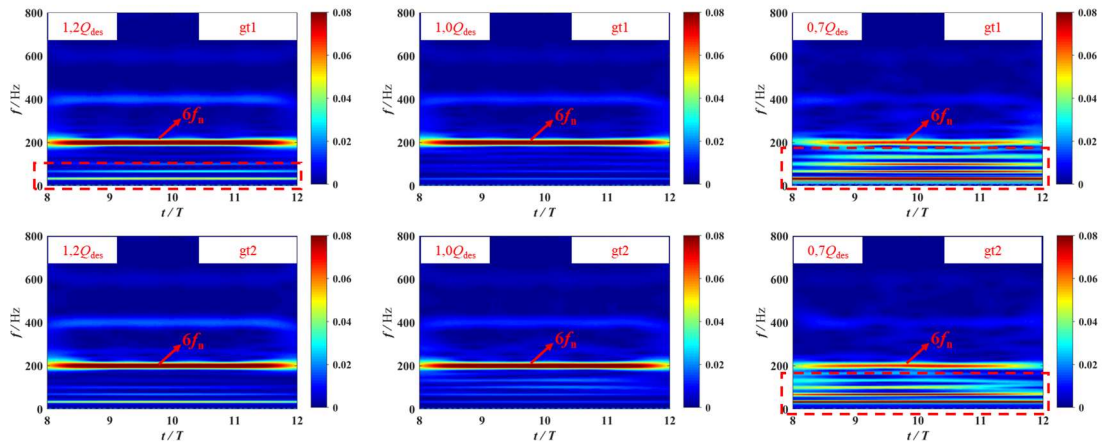


Fig. 7.42. Time-frequency characteristics of pressure pulsations in the gap of 6 blades tandem impeller under different flow conditions.

The influence of blade number variation on the time-frequency characteristics of pressure pulsations in the gap of the tandem impeller was also investigated. Fig. 7.43 presents the CWT results at monitoring points g_{t1} and g_{t2} under the design condition for different blade numbers. For all blade numbers, f_{BPF} exhibits the highest amplitude at both g_{t1} and g_{t2} . As the blade number decreases, the amplitudes of the f_{BPF} and its harmonics gradually increase. The f_{BPF} component remains temporally stable, while its harmonic components

display weaker temporal stability. At the same blade number, the f_{BPF} and f_n , along with their low-order harmonics, are consistently stronger at g_{t2} than at g_{t1} . When the blade number is reduced to 3, the number of pronounced low-frequency components increases significantly, and their temporal stability deteriorates. In particular, interference phenomena between low-order frequencies and the f_{BPF} can be clearly observed, especially at g_{t1} . Therefore, as the blade number decreases, the time-frequency characteristics of pressure pulsations in the gap region of the tandem impeller become more irregular, and the differences between g_{t1} and g_{t2} become increasingly pronounced. Combining the results from Figs. 7.42 and 7.43, it can be concluded that under the part-load condition with fewer blades, the unstable flow patterns in the gap lead to more complex and unsteady pressure pulsation characteristics, which can exert a detrimental influence on the safe and stable operation of the centrifugal pump.

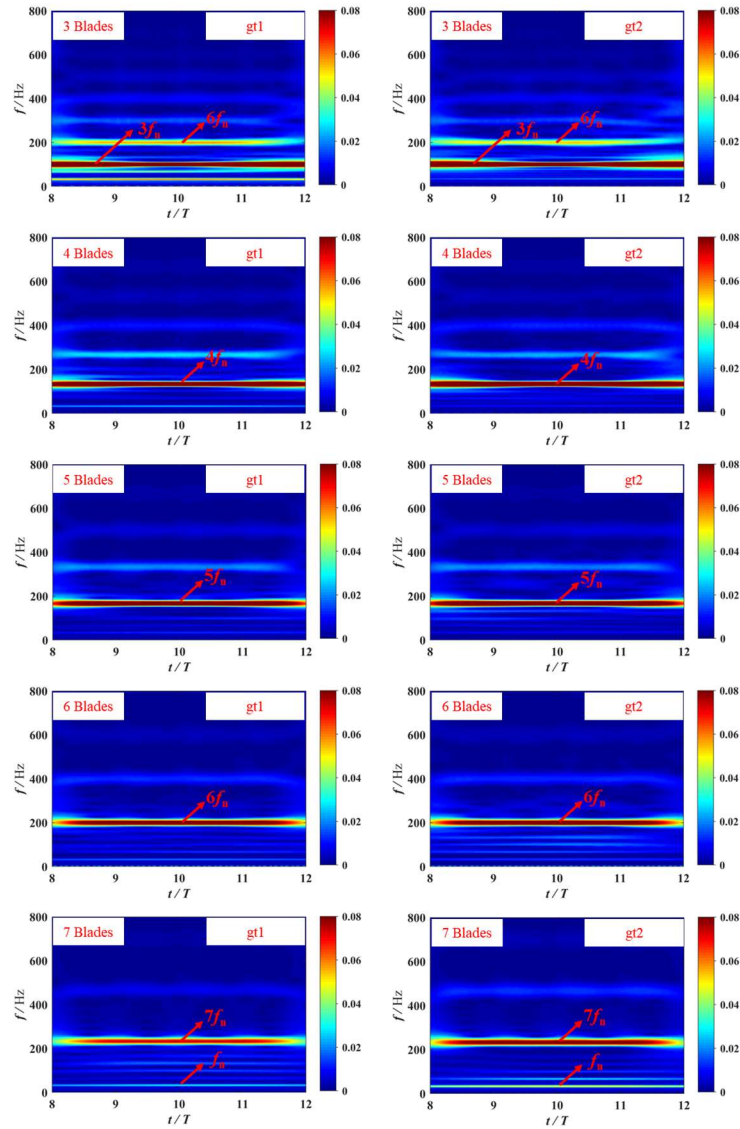


Fig. 7.43. Time-frequency characteristics of pressure pulsations in the gap of tandem impellers under different blade numbers ($1.0Q_{\text{des}}$).

7.3.2 Axial Force and Radial Force

In this study, the unsteady axial force F_A during the 8th to 12th revolutions of the impeller was statistically analyzed. Fig. 7.44 illustrates the variation of the peak-to-peak value of the axial force ΔF_A with respect to the number of blades for the two impeller configurations under three representative flow conditions. Under the design condition, both impellers exhibit small ΔF_A values with minimal fluctuation as the blade number changes. Under the overload condition, the ΔF_A of both impellers increases, accompanied by more pronounced fluctuations with blade number. However, under the part-load condition, the ΔF_A values of both impellers increase significantly compared with other conditions, and the fluctuations also become markedly larger. When comparing the two impellers, it is evident that under both the design and part-load conditions, the tandem impeller exhibits higher ΔF_A than the conventional impeller, with the difference being more pronounced under the part-load condition. Specifically, at the 5-blade configuration, the ΔF_A of the tandem impeller reaches 35.2N, approximately 1.5 times that of the conventional impeller. Under the overload condition, the tandem impeller shows a higher ΔF_A only at 4-blade and 5-blade configurations, while at other blade numbers, the ΔF_A is slightly smaller. Therefore, the unsteady characteristics of F_A in the tandem impeller do not deteriorate significantly under the overload and design conditions. However, under the part-load condition, the tandem impeller exhibits more severe unsteady axial force fluctuations across multiple blade numbers, which is closely related to the deterioration of internal flow patterns in the impeller.

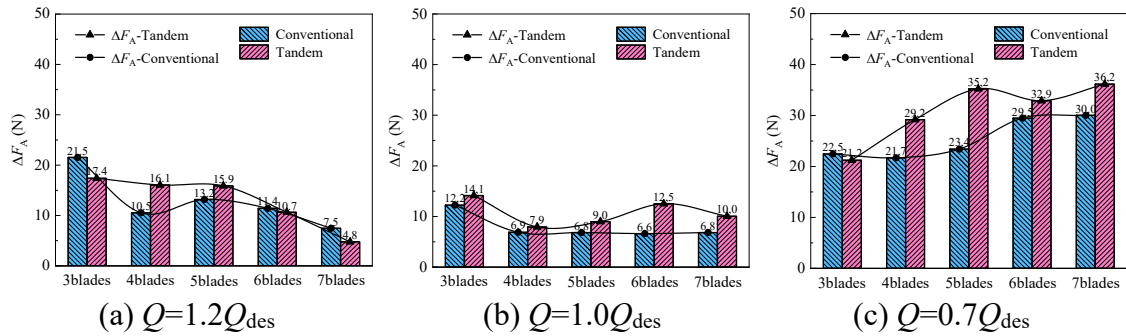
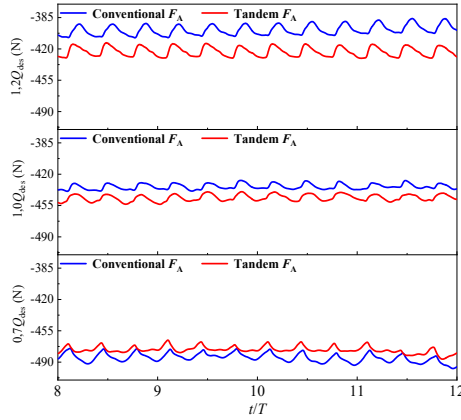


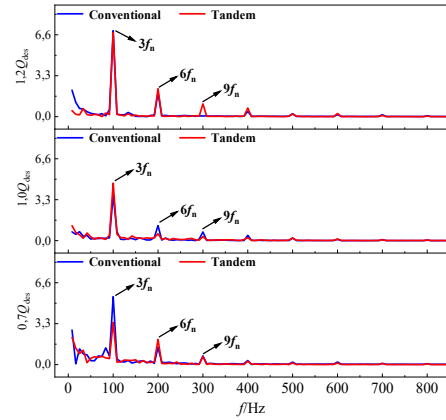
Fig. 7.44. Peak to peak values of axial thrust with conventional and tandem impellers under different blade numbers.

To further compare the unsteady characteristics of the F_A between the two impellers, Fig. 7.45 presents the time-domain and frequency-domain signals of F_A for five different blade numbers under three representative flow conditions. For all blade numbers, the F_A of both impellers increases progressively as the Q decreases. Similarly, as the blade number increases, the F_A under all three conditions also increases for both impellers. From the time-domain plots, it can be observed that the F_A of both impellers exhibits periodic fluctuations over time, with the number of oscillation periods corresponding to the blade

number. Under the design condition, the absolute F_A values of the tandem impeller remain consistently higher than those of the conventional impeller. Under the overload condition, when the blade number is fewer than 5, the tandem impeller also shows higher F_A amplitudes. However, under the part-load condition, except for the 5-blade case, the absolute F_A values of the tandem impeller are lower than those of the conventional impeller. In addition, under the part-load condition, the stability of the F_A time-domain signals for both impellers deteriorates, especially for the tandem impeller. Consequently, the ΔF_A in Fig. 7.44(c) increases significantly. From the frequency-domain plots, it can be seen that the f_{BPF} is the dominant frequency for F_A across all blade numbers and flow conditions, and higher harmonics of the f_{BPF} can also be observed. Under all blade numbers, the amplitude of the f_{BPF} is smaller under the design condition than under off-design conditions. Moreover, under the part-load condition, both impellers exhibit high amplitude low-frequency components, and the amplitude of these components increases further with increasing blade number, particularly in the tandem impeller. Under the part-load condition, the f_{BPF} amplitude in the tandem impeller weakens, while low-frequency components become more prominent. This indicates that flow pattern deterioration in the tandem impeller under the part-load condition significantly worsens the unsteady characteristics of the F_A , which is highly detrimental to the stable operation of the centrifugal pump.

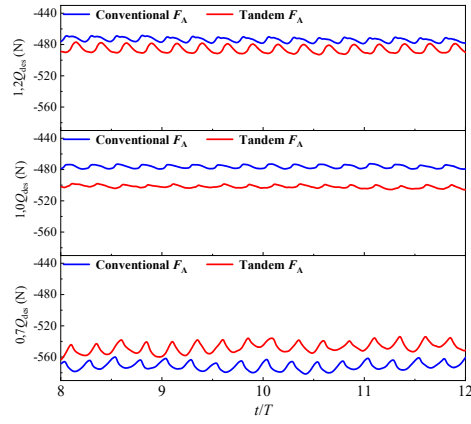


(a) 3 blades time domain

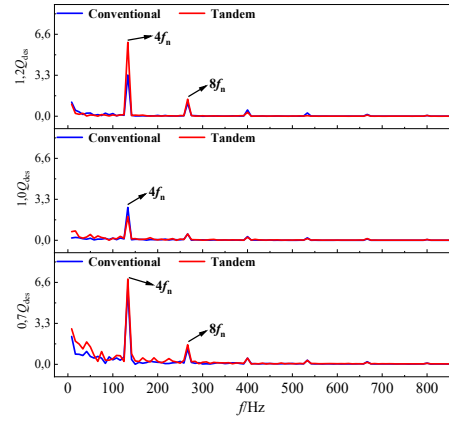


(b) 3 blades frequency domain

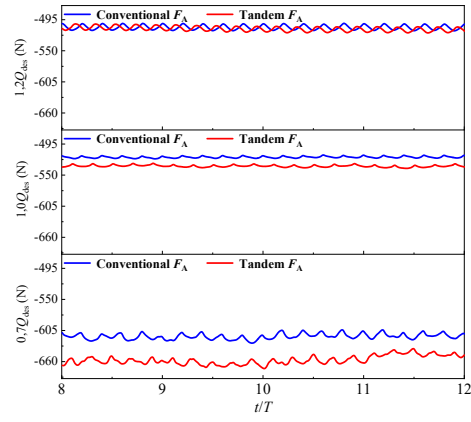
7 Comparison Analysis of Internal Flow Characteristics of the Centrifugal Pump



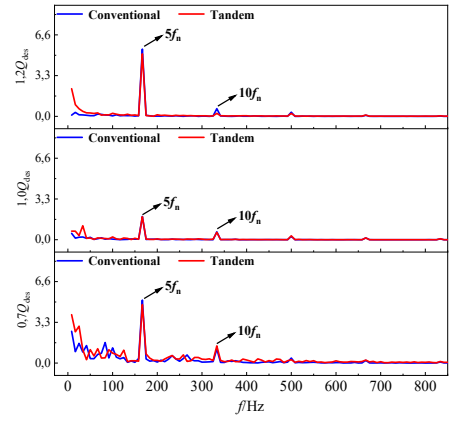
(c) 4 blades time domain



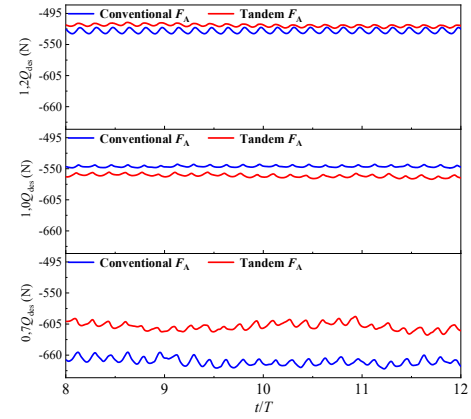
(d) 4 blades frequency domain



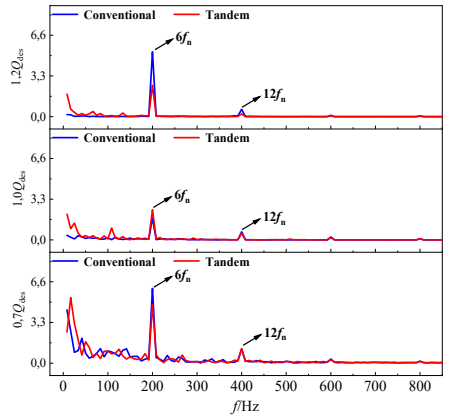
(e) 5 blades time domain



(f) 5 blades frequency domain



(g) 6 blades time domain



(h) 6 blades frequency domain

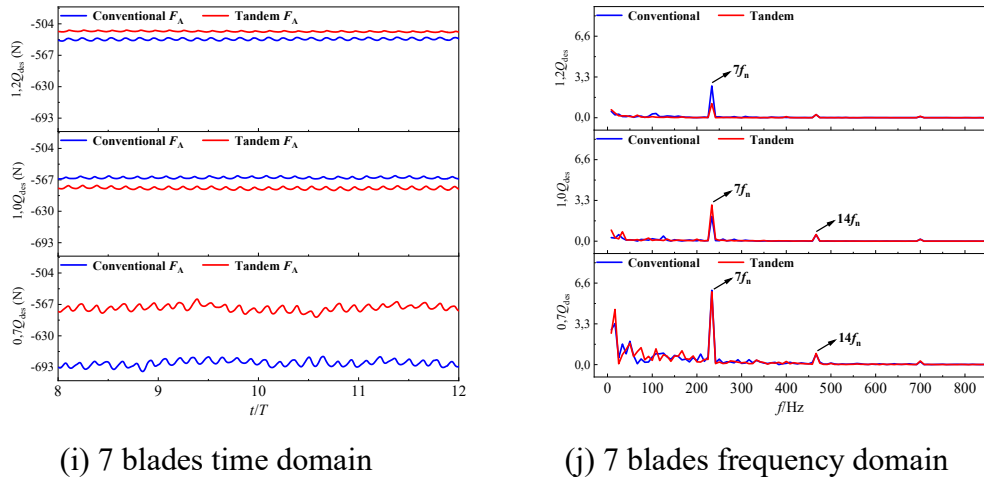


Fig. 7.45. Comparison of axial thrust time and frequency domain signals with conventional and tandem impellers under different blade numbers.

The time-frequency characteristics of the F_A were further analyzed, and the CWT results for the 6-blade conventional and tandem impellers under three flow conditions are shown in Fig. 7.46. Under the overload condition, both impellers exhibit a clear and stable f_{BPF} signal. However, the f_{BPF} amplitude in the tandem impeller is noticeably weaker, while low-frequency components begin to emerge. Under the design condition, the f_{BPF} remains stable for both impellers, though its amplitude in the conventional impeller is weaker compared with that under the overload condition. In contrast, the f_n and its low-order harmonics in the tandem impeller become more prominent. Under the part-load condition, the temporal stability of the f_{BPF} and its higher harmonics deteriorates significantly. A more distinct difference is observed in the low-frequency range, where multiple high amplitude characteristic frequencies appear, as shown in the red boxes. These low-frequency components fluctuate strongly and irregularly over time, and during certain time intervals, mutual interference among multiple frequency components is evident. Compared with the conventional impeller, the tandem impeller exhibits a greater variety of high amplitude characteristic frequencies and more severe interference phenomena among them.

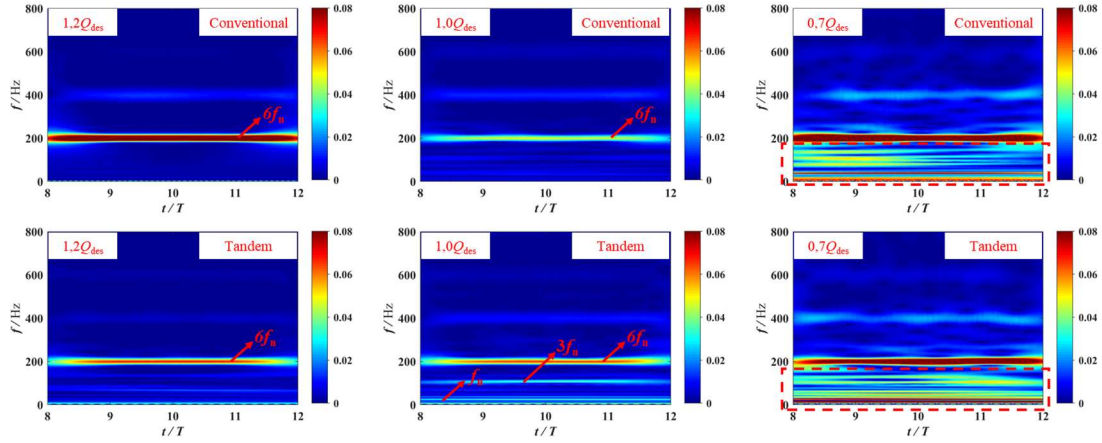


Fig. 7.46. Time-frequency characteristics of axial thrust with 6 blades conventional and tandem impellers under different flow conditions.

As shown in Fig. 7.46, the time-frequency characteristics of the F_A under the part-load condition are more complex. Therefore, to enable a more detailed comparison, Fig. 7.47 presents the CWT results of F_A for both impeller types under the part-load condition with different blade numbers. For all blade numbers, the f_{BPF} signals remain clearly identifiable and maintain high amplitudes over time. As highlighted by the red boxes, multiple high amplitude frequency components can be observed in the low-frequency range, exhibiting strong temporal fluctuations and instability. In this low-frequency region, the tandem impeller displays stronger amplitudes, larger oscillations, and more pronounced interference among different characteristic frequencies compared with the conventional impeller. As the number of blades increases, the time-frequency characteristics of F_A in the low-frequency band deteriorate more rapidly for the tandem impeller, which is attributed to the combined influence of multiple unsteady flow patterns in the impeller. This finding further confirms that, under the part-load condition, the application of the tandem impeller may reduce the operational stability of the centrifugal pump, especially when the blade number is relatively large.

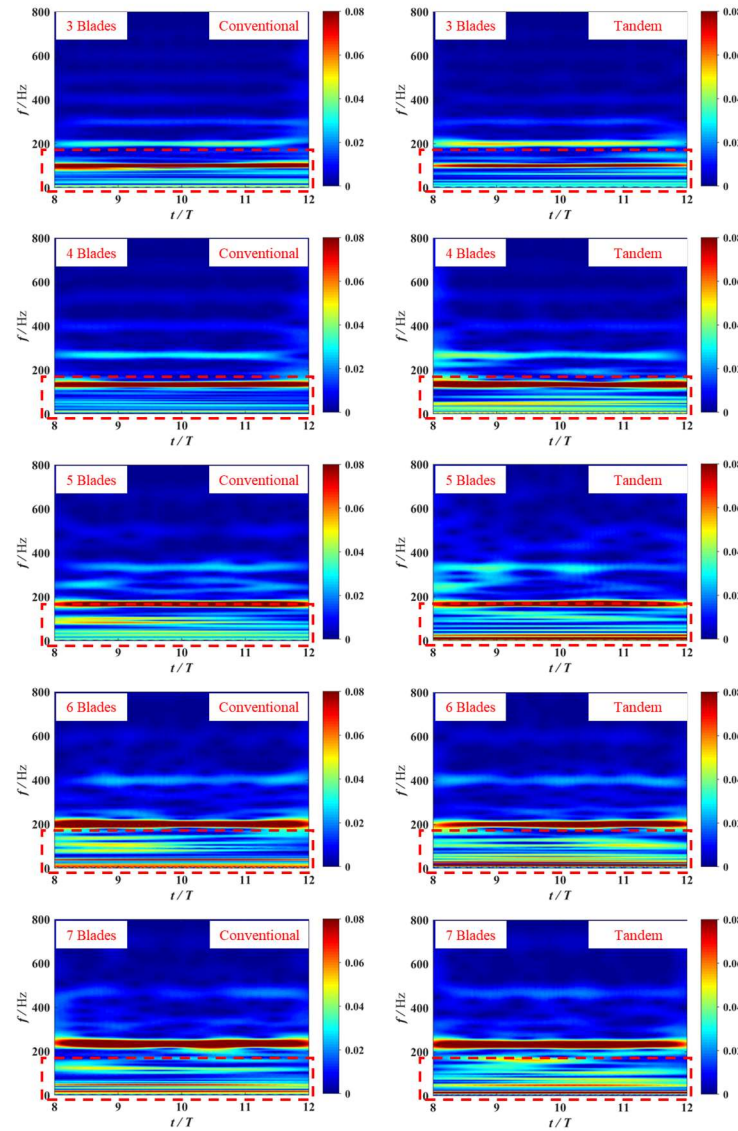


Fig. 7.47. Time-frequency characteristics of axial thrust with conventional and tandem impellers under different blade numbers ($0.7Q_{des}$).

In this study, the unsteady radial force F_R during the 8th to 12th revolutions of the impeller was monitored, and the peak-to-peak values ΔF_R of the two impellers under three representative flow conditions are presented in Fig. 7.48. Under the overload condition, the ΔF_R of both impellers gradually decreases with increasing blade number, with the rate of decrease becoming progressively slower. The tandem impeller consistently exhibits ΔF_R values that are 5-7 N lower than those of the conventional impeller, and the difference becomes larger as the blade number increases. Under the design condition, the ΔF_R values of both impellers are significantly lower than those under the overload condition. The ΔF_R of the conventional impeller continues to decrease as the blade number increases, whereas that of the tandem impeller shows a slight increasing trend when the blade number reaches 6-7. When the blade number is less than 6, the ΔF_R of the tandem impeller

remains smaller than that of the conventional impeller, but the difference gradually narrows with increasing blade number. Under the part-load condition, the variation trend of ΔF_R is opposite to that observed under overload and design conditions. The tandem impeller consistently shows larger ΔF_R than the conventional impeller, and the difference increases with blade number, reaching a maximum of about 12N. Therefore, the unsteady flow patterns in the impeller under the part-load condition can lead to an increase in F_R pulsation, particularly when the tandem impeller operates with a larger number of blades.

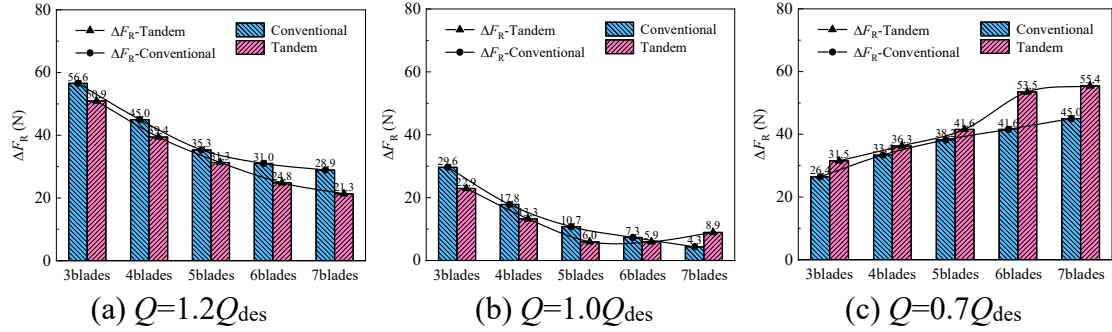


Fig. 7.48. Peak to peak values of radial force with conventional and tandem impellers under different blade numbers.

The time-domain and frequency-domain signals of the F_R for the 6-blade conventional and tandem impellers under three flow conditions are shown in Fig. 7.49. As illustrated in Fig. 7.49(a), the magnitude of F_R is significantly smaller than that of F_A , particularly under the design condition, where F_R remains very small and exhibits minimal fluctuations, indicating a stable internal flow pattern in the pump. Under off-design conditions, however, both the magnitude and fluctuation amplitude of F_R increase notably, with the largest fluctuations observed under the part-load condition, which is consistent with the trend shown in Fig. 7.48. Under the overload condition, the tandem impeller exhibits smaller F_R fluctuations in the time domain compared with the conventional impeller, whereas under the part-load condition, its fluctuations become larger. The frequency-domain results in Fig. 7.49(b) show that the dominant frequency of F_R for both impellers is the f_n , and its amplitude increases markedly under the part-load condition. Under the overload condition, the f_n amplitude of the conventional impeller is higher, whereas under the part-load condition, the f_n amplitude of the tandem impeller surpasses that of the conventional impeller.

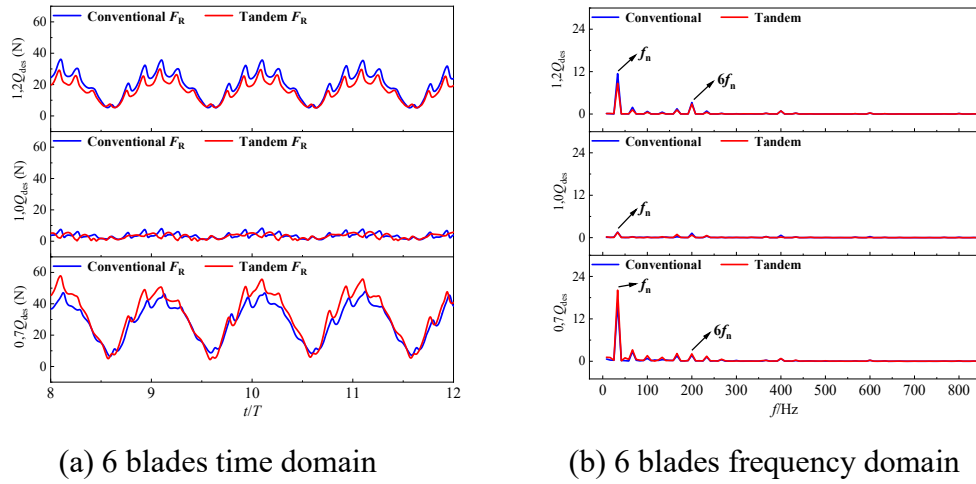


Fig. 7.49. Comparison of radial force time and frequency domain signals with 6 blades conventional and tandem impellers under different flow conditions.

Both Figs. 7.48 and 7.49 indicate that the unsteady characteristics of the F_R are more pronounced under the part-load condition. Therefore, Fig. 7.50 presents the time-domain and frequency-domain signals of F_R for the two impellers with 5 different blade numbers under the part-load condition. In the time-domain results, the mean value of F_R gradually increases with the number of blades, and the fluctuation amplitude also becomes larger. The tandem impeller consistently exhibits greater F_R fluctuations than the conventional impeller, and this difference becomes more significant as the blade number increases. The frequency-domain analysis shows that the dominant frequency of F_R is always the f_n , and the amplitude of f_n increases with the number of blades. In the F_R spectra, only f_n can be clearly identified, and its amplitude is consistently higher for the tandem impeller than for the conventional impeller.

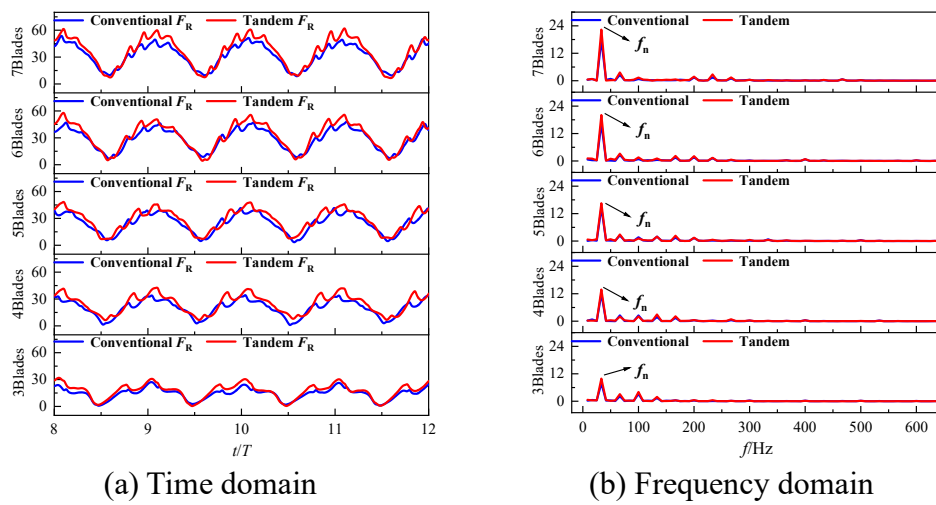


Fig. 7.50. Comparison of radial force time and frequency domain signals with conventional and tandem impellers under different blade numbers ($0.7Q_{des}$).

Finally, to further clarify the influence of different impeller structures on the unsteady characteristics of the radial excitation force components, Fig. 7.51 presents the distribution of the instantaneous radial excitation force vectors for the two 6-blade impellers under three flow conditions. As shown in Fig. 7.51(a), under the overload condition, the vector points of both impellers are mainly clustered in the negative-value region with a relatively clear distribution pattern. The conventional impeller exhibits a larger vector coverage area than the tandem impeller, indicating that its F_R is more unbalanced across directions. In Fig. 7.51(b), corresponding to the design condition, the distribution area of the radial force vector points for both impellers become significantly smaller, while the regularity of the distribution is reduced. The vector points of the tandem impeller are closer to the origin (O) and exhibit a smaller coverage area, implying that its F_R is more evenly distributed in different directions. As shown in Fig. 7.51(c), under the part-load condition, the vector points of both impellers are concentrated in the positive-value region, and their coverage area increases significantly. The vector points of the tandem impeller are farther from the O and cover a larger area than those of the conventional impeller. Therefore, the unsteady characteristics of F_R for the tandem impeller are considerably worse under the part-load condition.

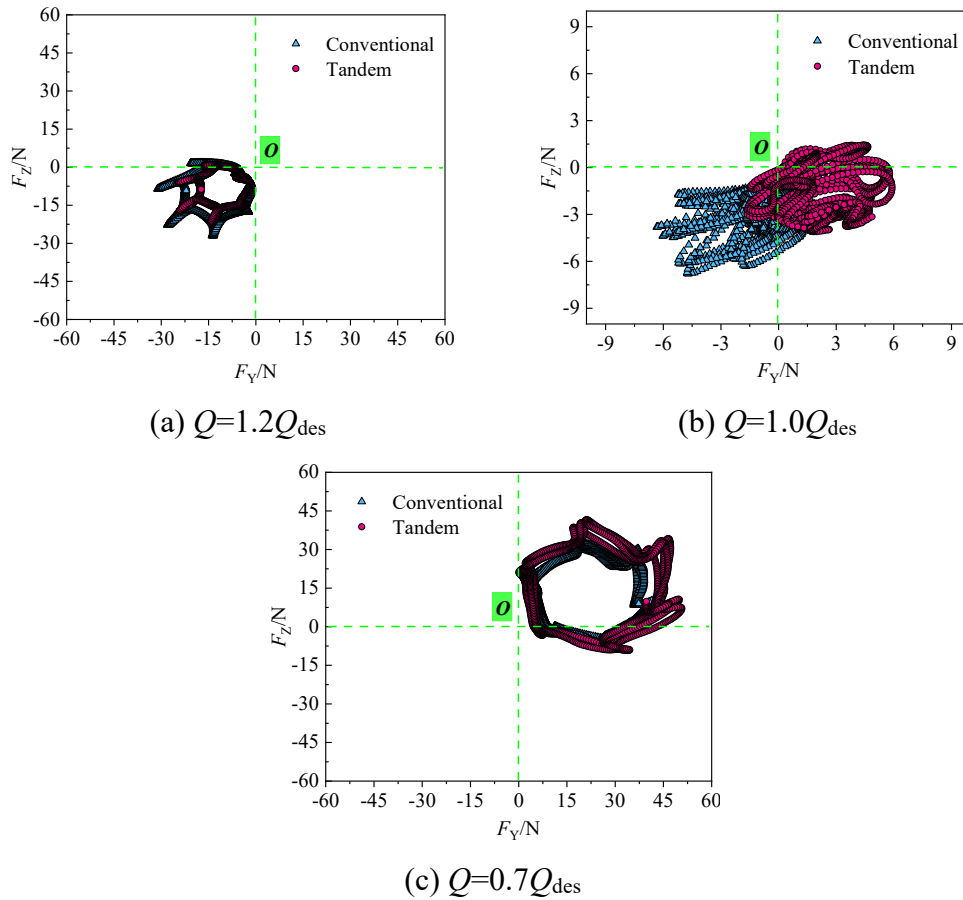


Fig. 7.51. Comparison of radial force vectors with 6 blades conventional and tandem impellers under different flow conditions.

8 Discussion

In this research, a comprehensive study has been conducted on the hydraulic design, numerical simulation, and experimental comparison of the tandem impeller and the conventional impeller. However, the obtained results still require further analysis and interpretation to reveal the underlying physical mechanisms and engineering significance. This chapter focuses on an in-depth discussion of the performance enhancement mechanism, internal flow evolution characteristics, and vibration excitation response of the tandem impeller. Its advantages and limitations are summarized, and its application potential in centrifugal pumps is explored, thereby providing theoretical support for future engineering-oriented design.

8.1 Performance Improvement of Tandem Impeller

The comparative analysis of the external characteristic parameters is primarily conducted to reveal the performance enhancement patterns of the tandem impeller relative to the conventional impeller in terms of H , η , and P . The systematic experimental and numerical investigations indicate that, under appropriate geometric configurations, the tandem impeller can achieve a significant increase in H while maintaining a stable η .

In the preliminary optimization of the conventional impeller, the effects of blade number and blade WA on centrifugal pump performance were comprehensively analyzed. The results show that increasing the blade number effectively improves H . However, when the blade number exceeds 6, the flow passage becomes narrower, leading to intensified local flow separation and causing the H improvement to approach saturation. Meanwhile, the 6-blade impeller exhibits the highest η and a relatively smooth performance curve under the design condition, thus achieving a balanced performance in terms of H , η , and P . On the other hand, increasing the WA enhances flow stability under part-load conditions but causes a significant increase in friction losses under overload conditions. By comparison, the impeller with WA=120° achieves the optimal trade-off between H and η . Therefore, based on the combined experimental and numerical results, the impeller configuration with 6 blades and WA=120° was identified as the optimal conventional impeller design and subsequently adopted as the baseline model for the design and comparative study of the tandem impeller.

Based on the optimized baseline configuration, the influence of the geometric parameters of the tandem impeller on centrifugal pump performance was further investigated. The results show that a reasonable circumferential offset of the rear blade, $D=0.13$ ($\Delta\varphi\approx7.45^\circ$), together with an appropriate rear-blade β_{21} , can effectively improve the continuity of energy transfer between the front and rear blades. When D is either too small or too large, fluctuations or reductions appear in the Q - H curve, whereas at the optimal offset, both the Q - H and Q - η curves exhibit smooth and stable variation trends. A moderate reduction in

β_{21} can further increase H , but an excessively small β_{21} leads to a slight decrease in η under part-load conditions. Ultimately, Case 3 ($D=0.13$, $\beta_{21}=10.8^\circ$) was identified as achieving the best balance between performance and stability.

Compared with the conventional impeller, the tandem impeller exhibits a higher H under most operating conditions. Both EXP and unsteady CFD show that the H improvement ranges from approximately 1-2m, with the most pronounced enhancement occurring under overload conditions. Although the P increases slightly, the gain in H effectively offsets the growth in energy consumption, resulting in an overall η comparable to that of the conventional impeller. This indicates that the tandem configuration achieves a higher hydraulic efficiency in energy transfer without introducing additional losses. Further adaptability studies show that the tandem impeller can improve external performance across different blade numbers, and the improvement in H becomes more significant as the blade number increases. Within the range of 3-7 blades, the maximum H increment $\Delta H_{\max}$ increases from 1.2m to approximately 2.0m, with the 6-blade and 7-blade configurations demonstrating the most prominent performance enhancement. In terms of η , the tandem impeller performs similarly to the conventional impeller under part-load and design conditions, while generally exhibiting higher η under overload conditions. The P shows minimal difference between the two impellers under part-load conditions and becomes slightly higher for the tandem impeller under overload conditions. However, its unit head energy consumption Ψ_H remains stable. In summary, the tandem impeller achieves a coordinated optimization of increased H , stable efficiency η , and balanced energy consumption, demonstrating a particularly notable comprehensive performance advantage under overload conditions.

From the perspective of internal flow characteristics, the tandem impeller alters the energy-loading pattern of the blades, resulting in flow structures and energy-transfer mechanisms distinct from those of the conventional impeller. In terms of the flow pattern, the tandem impeller forms a wider pressure-gradient region near the mid-span and TE sections of the blades, while the gap becomes a key zone for energy redistribution. Comparative analysis under different blade numbers shows that the vortex intensity and C_p in the gap region of the tandem impeller are significantly higher than those of the conventional impeller, and these enhancements become more pronounced with increasing Q and blade number. This indicates that the tandem configuration provides stronger work capacity and energy-loading capability in the mid-blade region. Analysis of the velocity field reveals that, when the blade number is small, the tandem impeller can effectively suppress the expansion of the low-velocity region at the blade inlet and improve inlet flow matching. As the blade number increases, the outlet velocity distribution becomes more uniform, with denser and more aligned streamlines, indicating a substantially enhanced flow-organizing ability. The discharge flow angle of the tandem impeller β_{22} increases by approximately 5° - 10° , leading to a higher $c_{u,22}$, which is identified as the primary

mechanism responsible for the observed H improvement.

Comparisons of vortex characteristics show that the conventional impeller generally exhibits large-scale SSV and TEV on the blade SS, whereas the tandem impeller can partially weaken the intensity of the mid-span SSV and delay its downstream propagation. However, due to the presence of an additional gap-flow region between the tandem blades, local backflow and adverse pressure-gradient peaks occur in the gap, which enhances local flow instability. This phenomenon becomes more pronounced under overload conditions and increased blade numbers, leading to increased local torque and energy loss. Vortex transport analysis further indicates that the RVS constitutes the dominant mechanism governing vortex generation and evolution in the centrifugal pump. The strong stretching deformation in the gap region of the tandem blades, coupled with the Coriolis force, intensifies the complexity of the vortex structures, representing the primary source of flow instability and the associated increase in local energy dissipation. In summary, the tandem-blade impeller demonstrates a coexistence of enhanced energy transfer, improved flow organization, and increased vortex-structure complexity within the internal flow pattern.

In terms of excitation characteristics, notable differences are observed between the tandem impeller and the conventional impeller with respect to pressure fluctuation intensity, dominant frequency behavior, and unsteady load distribution. The results indicate that the pressure fluctuation intensity in the impeller increases gradually from the LE toward the TE along the blade span, and the fluctuation amplitude is highest under part-load conditions. Compared with the conventional impeller, the tandem impeller exhibits higher fluctuation amplitudes at both TE of the SS and PS, with the difference most pronounced under part-load conditions and low blade numbers. The unstable vortex structures in the gap region are identified as the primary cause of this enhancement. As the blade number increases, the pressure fluctuation intensity at various monitoring points in the tandem impeller decreases overall and becomes more stable. The pressure fluctuation amplitude in the volute is mainly influenced by the transmission of pulsations from the impeller outlet. The tandem impeller shows slightly higher pulsation intensity on the small cross-sectional side of the volute, especially when the blade number is small. Despite the increased fluctuation amplitude, the dominant frequencies of both impeller configurations in the volute remain the f_{BPF} and its harmonics, indicating that the tandem-blade configuration does not alter the inherent pressure pulsation periodicity of the system. Under off-design conditions, particularly under part-load conditions, the dominant frequency of pressure pulsation in the impeller shifts toward low-order shaft-related frequencies, and the low-frequency components become more pronounced. The tandem impeller shows larger low-frequency amplitudes and stronger temporal fluctuation characteristics. This suggests that the unsteady evolution of the gap flow intensifies the low-frequency excitation behavior of the system, which may lead to a reduction in

operational stability.

Under different operating conditions, the F_A of both impellers exhibits regular fluctuations with changes in flow rate and blade number. Under part-load conditions, the peak-to-peak value of F_A for the tandem impeller increases significantly, reaching approximately 35N for the 5-blade configuration, which is about 1.5 times that of the conventional impeller. Frequency-domain analysis shows that the dominant frequency of F_A corresponds to the f_{BPF} and its harmonics, while the low-frequency components are enhanced under part-load conditions, indicating increased periodic disturbances induced by unsteady flow patterns. Time-frequency analysis further reveals that, under part-load conditions, the f_{BPF} component of the tandem impeller is weakened, the low-frequency amplitude is strengthened, and multi-frequency interference occurs, demonstrating a noticeable decrease in internal flow stability. For F_R , the fluctuations of the tandem impeller are slightly lower than those of the conventional impeller under overload conditions. However, under part-load conditions, the tandem impeller exhibits significantly higher F_R , with the maximum difference reaching approximately 12N. Radial force vector analysis shows that, under the design condition, the radial force vector distribution of the tandem impeller is more concentrated and closer to the origin, indicating better force balance. In contrast, under part-load conditions, the vector points become more dispersed and cover a wider area, indicating enhanced asymmetry in the internal flow pattern. In summary, the tandem impeller can maintain pressure fluctuation and load stability comparable to the conventional impeller under design and overload conditions. However, under part-load and low blade number conditions, the unstable vortex structures and energy coupling effects in the gap significantly intensify pressure pulsations and low-frequency excitations, leading to increased fluctuations in both F_A and F_R . This suggests that, although the tandem-blade impeller offers advantages in energy conversion, it still presents potential stability risks under off-design conditions.

8.2 Application Prospects and Feasibility Analysis

The tandem impeller configuration achieves graded energy loading within limited axial and radial dimensions, enabling higher head and improved energy conversion efficiency for centrifugal pumps across a wide range of operating conditions. The numerical and experimental results in this study demonstrate that the tandem structure can effectively enhance the velocity uniformity and pressure distribution at the impeller outlet under multiple operating conditions, thereby improving the pump performance. On this basis, the tandem impeller shows promising engineering application potential in fields such as energy equipment, chemical fluid transport, and circulation cooling systems. However, its practical implementation still requires comprehensive consideration of hydraulic performance, structural complexity, and manufacturing feasibility.

From the perspective of hydraulic performance, the tandem impeller enhances the work capacity and total pressure output by introducing a staged energy-loading mechanism

between the front and rear blades within a single impeller. This results in a significant performance improvement. For pump units requiring high H , large Q , or constrained installation space, the tandem impeller can serve as an alternative to conventional multistage pump configurations, thereby reducing system complexity and minimizing energy losses. More importantly, under the same H design requirement, the tandem impeller can substantially reduce the impeller outer diameter while maintaining or even improving performance. This leads to reduced material consumption, decreased rotor weight, shortened axial installation length, and a more compact overall pump structure. Such advantages not only reduce manufacturing and transportation costs but also provide a feasible approach for achieving miniaturization and lightweight design in high power-density pump systems. Therefore, in applications such as circulating cooling pumps in thermal power plants, petrochemical transport pumps, and primary coolant pumps in nuclear power systems, the tandem impeller can enhance operational stability and energy efficiency while ensuring compactness. Additionally, for renewable energy devices, such as pumped-storage units and ocean energy hydraulic conversion systems, the tandem impeller can expand the stable operating range by increasing the achievable H , thereby improving load adaptability.

From the perspective of unsteady characteristics, the pressure fluctuation amplitude of the tandem impeller under the design condition is comparable to that of the conventional impeller, and the periodic behavior remains stable. This indicates that the tandem configuration maintains good hydraulic stability under normal operating conditions. However, under off-design conditions, local vortices, backflow and low-frequency pulsations induced by gap flow lead to intensified pressure and F_A fluctuations at the impeller outlet, resulting in potential excitation risks. Therefore, the feasibility of applying the tandem impeller depends on the effective control of the gap-region flow. By appropriately reducing the gap length and optimizing the inlet angle and thickness distribution of the rear blade, the unsteady coupling effects can be significantly weakened, thereby achieving stable operation. In addition, proper matching of fundamental impeller design parameters, such as blade number and WA, is also essential. The results of this study indicate that, prior to designing a tandem impeller, a high-performance conventional impeller should be developed as the baseline configuration, ensuring that the tandem impeller can maintain both high performance and stable operation in practical engineering applications.

From a manufacturing perspective, the complex geometry of the tandem impeller imposes higher requirements on casting and machining, particularly in the blade-connection regions and the gap edges, where forming accuracy has a significant impact on performance. During fabrication, five-axis CNC machining, metal additive manufacturing, or hybrid processes combining integral casting with precision machining may be employed to ensure blade-surface accuracy and geometric consistency. For

applications sensitive to corrosion and wear, high-durability surface coatings, such as HVOF-sprayed WC-Co can be applied to the gap region and the blade SS to improve long-term reliability. In terms of maintenance, the tandem impeller introduces greater difficulty in flow-passage cleaning compared to conventional designs. However, maintenance costs can be effectively reduced through modular assembly and segmented replacement of wear-prone components.

From an energy-efficiency standpoint, the tandem impeller can achieve a higher head output per unit flow rate, thereby reducing the specific energy consumption of the pump unit. When applied in multi-pump parallel systems, it can decrease the operating load of individual pumps and improve part-load η . Although its manufacturing cost is slightly higher than that of the conventional impeller, the long-term energy-saving benefits and extended maintenance intervals provide an overall advantage in the pump system life-cycle cost. For large-scale installations, if the H improvement exceeds 5%, the additional manufacturing cost is expected to be recovered within approximately 2-3 years through energy savings.

In summary, considering hydraulic performance, structural feasibility, and operational stability, the tandem impeller configuration demonstrates notable performance enhancement and energy-efficiency potential under design and medium-to-over load conditions. However, targeted control strategies are still required for part-load conditions. By incorporating CFD-based optimization, experimental validation, and intelligent surrogate-model prediction during the design stage, an acceptable engineering balance between performance and stability can be achieved. Overall, the tandem impeller structure exhibits strong potential for practical deployment, particularly in pump systems requiring high H , limited installation space, or high energy-efficiency sensitivity. Moreover, it provides a new development pathway for achieving high η and lightweight design in conventional centrifugal pumps.

9 Conclusion and Outlook

This chapter provides a summary and outlook of the preceding research work. First, the major findings regarding the external characteristics, internal flow patterns, and excitation behavior of the tandem impeller centrifugal pump are systematically reviewed, and the key principles and innovative conclusions are distilled. Second, based on the research results and practical engineering requirements, the future development directions of tandem impeller technology and its application prospects under different conditions and pump types are discussed.

9.1 Main Research Conclusion

This study investigates the mechanism of hydraulic performance enhancement in centrifugal pumps equipped with tandem impellers. By integrating numerical simulations, experimental analysis, and theoretical modeling, the effects of the tandem impeller configuration on the external characteristics, internal flow patterns, and unsteady excitation forces of the pump are systematically examined. First, the design methodology of key hydraulic components in centrifugal pumps, particularly the tandem impeller configuration is introduced. The construction of the experimental system and the corresponding measurement procedures are then presented. Next, a high-fidelity three-dimensional unsteady flow simulation model is established based on CFD. Subsequently, the performance differences between the conventional and tandem impellers with various blade numbers and geometric parameters are analyzed through a combined evaluation of EXP and CFD results. Furthermore, the mechanisms of energy conversion and flow evolution of the tandem impeller are elucidated in depth from the perspectives of total pressure ratio distribution, relative velocity fields, potential rothalpy distribution, and vortex structures. Finally, the unsteady excitation characteristics, including pressure pulsations, F_A and F_R are compared for the two impeller configurations, thereby clarifying the stability behavior and excitation response of the tandem impeller under different blade number configurations and operating conditions. The main work and conclusions of this research are summarized in detail below.

Both the EXP and CFD of external characteristics indicate that the tandem impeller can significantly increase the pump H while maintaining the η essentially unchanged. The average H improvement reaches approximately 1-2m, corresponding to a relative increase of 5-8%, with the most pronounced enhancement observed under overload conditions. Through systematic optimization of the conventional impeller parameters, the configuration with 6 blades and $WA=120^\circ$ was identified as achieving the optimal balance among H , η , and P , and was therefore selected as the baseline for tandem impeller design. In the design of the tandem impeller, adjusting the β_{21} and selecting an appropriate D are critical. Further analysis showed that when the offset is set to $D=0.13$ and β_{21} is

moderately reduced to 10.8° , the energy transfer between the front and rear blades becomes more continuous, the Q - H curve remains smooth, and fluctuations in η are controlled within $\pm 1\%$ relative to the conventional impeller. Comparisons across different blade numbers indicate that the tandem impeller consistently provides higher H and more stable η than the conventional impeller, and the increase in H becomes more prominent as blade number increases. Among the tested configurations, the 6-blade tandem impeller demonstrates the best overall performance. Overall, by optimizing energy-loading distribution and improving flow uniformity, the tandem impeller achieves coordinated enhancement of H increase, η stability, and energy-consumption balance, demonstrating strong performance improvement potential and valuable engineering applicability.

The tandem impeller markedly modifies the energy transfer mechanism within the impeller flow passages. A wider pressure gradient region is formed near the mid-span and TE sections of the blades, while the gap region provides a secondary stage of energy loading, resulting in a more continuous energy transfer process. This staged working mechanism effectively enhances the local energy conversion capability of the impeller. Compared with the conventional impeller, the tandem impeller exhibits a more uniform velocity field and a larger β_{22} (increased by approximately 5° - 10°) under medium-to-over load conditions, leading to strengthened $c_{u,22}$ and improved energy output efficiency. This improvement enhances the momentum distribution at the impeller outlet and reduces flow field asymmetry, which constitutes the fundamental reason for the observed H enhancement in the tandem impeller.

Analysis of vortex characteristics indicates that the tandem impeller weakens the intensity of the SSV and suppresses part of the flow separation, but local backflow and adverse pressure gradient peaks occur in the gap region between the front and rear blades. These unstable flow structures induce locally increased torque and energy loss, thereby partially offsetting the positive effect of H enhancement on η . Vortex transport analysis further reveals that the RVS dominate the generation and development of vortex in the tandem impeller, while the CORF in the gap region intensifies vortex complexity. As the blade number increases, these effects become more pronounced, indicating a trend of strengthened unsteady characteristics for the tandem configuration under overload conditions.

Regarding hydraulic excitation characteristics, pressure fluctuations in the impeller are mainly concentrated near the blade TE and the impeller outlet, with the strongest pulsations occurring under part-load conditions. The tandem impeller exhibits generally higher-pressure fluctuation amplitudes at both the TE of SS and PS compared with the conventional impeller, with the differences being most pronounced under low blade numbers and part-load conditions. The vortex evolution in the gap region is identified as the primary cause of the enhanced pressure pulsations. As the blade number increases, the overall pressure-fluctuation intensity of the tandem impeller tends to decrease. In the

volute, the pressure fluctuation intensity is mainly influenced by the transmission of the pulsation originating from the impeller outlet. The tandem impeller shows slightly higher pulsations on the small cross-section side of the volute, although the frequency-domain characteristics remain unchanged. Under off-design conditions, low-order f_n components and their harmonics are amplified in the volute, and the tandem impeller exhibits larger low-frequency amplitudes, reflecting the strengthened unsteady effects induced by the gap flow. In terms of F_A , the peak-to-peak F_A of the tandem impeller increases significantly under part-load conditions, reaching up to approximately 1.5 times that of the conventional impeller. This behavior is mainly attributed to the instability induced by gap flow evolution. Both frequency-domain and time-frequency analyses indicate that under part-load conditions, the f_{BPF} of the F_A weakens, while low-frequency components are strengthened and multi-frequency interference appears, demonstrating a more complex unsteady F_A behavior. For F_R , the tandem impeller exhibits better force balance under design and overload conditions. However, under part-load conditions, the F_R fluctuations of the tandem impeller increase significantly, exceeding those of the conventional impeller by approximately 10-12N, and the radial force vector distribution becomes more dispersed. This again indicates that flow non-uniformity and unbalanced loading are more pronounced in the tandem impeller under part-load conditions. Therefore, under part-load conditions and higher blade numbers, the unsteady evolution of gap flow in the tandem impeller intensifies low-frequency excitation and increases the fluctuation amplitudes of both F_A and F_R , which may lead to potential operational stability issues.

9.2 Outlook

In this study, a centrifugal pump was investigated to examine the effects of the tandem-blade impeller on its performance and flow characteristics. The comparative analysis primarily focused on the performance, flow instability features, and excitation behavior of tandem impellers and conventional impellers with different blade numbers. However, due to limitations in research capacity, available time, and experimental resources, there are still certain shortcomings in the depth of analysis, methodological refinement, and experimental validation. The author believes that further improvements remain possible in this research field. Therefore, this section outlines several prospects and recommendations for future research efforts.

Building upon the findings of this study, future work may further explore the feasibility of a triple-blade configuration, with the aim of achieving sufficient energy exchange and higher H within limited spatial dimensions. In addition, the application potential of tandem-blade impellers in multi-stage centrifugal pump systems may also be investigated. Both configurations have the potential to enhance energy-transfer efficiency, improve interstage flow matching, and mitigate unsteady interference effects while maintaining a compact overall pump structure, thereby further improving the comprehensive pump

performance. However, attention should also be paid to potential challenges associated with multi-stage tandem arrangements, such as phase coupling, cumulative pressure pulsations, and possible reductions in operational stability. Future studies may integrate high-fidelity CFD simulations with flow visualization experiments to systematically clarify the interaction mechanisms of tandem blades in multi-stage flow patterns, thereby providing new design strategies for high-performance, high-head multi-stage pumps.

This study has verified the feasibility of the tandem-blade impeller in centrifugal pumps with low-medium n_q , while Fensterseifer [81] have applied the tandem-blade configuration to centrifugal pumps with medium-high n_q . In the future, it is necessary to extend the tandem-impeller design concept to centrifugal pumps with even higher and lower n_q , to further investigate its adaptability and performance variation under different flow characteristics. In addition, based on the application characteristics of tandem-blade impellers at various n_q levels, a rapid design methodology applicable to different pump types can be summarized and integrated into design software, enabling efficient parametric modeling of tandem-impeller structures. Furthermore, the applicability of the tandem impeller may be assessed under conditions involving gas-liquid mixtures, solid particle suspensions, or high-viscosity fluids, to demonstrate its performance stability and reliability in multi-phase pumps, liquefied-gas pumps, and chemical process pumps.

In the study of unsteady flow mechanisms, future work may incorporate the rigid vorticity transport equation, as well as POD and DMD modal decomposition methods, to conduct an in-depth analysis of vortex evolution in the gap region and at the blade TE, and to identify the dominant sources of pressure pulsation. This will help clarify the transformation pathway from flow instability to force response. Moreover, by systematically investigating the effects of geometric parameters such as gap width, installation phase, and rear-blade inlet angle, a multi-scale correlation model linking “structure-flow-excitation force” can be established, providing a theoretical basis for the design of tandem impellers with enhanced hydraulic stability.

To further verify the reliability of the numerical simulation results, future studies may employ flow pattern visualization techniques such as transient particle image velocimetry (TR-PIV), high-frequency dynamic pressure measurements, and Laser Doppler Velocimetry (LDV). These methods can be used to reconstruct experimental streamlines and conduct spatiotemporal correlation analysis in the impeller outlet and gap regions, thereby validating the numerically predicted unsteady flow structures.

Further investigation is required regarding the performance and stability of the tandem impeller under gas-entrainment and cavitation conditions. Previous studies have indicated that gas ingestion can significantly affect pump operating performance [119-121], while Furukawa et al. [122] suggests that the tandem-blade configuration may help mitigate such issues. In future research, the design framework established in this study may be combined with compressible multiphase-flow and cavitation models to systematically

evaluate the flow behavior and anti-cavitation capability of the tandem impeller under varying gas volume fractions and inlet pressure conditions. This will provide theoretical guidance and design references for the development of pumps with enhanced cavitation resistance and reliability.

References

- [1] Yang H, Jia H, Zhu Z C, et al. Review of the Hydraulic and Structural Design of High-Speed Centrifugal Pumps[J]. *Frontiers in Energy Research*, 2022, 10: 899093.
- [2] Zhao W Y, Ge J G, Ma D, et al. Vibration Analysis of Large Centrifugal Pump Rotors[C]. *IOP Conference Series: Materials Science and Engineering*, 2013.
- [3] Gao Z X, Zhu W R, Lu L, et al. Numerical and Experimental Study of Unsteady Flow in a Large Centrifugal Pump with Stay Vanes[J]. *Journal of Fluids Engineering*, 2014, 136(7): 1-10.
- [4] Bellary SAI and Samad A. Pumping Crude Oil by Centrifugal Impeller Having Different Blade Angles and Surface Roughness[J]. *Journal of Petroleum Exploration and Production Technology*, 2016, 6(1): 117-127.
- [5] Zhou J B, He D J, Zhang R, et al. Research on the Performance of a Centrifugal Aviation Fuel Pump Based on Response Surface Methodology[J]. *Processes*, 2023, 11(11): 3055.
- [6] Cavazzini G, Houdeline J B, Pavesi G, et al. Unstable Behaviour of Pump-Turbines and Its Effects on Power Regulation Capacity of Pumped-Hydro Energy Storage Plants[J]. *Renewable and Sustainable Energy Reviews*, 2018, 94: 399-409.
- [7] Ren Y, Zhu Z C, Wu D H, et al. Investigation of Flow Separation in a Centrifugal Pump Impeller based on Improved Delayed Detached Eddy Simulation Method[J]. *Advances in Mechanical Engineering*, 2019, 11(12): 1687814019897832.
- [8] Liu B Q, Yang W, Cheng L, et al. Numerical Study on Jet-wake Flow and Its Evolution in a Centrifugal Pump with Alternating Stall[J]. *Physics of Fluids*, 2023, 35(11): 115105.
- [9] Lv W L, Zhang Y, Zhang W B, et al. Experimental Research on Pressure Pulsation and Flow Structures of the Low Specific Speed Centrifugal Pump[J]. *Energies*, 2024, 17(18): 4730.
- [10] Zhao X R, Xiao Y X, Wang Z W, et al. Unsteady Flow and Pressure Pulsation Characteristics Analysis of Rotating Stall in Centrifugal Pumps Under Off-Design Conditions[J]. *Journal of Fluids Engineering*, 2018, 140(2): 021105.
- [11] Wu C S, Yang J, Yang S, et al. A Review of Fluid-Induced Excitations in Centrifugal Pumps[J]. *Mathematics*, 2023, 11(4): 1026.

- [12] Zhou J L, Jin G Y, Ye T G, et al. Fluid-induced Vibration Analysis of Centrifugal Pump Including Rotor System Based on Computational Fluid Dynamics and Computational Structural Dynamics Coupling Approach[J]. *Ocean Engineering*, 2023, 288(1): 115993.
- [13] Shim H S and Kim K Y. Design Optimization of the Impeller and Volute of a Centrifugal Pump to Improve the Hydraulic Performance and Flow Stability[J]. *Journal of Fluids Engineering*, 2020, 142(10): 101211.
- [14] Kim B, Siddique M H, Samad A, et al. Optimization of Centrifugal Pump Impeller for Pumping Viscous Fluids Using Direct Design Optimization Technique[J]. *Machines*, 2022, 10(9): 774.
- [15] Deshmukh P A, Deshmukh K D, Mandhare N A, et al. Performance Enhancement of Centrifugal Pump by Minimizing Casing Losses Using Coating[J]. *SN Applied Sciences*, 2020, 2(2): 252.
- [16] Yang G, Zhao X T, Zhang D S, et al. Hydraulic Components' Matching Optimization Design and Entropy Production Analysis in a Large Vertical Centrifugal Pump[J]. *Journal of Mechanical Science and Technology*, 2021, 35(11): 5033-5048.
- [17] Li Z L, Han G, Lu X E, et al. Improving the Operating Range Using a Centrifugal Compressor with a Tandem Impeller[J]. *Aerospace Science and Technology*, 2020, 96: 105548.
- [18] Roberts D A, Kacker S C, et al. Numerical Investigation of Tandem-Impeller Designs for a Gas Turbine Compressor[J]. *ASME Journal of Turbomachinery*, 2002, 124(1): 36-44.
- [19] Li C, Gao B, Zhang N, et al. Effects of Splitter Vanes on the Performance and Pressure Pulsation in a Centrifugal Pump[C]. *ASME JSME KSME Joint Fluids Engineering Conference Proceedings*, 2019: V03BT03A012.
- [20] Gülich J F. *Centrifugal Pumps*[M]. 4th ed. Springer, 2020.
- [21] Asuaje M, Bakir F, Kouidri S, et al. Inverse Design Method for Centrifugal Impellers and Comparison With Numerical Simulation Tools[J]. *International Journal of Computational Fluid Dynamics*, 2004, 18(2): 101-110.
- [22] Kim J H, Kim K Y. Analysis and Optimization of a Vaned diffuser in a Mixed Flow Pump to Improved Hydrodynamic Performance[J]. *Journal of Fluids Engineering*, 2012, 134(7): 071104.

-
- [23] Parikh T, Mansour M and Thévenin D. Maximizing the Performance of Pump Inducers Using CFD-Based Multi-Objective Optimization[J]. *Structural and Multidisciplinary Optimization*, 2022, 65: 9.
- [24] Wang W J, Sun J, Liu J, et al. Introducing Non-Hierarchical RSM and MIGA for Performance Prediction and Optimization of a Centrifugal Pump under the Nominal Condition[J]. *Processes*, 2022, 10(8): 1529.
- [25] Aliuly A, Amanzholov T, Seitov A, et al. Hydraulic Design and CFD-Based Parametric Study for Optimizing Centrifugal Pump Impeller Performance[J]. *Applied Sciences*, 2024, 14(22): 10161.
- [26] Elyamin G R H A, Bassily M A, Khalil K Y, et al. Effect of Impeller Blades Number on the Performance of a Centrifugal Pump[J]. *Alexandria Engineering Journal*, 2019, 58(1): 39-48.
- [27] Ding H C, Li Z K, Gong X B, et al. The Influence of Blade Outlet Angle on the Performance of Centrifugal Pump with High Specific Speed[J]. *Vacuum*, 2019, 159: 239-246.
- [28] Tan L, Zhu B S, Cao S L, et al. Influence of Blade Wrap Angle on Centrifugal Pump Performance by Numerical and Experimental Study[J]. *Chinese Journal of Mechanical Engineering*, 2014, 27: 171-177.
- [29] Jin J M, Jung K J, Kim Y H, et al. Effects of Gap Between Impeller and Volute Tongue on a Pressure Fluctuation in Double-Suction Pump[J]. *Journal of Mechanical Science and Technology*, 2020, 34: 4897-4903.
- [30] Zhou R, Yang J, Liu H L, et al. Effect of Volute Geometry on Radial Force Characteristics of Centrifugal Pump during Startup[J]. *Journal of Applied Fluid Mechanics*, 2022, 15(1): 25-36.
- [31] Carpenter W C. Effect of Design Selection on Response Surface Performance[J]. 1993.
- [32] Li Z, Ding H, Shen X, et al. Performance Optimization of High Specific Speed Centrifugal Pump Based on Orthogonal Experiment Design Method[J]. *Processes*, 2019, 7(10): 728.
- [33] Catur H, Yun J E, Shin J Y, et al. Optimal Design of a Liquid Hydrogen Centrifugal Pump Impeller[J]. *Energies*, 2024, 17(24): 6299.
- [34] Nishi Y, Konno K and Ono S. Multi-objective Optimization Design Method for Single-Blade Centrifugal Pump Impellers to Improve Performance and Reduce Radial Thrust[J]. *European Journal of Mechanics-B/Fluids*, 2025, 114: 204311.

- [35] Bezerra M, Santelli R, Oliveira E, et al. Response Surface Methodology (RSM) as a Tool for Optimization in Analytical Chemistry[J]. *Talanta*, 2008, 76(5): 965-977.
- [36] Kleijnen J. Kriging Metamodeling in Simulation: A Review[J]. *Journal of Operational Research*, 2009, 192(3): 707-716.
- [37] Heo M, Ma S, Shim H, et al. High-efficiency Design Optimization of a Centrifugal Pump[J]. *Journal of Mechanical Science and Technology*, 2016, 30: 3917-3927.
- [38] Aziz M, Tayarani-N. An Adaptive Memetic Particle Swarm Optimization Algorithm for Finding Large-Scale Latin Hypercube Designs[J]. *Engineering Applications of Artificial Intelligence*, 2014, 36: 222-237.
- [39] Chen W, Ma W S, Wang L, et al. Optimization of Centrifugal Pump Impeller Based on Neural Network and NSGA-II Algorithm[J]. *Journal of Physics: Conference Series*, 2024, 2752: 012120.
- [40] Lu J X, Zhou Y Z, Ge Y L, et al. Research into Prediction Method for Pressure Pulsations in a Centrifugal Pump Based on Variational Mode Decomposition-Particle Swarm Optimization and Hybrid Deep Learning Models[J]. *Sensors (Basel, Switzerland)*, 2024, 24(13): 4196.
- [41] Wang L, Asomani S, Yuan J, et al. Geometrical Optimization of Pump-As-Turbine (PAT) Impellers for Enhancing Energy Efficiency with 1-D Theory[J]. *Energies*, 2020, 13(16): 4120.
- [42] Pei J, et al. Multiparameter Optimization for the Nonlinear Performance Improvement of Centrifugal Pumps using a Multilayer Neural Network[J]. *Journal of Mechanical Science and Technology*, 2019, 33(6): 2681-2691.
- [43] Shi L, Zhu J, Tang F, et al. Multi-Disciplinary Optimization Design of Axial-Flow Pump Impellers Based on the Approximation Model[J]. *Energies*, 2020, 13(4): 779.
- [44] Liu Y Y, Yang G, Xu Y, et al. Effect of Space Diffuser on Flow Characteristics of a Centrifugal Pump by Computational Fluid Dynamic Analysis[J]. *PloS one*, 2020, 15(2): e0228051.
- [45] Yuan Y, Yuan S Q and Tang L D. Numerical Investigation on the Mechanism of Double-Volute Balancing Radial Hydraulic Force on the Centrifugal Pump[J]. *Processes*, 2019, 7(10): 689.
- [46] Nguyen D A, Wu K, Li X J, et al. Adjustable Tongue Vane for Improving the Energy Performance of a Submersible Single-Channel Pump for Wastewater Treatment[J]. *Heliyon*, 2025, 11(1): e41511.

-
- [47] Wu D H, Ren Y, Mou J G, et al. Unsteady Flow and Structural Behaviors of Centrifugal Pump under Cavitation Conditions[J]. Chinese Journal of Mechanical Engineering, 2019, 32(1): 1-15.
- [48] Zhang H H, You H L, Lu H S, et al. CFD-Rotordynamics Sequential Coupling Simulation Approach for the Flow-Induced Vibration of Rotor System in Centrifugal Pump[J]. Applied Sciences. 2020, 10(3):1186.
- [49] Mito R, Yamashita S. Prediction of Rotating Stall During Startup for Axial Compressors[C]. Turbo Expo: Power for Land, Sea, and Air, 2019, Phoenix, Arizona, USA, p.V02AT39A028.
- [50] You D H, Mittal R, Wang M, et al. Computational Methodology for Large-eddy Simulation of Tip-clearance Flows[J]. Aiaa Journal, 2004, 42(2): 271-279.
- [51] You D H, Wang M, Moin P, et al. Study of Flow in Tip-clearance Flow in Turbomachines Using Large-eddy Simulation[J]. Computing in Science & Engineering, 2004, 6(6): 38-46.
- [52] Khalifa A E, Al-Qutub A M, Ben-Mansour R. Study of Pressure Fluctuations and Induced Vibration at Blade-Passing Frequencies of a Double Volute Pump[J]. Arabian Journal for Science and Engineering, 2011, 36(7): 1333-1345.
- [53] Fu X L, Li D Y, Lv J W, et al. High amplitude Pressure Pulsations Induced by Complex inter-blade Flow During Load Rejection of Ultrahigh-head Prototype Pump Turbines[J]. Physics of Fluids, 2024, 36(3): 034115.
- [54] Pedersen N, Larsen P S, Jacobsen C B. Flow in a Centrifugal Pump Impeller at Design and Off-Design Conditions-Part I: Particle Image Velocimetry (PIV) and Laser Doppler Velocimetry (LDV) Measurements[J]. Journal of Fluids Engineering, 2003, 125(1): 61-72.
- [55] Sinha M, Pinarbasi A, Katz J. The Flow Structure During Onset and Developed States of Rotating Stall Within a Vaned Diffuser of a Centrifugal Pump[J]. Journal of Fluids Engineering, 2001, 123(3): 490-499.
- [56] Xu P, Liu Z P, Lu L, et al. Experimental Research on the Rotating Stall of a Pump Turbine in Pump Mode[J]. Water, 2019, 11(11): 2426.
- [57] Zhang S W, Chen H X, Ma Z, et al. Unsteady Flow and Pressure Pulsation Characteristics in Centrifugal Pump Based on Dynamic Mode Decomposition Method[J]. Physics of Fluids, 2022, 34 (11): 112014.

- [58] Ye C L, Tang Y, An D S, et al. Investigation on Stall Characteristics of Marine Centrifugal Pump Considering Transition Effect[J]. *Ocean Engineering*, 2023, 280: 114823.
- [59] Zhang Y N, Wu Y L. A Review of Rotating Stall in Reversible Pump Turbine[J]. *Proceedings of the Institution of Mechanical Engineers, Part C: Journal of Mechanical Engineering Science*, 2017, 231(7): 1181-1204.
- [60] Gunnar Backman. CFD Validation of Pressure Fluctuations in a Pump Turbine[D]. Lulea University of Technology, Lulea, 2008.
- [61] Luo Z Y, Feng Y, Sun X Y, et al. Research on the Unsteady Flow and Vortex Characteristics of Cavitation at the Tongue in Centrifugal Pump[J]. *Journal of Applied Fluid Mechanics*, 2024, 17(3): 646-657.
- [62] Wang C Y, Zeng Y S, Yao Z F, et al. Rigid Vorticity Transport Equation and its Application to Vortical Structure Evolution Analysis in Hydro-Energy Machinery[J]. *Engineering Applications of Computational Fluid Mechanics*, 2021, 15(1): 1016-1033.
- [63] Liao Z Y, Yang J, Liu X H, et al. Analysis of Unsteady Flow Structures in a Centrifugal Impeller Using Proper Orthogonal Decomposition[J]. *Journal of Applied Fluid Mechanics*, 2021, 14(1): 89-101.
- [64] Lu J X, Ning C Z, Wu F, et al. Dynamic Mode Decomposition based Investigation of Unsteady Flow Characteristics and Pressure Pulsations in Centrifugal Pumps Operating under Partial Load Conditions for Scientific Advancement[J]. *Flow Measurement and Instrumentation*, 2025, 103: 102850.
- [65] Ji L L, Li S, Li W, et al. Study on Passive Suppression Method of Rotating Stall in Mixed-Flow Pump: Using Different Impeller Rim Structures[J]. *Part A: Journal of Power and Energy*, 2023, 237(5): 965-984.
- [66] Li D Y, Qin Y L, Wang J P, et al. Optimization of Blade High-Pressure Edge to Reduce Pressure Fluctuations in Pump-Turbine Hump Region[J]. *Renewable Energy*, 2022, 181: 24-38.
- [67] Saha S L, Kurokawa J, Matsui J, et al. Suppression of Performance Curve Instability of a Mixed Flow Pump by Use of J-Groove[J]. *Journal of Fluids Engineering*, 2000, 122(3): 592-597.
- [68] Saha S L, Kurokawa J, Matsui J, et al. Passive Control of Rotating Stall in a Parallel-Wall Vaned Diffuser by J-Grooves[J]. *Journal of Fluids Engineering*, 2000, 123(3): 507-515.

-
- [69] Kurokawa J, J-Groove Technique for Suppressing Various Anomalous Flow Phenomena in Turbomachines[J]. *International Journal of Fluid Machinery and Systems*, 2011, 4(1): 1-13.
- [70] Ding H C, Lin F Y, Chang T, et al. Numerical Study on the Effect of Blade Trailing Edge Filing on Performance and Unsteady Pressure Pulsation in Low Specific Speed Centrifugal Pump[J]. *Journal of Vibration Engineering & Technologies*, 2024, 12: 233-245.
- [71] Wu C S, Zhang W Q, Wu P, et al. Effects of Blade Pressure Side Modification on Unsteady Pressure Pulsation and Flow Structures in a Centrifugal Pump[J]. *Journal of Fluids Engineering*, 2021, 143(11): 111208.
- [72] Wu C S, Li Q Q, Zheng F, et al. Improve of Unsteady Pressure Pulsation Based on Jet-Wake Suppression for a Low Specific Centrifugal Pump[J]. *Journal of Fluids Engineering*, 2021, 143(11): 111202.
- [73] Zhang J F, Liang Y, Yuan J P, et al. Numerical Simulation of Recirculation Control at Centrifugal Pump Inlet[J]. *Journal of Jiangsu University*, 2012, 33(4): 402-407.
- [74] Bloxham M J, Bons J P. A Global Approach to Turbomachinery Flow Control: Passage Vortex Control[J]. *Journal of Turbomachinery*, 2014, 136(4): 041003.
- [75] McGlumphy J, Ng W F, Wellborn S R, et al. Numerical Investigation of Tandem Airfoils for Subsonic Axial-Flow Compressor Blades[J]. *Journal of Turbomachinery*, 2009, 131(2): 021018.
- [76] MAO X C, WANG Y Y, DING X H, et al. Design Strategy and Relevant Flow Mechanisms of Highly Loaded 3D Compressor Tandem Cascades[J]. *Chinese Journal of Aeronautics*, 2024, 37(7): 220-235.
- [77] Böhle M, Frey T. Numerical and Experimental Investigations of the Three-Dimensional-Flow Structure of Tandem Cascades in the Sidewall Region[J]. *ASME Journal of Fluids Engineering*, 2014, 136(7): 071102.
- [78] Yuan S L, Wu Y F, Zhao S F, et al. Investigation on the Aerodynamic Performance and Flow Mechanism of Transonic Ultra-Highly Loaded Tandem-Rotor Stage[J]. *Aerospace*, 2024, 11(5): 389.
- [79] Shi X B, Lu J L, Zhao L M, et al. Investigations on the Influence of Tandem Blades on Inner Flow and Performance Characteristics of Centrifugal Pump[J]. *Proceedings of the Institution of Mechanical Engineers, Part E: Journal of Process Mechanical Engineering*, 2019, 234(1): 46-55.

- [80] PRITHVIR D, NARAYANA R. Studies on a Centrifugal Pump with Tandem Vanes[J]. Mechanical Engineering, 1974, 96(8): 63.
- [81] Fensterseifer M, Sterle L, Böhle M. Numerical and Experimental Investigation of the Performance of Tandem Vanes for a Centrifugal Pump[J]. ASME Journal of Fluids Engineering, 2022, 144(11): 111203.
- [82] Nicola Y, Fensterseifer M, Böhle M. Numerical and Experimental Investigation of Streamlines for Tandem Impeller in a Centrifugal Pump[J]. ASME Journal of Fluids Engineering, 2023, 145(4): 041202.
- [83] Eckardt D. Instantaneous Measurements in the Jet-Wake Discharge Flow of a Centrifugal Compressor Impeller[J]. Journal of Engineering for Power, 1975, 97(3): 337-345.
- [84] Kean W C, Thong S L, Winoto S H. Numerical Study of Inlet and Impeller Flow Structures in Centrifugal Pump at Design and Off-design Points[J]. International Journal of Fluid Machinery and Systems, 2011, 4(1): 25-32.
- [85] Li Y, Li X J, Zhu Z C, et al. Investigation of Unsteady Flow in a Centrifugal Pump at Low Flow Rate[J]. Advances in Mechanical Engineering, 2016, 8(12): 1.
- [86] Beomjun K, Keuntae P, Haecheon C, et al. Flow Characteristics in a Volute-Type Centrifugal Pump Using Large Eddy Simulation[J]. International Journal of Heat and Fluid Flow, 2018, 72: 52-60.
- [87] Wang Y Q and Gui N. A Review of the Third-Generation Vortex Identification Method and its Applications[J]. Chinese Journal of Hydrodynamics, 2019, 34(4): 413-429.
- [88] Tong Z M and Shang B Y. Investigating Three-Dimensional Vortex Evolution in Centrifugal Pump Under Rotating Stall Conditions Using Tomographic Particle Image Velocimetry[J]. Physics of Fluids, 2024, 36: 067101.
- [89] Shrestha P, Nottage C, Yu Y, et al. Stretching and Shearing Contamination Analysis for Liutex and Other Vortex Identification Methods[J]. Advances in Aerodynamics, 2021, 3: 8.
- [90] Liu C, Wang Y, Yang Y, et al. New Omega Vortex Identification Method[J]. Science China Physics, Mechanics & Astronomy, 2016, 59(8): 62-70.
- [91] Liu C, Gao Y, Tian S, et al. Rortex-A New Vortex Vector Definition and Vorticity Tensor and Vector Decompositions[J]. Physics of Fluids, 2018, 30(3): 035103.
- [92] Dong X, Dong G, Liu C. Study on Vorticity Structures in Late Flow Transition[J]. Physics of Fluids, 2018, 30(10): 104108.

-
- [93] Dong X, Tian S, Liu C. Correlation Analysis on Volume Vorticity and Vortex in Late Boundary layer transition[J]. *Physics of Fluids*, 2018, 30(1): 014105.
- [94] Liu C, Gao Y, Dong X, et al. Third Generation of Vortex Identification Method: Omega and Liutex/Rortex Based Systems[J]. *Journal of Hydrodynamics*, 2019, 31, 205-223.
- [95] Qian B, Cai Y H, Ding Q, et al. Investigation of Tip Leakage Vortex Structure and Trajectory in a Centrifugal Pump with a New Omega Vortex Identification Method[J]. *Applied Sciences*, 2022, 12(10): 5270.
- [96] Yang G, Shen X, Shi L, et al. Numerical Investigation of Hump Characteristic Improvement in a Large Vertical Centrifugal Pump with Special Emphasis on Energy Loss Mechanism[J]. *Energy*, 2023, 273: 127163.
- [97] Wu J Z, Ma H Y, and Zhou M D. *Vorticity and Vortex Dynamics*[M]. Springer-Verlag, 2006.
- [98] Meng Q, Shen X, Zhao X, et al. Numerical Investigation on Cavitation Vortex Dynamics of a Centrifugal Pump Based on Vorticity Transport Method[J]. *Journal of Marine Science and Engineering*, 2023, 11(7): 1424.
- [99] Guan X F. *Modern Pumps Theory and Design*[M]. China Astronautics Publishing House, Beijing, 2011.
- [100] Khaing C C, Lynn A Z and Nyi N. Design of Multistage Centrifugal Pump Impeller for High Head Applications[J]. *International Journal of Latest Technology in Engineering, Management & Applied Science*, 2019, 8(8): 115-120.
- [101] ASME PTC 19.1-2019 Test Uncertainty[S]. *Transactions of the ASME*, 2019.
- [102] Shen X, Zhang D S, Xu B, et al. Experimental Investigation of the Transient Patterns and Pressure Evolution of Tip Leakage Vortex and Induced-Vortices Cavitation in an Axial Flow Pump[J]. *Journal of Fluids Engineering*, 2020, 142(10): 101206.
- [103] Gourdain N, Sicot F, Duchaine F, et al. Large Eddy Simulation of Flows in Industrial Compressors: A Path From 2015 to 2035[J]. *Philosophical Transactions of The Royal Society A-Mathematical Physical and Engineering Sciences*, 2014, 372(2022): 20130323.
- [104] Parviz M and Krishnan M. DIRECT NUMERICAL SIMULATION: A Tool in Turbulence Research[J]. *Annual Review of Fluid Mechanics*, 1998, 30: 539-578.

- [105] Guo X, Zhu L, Zhu Z, et al. Numerical and Experimental Investigations on the Cavitation Characteristics of a High-speed Centrifugal Pump with a Splitter-blade Inducer[J]. *Journal of Mechanical Science and Technology*, 2015, 29(1): 259-267.
- [106] Tao R, Xiao R F and Wang F J, et al. Detached Eddy Simulations for High Speed Axial Flow Fuel Pumps with Swept and Straight Blade Impellers[J]. *Engineering Computations*, 2016, 33(8): 2530-2545.
- [107] Yang G, Shen X, Zhang D S, et al. Numerical Investigations on Performance Improvement Mechanism of a High-Power Vertical Centrifugal Pump with Special Emphasis on Hydraulic Component Matching[J]. *Journal of Hydrodynamics*, 2023, 35(4): 649-667.
- [108] Wilcox D C. Reassessment of the scale-determining equation for advanced turbulence models[J]. *AIAA Journal*, 1988, 26(11): 1299-1310.
- [109] Menter F R. Two-equation Eddy-viscosity Turbulence Models for Engineering Applications[J]. *Aiaa Journal*, 1994, 32(8): 1598-1605.
- [110] Shi S, Tang W, Huang X. Wake Flow Characteristics and Unsteady Performance of A Pumpjet Propulsor Under Hull Condition[J]. *Physics of Fluids*, 2022, 34(12).
- [111] Spalart P R. Strategies for Turbulence Modelling and Simulations[J]. *International Journal of Heat and Fluid Flow*, 2000, 21(3): 252-263.
- [112] Celik I B, Ghia U, Roache P J, et al. Procedure for Estimation and Reporting of Uncertainty due to Discretization in CFD Applications[J]. *ASME Journal of Fluids Engineering*, 2008, 130(7).
- [113] Stern F, Wilson R V, Coleman H W, et al. Comprehensive Approach to Verification and Validation of CFD Simulations-part 1: Methodology and Procedures[J]. *ASME Journal of Fluids Engineering*, 2001, 123(4): 793-802.
- [114] Li H, Chen Y, Bai L, et al. Assessing Energy Loss and Entropy Production in a Centrifugal Pump with Various Impeller Blade Trailing Edges[J]. *Journal of Applied Fluid Mechanics*, 2025, 18(2): 518-534.
- [115] Wang C Y, Wang F J, An D S, et al. A General Alternate Loading Technique and its Applications in the Inverse Designs of Centrifugal and Mixed-Flow Pump Impellers[J]. *SCIENCE CHINA Technological Sciences*, 2021, 64(4): 898-918.
- [116] Wang C Y, Wang F J, Wang H, et al. Computation Method and Control Strategy of Rotating Separation Flows in Hydraulic Machinery[J]. *Journal of Hydrodynamics*, 2022, 34(6): 1006-1020.

-
- [117] Torrence C, Compo G P. A Practical Guide to Wavelet Analysis[J]. Bulletin of the American Meteorological Society, 1998, 79(1): 61-78.
- [118] Ji L, Li W, Shi W, et al. Experimental Study of Pressure Pulsation in a Mixed-Flow Pump with Different Tip Clearances Based on Wavelet Analysis[J]. Shock and Vibration, 2020(9): 1-15.
- [119] Neumann M, Schäfer T, Bieberle A, et al. An Experimental Study on the Gas Entrainment in Horizontally and Vertically Installed Centrifugal Pumps[J]. ASME Journal of Fluids Engineering, 2016, 138(9): 091301.
- [120] Si Q R, Bois G, Liao M Q, et al. A Comparative Study on Centrifugal Pump Designs and Two-Phase Flow Characteristic under Inlet Gas Entrainment Conditions[J]. Energies, 2020, 13(1): 65.
- [121] Mansour M, Thévenin D. State of the Art on Two-Phase Non-Miscible Liquid/Gas Flow Transport Analysis in Radial Centrifugal Pumps Part B: Review of Experimental Investigations[J]. International Journal of Turbomachinery, Propulsion and Power, 2023, 8(4): 42.
- [122] Furukawa A, Ohshita S, Okuma K, et al. Development of Air/Water Two-Phase Flow Centrifugal Pump and Its Operating Characteristics[C]. Proceedings of the ASME/JSME 2003 4th Joint Fluids Summer Engineering Conference, 2003. 2: 1413-1418.

Publications and Conferences

Gang Yang, Xutao Zhao, Desheng Zhang, etc. Hydraulic components' matching optimization design and entropy production analysis in a large vertical centrifugal pump. *Journal of Mechanical Science and Technology*, 2021, 35 (11): 5033-5048.

Gang Yang, Desheng Zhang, Xueqi Yang, etc. Study on the flow pattern and pressure fluctuation in a vertical volute centrifugal pump with vaned diffuser under near stall conditions. *Journal of the Brazilian Society of Mechanical Sciences and Engineering*, 2022, 44 (4): 118.

Gang Yang, Xutao Zhao, Desheng Zhang, etc. Optimization design and unsteady flow analysis of stall characteristics in a vertical centrifugal pump. *Journal of Physics: Conference Series*, 2022, 2217: 012045.

Zhang D S, **Yang G**, Zhao X T, etc. Optimization Design of Vane Diffuser and Volute in Vertical Centrifugal Pump Based on Back Propagation Neural Network. *Transactions of the Chinese Society for Agricultural Machinery*, 2022, 53(4): 130-139.

Gang Yang, Xi Shen, Lei Shi, etc. Numerical investigation of hump characteristic improvement in a large vertical centrifugal pump with special emphasis on energy loss mechanism. *Energy*, 2023, 273: 127163.

Gang Yang, Xi Shen, Desheng Zhang, etc. Numerical investigations on performance improvement mechanism of a high-power vertical centrifugal pump with special emphasis on hydraulic component matching. *Journal of Hydrodynamics*, 2023, 35: 649-667.

Gang Yang, Xi Shen, Lei Shi, etc. Unsteady numerical investigations of the effect of guide vane openings on the hydrodynamic characteristics under stall conditions in a pump-turbine pump mode. *Energy Conversion and Management*, 2023, 293: 117499.

Gang Yang, Xi Shen, Bin Xu, etc. Numerical study on unsteady cavitation flow and vortex dynamics characteristics around the Delft Twist 11 hydrofoil. *Ocean Engineering*, 2023, 286: 115580.

Gang Yang, Xi Shen, Desheng Zhang, etc. Numerical investigation on vortex dynamics characteristics of a large vertical centrifugal pump operating in the hump region under cavitation conditions. *Journal of Physics: Conference Series*, 2024, 2707: 012020.

Gang Yang, Xi Shen, Qiang Pan, etc. Investigation on passive suppression method of hump characteristics in a large vertical volute centrifugal pump: Using combined diffuser vane structure. *Energy*, 2024, 304: 132067.

Gang Yang, Desheng Zhang, Xi Shen, etc. Investigation on flow instability in the hump region of the large vertical centrifugal pump under cavitation conditions based on proper orthogonal decomposition. *Physics of Fluids*, 2024, 36: 115134.

Gang Yang, Desheng Zhang, Hong Gao, etc. Transient flow characteristics and energy loss investigation in a desalination energy recovery device under rotor system axial sliding conditions: Focusing on the turbine side. *Physics of Fluids*, 2025, 37: 015196.

Gang Yang, Desheng Zhang, Qingzhao Pang, etc. Research on vortex characteristics and stall mechanism during runaway process of the large vertical centrifugal pump under power failure. *Physics of Fluids*, 2025, 37: 074123.

Gang Yang, Xi Shen, Qiang Pan, etc. Multi-objective optimization design for broadening the high efficiency region of hydrodynamic turbine with forward-curved impeller and energy loss analysis. *Renewable Energy*, 2025, 245: 122818.

Gang Yang, Desheng Zhang, Martin Böhle, etc. Mitigation of Unsteady Excitation in a Large Vertical Centrifugal Pump Under Flow Instability Via Novel Diffuser Configurations. *ASME Journal of Fluids Engineering*, 2025, 147: 121204.

Gang Yang, Desheng Zhang, Martin Böhle, etc. Transferability Investigation of the Tandem Impeller on the Performance of A Low Specific Speed Centrifugal Pump. *Physics of Fluids*, 2025, 37: 125175.

List of Supervised Student Thesis

J F. Ding	Study on the Performance Optimization of Multi-Operating Conditions and Unsteady Characteristics of Forward-Curved Blades Turbine	MA	2023
H. Gao	Study on the Influence of Axial Clearance on Internal Flow Characteristics and Axial Force of Seawater Desalination Turbine Energy Recovery Integrated Machine	MA	2023
J. Meng	Research on the Multi-Condition Performance Optimization and Internal Flow Characteristics of Forward-Curved Blade Turbine Based on Genetic Algorithm	MA	2024
W H. Luo	Study on the Transient Characteristics of Failure Shutdown Transition Process of Large Vertical Centrifugal Pumps	MA	2024

Curriculum Vitae

Personal data

Name:	Gang Yang
Nationality:	China
E-Mail	gang.yang@rptu.de

Education

09/2015-06/2019	Bachelor in Jiangsu University Jingjiang College, China.
09/2019-06/2022	Master in Jiangsu University, China.

Work Experience

09/2022-Present	Student Assistant Research Center of Fluid Machinery Engineering and Technology Jiangsu University
02/2024-02/2026	Research Assistant Department of Mechanical and Process Engineering, Institute of Fluid Mechanics and Turbomachinery RPTU University Kaiserslautern-Landau

