Analysis of vortices formed in flow passage of a five-bladed centrifugal water pump by means of PIV method

Cite as: AIP Advances 9, 075011 (2019); https://doi.org/10.1063/1.5099530
Submitted: 11 April 2019 . Accepted: 07 June 2019 . Published Online: 16 July 2019

Yanping Wang, Hui Yang, Bo Chen, Panlong Gao, Hui Chen, and Zuchao Zhu

ARTICLES YOU MAY BE INTERESTED IN

Optical polarization technique—for enhancement of image quality—in speckle contrast-based perfusion imaging: A characterization study
AIP Advances 9, 075003 (2019); https://doi.org/10.1063/1.5087228

Detection of radiation torque exerted on an alkali-metal vapor cell
AIP Advances 9, 075002 (2019); https://doi.org/10.1063/1.5097258

Sample-based calibration for cryogenic broadband microwave reflectometry measurements
AIP Advances 9, 075005 (2019); https://doi.org/10.1063/1.5097897
Analysis of vortices formed in flow passage of a five-bladed centrifugal water pump by means of PIV method

Cite as: AIP Advances 9, 075011 (2019); doi: 10.1063/1.5099530
Submitted: 11 April 2019 • Accepted: 7 June 2019 • Published Online: 16 July 2019

Yanping Wang,1 Hui Yang,1,a) Bo Chen,1 Panlong Gao,1 Hui Chen,2 and Zuchao Zhu1

AFFILIATIONS
1 State-Province Joint Engineering Lab of Fluid Transmission System Technology, Zhejiang Sci-Tech University, Hangzhou 310018, China
2 Xi’an Aerospace Propulsion Institute, China Aerospace Science and Technology Corporation, Xi’an 710100, China

a)Address all correspondence to this author Email: yanghui@zstu.edu.cn

ABSTRACT
Detailed optical measurements for the flow inside rotating passages of a five-bladed centrifugal impeller were performed by particle image velocimetry (PIV). The flow in mid-plane perpendicular to the pump axis was measured at 1400 r/min rotation speed. Seven flow rates, namely, 1.2, 1.0, 0.8, 0.6, 0.4, 0.2, and 0 Qd, were surveyed. The averaged PIV velocity maps and streamline were analyzed. Results show that when flow rate decreases to 0.8 Qd, the flow separation forms initially at the blade suction side in passage 1. With decreasing of flow rate, the flow separation appears in much more passage, the separation region enlarges, and the flow at the pressure side begins to form flow separation. All passages are gradually occupied by the vortices generated by flow separation, until the passages are finally blocked. The scale of vortex along the stream-wise direction at the suction side is larger than that at the pressure side, whereas the scale of vortex along the span-wise direction is smaller than that at the pressure side. With the decrease in flow rate, the scales of vortices at the suction and pressure sides increase, and the vortices at both sides move toward the inlet and outlet, respectively. Moreover, the effect of vortices on the tangential and radial components of the absolute velocity was analyzed.

© 2019 Author(s). All article content, except where otherwise noted, is licensed under a Creative Commons Attribution (CC BY) license (http://creativecommons.org/licenses/by/4.0/). https://doi.org/10.1063/1.5099530

I. INTRODUCTION

The flow inside a centrifugal impeller is complicated due to the curvature and rotation of the blade,1–6 and the theoretical and experimental study on internal flow is advancing.7–13 Particle image velocimetry (PIV) has been widely used as a measuring approach to investigate the internal flow of centrifugal pumps for its nondisturbance on flow velocity. Keller et al.14 adopted PIV to analyze the unsteady flow field near the volute tongue region at high flow rate. They obtained the shedding of wake vortex at the trailing edge of the impeller blade, its influence on the volute tongue, and its evolution process. Sinha and Katz15 measured the flow field of centrifugal pump near the design flow rate. They acquired the average velocity, vorticity field, and corresponding turbulent stress distribution at different impeller blades. Pedersen et al.16 used PIV to measure the flow field at design and low flow rates, analyzed the instantaneous and average velocity fields of the flow field, and compared the experimental results with LDV data. Sinha et al.17 applied PIV to study the occurrence and development of the internal rotating stall of centrifugal pump. Li et al.18 examined the rotor-stator interaction mechanism in the impeller and vanes of mixed-flow pump under part loading condition on the basis of PIV. Bai et al.19 used 2D PIV to analyze the secondary flow in a U-shaped open channel with long straight inflow/outflow reaches and gain deep insight into the evolution of secondary flows in open-channel bend. At present, studies on the use of PIV in investigating internal flow in pumps are mainly focused on flow pattern and turbulence statistics in single or dual flow passages at a certain flow rate.20–22 The study on the variation of vortex motion with flow rate and the relationship between the vorticity and pump head or flow rate in the passage is limited. Considering the vortex is strongly related with pump performance,23 in the present work,
PIV is utilized to measure the flow field in a five-bladed centrifugal pump, and the velocity field in the impeller is obtained. By analyzing the streamline of the velocity field, the variation of the scale and position of vortex in the flow passage with flow rate is evaluated under the same rotation speed, and the effects of vorticity on the pump head and impeller passage through-flow are investigated.

II. TEST PUMP AND TEST RIG

Fig. 1 displays the real pump, its 3D model, and the blade-to-blade and meridional views. The impeller and rear cover of the experimental pump are made of plexiglass. For efficient and easy visual observation, the surface is polished, and the non-transparent areas of the cover and shroud of the impeller are blackened to reduce background noise. The impeller has a 2D geometry and thus is fixed in the axial direction. The water flows in along the axis and then flows out through the eight diversion holes on the pump wall. For a clear and convenient description, the impeller passages are labeled with numbers (1, 2, 3, 4, and 5), and numbered passages are fixed in space. The geometric parameters of the impeller are as follows: inlet diameter \( D_1 = 56 \text{ mm} \), outlet diameter \( D_2 = 142 \text{ mm} \), inlet inclination angle \( \beta_1 = 18^\circ \), outlet inclination angle \( \beta_2 = 40^\circ \), and number of blades \( Z = 5 \).
The experimental platform is a closed test rig, as shown in Fig. 2. The variable frequency control cabinet is used to regulate the motor. Two control valves are positioned at the inlet and outlet of the experimental pump to control the flow rate. The electromagnetic flowmeter is used to measure the flow rate. The head is measured by two pressure transmitters. The power is gauged by a three-phase PWM special tester. On the basis of the experimental platform, the flow rate, head, and torque of the experimental pump are collected using a pump parameter measuring instrument. The PIV used in Fig. 2 is a commercial system produced by TSI Corporation (United States), which consists of a laser, cross-frame CCD camera, time synchronization controller, phase-locked synchronization system, and INSIGHT 3G software platform. The resolution of the CCD camera is 2048 pixel $\times$ 2048 pixel and the gray scale is 12 bit. The test section is mid-plane between the shroud and hub of the impeller and is perpendicular to the rotation axis. Fig. 3 shows the measuring area of the PIV. The tracer particles are SiO$_2$ with good followability and scattering properties. The particle size and density of tracer particles are 20–60 $\mu$m and 1050 kg/m$^3$, respectively.

III. EXPERIMENTAL RESULTS AND DISCUSSION

The external characteristic of the experimental pump was characterized before the PIV experiment. The experimental rotation speed $n$ is 1400 r/min. The Re value (formula: \( Re = \frac{\Omega D^2}{v} \)), where $\Omega$ is the angular velocity, $D$ is the impeller outlet diameter, and $v$ is the kinetic viscosity of water, which is equal to $1.0 \times 10^{-6}$ m$^2$/s, therefore, the internal flow of the impeller is turbulent. The velocity fluctuation of turbulence is between 5% and 10%. Fig. 4 show the external characteristic curve. The optimal flow rate $Q_d$ is 2.5 m$^3$/h, the head is 5.1 m, and the efficiency is 39.5%.

The PIV experiment was conducted at 1.2, 1.0, 0.8, 0.6, 0.4, 0.2, and 0 $Q_d$. The flow rate was adjusted by regulating the outlet valve. To improve the phase average velocity field, 300 pictures were taken under each flow rate condition. The phase-locked device ensures that the pump is always in the same phase position when they are recorded every time. INSIGHT 3G software was used for image processing. The images were normalized before processing, and multi-pass correlation interrogation and image deformation methods were utilized. The interrogation region gradually decreased from 64 pixels to 32 pixels, and the final pass interrogation regions were overlapped by 50%. According to Keller et al.,[14] the associated uncertainty for the PIV system is approximately 1%–3% of the measured velocity for instantaneous velocities, and the uncertainty of the mean flow reduces at approximately an order of magnitude after phase-averaging. As the present study focused on the mean flow and detection of large-scale structures, this deficiency was found insignificant. The distance between adjacent vectors was 1.56 mm. The entire system could ensure security and decrease the interaction times. At all flow rates, the import gauge pressure was between 0.0151 MPa and 0.0133 MPa. The calculated cavitation allowance was from 11.5 m to 11.7 m, which was considerably larger than the essential cavitation allowance. Moreover, we observed the inlet flow field under different flow rates before the PIV experiment by using a high-speed camera, and no cavitation was found. Thus, in the following discussion, cavitation is neglected.

All subsequent results were obtained from 300 images. Fig. 5 depicts the direction of the coordinate system. $x$ and $y$ represent the horizontal and vertical directions, respectively. The measurement results are functions of position $x$, $y$ coordinates, and time $t$, including phase average and pulsation:
A. Flow structure of relative velocity field in the impeller

Fig. 6 shows the streamline of mid-plane relative velocity under different flow rates. The impeller flow passages are labeled with numbers in Fig. 6(g) on the basis of Fig. 1. The rotation direction of the impeller is clockwise. Fig. 6 displays the relative velocity field follows the blade curvature along the impeller passage when the flow rate is greater than or equal to 1.0 \( Q_d \). No significant separation occurs, and the flow field distribution in each flow passage is almost identical. A low velocity region is formed at the pressure side of the blade, and it extends approximately 1/3 downstream. When the flow rate is less than or equal to 0.8 \( Q_d \), the water enters the impeller flow passage at a large incident at the blade leading edge. With the increase of incident, the water entering the flow passage cannot follow the blade, that is, the water fluid and blade are separated. Moreover, the low velocity region begins to form at the blade suction side. When the flow rate is 0.8 \( Q_d \), a low velocity region close to the blade suction side appears in flow passage 5, which extends from 1/4 location of the suction side inlet to outlet accompanied by a clockwise separation vortex. The inlet fluid approaches the pressure side due to the blocking effect of the vortex, and the low velocity region of the pressure side is pressed toward the outlet.

A relative low velocity region appears at the suction side outlet in passages 1 and 3. No apparent low velocity region appears at the suction side of passages 2 and 4, and the flow field distribution in them is consistent with that at 1.0 and 1.2 \( Q_d \). Fig. 6 reveals that with increasing of low velocity region area and velocity, the inlet velocity and high velocity region area are decreased, which indicates that the vortex is caused by the separation of inlet fluid from the blade. With decreasing of flow rate, the velocity value and high velocity region area of the flow passage inlet decrease relatively. When the flow rate is 0.6 \( Q_d \), in addition to passage 4, a low velocity clockwise vortex appears at the suction side of other passages. Meanwhile, a counterclockwise vortex forms in the low velocity region of the pressure side outlet in flow passage 5, and its flow is in the opposite direction of the impeller rotation, which reduces the through-flow in the passage. Pedersen et al. observed a similar vortex. The low velocity region at the pressure sides of other flow passages approaches the outlet, the vortex on the blade suction side in flow passage 5, and its flow is in the opposite direction of the impeller rotation, which reduces the through-flow in the passage. Pedersen et al. observed a similar vortex. The low velocity region at the pressure sides of other flow passages approaches the outlet.

B. Development of separation vortex in impeller flow passage

As shown in Fig. 6, the vortex is initially generated in flow passage 5. Therefore, flow passage 5 is selected to analyze the development characteristic of the vortex in flow passage. Fig. 7 reveals the scale distribution of the separation vortex along the stream-wise and span-wise directions in flow passage 5. As shown in Fig. 7, the clockwise vortex at the suction side is larger than the counterclockwise vortex at the pressure side along the flow direction, but it is smaller than that along the span-wise direction, which reflects the generation condition of clockwise and counterclockwise vortices. The clockwise vortices are caused by the increase in incident. As a result of obstruction caused by the clockwise vortex, the fluid is brought close to the pressure side under the action of a large Coriolis force, thereby generating the counterclockwise vortices. With the decreasing of flow rate, the scale of vortices at the suction and pressure sides increases regardless of the direction (stream-wise or span-wise). In the stream-wise direction, the structure of vortex at the pressure side is consistent when the flow rate is at 0.2–0.6 \( Q_d \). When the flow rate is 0, the scale of vortex increases significantly, and the suction force is consistent with that when flow rate is at 0.2–0.4 \( Q_d \). The scale of vortex changes greatly when the flow rates are 0.6–0.4 and 0.2–0 \( Q_d \). In the span-wise direction, the scale of vortex at the pressure side increases rapidly, but that at the suction side increases slowly, as the flow rate decreases.

Fig. 8 presents the center position distribution of the separation vortex in flow passage 5, where \( r \) is the distance from the...
FIG. 6. Velocity streamline diagram under different flow rates.
FIG. 7. Scale of separation vortices in passage 5.

FIG. 8. Center position of separation vortices.

FIG. 9. Velocity distribution along different radii ($r/r_m$).
center of the separation vortex to the impeller axis and \( r_m \) is the impeller outlet radius. Fig. 8 shows that as flow rate decreases, the counterclockwise vortex at the pressure side gradually moves toward the outlet direction, and the clockwise vortex at the suction side gradually moves toward the inlet direction. Hence, the flow passage blocking effect is intensified as flow rate decreases, and the separation of fluid from the blade is intensified whether at the suction or pressure surface. Consequently, the clockwise and counterclockwise

**FIG. 10.** Contour plot of the absolute tangential velocity \((C_t)\).
vortices are enhanced, and their strength is close to each other as flow rate increases. A repulsive phenomenon occurs between the vortices in the same flow passage, thereby causing the clockwise vortex to move toward the passage inlet and the counterclockwise vortex to move toward the outlet.

C. Detailed velocity distribution

Fig. 9 shows the continuous distribution of relative velocity ($W$) in passage 5 from blade inlet to blade outlet. $W/L = 0$ is at suction side (labeled as SS), whereas $W/L = 1$ is at the pressure side (labeled as PS). For flow rates of 1.2 and 1.0 $Q_d$, along the meridional distance of $r/r_m = 0.4–0.8$, the relative velocity at the suction side is higher than that at the pressure side due to the effect of potential vortex. The result is consistent with Ref. 24. For a flow rate of 0–0.8 $Q_d$, along the meridional distance of $r/r_m = 0.4–0.8$, the relative velocity at the pressure side is higher than that at the suction side due to the appearance of clockwise (at the suction side) and counterclockwise (at the pressure side) vortices, respectively (Fig. 6). At the outlet location, the distributions of relative velocity for all flow rates are similar and displace a jet–wake flow pattern, that is, the relative velocity at the pressure side is considerably higher than that at the suction side. The phenomenon has been discussed in many studies (e.g., Refs. 25–27).

D. Tangential and radial components of absolute velocity

Fig. 10 shows the distributions of absolute tangential velocity ($C_t$). $C_t$ represents the head obtained by the flow. The distribution trends corresponding to each flow rate are generally the same. The magnitude of $C_t$ increases with the increase in meridional distance, which results from the increase in Coriolis forces. Near the outlet, the magnitude of $C_t$ at the suction side is larger than that at the pressure side, which is in account of the jet–wake flow.

Fig. 11 presents the detailed distributions of $C_t$ in passage 5. As no vortex appears in the passage, the distributions of $C_t$ at flow rates of 1.2 and 1.0 $Q_d$ are different from those at other flow rates.
At the inlet, \( C_t \) at the pressure side is larger than that at the suction side. This phenomenon is caused by the curvature effect of the blade leading edge. For large radii of \( r/r_m = 0.6 \) and 0.8, \( C_t \) at the pressure side is larger than that at the suction side. This result is due to the effect of potential vortex. For a flow rate of 0–0.8 \( Q_d \), the distribution of \( C_t \) corresponding to each flow rate is almost the same as the vortices are formed. At the inlet, the magnitude of \( C_t \) changes slightly along the curve of \( r/r_m = 0.4 \). For \( r/r_m = 0.6 \), \( C_t \) at the suction side is larger than that at the pressure side, indicating that the clockwise vortex is formed at the suction side (Fig. 6). For \( r/r_m = 0.8 \), \( C_t \) initially decreases and then increases from the suction side to the pressure side in the same passage due to the radius across the clockwise and counterclockwise vortices. Under all flow rate conditions, \( C_t \) increases as the flow rate decreases.

**FIG. 13.** Contour plot of absolute radial velocity (\( C_t \)).
which can be found from the distribution of mean absolute tangential velocity \( \overline{C}_t \) (Fig. 12). The mean absolute tangential velocity is the average value of \( C_t \) along the same radius. Thus, the clockwise and counterclockwise vortices can increase the head obtained by the flow.

The scale of vortex at the suction side is larger than that at the pressure side. For a flow rate of 1.2 \( Q_d \), the pressure side near the suction side along a meridional distance of \( r/r_m = 0.44–0.58 \). With increasing of Coriolis force, \( C_t \) decreases gradually along the radius of \( r/r_m = 0.58–1 \). \( C_t \) near the pressure side is considerably smaller than that near the suction side along a meridional distance of \( r/r_m = 0.44–0.58 \). Zhang et al.\(^{28}\) observed the same phenomenon. The blade leading edge has a small negative \( C_t \) region, which means that the reverse flow appears. For a flow rate of 1.0 \( Q_d \), the large \( C_t \) region near the suction side decreases, and the value of negative \( C_t \) near the pressure side increases. For a flow rate of 0.8 \( Q_d \), the distribution of \( C_t \) in passage 5 changes remarkably. For flow rates of 1.0 and 1.2 \( Q_d \), the large \( C_t \) region near the suction side moves toward the pressure side and merges with the large \( C_t \) region near the pressure side, and negative \( C_t \) appears near the suction side, which is in accordance with the clockwise vortex (Fig. 6). For the other passages, the large \( C_t \) regions near the suction side decrease continuously, and the large \( C_t \) regions near the pressure side appear in all passages. As flow rate decreases, the negative \( C_t \) region forms near the suction side at the outlet, which is in accordance with the counterclockwise vortex near the pressure side (Fig. 6), and the large \( C_t \) region near the suction side disappears gradually, thereby decreasing the through-flow. This phenomenon can be seen from the distribution of mean absolute radial velocity \( \overline{C}_r \) (Fig. 14). The mean absolute radius velocity is the average value of \( C_r \) along the same radius. Thus, the clock and counterclockwise vortices decrease the through-flow in the impeller passage.

**IV. CONCLUSION**

An external characteristic experiment was conducted on a five-bladed centrifugal pump, and the external characteristic curve under 1400 r/min rotational speed was obtained. On this basis, seven test flow rates between 0 and 1.2 \( Q_d \), that is, 1.2, 1.0, 0.8, 0.6, 0.4, 0.2, and 0 \( Q_d \), of the PIV experiment was selected. By analyzing the vortex motion in the flow passage of a centrifugal impeller, the conclusions are summarized as follows:

1. As flow rate decreases, the flow separation initially occurs at the suction side of a certain flow passage. When the flow rate continues to decrease, the flow separation gradually occurs in other flow passages, that is, at the suction and pressure sides. As the flow decreases continuously, the counterclockwise vortex at the pressure side and the clockwise vortex at the suction side are enhanced. Finally, all flow passages are occupied by the vortex, and the flow passage is blocked.

2. The scale of vortex at the suction side is larger than that at the pressure side along the flow direction, but it is smaller than that along the span direction. As flow rate decreases, the scales of vortices at the pressure and suction sides increase, and the vortices at the suction and pressure sides move toward the inlet and outlet directions, respectively.

3. The vortices at the suction and pressure sides can increase the absolute tangential velocity and decrease the radial velocity, thereby increasing the head obtained by the flow and decreasing the through-flow through the impeller passage.

**ACKNOWLEDGMENTS**

This work was supported by the National Natural Science Foundation of China (Grant No. 51536008), National Basic Research Program of China (Grant No. 6.3321), and Public Projects of Zhejiang Province (2017C31075). The assistance is gratefully acknowledged.

**NOMENCLATURE**

- \( Q_d \) nominal flow rate, \( \text{m}^3/\text{h} \)
- \( H \) head, m
- \( \eta \) efficiency, \%
- \( n \) rotation speed, r/min
- \( D_1 \) impeller inlet diameter, mm
- \( D_2 \) impeller outlet diameter, mm
- \( \beta_1 \) blade inlet angle, \(^\circ\)
- \( \beta_2 \) blade outlet angle, \(^\circ\)
- \( Z \) blade number

**REFERENCES**

1. C. Wang, X. K. He, W. D. Shi, X. K. Wang, X. L. Wang, and N. Qiu, "Numerical study on pressure fluctuation of a multistage centrifugal pump based on whole flow field," AIP Advances 9, 035118 (2019).
2. T. Shigemitsu, J. Fukutomori, K. Kaji, and T. Wada, "Unsteady internal flow conditions of mini-centrifugal pump with splitter blades," J. Therm. Sci. 22, 86–91 (2013).
3. T. Shigemitsu, J. Fukutomori, T. Wada, and H. Shinohara, "Performance analysis of mini centrifugal pump with splitter blades," J. Therm. Sci. 22, 573–579 (2013).
4. D. Y. JI, Y. L. Qin, Z. G. Zuo, H. J. Wang, S. H. Liu, and X. Z. Wei, "Numerical simulation on pump transient characteristic in a model pump turbine," J. Fluids Eng. 141, 111101 (2019).
C. Wang, W. D. Shi, X. K. Wang, X. P. Jiang, Y. Yang, W. Li, and L. Zhou, “Optimal design of multistage centrifugal pump based on the combined energy loss model and computational fluid dynamics,” Applied Energy 187, 10–26 (2017).

X. J. Li, B. X. Yu, Y. C. Ji, J. X. Lu, and S. Q. Yuan, “Statistical characteristics of suction pressure signals for a centrifugal pump under cavitation conditions,” J. Therm. Sci. 26, 47–53 (2017).

J. J. Feng, F. K. Renra, and X. Q. Luo, “Experimental investigation on turbulence fields in a radial diffuser pump using PIV technique,” Adv Mech Eng. 6, 702318 (2015).

X. M. Ren, H. G. Fan, Z. F. Xie, and B. Liu, “Stationary stall phenomenon and pressure fluctuation in a centrifugal pump at partial load condition,” Heat Mass Transfer 11, 1–12 (2019).

X. J. Li, P. L. Gao, Z. C. Zhu, and Y. Li, “Effect of the blade loading distribution on hydrodynamic performance of a centrifugal pump with cylindrical blades,” J. Mech. Sci. Technol. 32, 1161–1170 (2018).

D. Y. Li, X. L. Fu, Z. G. Zuo, H. J. Wang, Z. G. Li, S. H. Liu, and X. Z. Wei, “Investigation methods for analysis of transient phenomena concerning design and operation of hydraulic-machine systems—A review,” Renew. Sust. Energ. Rev. 101, 26–46 (2019).

C. Wang, X. K. He, D. S. Zhang, B. Hu, and W. D. Shi, “Numerical and experimental study of the self-priming process of a multistage self-priming centrifugal pump,” Int. J. Energ. Res. 1–19 (2019).

C. Wang, X. X. Chen, N. Qu, Y. Zhu, and W. D. Shi, “Numerical and experimental study on the pressure fluctuation, vibration, and noise of multistage pump with radial diffuser,” J. Braz. Soc. Mech. Sci. 40, 481 (2018).

Y. N. Zhang, T. Chen, J. W. Li, and J. X. Yu, “Experimental study of load variations on pressure fluctuations in a prototype reversible pump turbine in generating mode,” ASME J. Fluids Eng. 139, 074501 (2017).

J. Keller, E. Blanco, R. Barrio, and J. Parrondo “PIV measurements of the unsteady flow structures in a volute centrifugal pump at a high flow rate,” Exp. Fluids 55, 1820 (2014).

M. Sinha and J. Katz, “Quantitative visualization of the flow in a centrifugal pump with diffuser vanes—I: On flow structures and turbulence,” J. Fluids Eng. 122, 97–107 (1999).

N. Pedersen, P. S. Larson, and C. B. Jacobsen, “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. Fluids Eng. 125, 61–72 (2003).

M. Sinha, A. Pinarbarisi, and J. Katz, “The flow structure during onset and developed states of rotating stall within a vaned diffuser of a centrifugal pump,” J. Fluids Eng. 123, 490–499 (2001).

W. Li, L. Zhou, W. D. Shi, L. L. Ji, Y. F. Yang, and X. F. Zhao, “PIV experiment of the unsteady flow field in mixed-flow pump under part load condition,” Exp. Therm. Fluid Sci. 83, 191–199 (2017).

R. N. Bai, D. J. Zhu, H. Chen, and D. X. Li, “Laboratory study of secondary flow in an open channel bend by using PIV,” Water 11, 659 (2019).

L. Zhou, L. Bai, W. Li, W. D. Shi, and C. Wang, “PIV validation of different turbulence models used for numerical simulation of a centrifugal pump diffuser,” Eng. Computation 35, 2–17 (2018).

L. Zhou, W. D. Shi, W. Li, and R. Agarwal, “Particle image velocimetry measurements and performance experiments in a compact return diffuser under different rotating speed,” Exp. Techniques 40, 245–252 (2016).

L. Zhou, W. D. Shi, W. D. Cao, and H. B. Yang, “CFD investigation and PIV validation of flow field in a compact return diffuser under strong part-load conditions,” Sci. China (Tech. Sci.) 58, 405–414 (2015).

Y. N. Zhang, X. Qiu, F. P. Chen, K. H. Liu, Y. N. Zhang, X. R. Dong, and C. Q. Liu, “A selected review of vortex identification methods with applications,” I. Hydrodyn. 30, 767–779 (2018).

R. W. Westra, L. Broersma, K. Van Andel, and N. P. Kruyt, “PIV measurements and CFD computations of secondary flow in a centrifugal pump impeller,” J. Fluid. Eng.-T. ASME 132, 061104 (2010).

D. Eckardt, “Detailed flow investigations within a high-speed centrifugal compressor impeller,” J. Fluids Eng. 98, 390–399 (1976).

K. H. Rohne and M. Banzhaf, “Investigation of the flow at the exit of an unshrouded centrifugal impeller and comparison with the classical jet-wake theory,” ASME J. Turbomach. 113, 0654–659 (1990).

Z. G. Sun, L. J. Hu, P. He, W. Li, and C. Q. Tan, “Investigation on the secondary flow structures and jet-wake structure of the Eckardt’s impeller,” J. Eng. Thermophys. 32, 2017–2021 (2011).

N. Zhang, B. Gao, Z. Li, W. Li, and C. Q. Tang, “Unsteady flow structure and its evolution in a low specific speed centrifugal pump measured by PIV,” Exp. Therm. Fluid. Sci. 97, 133–144 (2018).