Research Article

Experimental Investigation on Turbulence Fields in a Radial Diffuser Pump Using PIV Technique

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Particle image velocimetry (PIV) technique has been successfully employed to measure the unsteady periodic flow in a low-specific speed radial diffuser pump, in order to obtain turbulence intensity fields caused by rotor-stator interaction at different operation conditions. The PIV measuring region covers a complete impeller channel and a complete diffuser channel, enabling the investigation of turbulence behavior in both the impeller and diffuser region simultaneously, and the turbulence transportation from the upstream impeller wake to the diffuser region. In addition, the flow rate effect on the turbulence field has also been investigated by comparing measured turbulence fields at different flow rates in both qualitative and quantitative manners. This work can enhance the understanding of turbulence generation mechanism, transport behavior, and development characteristics in rotor-stator interaction in vaned radial diffuser pumps.

1. Introduction

Radial pumps are turbomachines for transporting liquids by raising a specified volume flow to a specified pressure level, which have been widely applied in industry. The internal flow developing in radial pumps is extremely complicated and highly turbulent, caused by streamline curvatures, system rotation, rotor-stator interaction, and turbulence effects [1, 2].

Particle image velocimetry (PIV) is a measurement technique for obtaining instantaneous velocity fields, in which the property actually measured is the distance travelled by particles in the flow within a known time interval. PIV is a powerful alternative or supplement to laser Doppler velocimetry (LDV), which offers more information on the instantaneous spatial flow structures and reduces considerably acquisition time. As an effective method of nonintrusive and instantaneous measurement, PIV has been widely used to capture flow structures in turbomachines, such as the work conducted in compressors [3–5], turbines [6, 7], and fans [8, 9].

PIV conducted in radial pumps to investigate the unsteady flow field can be summarized as follows. Akin and Rockwell [10] used PIV to study the wake from a model impeller and its interaction with a stationary diffuser blade. Wernet [11] provided an overview of applications of PIV to rotating machinery. Sinha et al. [12, 13] used PIV measurements to identify the unsteady flow structure and turbulence in a transparent radial pump with a vaned diffuser. The results demonstrated that the entire flow was dominated by a series of wakes, generated by the impeller blades, diffuser vanes and unsteady separations. In addition, Sinha et al. [14] and Krause et al. [15] used two-dimensional PIV to investigate the onset and development of rotating stall within radial pumps. Some other work can be found by Wuibaut et al. [16, 17], Dupont et al. [18], Feng et al. [19, 20], and Cavazzini et al. [21].

All the above work contributes to the understanding of the complex unsteady flow in radial diffuser pumps. However, most attention was paid on phase-averaged velocity fields and velocity-related flow structures rather than on turbulence fields. However, the turbulence field plays also an important role in the interior flow in radial pumps, therefore deserving being precisely measured and investigated in detail.

In this paper, PIV has been applied to obtain turbulence intensity fields at different operation conditions inside a
Table 1: Geometric data and operating condition.

| Impeller          | Diffuser         |
|-------------------|------------------|
| Number of blades  | Number of vanes  |
| $Z_i$             | $Z_d$            |
| 6                 | 9                |
| Inlet radius      | Inlet radius     |
| $R_1$             | $R_3$            |
| 40 mm             | 77.5 mm          |
| Outlet radius     | Outlet radius    |
| $R_2$             | $R_4$            |
| 75.25 mm          | 95 mm            |
| Blade height      | Vane height      |
| $b_i$             | $b_d$            |
| 12.7 mm           | 14 mm            |
| Inlet throat width| Inlet throat width|
| $W_1$             | $W_3$            |
| 17.9 mm           | 5.7 mm           |
| Inlet blade angle | Inlet blade angle|
| $\beta_1$         | $\alpha_3$      |
| 17.9 deg          | 9 deg            |
| Outlet throat width| Outlet throat width|
| $W_2$             | $W_4$            |
| 23.8 mm           | 7.5 mm           |
| Outlet blade angle| Outlet blade angle|
| $\beta_2$         | $\alpha_4$      |
| 22.5 deg          | 19.7 deg         |
| Design operating point | Volume flow rate |
| $Q_{\text{DES}}$  | 0.0045 m$^3$/s   |
| Rotating speed    | $n_{\text{DES}}$ |
| $r_{\text{DES}}$  | 1450 rpm         |
| Delivery head     | $H_{\text{DES}}$ |
| $H_{\text{DES}}$  | 7 m              |

2. Experiment Set-Up and Procedure

The pump stage under investigation is a low-specific speed radial pump, consisting of a shrouded impeller, a vaned diffuser, and a vaned return channel. The cross-section of the pump is given in Figure 1. The impeller has six strongly backswept blades, and the diffuser has nine vanes. Both the impeller blades and diffuser vanes are totally designed in two dimensions. The radial gap between the impeller trailing edge and the diffuser leading edge is 2.25 mm. The whole pump is manufactured out of Plexiglas to provide optical access for optical measurements. A view of the impeller and diffuser is shown in Figure 2. The main parameters of the pump and the design operating point are listed in Table 1.

The pump test stand equipped with PIV system is shown in Figure 3. An electronic motor through a belt is applied to drive the pump, and the rotating speed of the motor can be increased gradually with a step of 0.1 Hz. A tank with about 3 m$^3$ water is used to feed the water into the pump and to recollect the water out of the pump stage. An electromagnetic flow meter from KROHNE Inc. is installed on the pipe behind the pump to measure the flow rate of the pump.

The laser light source in the PIV measurements is a Solo PIV Nd:YAG laser system from New Wave Research Inc. It is a double-cavity 532 nm Nd:YAG laser with maximum energy output of 120 mJ/pulse. The laser beam with a diameter of 5 mm is guided by a flexible guiding arm. Cylindrical optics and a thickness adjuster are connected to the output end of the guiding arm. The optics produces a light sheet with a thickness of approximately 1 mm and a divergence angle of 15 deg, to illuminate the particles added to the water. The water is seeded with polyamide particles with average diameter of 20$\mu$m and density of 1016 kg/m$^3$. The particle density is very close to the density of water so that gravity force does not play a major role when comparing particle paths with streamlines.

The images are recorded by a CCD camera with a resolution of 1024 $\times$ 1280 pixels and with a maximum double frame rate of 4.5 Hz, which is aligned vertically to the laser sheet and mounted on a two-dimensional traverse system. An optical filter is installed in front of the camera in order to transmit only the fluorescent light from particles to the camera sensor. The laser and the camera are synchronized with the impeller blade orientation by using a magnetic sensor installed on the pump shaft used as a shaft encoder. This sensor generates one pulse per impeller revolution for the input of the PIV processor at a preselected impeller position. Other measuring positions can be obtained by a time delay based on it. The time delay between two laser illumination pulses should be long enough to determine the displacement between the images of the tracer particles with sufficient resolution and short enough to avoid particles with an out-of-plane velocity component leaving light sheet between subsequent illuminations. It is set to 30 ms after a thorough investigation with different time delays, which is
corresponding to a rotating angle change of 0.26 deg under the design operating speed.

The measuring region for the PIV measurement is indicated in Figure 4, with the size of 72.5 mm × 58 mm, covering a complete impeller channel and a complete diffuser channel as well. Before the measurement, a predefined target with scales has been placed in the measuring area inside the pump with water, to get the scale factor between the object and the image. In the measurements, 17 relative impeller positions have been considered. For each relative impeller position, 200 double-frame images are recorded. It is found that increasing the number of PIV maps does not have evident change in phase-averaged velocity and turbulence field.

3. Results and Discussion

The measurements are conducted only in two dimensions (x and y), and the axial velocity in the z-axis direction is not measured (see Figure 4). The relative impeller position \( \varphi \) is defined to associate the impeller circumferential position between the impeller and the diffuser. \( \varphi = 0 \) deg is defined as the relative position when the preselected impeller blade trailing edge (the lower one in Figure 4) begins to approach one preselected diffuser leading edge (the middle one in Figure 4), and all other impeller positions are based on this definition. The measured velocity components (\( u \) and \( v \)) in two orthogonal directions (\( x \) and \( y \)) each can be decomposed into two parts: a phase-averaged component depending on the measuring point position and the impeller circumferential position and a fluctuating component representing the turbulence effects, as denoted in the following:

\[
\begin{align*}
    u_i (x, y, \varphi) &= \bar{u}(x, y, \varphi) + u'_i (x, y, \varphi), \\
    v_i (x, y, \varphi) &= \bar{v}(x, y, \varphi) + v'_i (x, y, \varphi),
\end{align*}
\]

\( i = 1, \ldots, N \).

A good estimation of phase-averaged absolute velocity component at a certain impeller position \( \varphi \) can therefore be obtained by

\[
\begin{align*}
    \bar{u}(x, y, \varphi) &= \frac{1}{N} \sum_{i=1}^{N} u_i (x, y, \varphi), \\
    \bar{v}(x, y, \varphi) &= \frac{1}{N} \sum_{i=1}^{N} v_i (x, y, \varphi).
\end{align*}
\]

The turbulence intensity \( Tu \) is calculated in (3) based on turbulent components and normalized by the impeller tip speed \( U_2 \):

\[
Tu(x, y, \varphi) = \frac{1}{U_2^2} \sqrt{\frac{1}{2N} \sum_{i=1}^{N} \left[ u'^2_i (x, y, \varphi) + v'^2_i (x, y, \varphi) \right]}.
\]

Here \( N = 200 \) is the number of instantaneous vector maps at the impeller position \( \varphi \).

Notice that only two velocity components are used to calculate \( Tu \), since the third one is not available by two-dimensional PIV measurement. However, \( Tu \) defined here is representing a kind of averaged turbulence intensity in two-axis directions by using the factor of 1/2 in (3).

The following statistical uncertainties have been estimated based on 95% confidence level: 1% in velocity measurement, 2% in phase-averaged components, and 5% in turbulence intensity.

All results presented here are at midspan, that is, at half blade height plane. The flow inside the impeller and diffuser is nearly two-dimensional due to the pump geometry designed on purpose (the impeller and diffuser have totally two-dimensional blades, and the hub and shroud are parallel and perpendicular to the rotating axis), which has already been validated by CFD simulations [22]. Therefore, the flow structure at midspan can represent the main flow field inside the pump to a great extent.

3.1. Design Operating Point (\( Q_{DES} \)). Figure 5 shows the turbulence intensity (\( Tu \)) and phase-averaged relative velocity field at the impeller position \( \varphi = -4 \) deg. The impeller rotation sense is clockwise. For the turbulence field in Figure 5(a), it is observed that the measurement result is generally quite good,
Despite some hotspots near the measuring region boundary and some noise due to the reflection from solid surfaces of the blades, main features of turbulence intensity can be easily identified. In the impeller region, high turbulence is observed on the impeller suction side, where $Tu$ is about 7-8%, based on the impeller tip speed $U_2$. This high turbulence is caused by the interaction between the leakage flow in the front side chamber and the main flow in the impeller passage, confirmed by comparing the CFD result including side chambers with the one without considering side chambers [22]. Another high turbulence region is found just behind the impeller trailing edge caused by the impeller wake with a magnitude of 7-8%, where low relative velocity can be observed from Figure 5(b).

In the diffuser region, high turbulence is located around the diffuser leading edge and on the diffuser suction side near the diffuser leading edge. This is due to the impingement of the high turbulence, carried out by the impeller wake near the trailing edge and the diffuser leading edge. Furthermore, the diffuser wake with relatively high turbulence intensity is also very clear behind the diffuser trailing edge.

When associating the turbulence with the relative velocity field, it can be observed that the turbulence is generally high when the phase-averaged velocity is low. This is because high phase-averaged velocity is assumed to be more stable than low velocity in the flow and hence a lower turbulence can be expected.

The turbulence intensity at $\varphi = 10$ deg is shown in Figure 6 to examine the influence of relative impeller position on turbulence field. Compared with Figure 5(a) for $\varphi = -4$ deg, similar features can be observed. High turbulence near the suction side of the left impeller blade is visible for this impeller position, suggesting that the high turbulence due to the interaction between the leakage flow in the side chambers and the impeller main flow exists in all impeller passages. Furthermore, some segment of the impeller wake chopped by the diffuser leading edge has been pushed downstream along the suction side of the diffuser vane. However, the movement of high turbulence on the vane suction side is slower than the angular velocity of the impeller.

In order to present the turbulence intensity in a quantitative manner, six sections are located in the radial pump. Four radii starting from near the impeller inlet radius ($r/R_2 = 0.60$) to behind the impeller outlet radius ($r/R_2 = 1.01$) are indicated in Figure 5(a). In addition, the diffuser inlet and outlet throat are indicated by L3 and L4, respectively. $S^*$ is defined as the normalized circumferential coordinate; $\theta^* = 0$ and $\theta^* = 1$ represent the circumferential position on the impeller suction side (SS) and on the pressure side (PS), respectively. For comparison in diffuser region, $S^*$ indicates the dimensionless distance from the diffuser vane suction side to the corresponding pressure side. $S^* = 0$ is on the suction side and $S^* = 1$ is on the pressure side of the diffuser vane.

The turbulence intensity at $\varphi = -4$ deg is compared in Figure 7 at different radii. The lowest $Tu$ in the impeller is about 3%, occurring near the pressure side. $Tu$ on
the pressure side is smaller than on the suction side. Figure 8 gives the turbulence intensity distribution at the diffuser inlet and outlet throat, indicating that Tu varies from 5% to 7.3% at the diffuser inlet throat. At the outlet throat, Tu holds the value of 7.8% due to the diffuser wake near the suction side, and it decreases from the suction side to the pressure side.

3.2. Off-Design Operating Conditions. In order to investigate the effect of operating condition on turbulence intensity, the turbulence fields in three off-design conditions are compared, one overload operating point \( Q/Q_{DES} = 1.15 \), and two part-load operating points \( Q/Q_{DES} = 0.75 \) and \( Q/Q_{DES} = 0.5 \).

Figure 9 illustrates the turbulence intensity contours for \( Q/Q_{DES} = 1.15 \), at the same relative impeller positions in Figure 4 for \( Q/Q_{DES} \), and the same legend is chosen for a good qualitative comparison. Generally speaking, the turbulence intensity at \( Q/Q_{DES} = 1.15 \) is bigger than at \( Q/Q_{DES} \) possibly due to the increasing leakage flow effect and the increase of incidence at the diffuser leading edge. The high turbulence in the impeller passage, impeller wake, and diffuser wake is now becoming more evident.

Figure 10 gives the turbulence intensity contours at \( Q/Q_{DES} = 0.75 \). When compared with \( Q/Q_{DES} \), the turbulence intensity in the diffuser has become generally higher. For example, the turbulence intensity around the diffuser vane leading edge is clearly higher than that of \( Q/Q_{DES} \). In addition, segments with high turbulence near the diffuser inlet throat caused by the impingement of impeller wake and the diffuser leading edge are becoming clearer and in a higher magnitude compared with at \( Q/Q_{DES} \).

When reducing further the flow rate from \( Q/Q_{DES} \) to \( Q/Q_{DES} = 0.5 \), flow separation occurs in impeller due to increasing incidence at impeller leading edges. The phase-averaged relative velocity field is shown with streamlines for \( \phi = 10 \) deg in Figure 11, and flow separation can be clearly observed.

The turbulence intensity contours at \( Q/Q_{DES} = 0.5 \) are illustrated in Figure 12. High turbulence near the impeller leading edge on the impeller suction side is dominant, higher than 10%, which is caused by the unsteady flow separation shown in Figure 11. Flow separation with low velocities is commonly assumed to have more unstable and nonperiodic features, hence generally corresponding to high turbulence. The turbulence on the impeller pressure side near the impeller outlet is now becoming much higher than that of all other three operating points. In addition, the turbulence in the semivaned region of the diffuser is becoming higher, possibly due to the fact that the impeller outflow to the diffuser becomes more unsteady and nonuniform.

For showing the turbulence field in a quantitative manner, Figure 13 illustrates the comparison of turbulence intensity at the same impeller position \( \phi = -4 \) deg at some selected radii in the impeller, at the diffuser inlet and outlet throat, among different operating conditions of the pump stage.

For the comparison in the impeller region, at the radius of \( r/R_2 = 0.60 \) (Figure 13(a)), the turbulence intensity on the impeller blade pressure side is about 4–5.5%. It is found that the operating point \( Q/Q_{DES} = 1.15 \) generally holds the biggest turbulence intensity, and the lowest turbulence intensity occurs at \( Q/Q_{DES} = 0.5 \). This trend denotes that the high turbulence caused by the leakage flow increases with increasing flow rate. The highest Tu for each operating point is decreasing with decreasing flow rate: about 11% for \( Q/Q_{DES} = 1.15 \), 9% for \( Q/Q_{DES} = 1.05 \), 8% for \( Q/Q_{DES} = 0.75 \), and 7% for \( Q/Q_{DES} = 0.5 \). For all four operating points, Tu generally decreases from the middle of the impeller passage to the pressure side. This trend keeps developing for the next radius \( r/R_2 = 0.75 \) (Figure 13(b)), except near the impeller suction side (between the circumferential range \( \theta^* = 0.05 \) and \( \theta^* = 0.2 \)). Tu keeps decreasing due to the weakening effect from leakage flow in the side chambers, except for \( Q/Q_{DES} = 0.5 \) near the impeller pressure side (Figure 13(c)). Near the impeller outlet \( r/R_2 = 1.01 \) (Figure 13(d)), the highest Tu is observed for \( Q/Q_{DES} = 0.5 \), and Tu at \( Q/Q_{DES} \) is generally the lowest, indicating that the smallest unsteadiness
Figure 9: Turbulence intensity field at $Q/Q_{DES} = 1.15$.

Figure 10: Turbulence intensity field at $Q/Q_{DES} = 0.75$.

occurs at the design operating point $Q_{DES}$. For the off-design conditions, $Tu$ increases with decreasing flow rate.

For the comparison at the diffuser inlet throat, shown in Figure 13(e), $Tu$ on the pressure side is higher than on the suction side for all given conditions, due to the high $Tu$ near the diffuser leading edge. $Tu$ is generally in the range between 5% and 8%. At the diffuser outlet throat in Figure 13(f), higher $Tu$ is observed near the suction side of the diffuser vane influenced by the diffuser wake, decreasing with flow rate: 9.2% for $Q/Q_{DES} = 1.15$, 7.8% for $Q_{DES}$, 6.5% for $Q/Q_{DES} = 0.75$, and 6% for $Q/Q_{DES} = 0.5$. However, in the region near the pressure side, the design operating point $Q_{DES}$ holds the lowest turbulence, due to the smallest incidence.

4. Conclusions

Two-dimensional PIV has been successfully employed to measure turbulence intensity fields in a low-specific speed
radial pump equipped with a vaned diffuser at different operation conditions. The turbulence field inside the pump stage has been investigated qualitatively and quantitatively. Main conclusions can be drawn as follows.

1. At design operating point $Q_{DES}$, high turbulence regions can be observed behind the impeller trailing edge due to the impeller wake, on the impeller suction side due to the leakage flow, around the diffuser leading edge caused by the impingement of the impeller wake and the diffuser leading edge, and on the diffuser suction side and behind the diffuser trailing edge caused by the diffuser wake. The turbulence caused by the leakage flow is in the same level with the one in the impeller wake, with the turbulence intensity of about 7-8% (based on the impeller tip speed $U_2$). The impeller wake is chopped by the diffuser leading edge and transported along the diffuser vane suction side, producing a local high turbulence region on diffuser suction side after the diffuser throat. In addition, high velocity is assumed more stable than low velocity in the flow and hence a lower turbulence is expected.

2. The design operating point $Q_{DES}$ holds the smallest turbulence level in the impeller wake region.

3. The turbulence in impeller passage generally decreases with decreasing flow rate of the operation condition.

4. At strong part-load operating condition $Q/Q_{DES} = 0.5$, high turbulence is produced by unsteady flow separation near the impeller blade leading edge. In addition, the turbulence in the semivaned region of the diffuser increases due to the upstream increasing unsteady flow effect.

The present work can provide a good database for turbulence field both in the rotating impeller and in the vaned diffuser for radial pumps, in qualitative and quantitative manner, which can enhance the understanding of turbulence generation mechanism, transport behavior, and development characteristics in case of rotor-stator interaction in vaned radial diffuser pumps.

**Nomenclature**

- $\theta^*$: Normalized circumferential coordinate
- $\phi$: Impeller circumferential position
- $Q$: Flow rate
- $r, R$: Radius
- $S^*$: Dimensionless distance
- $T$: Period of the pump
- $Tu$: Turbulence intensity
- $u$: Absolute velocity in $x$-axis direction
- $U$: Circumferential velocity
- $v$: Absolute velocity in $y$-axis direction
- $W$: Relative velocity.

**Abbreviations**

- PS: Pressure side
- SS: Suction side.

**Subscripts**

- 1: Impeller inlet
- 2: Impeller outlet
- 3: Diffuser inlet
- 4: Diffuser outlet
- DES: Design
- $i$: Impeller.

**Conflict of Interests**

The authors declare that there is no conflict of interests regarding the publication of this paper.
Figure 13: Turbulence intensity comparison for different flow rates, $\phi = -4$ deg.

(a) $r/R_2 = 0.60$

(b) $r/R_2 = 0.73$

(c) $r/R_2 = 0.86$

(d) $r/R_2 = 1.01$

(e) At diffuser inlet throat (L3)

(f) At diffuser outlet throat (L4)
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