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Original Paper

Correction and Experimental Verification of Velocity Circulation in a Double-blade Pump Impeller Outlet

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Abstract

It is difficulty to calculate velocity circulation in centrifugal pump impeller outlet accurately. Velocity circulations of a double-blade pump impeller outlet were calculated with Stodola formula, Weisner formula and Stechkin formula. Simultaneously, the internal flow of impeller for the double-blade pump were measured with PIV technology and average velocity circulations at the 0.8, 1.0 and 1.2 times of design flow were obtained. All the experimental values were compared with the above calculation values at the three conditions. The results show that calculation values of velocity circulations with Weisner formula is close to the experimental values. On the basis of the above, velocity circulations of impeller outlet were corrected. The results of experimental verification show that the corrected calculation errors, whose maximum error is 3.65%, are greatly reduced than the uncorrected calculation errors. The research results could provide good references for establishment of theoretical head and multi-condition hydraulic optimization of double-blade pumps.

Keywords: Double-blade pump, Impeller, Velocity circulation, PIV, Correction.

1. Introduction

The theoretical head of centrifugal pump can be described with velocity circulation of the impeller inlet and impeller outlet [1-3]. When impeller blade number is infinite, velocity circulation of centrifugal pump impeller outlet is easy to be obtained, but in practical application, the impeller blade number is limited, the impeller outlet relative velocity will produce slip. That is velocity slip phenomenon. Due to the limited blade number, the velocity circulation of centrifugal pump impeller outlet is very difficult to be calculated, resulting in theoretical head of pump not being calculated accurately. Therefore, the centrifugal pump impeller outlet circulation is one of the key to solve the theoretical head of pump.

Particle Image Velocimetry (PIV) technology is a non-contact transient flow field measuring technology, which was developed in the late 1980s and matured gradually. It can capture velocity information of whole flow fields and does not interfere with the flow field. In recent years, PIV technology has been widely used to measure absolute velocity distributions and relative velocity distributions in pumps[4-12].

The structure of double-blade pump impeller is special, resulting in velocity slip being more obvious. Therefore, we choose a double-blade pump as research object to calculate velocity circulation of impeller outlet under three conditions with Stodola formula, Weisner formula and Stechkin formula. At the same time, PIV experiments were carried out and the calculation formula with small errors are corrected.

2. Performance Characteristic Test of Double-blade Pump

2.1 Double-blade Pump

Semi-spiral suction chamber of the double-blade pump is made of stainless steel. Impeller and volute are made of Perspex glass. The volute is designed by using equivalent velocity moment method, whose cross section is a rectangle and type line is the logarithmic spiral. Main design parameters of the double-blade pump are shown in Table 1.

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Table 1 Main design parameters

Description	Parameter	Value
Design Flow rate ($m^3 \cdot h^{-1}$)	Q_d	34.5
Head (m)	H	4.4
Rotational speed ($r \cdot min^{-1}$)	n	1000
Specific speed *	n_s	117.6
Suction chamber inlet diameter	D_s	80
Impeller inlet diameter (mm)	D_j	90
Impeller outlet diameter (mm)	D_2	201.2
Impeller outlet width (mm)	b_2	44.6
Blade inlet angle (°)	β_1	21.4
Blade outlet angle (°)	β_2	23.8
Blade wrap angle (°)	φ	177.5
Volute inlet diameter (mm)	D_3	212
Volute inlet width (mm)	b_3	77
Volute outlet section(mm×mm)	L×W	80×6

* $n_s = \frac{3.65n\sqrt{Q}}{H^{3/4}}$, where units of flow rate, rotational speed and head are m^3/s, r/min and m, respectively.

1.2 Test Bench

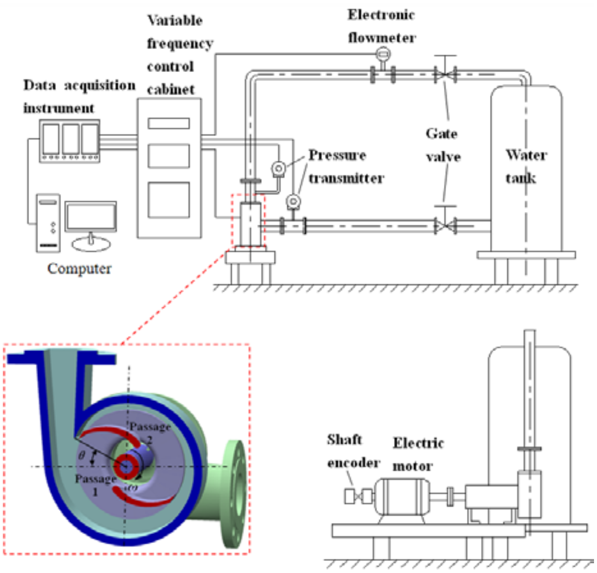


Fig. 1 Sketch of test bench

The performance characteristics test bench and the pump system are shown in Fig. 1. Main performance test equipments include a variable frequency control cabinet, a three-phase asynchronous motor, an electrical flowmeter, a pressure transmitter, a three-phase PWM digital power meter which is installed in the variable frequency control cabinet, etc. During the test, the motor speed is adjusted by the variable frequency control cabinet, the flow rate is measured by the electrical flowmeter, head is received by mercury manometer, power is obtained by three phase PWM digital power meter and all of experimental data are managed and analyzed by data acquisition instrument.

1.3 Test Result

Energy performance curve of the double-blade pump is shown in Fig. 2. The test results show that the head at the design flow rate is 4.5m and its efficiency is 58.48%, which is higher than the machinery industry standard of P.R. China (JB/T 8857-2000, 55.8% pump efficiency is required).

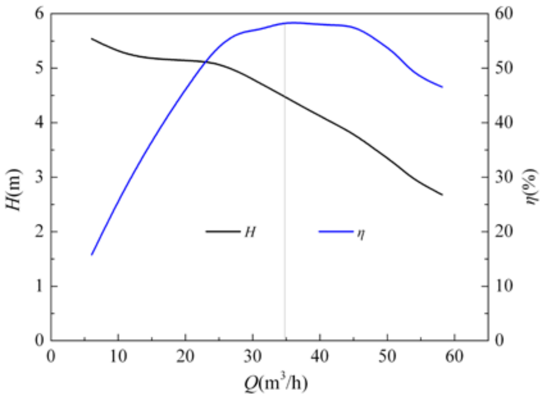


Fig. 2 Energy performance curves

3. Theory Analysis of Velocity Circulation in the Double-blade Pump Impeller Outlet

Velocity circulation computational formula in the double-blade pump impeller outlet is as following.

$$\Gamma_2 = 2\pi R_2 v_{u2} \tag{1}$$

where v_{u2} is circular velocity component in the impeller outlet, R_2 is impeller outlet radius.

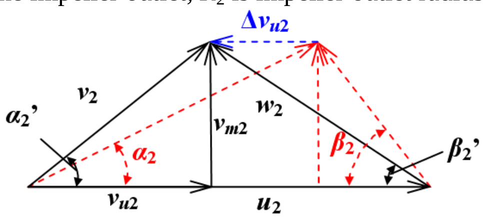


Fig. 3 Velocity triangle of impeller outlet

From velocity triangle of impeller outlet (Fig. 3), we can know the calculation formula of v_{u2}.

$$v_{u2} = \sigma u_2 - \frac{v_{m2}}{\tan\beta_2} \tag{2}$$

$$u_2 = \frac{D_2 \pi n}{60} \tag{3}$$

$$v_{m2} = \frac{Q}{\eta_v \pi D_2 b_2 \psi_2} \tag{4}$$

where σ is slip factor, u_2 is circular velocity in the impeller outlet, v_{m2} is meridional velocity in the impeller outlet, η_v is volumetric efficiency, Ψ_2 is the blockage coefficient of blade outlet.

$$\eta_v = \frac{Q}{Q+q} \tag{5}$$

$$q = \mu f \sqrt{2g\Delta H_m} \tag{6}$$

$$\Psi_2 = 1 - \frac{zS_{u2}}{D_2 \pi} \tag{7}$$

where f is the sectional area of the sealing gap defined as f=2$\pi R_m b$, R_m is radius of the sealing ring, b is sealing gap width, μ is velocity coefficient of sealing gap defined as $\mu = \dfrac{1}{\sqrt{1+0.5\zeta+\dfrac{\lambda l}{2b}}}$, ζ is coefficient of fillet radius with ζ=0.5 to 0.9, λ is hydraulic resistance coefficient with λ = 0.04 to 0.06, l is sealing gap length, ΔH_m is pressure difference over the seal (ΔH_m=0.6H at $n_s \leq 100$, while ΔH_m=0.7H at $n_s \geq 100$), S_{u2} is the circumference thickness of blade outlet.

Velocity circulations in the double-blade pump impeller outlet were calculated with three common slip factors, as shown in Fig. 4. Wherein Stechkin slip factor, Ψ=π/3, R_1 is the radius of impeller blade inlet. It can be seen form Fig. 4 that velocity circulations in the impeller outlet calculated with Weisner slip factor are obviously bigger than that with Stodola slip factor and Stechkin slip factor. Velocity circulation values in the impeller outlet calculated with Stodola slip factor are least. But change rule of velocity circulation is the same. That is velocity circulation in the impeller outlet decreases with the increase of flow rate.

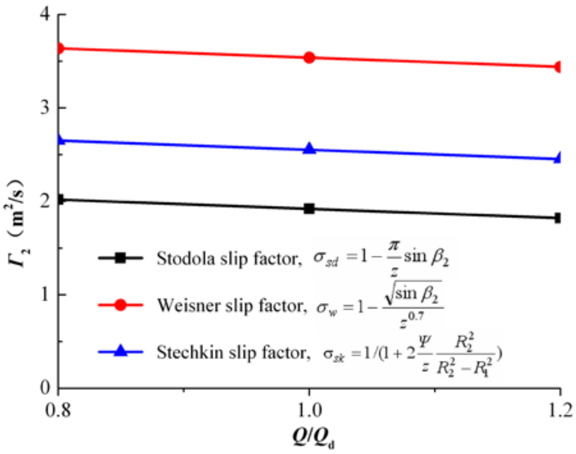


Fig. 4 Comparison with three calculation values

4. Experiment Research of Velocity Circulation in the Impeller Outlet

4.1 PIV System

Fig. 5 shows the PIV system in this experiment. It mainly includes NewWave YAG200-NWL pulse laser, the 610035 laser pulse^{TM} synchronizer, the 630059 power view^{TM} plus 4M PIV camera, Insight 3G, the 610015 light arm, light source lens, the external trigger synchronization system[9], and so on.

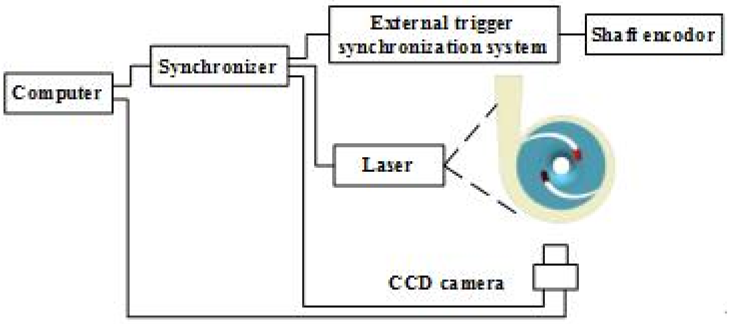


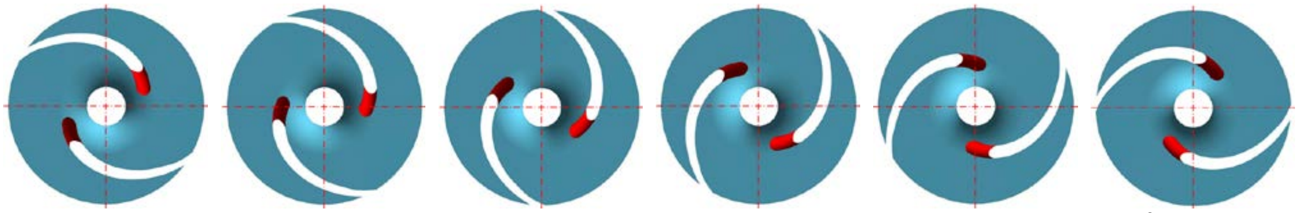
Fig. 5 Sketch of PIV systems

4.2 Measurement Region

The middle section of the impeller is selected as the measurement region, including the impeller passage 1 and passage 2 (Fig. 1). The angle between the blade suction surface near tongue and the horizontal axis θ is 28 °.

4.3 Arrangement of Blade Phase

Six different phase conditions (shown in Fig. 6) are measured, that is, t = 0, T/6, T/3, T/2, 2T/3, 5T/6 with a phase interval being 30°. When t=0, θ is 28°. The phase difference is set through the pulse delay time in the Insight 3G code and the pulse delay time is 5ms if blades rotate 30° at n=1000r/min.



(a) t=0 (b) t=T/6 (c) t=T/3 (d) t=T/2 (e) t=2T/3 (f) t=5T/6

Fig. 6 Sketch of blade phase

In this experiment, Al_2O_3 powders are chosen as tracer particles.

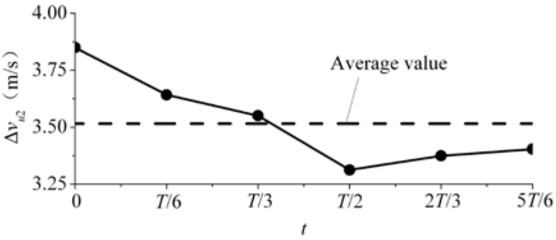
4.4 Data Acquisition and Processing Method

(1)Set related parameters in the Insight 3G software, pulse separation is 400μs, delay time is 100μs, the interrogation window size is 32pixel × 32pixel. Total 100 group instantaneous velocity fields were measured in each condition.

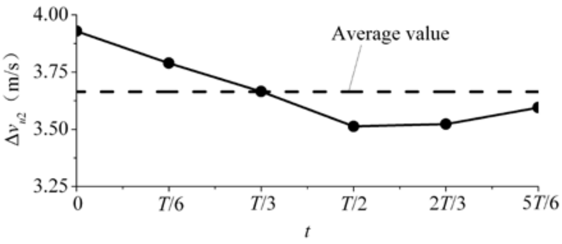
(2) Use cross-correlation technique to process 100 group images in different conditions and amend these images with “standard deviation”, “local mean”, “median test” and “secondary peak” technology.

(3) In order to obtain circular velocity component and slip velocity in the impeller outlet, compile PIV velocity processing program of centrifugal pump by using visual c++ 2010.

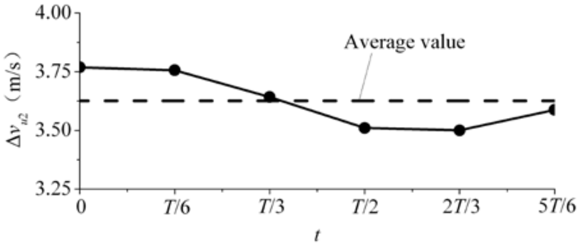
4.5 Experiment Results and Analysis



(a) $0.8Q_d$



(b) $1.0Q_d$



(c) $1.2Q_d$

Fig. 7 Slip velocities at the three conditions

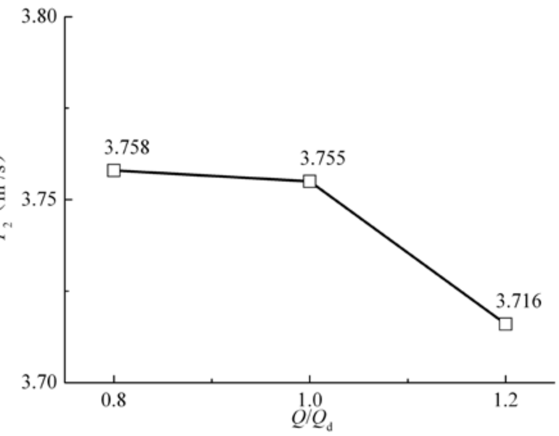


Fig. 8 Average velocity circulation of impeller outlet

Fig.7 shows slip velocities at the three conditions. From 0 to $5T/6$, slip velocity decreases first, and then increases. But slip velocity values of $t=5T/6$ are smaller than that of $t=0$. In order to analyze slip velocities in the impeller outlet, average slip velocities under the six phases are made. From Fig. 7, average slip velocity at the $1.0Q_d$ condition is the largest, namely 3.66m/s.

Fig. 8 indicates the change of average velocity circulation of impeller outlet. It can be seen from this figure that average velocity circulation of impeller outlet decreases with the increase of flow rate. At the $0.8Q_d$ condition, average velocity circulation of impeller outlet is maximum, whose value is 3.758 m^2/s. At the $1.2Q_d$ condition, average velocity circulation of impeller outlet is minimum, whose value is 3.716 m^2/s.

5. Correction and Experimental Verification of Velocity Circulation in the Impeller Outlet

5.1 Correction

Table 2 shows errors between with calculation values and experiment values of velocity circulation of impeller outlet under the three conditions. Calculation values of velocity circulations with Weisner formula is close to the experimental values, the maximum error is - 7.48% at the $1.2Q_d$ condition. Calculation errors of velocity circulation with Stodola formula are the biggest, the maximum error reaches up to -51%. Accordingly, Stodola formula and Stechkin formula are not apply to calculate velocity circulation of double-blade pump impeller outlet.

Table 2 Comparison with calculation values and experiment values

		$0.8Q_d$	$1.0Q_d$	$1.2Q_d$
Experimental value(m^2/s)		3.758	3.755	3.716
Stodola	Calculation value(m^2/s)	2.019	1.920	1.821
	Error(%)	-46.28	-48.87	-51.00
Weisner	Calculation value(m^2/s)	3.636	3.537	3.438
	Error(%)	-3.25	-5.80	-7.48
Stechkin	Calculation value(m^2/s)	2.651	2.552	2.453
	Error(%)	-29.47	-32.04	-34.00

Based on Weisner formula, velocity circulations of impeller outlet are corrected. Correction coefficient k_Γ is as following.

$$k_\Gamma = \Gamma_{2,Experiment} / \Gamma_{2,\sigma_w} \tag{8}$$

So correction coefficient k_Γ under $0.8Q_d$, $1.0Q_d$ and $1.2Q_d$ is 1.0335, 1.0617 and 1.0807, respectively.

5.2 Experimental Verification

In order to verify the feasibility of correction method, a double-blade impeller is made. Its geometrical parameters are shown in table 3. In addition to the geometric parameters in table 3 being different form table 1, other parameters of the impeller are the same.

Table 3 Main geometric parameters of impeller

Description	Parameter	Value
Impeller outlet diameter (mm)	D_2	200
Impeller outlet width (mm)	b_2	47
Blade inlet angle (°)	β_1	15.1
Blade outlet angle (°)	β_2	30
Blade wrap angle (°)	φ	140

Energy performance curve of the new double-blade impeller is shown in Fig. 9. The test results show that the head at the design flow rate is 4.44m and its efficiency is 57.37%.

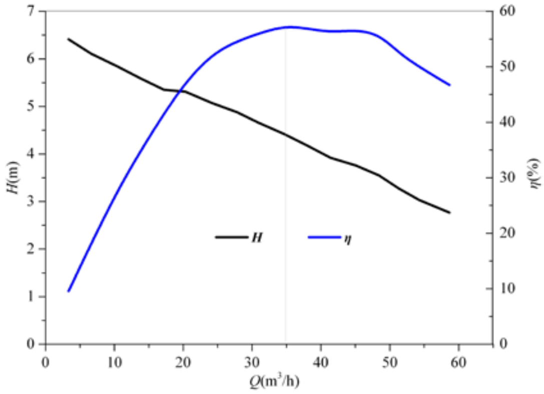


Fig. 9 Energy performance curves of the new double-blade impeller

According to the above correction formula $\Gamma_2 = k_\Gamma 2\pi R_2 v_{u2}$, Velocity circulation of impeller outlet are calculated, which shown in table 4.

Table 4 Correction results

	$0.8Q_d$	$1.0Q_d$	$1.2Q_d$
Experimental value(m^2/s)	3.660	3.625	3.543
Calculation value(m^2/s)	3.412	3.334	3.268
Error(%)	-6.78	-7.85	-7.74
Correction value(m^2/s)	3.526	3.546	3.532
Error(%)	-3.65	-2.16	-0.29

From table 4, calculation errors with the correction formula are greatly reduced, especially at the $1.2Q_d$ condition. At the $0.8Q_d$ condition, corrected calculation error of velocity circulation of impeller outlet is -3.65%, which reduced by 3.13%. At the $1.0Q_d$ condition, corrected calculation error of velocity circulation of impeller outlet is -2.16%, which reduced by 5.69%. At the $1.2Q_d$ condition, corrected calculation error of velocity circulation of impeller outlet is -0.29%, which reduced by 7.45%. Therefore, this correction method is feasible and effective.

6. Conclusion

Velocity circulations of a double-blade pump impeller outlet were calculated with Stodola formula, Weisner formula and Stechkin formula. Results show that calculation values of velocity circulations with Weisner formula is close to the experimental values.

The internal flow of impeller for the double-blade pump were measured with PIV and average velocity circulations at the $0.8Q_d$, $1.0Q_d$ and 1. $2Q_d$ conditions were obtained. Result shows that average velocity circulation of impeller outlet decreases with the increase of flow rate.

Velocity circulations of impeller outlet were corrected. A double-blade impeller is redesigned and its inner flow is measured with PIV. The comparison results show that the maximum corrected calculation error is 3.65%, which greatly reduced than the uncorrected calculation error.

Acknowledgments

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