Design and analysis of a radial diffuser in a single-stage centrifugal pump

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ABSTRACT
Radial diffusers can improve the flow uniformity in pumps and affect the hydraulic performance of centrifugal pumps directly. The diffusion coefficient \( d \) is an important parameter in fluid machinery but it has seldom been used in the diffuser design of single-stage centrifugal pumps. To improve the design method of radial diffuser use in centrifugal pumps, the diffusion coefficient was introduced into the design of radial diffusers based on a single-arc hydraulic design method and it was found that the vane outlet angle, vane outlet thickness and vane number have a significant impact on the design results. A single-stage centrifugal pump with a radial diffuser was selected as the research model. The inner flow was simulated using the commercial computational fluid dynamics (CFD) program CFX and verified by experiment. The results indicate that the head and efficiency of the pump are best when the vane outlet angle is 6°. The flow area decreases and the flow velocity at radial diffuser outlet increase when the outlet thickness is greater than 2 mm. The hydraulic loss is minimum and the head and efficiency are better when the vane number is 8 at different flow rates. So, the optimal range of the diffusion coefficient for the model pump is around 1.6 to 2. The study indicates that it is feasible to design radial diffusers according to the diffusion coefficient.

ARTICLE HISTORY
Received 27 December 2015
Accepted 21 June 2016

KEYWORDS
Centrifugal pump; radial diffuser; diffusion coefficient; energy characteristic; numerical simulation

1. Introduction

The radial diffuser is an important collector in centrifugal pumps and is widely used in large single-stage and multistage pumps. In large single-stage pumps, although the radial diffuser does not have return guide vanes it still can make the impeller outlet flow uniform. Recently, with the rapid development of modern industry, the application of large centrifugal pumps has become more diverse and the use of radial diffusers in large single-stage centrifugal pumps to improve flow uniformity is also more widespread. To some extent, radial diffusers can have a significant impact on the performance of large single-stage centrifugal pumps.

The performance of diffusers has been an important field of research for many years. Reneau, Johnston, and Kline (1967) found that the performance of 2D diffusers is greatly affected by the inlet conditions. High recovery occurs at high area ratios (up to 5) and the minimum head loss occurs at \( 2\theta < 7 \) deg.

Much attention is still devoted to research on radial diffusers (Lugovaya, Olshtynsky, Rudenko, & Tverdokhleb, 2012; F. Shi, 2001). Goto and Zangeneh

Nomenclature

- \( \beta_1 \): Inlet angle of impeller (°)
- \( \beta_2 \): Outlet angle of impeller (°)
- \( \beta_3 \): Inlet angle of radial diffuser (°)
- \( \beta_4 \): Outlet angle of radial diffuser (°)
- \( b_2 \): Outlet width of impeller (mm)
- \( b_3 \): Inlet width of radial diffuser (mm)
- \( b_4 \): Outlet width of radial diffuser (mm)
- \( D_2 \): Outlet diameter of impeller (mm)
- \( D_3 \): Inlet diameter of radial diffuser (mm)
- \( D_4 \): Outlet diameter of radial diffuser (mm)
- \( H \): Head (m)
- \( n \): Rotational speed (r/min)
- \( n_s \): Specific speed (\( n_s = 3.65 n \sqrt{Q/H^{3/4}} \))
- \( N \): A positive integer
- \( P \): Total pressure (Pa)
- \( Q \): Flow rate (m³/h)
- \( T \): Vane outlet thickness of radial diffuser (mm)
- \( V \): Absolute velocity (m/s)
- \( Z_B \): Vane number of impeller
- \( Z_R \): Vane number of radial diffuser

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(2002) presented a new approach to optimizing a pump diffuser based on a three-dimensional inverse design method and computational fluid dynamics (CFD). Other researchers designed a twisted return guide vane for a submersible multistage pump, the results of their work demonstrating that return guide vanes with a twisted inlet can reduce the flow loss in radial diffusers. The results show that the pressure changes little at the helix section along the direction of the flow then increases more and more at the diffuser section and hits its maximum at the end of the diffuser section (W. D. Shi; Zhang, & Lu, 2006; J. C. Zhang & Li, 2006). Zhou, Shi, Lu, Xu, and Wang (2011) optimized the guide vane by using an orthogonal test method and found that the wrap angle and inlet angle of the radial diffuser have a greater impact on the pump head and efficiency. The results indicate that the three-dimensional guide vane has a smaller hydraulic loss and a higher pressure conversion capacity. And at the same time, they studied the performance of the deep-well centrifugal pump with the new designed guide vane (Zhou, Shi, & Lu, 2011).

Liu, Cui, Tan, Wu, and Xu (2013) studied the effect that the relative position between the guide vane and the volute tongue has on the characteristics of a centrifugal pump and found that the head and efficiency are at maximum when the angle between the guide vane and tongue is about 20°. Hou, Zhang, Li, Zhou, and Wang (2014) simulated the full flow in a centrifugal pump and found that pre-whirl regulation at the inlet of the guide vane can improve the hydraulic performance of the pump at off-design conditions. In order to study the effect of the outlet diameter of radial diffusers on the performance of centrifugal pumps, Jiang, Li, and Liu (2014) simulated the flow in three pumps and found that the performance of a pump with a diffuser of smaller outlet diameter is better under all operating conditions.

Recently, CFD has been widely used to research pump performance. Huang, Mou, and Zheng (2012) optimized the match between the guide vane outlet and the impeller inlet in a multistage centrifugal pump using CFD and found that the static pressure is highest when the outlet angle of the guide vane is 90°. Alemi, Nourbakhsh, Raisee, and Najafi (2015) used CFD to study the effect of casings and found that the triple-volute is the most appropriate volute geometry under off-design conditions.

Many studies have also been carried out on radial diffusers in multistage pumps. Some studies presented a numerical investigation of the water flow in the first stage of a two-stage centrifuge pump (Segala et al., 2011; Stel et al., 2013). Q. H. Zhang et al. (2013) designed a circumferential crankle guide vane based on the twisted centrifugal impeller design method to match the compact structure of multistage pumps.

Experimental method was widely used into researches of internal flow for verifying the reliability of the numerical simulation. Sinha and Katz (1999) adopted particle image velocimetry measurements to identify the unsteady flow structures and turbulence in a centrifugal pump with a vane diffuser.

The diffusion coefficient \( d \) is an important parameter in both fluid machinery and pump design. Although many studies have investigated the design of radial diffusers for pumps with respect to the diffusion coefficient \( d \), most of them focus on the radial diffusers in multistage pumps, which have return guide vanes. In the present work, a design method for a radial diffuser without return guide vanes for single-stage centrifugal pumps was constructed and studied. First, the diffusion coefficient was introduced into the design of a radial diffuser without return guide vanes and the single-arc hydraulic design method was established. Next, the effects that important parameters – such as the vane outlet angle, vane outlet thickness and vane number of the radial diffuser – have on the performance of centrifugal pumps were analyzed using numerical methods. Finally, the design method was verified by experiments. The research in this paper provides preliminary reference to the diffuser design according to the diffusion coefficient \( d \).

2. The design of radial diffusers

The structure of a single-stage centrifugal pump with a radial diffuser is similar to that of a centrifugal compressor. The effect of a radial diffuser on a single-stage centrifugal pump is similar to the effect that a vane diffuser has on a centrifugal compressor. In compressors, the vane diffuser is considered to be a ring cascade that transfers most of the kinetic energy at the inlet impeller into pressure energy, and the diffusion coefficient is an important parameter for the vane diffuser design. For incompressible fluid, the diffusion coefficient is defined by the area ratio \( A_2/A_1 \) or the velocity ratio \( V_1/V_2 \) between the inlet and outlet of the vane diffuser. Radial diffusers in single-stage centrifugal pumps also reduce the velocity and increases the pressure, so the diffusion coefficient was introduced to evaluate the performance of the radial diffuser. The diffusion coefficient is defined as:

\[
   d = \frac{V_3}{V_4},
\]

where \( V_3 \) is the inlet velocity of the radial diffuser and \( V_4 \) is the outlet velocity of the radial diffuser. According to this definition, the bigger the diffusion coefficient, the better the performance of the radial diffuser.
One-dimensional theory was used to deduce the diffusion coefficient of radial diffusers for single-stage centrifugal pumps. The velocity triangles at the impeller outlet and the inlet and outlet of the radial diffuser are shown in Figure 1, and the deduction process is detailed below.

The circumferential velocity $U_2$ at the impeller outlet is:

$$U_2 = \frac{\pi n D_2}{60}.$$  

(2)

The meridional velocity $V_{m2}$ at the impeller outlet is:

$$V_{m2} = \frac{Q_d}{\pi D_2 b_2 \varphi_2},$$  

(3)

where $\varphi_2$ is the blockage factor at the impeller outlet.

The circumferential component of the absolute velocity $V_{U2}$ at the impeller outlet is

$$V_{U2} = U_2 - \frac{V_{m2}}{\tan \beta_2} = \frac{\pi n D_2}{60} - \frac{Q_d}{\pi D_2 b_2 \varphi_2 \tan \beta_2},$$  

(4)

The meridional velocity $V_{m3}$ at the diffuser inlet is:

$$V_{m3} = \frac{Q_d}{\pi D_3 b_3 \varphi_3},$$  

(6)

where $\varphi_3$ is the blockage factor at the diffuser inlet and can be written as:

$$\varphi_3 = 1 - \frac{b_3 Z_R}{D_3 \pi \sin \beta_4}.$$  

(7)

The absolute velocity $V_3$ at the diffuser inlet is:

$$V_3 = \sqrt{V_{U3}^2 + V_{m3}^2} = \sqrt{\left(\frac{\pi n D_2}{60D_3} - \frac{Q_d}{\pi D_3 b_2 \varphi_2 \tan \beta_2}\right)^2 + \left(\frac{Q_d}{\pi D_3 b_3 \varphi_3}\right)^2}. $$  

(8)

The meridional velocity $V_{m4}$ at the diffuser outlet is:

$$V_{m4} = \frac{Q_d}{\pi D_4 b_4 \varphi_4},$$  

(9)

where $\varphi_4$ is the blockage factor at the diffuser outlet.

The absolute velocity $V_4$ at the diffuser outlet is:

$$V_4 = \frac{V_{m4}}{\sin \beta_4} = \frac{Q_d}{\pi D_4 b_4 \varphi_4 \sin \beta_4},$$  

(10)

where $\beta_4$ is the outlet angle of the radial diffuser.

Based on with Equations (8) and (10), the diffusion coefficient $d$ can be written as:

$$d = b_4 \varphi_4 \left(\frac{D_4}{D_3}\right) \sin \beta_4 \times \sqrt{\left(\frac{\pi^2 n D_2^2}{60 Q} - \frac{1}{b_2 \varphi_2 \tan \beta_2}\right)^2 + \left(\frac{1}{b_3 \varphi_3}\right)^2},$$  

(11)

As shown in Equation (11), the diffusion coefficient $d$ of the radial diffuser is related to the design parameters of centrifugal pumps and the structural parameters of the impeller and radial diffuser, so it can be used as the design criterion for the radial diffuser.

In order to display the relations between $\beta_4$, $b_4$, $Z_R$ and $d$ directly, based on Equation (7), Equation (11) can be written as:

$$d = b_4 \varphi_4 \left(\frac{D_4}{D_3}\right) \sin \beta_4 \times \sqrt{\frac{\pi^2 n D_2^2}{60Q} - \frac{1}{b_2 \varphi_2 \tan \beta_2}} + \frac{1}{b_3 \varphi_3} - \frac{D_3 \pi \sin \beta_3}{b_3^2 Z_R},$$  

(12)

where $t$ is defined as the vane outlet thickness instead of $b_4$.

Through careful analysis of Equation (12), it can be seen that the vane outlet angle $\beta_4$ and the blockage factor at the outlet $\varphi_4$ of the radial diffuser have a significant impact on the diffusion coefficient $d$. So, the vane outlet angle, the vane outlet thickness and the vane number that influences the blockage factor at the vane outlet are meticulously investigated in this paper.

### 3. The validation model

#### 3.1. The parameters and structure of the model

A single-stage, single-suction centrifugal pump with a radial diffuser was used as the research model to validate
the above design method. The model pump consists of an impeller, a radial diffuser and a volute. The hydraulic structure is shown in Figure 2.

The flow rate of the model pump $Q = 32\, m^3/h$, the head $H = 7.5\, m$, the efficiency $\eta = 75\%$ and the rotating speed $n = 1450\, r/min$, with the specific speed $n_s = 110$. The main geometric parameters of the model pump are summarized in Table 1.

### 3.2. The design of the new radial diffuser

Keeping the outlet and inlet diameters of the original radial diffuser constant and using the diffusion coefficient as the design criterion, the single-arc hydraulic design method for the radial diffuser (as shown in Figure 3) is presented based on the traditional method. In this new method, the pressure side and suction side are two different sides of the radial diffuser are designed to be a single arc which can easily control the vane shape. Also, this new method can make the design and production of the radial diffuser more fast and convenient.

### 4. Numerical simulation

Numerical simulation was used to study the effect of the vane outlet angle $\beta_4$, the vane outlet thickness $t$ and the vane number $Z_R$.

First, Pro/E, a commercial 3D software, was used to build and assemble the computational domains. As shown in Figure 4, the whole computational domain consists of a suction chamber, a radial diffuser, an impeller, a volute and an outlet pipe. The outlet pipe can reduce the effect of the boundary conditions on the inner flow and its length is four times that of the pump outlet diameter.

Second, Gambit, a mesh generating software, was used to generate the mesh of the computational domain. The suction chamber, radial diffuser and outlet domain were generated as hexahedral cells, while the impeller and volute were generated as a tetrahedral mesh due to their complex structure. To check the mesh size independence, six grid topologies were generated. The head and efficiency under the design flow rate were used as the criteria.

![Figure 2. Hydraulic structure of the model pump.](image)

![Figure 3. Single-arc design method for the radial diffuser.](image)

![Figure 4. 3D model of the flow field calculation domain.](image)
and the simulation results (Table 2) show that the maximum difference of the heads is just 0.16% and that of the efficiencies is just 0.25%. Therefore, taking the calculation resources and CFD post-processing into consideration, 4 was selected as the number for the mesh size.

Finally, CFX, a commercial computational fluid dynamics code, was applied to perform a steady numerical simulation of the internal flow in the model pump and the re-normalisation group $k$-$\varepsilon$ model was selected as the turbulence model. The impeller was set to be the rotating domain and its speed was set to 1450 r/min with the other domains left static. There were four domain interfaces in total and the interface model was set to be a general connection. The total pressure was used as the inlet boundary condition and the inlet pressure was 1 atm. The outlet was set to be a mass flow and the velocity was calculated by the flow rate. The wall boundary was set to be a no-slip boundary and the roughness of all surfaces was set as 0.025 mm.

5. Results and analysis

5.1. The effect of the vane outlet angle on performance

Six radial diffusers with different vane outlet angles were designed using the single-arc hydraulic design method. The vane outlet angle $\beta_4$ of the radial diffuser was set as $3^\circ$, $6^\circ$, $10^\circ$, $15^\circ$, $25^\circ$ and $40^\circ$ and the diffusion coefficient $d$ was set as 1.02, 2.05, 3.40, 5.07, 8.27 and 12.85, respectively according to Equation (12). The internal flow in the model pump under $0.6Q_d$, $0.8Q_d$, $1.0Q_d$, $1.2Q_d$ and $1.4Q_d$ was simulated and the prediction performance curves are shown in Figures 5 and 6.

It can be seen from Figure 5 that for different values of the vane outlet angle $\beta_4$ the head is obviously different under the same flow rate. With an increase in flow rate the difference of the head curves becomes greater. At the design flow rate, the maximum head difference is 1.83 m. Under all operating conditions, the head is at maximum when the vane outlet angle $\beta_4$ is $6^\circ$ and the head is at minimum when the $\beta_4$ is $40^\circ$. The head is close to the design requirement when the vane outlet angle $\beta_4$ is at $6^\circ$ at the design flow rate.

Figure 6 shows the efficiency curves of the model pump under different radial diffuser vane outlet angles $\beta_4$. The shape of each efficiency curve is very similar and the maximum of each efficiency curve is close to the $1.0Q_d$ condition, but under the same flow rate the efficiency of each curve has significant differences. With an increase in flow rate the efficiency difference between the curves becomes greater, with a maximum efficiency difference of 14.5% at the design condition. Under all operating conditions, the efficiency is at maximum when $\beta_4 = 6^\circ$ and minimum when $\beta_4 = 40^\circ$.

The hydraulic loss in the radial diffuser under different vane outlet angles is shown in Figure 7. The hydraulic loss in the diffuser can be obtained by $\Delta h = (P_{out} - P_{in})/(\rho g)$, where $P_{out}$ is the outlet pressure and $P_{in}$ is the inlet pressure. For all hydraulic loss curves under different vane outlet angles $\beta_4$ it can be seen that the hydraulic loss gradually decreases as the flow increases from $0.6Q_d$ to $1.0Q_d$. From $1.0Q_d$ to $1.2Q_d$, the hydraulic loss decreases when $\beta_4$ is greater than or equal
Figure 7. Hydraulic loss in the radial diffuser for different vane outlet angles $\beta_4$. 

To $15^\circ$ but increases when $\beta_4$ is less than $15^\circ$. It can also be observed that the hydraulic loss always increases when the vane outlet angle $\beta_4$ is too small ($3^\circ$) or too big ($40^\circ$). The explanation for this is that smaller values of $\beta_4$ result in a greater dynamic loss while bigger values of $\beta_4$ lead to a greater diffusion loss. At the design flow rate, the hydraulic loss is at minimum when the vane outlet angle $\beta_4 = 6^\circ$.

At the design flow rate, the distribution of absolute velocity in the middle plane of the model pump is shown in Figure 8. The velocity at the impeller outlet is at maximum due to the rotation of the impeller. The diffusion coefficient $d$ is just 1.02 when the vane outlet angle $\beta_4 = 3^\circ$. In this case, the velocity at the radial diffuser outlet is high and the flow field is very smooth. When the vane outlet angle $\beta_4$ is $6^\circ$ or $10^\circ$, the velocity at the radial diffusion outlet decreases as the diffusion coefficient $d$ increases, and the flow field is still good. When the vane outlet angle $\beta_4$ is greater than $10^\circ$ there are a lot of low-velocity areas in the radial diffuser and volute with an increase in the diffusion coefficient $d$. Under these circumstances, unsteady flow occurs easily, such as flow separation and back flow.

At the design condition, the total efficiency in the pump and the integral hydraulic loss in the radial diffuser and the volute are shown in Figure 9. As can be seen, the total hydraulic loss decreases as the diffusion coefficient increases from 1.02 to 2.05, then gradually increases as it changes from 2.05 to 8.27, and finally sharply increases beyond the value of 8.27. The total efficiency change in the pump corresponds to the integral hydraulic loss in the radial diffuser and volute.

When the diffusion coefficient is 1.02, the friction loss in the diffusers is large because of the narrow channel and the high outlet velocity of the radial diffuser. When the diffusion coefficient is 2.05, the radial diffuser plays a role in reducing the velocity and increasing the pressure, which leads to a decrease in the flow rate in the volute. Consequently, the hydraulic loss decreases and the efficiency increases at the same time. With future increases in the diffusion coefficient, the diffusion of the radial diffuser channel gets bigger and a low-velocity area appears on the suction surface and in the volute diffuse section, meaning that the hydraulic loss increases and the efficiency decreases. As the diffusion coefficient continues to increase, a vortex occurs in the radial channel and the efficiency decreases sharply. To summarize, there is a suitable range for the diffusion coefficient $d$. The radial diffuser loses the effect of reducing the velocity and increasing the pressure if $d$ is too small, and the diffusion and flow loss increase if it is too large. Based on the

Figure 8. Velocity distribution in the pump middle plane at the design flow rate for: (a) $\beta_4 = 3^\circ$, (b) $\beta_4 = 6^\circ$, (c) $\beta_4 = 10^\circ$, (d) $\beta_4 = 15^\circ$, (e) $\beta_4 = 25^\circ$, and (f) $\beta_4 = 45^\circ$.

Figure 9. Total efficiency in the pump and the integral hydraulic loss in the radial diffuser and the volute at the design condition.
research model, it is best for the diffusion coefficient $d$ to stay around a value of 2 for different vane outlet angles.

According to the above analysis, it can be concluded that a vane outlet angle $\beta_4$ of $6^\circ$ is the best for the model pump.

### 5.2. The effect of the vane outlet thickness on performance

Based on a vane outlet angle $\beta_4$ of $6^\circ$, four radial diffusers were designed with a vane outlet thickness $t$ of 2 mm, 5 mm, 8 mm and 10 mm and a diffusion coefficient $d$ of 1.63, 1.19, 0.76 and 0.54, respectively according to Equation (12).

The internal flow in the pump was simulated for the four different radial diffusers under $0.6Q_d$, $0.8Q_d$, $1.0Q_d$, $1.2Q_d$ and $1.4Q_d$. The predicted curves of head and efficiency are shown in Figures 10 and 11.

As is shown in the Figure 10, although the values of vane outlet thickness $t$ are different, the head on each curve gets smaller as the flow rate increases. However, under the same flow rate, the head shows significant differences for different head curves, especially under larger or smaller flow rates. At $1.4Q_d$, the head difference is at maximum and at about 1.7 m. Under all operating conditions, the head is at maximum when the vane outlet thickness $t = 2$ mm and decreases as the vane outlet thickness $t$ increases and the diffusion coefficient $d$ decreases.

Figure 11 shows the efficiency curves for different values of vane outlet thickness $t$. It can be seen that the best efficiency occurs close to $1.0Q_d$ for each efficiency curve. At $0.6Q_d$ the efficiency difference between these curves is the smallest, about 3%, and the difference becomes gradually greater with an increase in flow rate. At $1.4Q_d$, the efficiency difference is about 20% and from $0.8Q_d$ to $1.4Q_d$, the efficiency is highest when the vane outlet thickness $t = 2$ mm and it decreases as the vane outlet thickness $t$ increases.

The hydraulic loss in the radial diffuser under different values of vane outlet thickness $t$ is presented in Figure 12. It can be seen that from $0.6Q_d$ to $1.0Q_d$ the hydraulic loss in the radial diffuser with different values of $t$ gradually decreases, but if the flow rate increases further then the hydraulic loss increases at the same time. At a small flow rate, the main hydraulic loss in the radial diffuser is diffusion loss and this increases as the vane outlet thickness $t$ decreases. At a large flow rate, the dynamic loss increases, which leads to an increase in the total hydraulic loss, and the dynamic loss increases as the vane outlet thickness increases. It was found that the hydraulic loss is at minimum when the vane outlet thickness $t$ is 2 mm from $0.8Q_d$ to $1.4Q_d$.

At the design flow rate, the velocity distribution in the middle pump plane is presented in Figure 13. It can be clearly seen that there are low-velocity zones in the volute and radial diffuser when the vane outlet thickness $t = 2$ mm. With an increase in the vane outlet thickness $t$, the diffusion coefficient $d$ decreases and the area

![Figure 10. $H$-$Q$ curves for different vane outlet thicknesses $t$.](image1)

![Figure 11. $\eta$-$Q$ curves for different vane outlet thicknesses $t$.](image2)

![Figure 12. Hydraulic loss in the radial diffuser for different vane outlet thicknesses $t$.](image3)


5.3. The effect of the vane number on performance

Based on the above results, six radial diffusers with different vane numbers were designed with the same vane outlet angle $\beta_4 = 6^\circ$ and vane outlet thickness $t = 2\, \text{mm}$. The vane number was 6, 7, 8, 9, 11 and 12 and the diffusion coefficient $d$ was 1.72, 1.67, 1.63, 1.56, 1.46 and 1.41, respectively based on Equation (12).

The internal flow in the pump with the different radial diffusers was simulated for $0.6Q_d, 0.8Q_d, 1.0Q_d, 1.2Q_d$ and $1.4Q_d$. The predicted head and efficiency curves are shown in Figures 15 and 16. It can clearly be seen from the Figure 15 that the head on each curve gets smaller as the flow rate increases. When the vane number is 7 or 8, the heads are close to each other and all are higher than the other. The head with 6 vanes is lowest. Under all flow rates, the head difference between the maximum and minimum is 0.8 m. In Figure 15, it can be seen that the head under all flow rates decreases if the vane number $Z_R$ is too big or too small.

The pump efficiency curves under different vane numbers are shown in Figure 16. It can be seen that the
difference in efficiency is small at the design flow rate and that the maximum difference is 2%. With the flow increasing or decreasing, the difference in efficiency increases for off-design conditions. At 0.6Q_d, the efficiency gradually increases as the vane number increases and the maximum difference is 6%. At 1.4Q_d, the efficiency is at maximum for Z_R = 7 and at minimum for Z_R = 12, and the maximum difference is 8%. A vane number that is too great or too small results in reduced efficiency. From 0.9Q_d to 1.1Q_d, the best Z_R value for the model pump is 8.

Figure 17 presents the hydraulic loss curves in the radial diffuser for different vane numbers. It can be seen that as the flow rate increases from 0.6Q_d to 1.0Q_d, the hydraulic loss on each curve decreases gradually, but if the flow rate increases further, the hydraulic loss increases. At a low flow rate, the main hydraulic loss is diffusion loss in the radial diffuser, which it increases as the vane number decreases. As the flow rate and vane number Z_R increase, the diffusion coefficient d decreases, and the friction loss and dynamic loss increase as the diffusion loss decreases. Therefore, there is a minimum for hydraulic loss; it is clear that the hydraulic loss is lowest under all flow rates when the vane number Z_R is 8.

The velocity distribution in the pump’s middle plane at the design flow rate is shown in Figure 18. It can be seen that there is an obvious low-velocity zone in the radial diffuser when the vane number Z_R is less than 11 and the flow velocity is small. The low-velocity zone in the radial diffuser results in a good pressure rise, but there is a large diffusion loss due to the big diffusion coefficient. When the vane number Z_R is greater than 9, the low-velocity zone in the radial diffuser disappears, which means that the total velocity is very large and this leads to a large amount of friction loss and dynamic loss. So, the above theory analysis of Figure 17 agrees well with the flow field analysis of Figure 18.
The total efficiency in the pump and the integral hydraulic loss in the radial diffuser and the volute at the design condition are shown in Figure 19. It can be seen that the hydraulic loss decreases as the diffusion coefficient $d$ increases and reaches its minimum when $d = 1.63$, while the efficiency changes in accordance with the hydraulic loss. The diffusion coefficient $d$ is large when the vane number is low, which results in an increase in diffusion loss, while the whole hydraulic loss is small because the friction loss is small. The flow becomes better as the vane number $Z_R$ increases but the friction loss also increases. Therefore, the hydraulic loss of the pump as the vane number increases (and the diffusion coefficient $d$ decreases). Thus, it is concluded that the best diffusion coefficient is about 1.6 for different vane numbers.

In summary, based on the presented investigations, the best vane number for this model is 8.

6. Experimental verification

6.1. Experimental system

According to the numerical simulation results, a new radial diffuser was designed with a vane outlet angle $\beta_A$ of 6°, a vane outlet thickness $t$ of 2 mm and a vane number $Z_R$ of 8. The structures of the new and old radial diffusers are shown in Figure 20. It can be seen that the outlet angle and outlet thickness of the new radial diffuser all get smaller and the diffusion parts become longer, which means that more kinetic energy can be transformed into pressure energy. In this case, the friction loss of the pump decreases as the velocity decreases.

In order to verify the design, the new radial diffuser (Figure 21) was constructed using rapid prototyping manufacturing and used in the experiment. The model pump and test rig are shown in Figures 22 and 23.

As can be seen, the pump was tested in a closed test rig which consisted of the model pump, a torque meter, a motor, a pressure transfer, turbine flow meters, a pressurizer tank, a cavitation tube, a vacuum pump, piping and valves. Considering that the material of impeller is polymethylmethacrylate, whose strength is weaker compared with metal, the inverter was used to start the pump and gradually increase the speed until the rated speed was achieved. The shaft torque was measured using a torque meter with an accuracy of ±0.10%. A pressure transfer was used to measure the static pressure values at the inlet and outlet of the pump, again to an accuracy of ±0.10%. The flow rate was measured by a turbine flow meter to an accuracy of ±0.14%.

6.2. Comparisons between the simulated and experimental results

Figure 24 shows the performance of the model pump with the new radial diffuser. It can be seen that the head increases about 1 m under all operating conditions. The efficiency also increases to different extents, increasing 8% on average at low flow rates and 5% on average at high flow rates. The power of the pump with the new radial diffuser is less than that of the old radial diffuser at low flow rates and greater at high flow rates. The increase of the head and the efficiency results in a decrease in hydraulic loss. In summary, the experiment results indicate that it is feasible and practical to design a radial diffuser using the diffusion coefficient $d$. 
7. Conclusions

In this paper, a single-stage, single-suction centrifugal pump with a radial diffuser was used as the research model. The diffusion coefficient \( d \) was applied to design a radial diffuser based on a single-arc hydraulic method and the effect of the outlet angle, outlet thickness and vane number were investigated via CFD numerical simulation. An experiment was then conducted to verify the simulation results and the following conclusions are drawn:

1. The diffusion coefficient of the radial diffuser and the single-arc hydraulic design method have been defined. The outlet angle, outlet thickness and vane number of the radial diffuser are the main parameters that affect pump performance.

2. The head and efficiency under different outlet angles \( \beta_4 \) are clearly different. Under all operating conditions, the head and efficiency are at maximum when the outlet angle \( \beta_4 = 6^\circ \) and at minimum when \( \beta_4 = 40^\circ \). The difference of the head and the efficiency increase as the flow rate increases, the difference of the head and the efficiency are 1.83 m and 14.5%, respectively at the design condition. The radial diffuser cannot reduce the velocity and increase the pressure when \( \beta_4 < 6^\circ \). The diffusion level of the radial diffuser channel increases when \( \beta_4 > 6^\circ \), which can often result in the appearance of a vortex.

3. Under all operating conditions, the head and the efficiency are at maximum when the outlet thickness \( t = 2 \) mm. The values of the head and the efficiency decrease as the flow rate increases. At \( 1.4Q_d \), the head when \( t = 10 \) mm is about 26.5% of that when \( t = 2 \) mm, and the efficiency is only about 19%. As the outlet thickness increases, the diffusion coefficient increases and the hydraulic loss increases, so the pump efficiency decreases.

4. The highest head difference is 0.8 m under different vane numbers. The head and efficiency decrease as the vane number of the radial diffuser \( Z_R > 8 \) or \( Z_R < 8 \). At low flow rates, the diffusion loss in the channel of the radial diffuser is great and at high flow rates, the friction loss increases. The hydraulic loss in the diffuser and the volute is lowest when the vane number of the diffuser is 8.

5. In summary, the optimal range of the diffusion coefficient is around 2.0 for different outlet angles, and from 1.6 to 1.7 with different outlet thicknesses and vane numbers. Thus it is concluded that the best range of the diffusion coefficient for the model pump is from 1.6 to 2.0.

Disclosure statement

No potential conflict of interest was reported by the authors.
Funding

This work was supported by the National Natural Science Foundation of China [grant numbers 51509109 and 51309119]; the Foundation of Jiangsu Province [grant numbers BY2014123-07, ZBZZ-040, and BE2014116]; and the Priority Academic Program Development of Jiangsu Higher Education Institutions (PAPD).

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