Effects of radial diffuser hydraulic design on a double-suction centrifugal pump

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Abstract. In order to study effects of radial diffuser on hydraulic performance of crude oil pump, the steady CFD numerical method is applied and one large double-suction oil pump running in long-distance pipeline is considered. The research focuses on analysing the influence of its diffuser vane profile on hydraulic performance of oil pump. The four different types of cylindrical vane have been designed by in-house codes mainly including double arcs (DA), triple arcs (TA), equiangular spiral line (ES) and linear variable angle spiral line (LVS). During design process diffuser vane angles at inlet and outlet are tentatively given within a certain range and then the wrapping angle of the four types of diffuser vanes can be calculated automatically. Under the given inlet and outlet angles, the linear variable angle spiral line profile has the biggest wrapping angle and profile length which is good to delay channel diffusion but bring more friction hydraulic loss. Finally the vane camber line is thickened at the certain uniform thickness distribution and the 3D diffuser models are generated. The whole flow passage of oil pump with different types of diffusers under various flow rate conditions are numerically simulated based on RNG $k$-$\epsilon$ turbulent model and SIMPLEC algorithm. The numerical results show that different types of diffusers can bring about great difference on the hydraulic performance of oil pump, of which the ES profile diffuser with its proper setting angle shows the best hydraulic performance and its inner flow field is improved obviously. Compared with the head data from model sample, all designed diffusers can make a certain improvement on head characteristic. At the large flow rate conditions the hydraulic efficiency increases obviously and the best efficiency point shift to the large flow rate range. The ES profile diffuser embodies the better advantages on pump performance which can be explained theoretically that the diffuser actually acts as a diffusion device and is good to transform the dynamic energy to pressure energy. Then through the hydraulic loss analysis of each pump component for all diffusers, it shows that the impeller takes up the biggest part of the whole loss about 8.19% averagely, the radial diffuser about 3.70% and the volute about 1.65%. The hydraulic loss of impeller is dominant at the large flow rate while the radial diffuser is at the small flow rate. Among all diffusers, the ES profile diffuser generates the least loss and combined to the distribution of velocity vector and turbulent kinetic energy for two kinds of diffusers it also shows that ES profile is fit to apply in radial diffuser. This research can offer a significant reference for the radial diffuser hydraulic design of such centrifugal pumps.

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1. Introduction
Large double-suction oil pump, as one key device in long distance oil pipeline, has widely application for its centrifugal structure with large flow rate, stable running operation and convenient maintenance. Until now, most of the running large double-suction oil pumps online in domestic still depend on import, and not only its high price cost, long delivery time and inconvenient maintenance, but its average efficiency index beyond domestic over 2.5% [1]. In 2015 the ministry of industry and information technology has declared to launch and start the project of Special Action Strong Industrial Base which required improving the control abilities to the key technologies of energy equipment, and thus the study of large double-suction oil pump possesses great value in future.

As the core component in centrifugal pump, the impeller has been researched in many aspects so far. Bi Zhigao [2] numerically studied how blade profile influence centrifugal oil pump and the results indicated that increasing wrapping angle of blade will steepen the head vs. flow rate curve and under the same flow rate condition the head becoming lower, moreover, such characteristic is not effected by medium viscosity. Chen Xiaoling [3] experimentally studied the influence of geometrical parameters of impeller on oil pump in transferring viscous oil and showed that large blade outlet angle can promote transporting ability and less blade number is good to transport high viscous crude oil. By contrast to another important component is radial diffuser, and also as a key component in pump with dual characteristics of suction chamber and pressure chamber, the diffuser generates large hydraulic loss for high velocity in flow channel and thus proper design will increase pump performance and decrease hydraulic loss. Mostly the investigations in open literatures mainly adopt experiment and simulation method to study the influence of radial diffuser on pump performance. Experimental test considers more the hydraulic loss in diffuser and measure its inner flow field and numerical simulation focuses on rotor-stator interaction between impeller and diffuser and rotating stall phenomenon. Ubaldi [4] conducted experiments to study the radial gap between impeller and diffuser on unsteady flow. Shi [5] numerically investigated pressure fluctuations in the entire stage of a diffuser pump due to the interaction between impeller and diffuser vanes. The vane number configuration on the unsteady flow by Feng [6] due to impeller-diffuser interaction in radial diffuser pumps were studied that decreasing diffuser vane number can lead to an increase in pressure fluctuation. Torbergsen [7] and Sano [8] researched the mechanism of rotating stall phenomena in diffuser with numerical simulation.

Inspired by the design methods of cylindrical pump blades [9, 10], the authors in this study totally have designed four kinds of radial diffusers with different camber lines using self-own program and they are double arcs (DA), triple arcs (TA), equiangular spiral line (ES) and linear variable spiral line (LVS). During the design process the inner and outer diameters of radial diffuser still keep unchanged and by changing the inlet angle and outlet angle of vane to make good pump performance. Considering vane number on flow passage diffusion, this study at the last discusses how vane number influence pump hydraulic performance, and in this part the relative circumferential mid-positon between radial diffuser and volute tongue is fixed and the total expelling ratio is constant by adjusting vane thickness distribution.

2. Research model description
The research sample model in this paper is one large double-suction oil main-pump from Shanlan-Lanzhou long distance oil pipeline. This pump is characterized of centrifugal volute consisting of impeller, radial diffuser and volute, and the main characteristic parameter in transferring pure water are rotating speed \(n\) of 2980r/min, design flow rate \(Q_d\) of 2806m³/h and head \(H\) of 139m. The main geometric sizes of corresponding components are listed in Table 1. According to viscous conversion relationship from KSB company [11] and when dynamic viscosity of crude oil \(\mu\) is 150mPa * s, the correction ratio of flow rate \(C_Q\) is 0.998, of head \(C_H\) is 0.975 and of efficiency \(C_E\) is 0.88, and therefore the hydraulic performance of pump in transferring pure water can be used to obtain that in transferring crude oil.
Table 1. Main geometrical size of pump components (Units: Length-mm; angle-°)

| Impeller       | Value | Radial diffuser | Value | Volute       | Value |
|----------------|-------|-----------------|-------|--------------|-------|
| Inlet diameter | 255   | Inlet diameter  | 448.4 | Base circle diameter | 750   |
| Outlet diameter| 444.4 | Outlet diameter | 738   | Tongue angle | 29    |
| Blade number   | 5     | Inlet width     | 92    |              |       |
| Outlet Width   | 87    | Outlet width    | 99    |              |       |
| Wrapping angle | 135   | Vane number     | 8     |              |       |

3. Hydraulic design of radial diffuser

The radial diffuser in double-suction pump usually adopts cylindrical style which is common to see in low specific speed centrifugal pump. The basic element profiles are arc line and spiral line. In this study the DA profile, TA profile, ES profile and LVS profile are generated to construct radial diffusers and by equipped them to investigate how different profiles influence hydraulic performance of double-suction pump. The arc and spiral style of diffuser profiles are designed as follows.

Figure 1. Arc line style profile.

Figure 2. Spiral line style profile.

Figure 1 illustrates the design process of arc line style profile. Taken one arc in multi-arcs as example, MN is one basic arc and its centre is point O and the radius is $r_0$. The centre of cylindrical diffuser is point O and the inner and outer radius are $r_1$ and $r_2$ respectively. Point P is one moving point in profile. $\beta_1$ and $\beta_2$ are inlet and outlet angle.

\[
OO^2 = r_0^2 + r_1^2 - 2r_0r_1 \cos \beta_1 \tag{1}
\]

\[
OO^2 = r_0^2 + r_2^2 - 2r_0r_2 \cos \beta_2 \tag{2}
\]

By solving simultaneous equation (1) and equation (2) can obtain $r_0 = (r_2^2 - r_1^2) / (2(r_2 \cos \beta_2 - r_1 \cos \beta_1))$. Thus the generating process can be considered as like this: Rotating point O of angle $\beta_1$ in clockwise to get point O' and the distance between O' and M is length of $r_0$ at symmetrical axis. Point P is obtained by rotating point M in anticlockwise of angle $\theta$ and finally the setting angle $\beta$, radius $r$, profile length and wrapping angle $\varphi$ can be easily calculated by point-by-point integration method.

Figure 2 shows how spiral line style profile draws. In polar coordinates system the spiral equation can be expressed as in equation (3).

\[
r = re^{\theta \tan \alpha} \tag{3}
\]

Here $\theta$ is still wrapping angle at point P. ① When $\alpha$ is constant from 0° to 90° , equation (3) represent equiangular spiral line equation and boundary conditions: $\theta=0$, $r=r_1$; $\theta=\varphi$, $r=r_2$. It is to find that the setting angle $\beta$ is equal to $\alpha$ and in the design process $\beta$ is set as $\beta=0.5(\beta_1 + \beta_2)$ with the corresponding wrapping angle of $\varphi=\ln(r_2/r_1)/\tan \beta$. ② When the setting angle is $\beta=a\theta+b$, \[
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equation (3) represent linear variable angle spiral line equation and boundary conditions: \( \theta = 0, \beta = \beta_1; \theta = \varphi, \beta = \beta_2 \).

\[
\varphi = \left( \ln \left( \frac{r_2}{r_1} \right) / \ln \left( \cos \beta_1 / \cos \beta_2 \right) \right) \left( \beta_2 - \beta_1 \right), a = \left( \beta_2 - \beta_1 \right) / \varphi, b = \beta_1
\]

\[
r = r_1 \left( \cos \beta_1 / \cos \left( a \theta + b \right) \right)^{\frac{1}{\alpha}}
\]

(4)

It is worth mentioning that as for different profiles of radial diffuser, they must be matched with proper inlet angle and outlet angle which promote good hydraulic performance of pump. That is to say that under the same inlet angle and outlet angle, all designed radial diffuser with its profile cannot match the pump well. Therefore after constant try and error, the proper angle for each kind of diffuser has been obtained as shown in Table 2. The theoretical analysis tells that once the radius and angle at inlet and outlet for each diffuser are given, the setting angle at each point in profile line, profile length and vane wrapping angle are all fixed. From Table 2 it reads that the inlet angle of diffuser is around 18° and the outlet angle differs not much. The ES profile has the smallest wrapping angle and the shortest profile length.

### Table 2. Proper parameters for each radial diffuser

| Items            | DA  | TA  | ES  | LVS |
|------------------|-----|-----|-----|-----|
| Inlet angle \( \beta_1^{\circ} \) | 18  | 18  | 24.5| 18  |
| Outlet angle \( \beta_2^{\circ} \) | 26  | 22  | 24.5| 20  |
| Wrapping angle \( \varphi^{\circ} \) | 69.54| 77.73| 62.64| 82.90|
| Profile length /m | 0.376 | 0.417 | 0.320 | 0.421 |

Figure 3 and figure 4 show the distribution of vane angle of diffusers and their profile camber lines. At the certain given angle and diameter of each diffuser profile, the vane angle of the ES profile line from inlet to outlet keeps constant while others gradually grow and LVS profile changes very slowly, by contrast, the vane angle of DA profile changes not quite uniformly which may induce flow separation at the exit of its first arc. Figure 4 also tells that from the profile line of DA to LVS, the flow channel length becoming increasing and this is good to delay channel diffusion and good to realize the stable conversion of dynamic energy to pressure energy, however quite long channel length will bring about extra hydraulic loss.

**Figure 3.** Distribution of setting angle with radius.

**Figure 4.** Different diffuser profile lines.

### 4. Numerical methods and model description

As shown in figure 5, the main computational domain of the pump consists of six parts including inlet extent pipes, impeller, radial diffuser, volute casing and outlet extent pipe. Then the commercial software of ANSYS Meshing is used to generate unstructured tetrahedral grid considering the complexity and skewness of pump components shown in figure 5(b). After this, the solver of ANSYS Fluent 14.0 is used to calculate inner fields for steady,
viscous and incompressible flow. The RNG $k$-$\varepsilon$ model and SIMPLEC algorithm were applied to solve RANS equations. The velocity is applied for inlet boundary condition by assuming that velocity at inlet cross-section is uniform. The turbulent intensity and hydraulic diameter are 5% and 145mm respectively. The outflow is used for outlet boundary condition. The non-slip condition is given at solid walls. The Multiple Reference Frame model is applied to take into account the interaction between stationary diffuser and rotating impeller with interface pairs. The standard wall function based on the logarithmic law has been used. PRESTO! Scheme is used for pressure term and second order upwind discretization scheme is used for convection terms. The convergence criterion of numerical simulation is set as a residual of $1e^{-4}$.

Before the real numerical simulation and in order to reduce the computational time and improve the reliability of calculation accuracy, the grid independence under the design operating condition has been investigated. The number of each component keeps at the relative constant proportion compared to the whole grid. And during this process, the characteristic performance of head and hydraulic efficiency are used to evaluate grid effect on final solution. The investigation displays that the head and hydraulic efficiency become stable with the total grid number till 3.04 million cells and thus this refining mesh scheme is considered not to generate value fluctuation. Then the mesh with 3.04 million cells is adopted for the next calculations.

After the inner fields of pump are simulated, the characteristic parameters and the corresponding hydraulic loss efficiency of each component can be defined as follows. 

\[ H = \left( \frac{p_{\text{out}} - p_{\text{in}}}{\rho g Q} \right) \]  
\[ \eta_h = \left( \frac{\rho g Q}{M \omega} \right) \times 100\% \]  
\[ \Delta \eta_{h-i} = \left( \frac{P - (p_{\text{im-out}} - p_{\text{im-in}}) Q}{P} \right) \times 100\% \]  
\[ \Delta \eta_{h-r} = \left( \frac{p_{\text{rd-in}} - p_{\text{rd-out}}) Q}{P} \right) \times 100\% \]  
\[ \Delta \eta_{h-v} = \left( \frac{p_{\text{vo-in}} - p_{\text{vo-out}}) Q}{P} \right) \times 100\% \]  

Here in the equations above, $p_{\text{in}}$ and $p_{\text{out}}$ are total pressure of pump at inlet and outlet section, $P$ is shaft power, $M$ is torque and $\omega$ is angular velocity. $p_{\text{im-out}}$, $p_{\text{rd-in}}$ and $p_{\text{vo-in}}$ are total pressure at inlet section of impeller, radial diffuser and volute. $p_{\text{im-out}}$, $p_{\text{rd-out}}$ and $p_{\text{vo-out}}$ are total pressure at outlet section of impeller, radial diffuser and volute. $\eta_h$ is hydraulic efficiency of pump. $\Delta \eta_{h-i}$, $\Delta \eta_{h-r}$ and $\Delta \eta_{h-v}$ are hydraulic loss efficiency of impeller, radial diffuser and volute.

5. Results and discussions

5.1. The influence of diffuser profiles on hydraulic performance of double-suction pump

Figure 6 displays the hydraulic performance of double-suction pump equipped with different profiles of radial diffusers under its own proper inlet angle and outlet angle. Figure 6 (a) shows the head vs. flow rate curves and it reads that compared with the head data from model sample, all designed
diffusers can make a certain improvement on head. The head vs. flow rate curve with DA profile diffuser is very close to that with model design diffuser because the measured data from the sample is quite similar to DA profile. The ES profile diffuser contributes more on head vs. flow rate curve and offers the highest head value in all flow rate conditions.

Figure 6 (b) shows the hydraulic efficiency vs. flow rate curves. Totally to say, all designed radial diffusers can improve the hydraulic efficiency of double-suction pump at the large flow rate conditions and the best efficiency point shift to the large flow rate range compared to that the pump data with model sample diffuser. Just like the head vs. flow rate curve, the hydraulic efficiency vs. flow rate curve with DA profile diffuser is very close to that with model design diffuser. In contrast to model design data, the efficiency curves of pump equipped with TA, ES and LVS profile diffusers all performs better and their best efficiency points shift to the left but are still inclined to the right side compared with the data of model sample. The ES profile diffuser generate the best hydraulic efficiency as well as head and this can be explained theoretically that the diffuser actually acts as a diffusion device conversing the dynamic energy to pressure energy just like a volute and the fluid flowing through keeps the constant velocity moment under no external force applying on and without considering the viscosity of fluid. Besides, that the high efficiency area of pump hydraulic performance with ES profile diffuser cannot reach and cover that with model sample diffuser may because the unreasonable design of volute casing seen in Ref. [12, 13] and by decreasing the throat area is able to effectively shift the best efficiency point to small flow rate range.

5.2. The hydraulic loss efficiency analysis of each component for double-suction pump

In order to quantitatively evaluate the hydraulic loss of each component in double-suction pump and according to the definitions in part 4, the corresponding hydraulic loss efficiency has been calculated out under the designed condition $Q_d$ listed in Table 3.

| Items  | Impeller\(\Delta \eta_{\text{im}}\) (%) | Radial diffuser\(\Delta \eta_{\text{rd}}\) (%) | Volute\(\Delta \eta_{\text{vo}}\) (%) | Total \(\Delta \eta\) (%) |
|--------|----------------------------------|---------------------------------|-----------------|-----------------|
| DA     | 8.32                             | 4.20                            | 1.80            | 14.32           |
| TA     | 8.13                             | 3.25                            | 1.85            | 13.23           |
| ES     | 8.03                             | 3.28                            | 1.56            | 12.87           |
| LVS    | 8.27                             | 4.06                            | 1.41            | 13.74           |

It reads that among the four pumps with different radial diffusers, generally the hydraulic loss efficiency of the impeller takes up the biggest part of 8.19% averagely, the radial diffuser of 3.70% and the volute takes up about 1.65%. Thus in a sense, the radial diffuser offers a potential ability for reducing hydraulic loss for pump and by improving the geometry of diffuser can realize redistribution of inner flow fields of the total flow passage. Furthermore, among the four different radial diffusers
under its own proper inlet and outlet angle, the ES profile diffuser accounts for the least hydraulic loss and the corresponding hydraulic loss efficiencies of each component are also the least of about 8.03%, 3.28% and 1.56. Compared with DA profile diffuser, the component hydraulic loss is decreased obviously by 1.45%, which proves that ES profile is fit to apply in radial pressure diffusion and use in double-suction pump as diffuser profile.

5.3. The inner flow fields of radial diffuser under different flow conditions

The figure 7 presents the distribution of velocity vector at mid-span of DA and ES profile diffusers under three flow rate conditions. It can be seen that the velocity at inlet of diffuser is big and then gradually decrease which indicates that the diffuser can effectively converse the dynamic energy to pressure. With the flow rate increasing, the inner flow improves to some extent. However, there are some obvious flow separations in flow channel of DA profile diffuser while it is rarely to see in ES profile diffuser. Therefore under different flow rate conditions, the ES profile diffuser can better adapt the flow but it is also not so well at the big flow rate condition.

Figure 7. The distribution of velocity vector at mid-span of diffuser.

Figure 8 shows the distribution of turbulent kinetic energy at mid-span DA and ES profile diffusers under three flow rate conditions. It reads from this figure that totally where the flow separation is more serious, the turbulent kinetic energy is quite higher which implies that the flow loss in this area is quite bigger. At the flow rate of 0.8 \( Q_d \) condition, the flow separation happens seriously in each channel just like the velocity vector distribution in figure 7. In general, the mass weighted average value of turbulent kinetic energy \( k \) for DA profile diffuser is bigger than that for ES profile diffuser and this exactly indicates that the distribution of turbulent kinetic energy for ES profile diffuser is more reasonable. However there is not good under the flow rate of 1.2 \( Q_d \) which mainly because the thickness distribution is uniform from leading to trailing edge.

Figure 8. The distribution of turbulent kinetic energy at mid-span of diffuser.

Table 4. The hydraulic loss efficiency of each component with ES diffuser (%)

| Items | Impeller | Radial diffuser | Volute | Total |
|-------|----------|-----------------|--------|-------|
| \( \Delta \eta_{\text{in}} \) | \( \Delta \eta_{\text{im}} \) | \( \Delta \eta_{\text{rd}} \) | \( \Delta \eta_{\text{vo}} \) | \( \Delta \eta \) |
| 0.8\( Q_d \) | 8.96 | 6.79 | 1.53 | 17.28 |
| 1.0\( Q_d \) | 8.03 | 3.28 | 1.56 | 12.87 |
| 1.2\( Q_d \) | 9.80 | 3.43 | 1.62 | 14.85 |

What’s more, the hydraulic loss efficiency of each component equipped with ES profile diffuser under three different flow rate conditions is presented in Table 4. Under the three flow rate conditions, the hydraulic loss proportion of volute casing is varied not much with the average level of 1.57% while the impeller and radial diffuser changes obviously. The hydraulic loss of impeller is dominant at large flow rate condition and radial diffuser at small flow rate condition. Taking the radial diffuser for granted at the small flow rate condition, the flow separations in flow channels and the inlet incidence phenomenon is quite evident compared with the other two conditions and this is easy to tell in figure7.
6. Conclusions
As the important device in long-distance pipeline for crude oil, the double-suction pump and its component radial diffuser have been focused on in this study by the steady CFD numerical simulation. The diffuser profile is chosen as the key point to investigate and some conclusions can be drawn as follows.

① The four kinds of diffuser profiles are designed by in-house codes consisting of double arcs, triple arcs, equiangular spiral line and linear variable angle spiral line. And after constant try and error, the proper angles for each profile are given.

② Compared with the model pump, all designed diffusers can make improvement on pump hydraulic performance while the best efficiency point shifts to the large flow rate range.

③ Through the hydraulic loss analysis of each component, the impeller takes up the biggest part of the whole loss about 8.19%, the radial diffuser about 3.70% and the volute about 1.65. The hydraulic loss of impeller is dominant at the large flow rate while the radial diffuser at the small flow rate.

④ Combined to the distribution of velocity vector and turbulent kinetic energy for DA and ES profile diffusers, the ES profile is fit to apply in radial diffuser.

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References
[1] Wang Y 2012 Analysis of blade inlet edge design parameters variation impact on performance of double suction pump, MS thesis, Lanzhou University of Technology.
[2] Bi Z H 2014 Influence of blade profile on performance of centrifugal oil pump, Journal of Lanzhou University of Technology. 40 62.
[3] Chen X H, Zhang W G 1999 Influence of geometric parameters of impeller on centrifugal pump performance in transferring viscous oil, CPM. 27(2) 19.
[4] Ubaldi M 1994 Nonintrusive measurements of the unsteady flow in the radial gap between centrifugal impeller and vaned diffuser, the 12th Symposium on Measuring Techniques for Transonic and Supersonic Flow in Cascades and Turbomachines.
[5] Shi F, Tsukamoto H 2001 Numerical study of pressure fluctuations caused by impeller-diffuser interaction in a diffuser pump stage, Journal of Fluids Engineering. 123 466.
[6] Feng J J, Friedrich-Karl B and Hans J D 2007 Numerical investigation on pressure fluctuations for different configurations of vaned diffuser pumps, International Journal of Rotating Machinery.
[7] Torbergsen E A 1998 Impeller/Diffuser interaction forces in centrifugal pumps, Doctor thesis, Norwegian University of Science and Technology.
[8] Sano T 2000 Alternate blade stall and rotating stall in vaned diffuser-part1: effects of impeller/diffuser clearance, Trans. Jpn. Soc. Mech. 66 650.
[9] Xu H Y 1990 Theoretical research on profile of circular arc blade, Fluid Machinery. 8 7.
[10] Xu H Y 1989 Draw method of variable spiral line for cylindrical blade, Fluid Machinery. 7 6.
[11] KSB Aktiengsllschaft 1990 Centrifugal pump lexicon, 3rd Edition. 344.
[12] Kim J H, Lee H C and Kim J H 2015 Design techniques to improve the performance of a centrifugal pump using CFD, Journal of Mechanical Science and Technology. 29 215.
[13] Yang S, Kong F and Chen B 2011 Research on pump volute design method using CFD, International Journal of Rotating Machinery.