Flow Characteristics in Volute of a Double-Suction Centrifugal Pump with Different Impeller Arrangements

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Abstract: As an important type of centrifugal pump, the double-suction pump has been widely used due to its high efficiency region and large flow rate. In the present study, the complex flow in volute of a double-suction centrifugal pump is investigated by numerical simulation using a re-normalization group (RNG) $k$-$\varepsilon$ model with experimental validation. Axial flows are observed in volute near the impeller outlet and compared with four staggered angles. The net area-weighted average axial velocities decrease as the staggered angle increases. The axial flows are mainly caused by the different circumferential pressure distribution at the twin impeller outlet. The dominant frequencies of the axial velocities for different staggered angles are $f_{BP}$ and its harmonic. The pressure fluctuations in most regions of the volute are obtained by superimposing the pressure generated by the two impellers.

Keywords: double-suction pump; axial flow; staggered angle; pressure difference; pressure fluctuation

1. Introduction

Double-suction pumps are an important type of centrifugal pump and are used for large flow rate. Double-suction pumps have two suction chambers. Due to the back-to-back arrangement of the twin impellers, the average axial force in double-suction pumps is zero. However, these pumps usually involve high energies and suffer from more complex pressure fluctuations than other centrifugal pumps [1,2]. The inlet fluid is separated by the two suctions and flows into impellers, where the two fluids are independent of each other. Because there is no partition in the middle of the volute, the flow from the two impellers will mix in the volute. This makes the pressure distribution in the volute of the double-suction pump uneven.

With the development of computational fluid dynamics, more and more researchers predicted the internal flow and pressure fluctuation in pumps by numerical simulation. New calculation models and numerical methods have been developed and verified [3–6]. The results from numerical simulation can reveal detailed fluid information in impeller and volute, including pressure and streamline distributions, which are difficult to obtain in experiments. Liu et al. [7,8] investigated the effects of different tip clearance sizes in a mixed flow pump. As the tip clearance increased, the development of
a leakage vortex was observed. For different tip clearances, the trajectory of leakage vortex remained in the direction along the blade suction side. The results show that, as tip clearance size increased, the intensity of the leakage vortex and the separation between the other leakage vortex was enhanced. Liu et al. [9,10] investigated the influence of the inlet guide vanes of a centrifugal pump using a re-normalization group (RNG) $k$-$\varepsilon$ turbulence model. The flow patterns and pressure fluctuations with 2D and 3D inlet guide vanes were studied at different flow rates for various angles.

For a centrifugal pump with vaneless volute, the non-uniform flow from the impeller becomes more unsteady in the volute with fluctuations associated with the blade-passing frequency and its harmonics [11,12]. The amplitude of the fluctuations usually reaches the maximum value at the tongue region. The huge energy loss makes the flow in volute more complex and uncontrollable than that in the impeller. Tan et al. [13,14] analyzed the complicated flow field in the pump at part-load condition. The dominant frequencies for the pressure fluctuations on monitoring points in volute were compared with non-cavitation and cavitation conditions. The maximum amplitudes in serious cavitation conditions were twice that of those in non-cavitation conditions. Kye et al. [15] investigated the flow losses caused by unsteady impeller-volute interactions in volute, especially in the off-design condition. In this condition, some high-pressure fluid from the volute outlet re-entered into the volute near the volute tongue. This pressurized fluid resulted in a high adverse pressure gradient on the blade pressure side. In addition, the leakage near the volute tongue was induced by a high-pressure gradient. Barrio et al. [16] used a verified numerical model to study the flow pulsations caused by blade-tongue interaction at different flow rates. The leakage flow evolution was obtained.

Due to double impellers and volutes, double-suction pumps suffer from not only complex flow in ordinary centrifugal pumps, such as secondary flow and reverse flow, but also some special flow phenomena. González et al. [17,18] analyzed the inner flow to discuss the design parameters of a double-suction pump. The unsteadiness of impeller inlet and volute were studied in terms of both transient and average pressure. Also, the flow characteristics in the turbine mode were explored. Kyung et al. [19] investigated the velocity and pressure distributions in the impeller in a double-suction centrifugal pump at different flow rates with an impeller-only model and a full pump model. Fernández et al. [20] studied the effect of inlet nonuniformities by comparing the features of the flow patterns near the volute tongue. The jet-wake structure was analyzed. The results show that the nonlinear interaction between the impeller and the volute tongue can be quantified by an unsteady component. Yao et al. [21] investigated the influence of impeller types on pressure fluctuations in double-suction centrifugal pumps. The results show that the impeller with staggered blades can reduce pressure fluctuation in volute. The impeller with splitter blades can decrease the pressure fluctuations in volute and weaken the pressure fluctuation in the suction chamber at rated flow rate. Zou et al. [22] analyzed the evolution of impeller radial force in a double-suction pump during startup. The numerical results show the relationship of the maximum radial force value between startup and rated condition. The evolution of radial force during startup was predicted by a transient formula. Rodriguez et al. [23] carried out theoretical analysis of radial force in a qualitative way. The relationship between the number of blades and guide vanes and dominant frequency was analyzed. The vibration was interpreted as a result of the modulation in the amplitudes of the interactions. Spence et al. [24,25] investigated the effect of various geometrical features on the pressure fluctuations and performance characteristics of a double-suction pump. The investigation covered four geometric parameters. The results show that the vane arrangement had a noticeable effect on performance characteristics. The head and power are reduced slightly from an inline to a staggered arrangement. The staggered arrangement decreased the pressure fluctuations at the impeller blade’s tip and shroud locations at all flow rates. The vane arrangement had a great effect on the pressure fluctuations at the leakage flow and volute locations. Song et al. [26] studied the radial force and pressure fluctuation in a double-suction centrifugal pump under different staggered angles. The results show that the amplitude of radial force fluctuation is reduced as the staggered angle increases. The phase angle of the radial force vector in
different cases is equal to the staggered angle. In addition, the staggered arrangement can suppress
the amplitude of pressure fluctuation in the mid-span of the volute.

Previous studies on double-suction pumps show that the staggered impeller can suppress the
pressure and radial force fluctuations. However, the staggered arrangement will also result in different
pressure distribution at the outlet of the twin impeller. This difference in pressure will cause axial
flow in the pump, which does not exist in other types of centrifugal pump. Thus, the staggered
impeller-induced axial flow and pressure fluctuation in the pump should be investigated. In the
present article, the systematic investigation of flow characteristics in the pump is conducted for four
double-suction impellers with different staggered angles, and the effect of axial flow on pressure
fluctuation is analyzed.

2. Physical Model and Computational Domain

2.1. Geometry and Operation Condition of Pump

The investigated pump in the present work is a double-suction centrifugal pump, with two twin
impellers built together with six blades. Figure 1 shows the pump in experimental measurement. The main design parameters of the pump are listed in Table 1.

![-tested double-suction centrifugal pump](image)

**Figure 1.** Tested double-suction centrifugal pump.

**Table 1.** Parameters for the pump.

| Description                  | Parameter | Value |
|------------------------------|-----------|-------|
| Rated flow rate (m³/h)       | $Q_n$     | 864   |
| Rated head (m)               | $H$       | 17.8  |
| Rotational speed (r/min)     | $n$       | 1480  |
| Impeller period (s)          | $T$       | 0.04054 |
| Specific speed (-)           | $n_s$     | 215.9 |
| Number of impeller blades (-) | $z$       | 6     |
| Diameter of impeller inlet (mm) | $D_1$   | 198   |
| Diameter of impeller outlet (mm) | $D_2$  | 286   |
| Diameter of inlet straight pipe (mm) | $D_{in}$ | 300   |
| Diameter of outlet straight pipe (mm) | $D_{out}$ | 250 |

The entire flow domain in the computational model includes the inlet straight pipe, double-suction
chambers, twin impellers, double volute and outlet straight pipe, as shown in Figure 2a. The rotation
angle between the two impellers is defined as staggered angle $\alpha$, shown in Figure 2b. Four different
impeller arrangements are investigated in the present work, including an inline arrangement ($\alpha = 0^\circ$)
and three specific position staggered arrangements ($\alpha = 10^\circ$, $\alpha = 20^\circ$, $\alpha = 30^\circ$), named as case0, case10,
case20 and case30. For all cases, impeller1 remains unchanged, and impeller 2 is rotated according to the staggered angle $\alpha$. The sketch of the pump is shown in Figure 2c.

2.2. Computational Mesh of Centrifugal Pump

A 3-D computational model is established for the pump and then discretized into meshes. The structured mesh for impellers is generated using CFX-Turbogrid 14.5. The unstructured meshes for volute and suction are generated using ICEM 14.5. The volute mesh is refined near the tongue region. The meshes of impellers, suction and volute are shown in Figure 3.

Figure 2. The geometry of the pump. (a) The computational domain; (b) Staggered angle; (c) Sketch of the pump.

Figure 3. Mesh of different parts in pump. (a) Impellers mesh of case0; (b) Suction mesh; (c) Volute mesh.
3. Numerical Method and Validation

Commercial software ANSYS CFX 17.0 was used to simulate the inner flow of the pump by solving the Reynolds-Averaged Navier–Stokes equations. A turbulence of RNG $k$-$\varepsilon$ model was utilized in the solution of the motion equation. This turbulence model can predict the rotating and curvature flow in the flow passage components accurately, which is validated by several previous works [8–11,27]. Total pressure is set as inlet boundary condition and mass flow at outlet is set as the outlet boundary condition. A no-slip wall was applied to all pump walls. The frozen-rotor and transient-rotor stator interface options were modelled on the interfaces between rotating and stationary frames for steady and transit calculations, respectively. High-resolution discretization was used for the convective term. The time-dependent term scheme was the second order back Euler. The convergence criterion is defined as that the root-mean-square residual for all equations at the end of each time step is below $1 \times 10^{-5}$ [28,29].

Mesh number has an influence on the results of numerical simulation. In the present work, five sets of meshes were used to validate the grid independence, as shown in Table 2. The computed results show that the relative error of pump head $\Delta H/H_1$ and efficiency $\Delta \eta/\eta_1$ is less than 0.002. Considering computational cost and accuracy, Mesh 3 was chosen for the following calculations.

Table 2. Mesh sizes for mesh independency test.

| Item           | Mesh 1 | Mesh 2 | Mesh 3 | Mesh 4 | Mesh 5 |
|----------------|--------|--------|--------|--------|--------|
| Inlet pipe     | 141777 | 141777 | 141777 | 141777 | 141777 |
| Outlet pipe    | 90068  | 90068  | 90068  | 90068  | 90068  |
| Impeller       | 531036 | 1013796| 1492992| 2018520| 2500380|
| Volute         | 1511542| 1511542| 1511542| 1511542| 1511542|
| Suction        | 1161816| 1161816| 1161816| 1161816| 1161816|
| Whole passage  | 3436239| 3918999| 4398195| 4923723| 5405583|
| $H/H_1$        | 1      | 1      | 0.99956| 0.99834| 0.99678|
| $\eta/\eta_1$ | 1      | 1      | 0.99849| 0.99849| 0.99849|

Three time steps were used to validate the time independence, including $\Delta t = 4.2230 \times 10^{-4}$ s, $\Delta t = 2.1115 \times 10^{-4}$ s, $\Delta t = 1.0557 \times 10^{-4}$ s (equivalently 16, 32, 64 time steps per blades passage). Figure 4 shows the histories of pressure on point V1, V2 and V3 for three time steps, which is near the volute tongue, near the middle and outlet of the volute. As shown in Figure 4, except for $\Delta t = 4.2230 \times 10^{-4}$ s, the results do not vary significantly. Finally, the time step $2.1115 \times 10^{-4}$ s was chosen for the present work.

A comparison of the CFD predictions with the experimental measurements [26] of head and efficiency are shown in Figure 5 at eight different flow rates. The performance curves coincide well in a wide region, which reveals the accuracy and reliability of the numerical method. The relative difference of efficiency at design flow rate is below 2%, and it is below 5% at off-design conditions. The error may be attributed to strong recirculation and complex backflow. For the head, a good agreement is indicated at low flow rate, whereas the relative error is a little larger at high flow rates, which may be caused by increased leakage near the shroud of the impellers. According to the above analysis, the accuracy and reliability of the numerical method is validated.
4. Results and Discussion

When the staggered angle is set to a different value, the circumferential pressure distributions on the middle plane of the twin impeller outlet do not match. This pressure difference will result in a mixture effect in the volute. This phenomenon is investigated in detail in the following section.

4.1. Flow Pattern at Impeller Outlet

Figure 6 shows the pressure distribution on the outlet of the impellers at rated flow rate. The upper one is impeller1, and the lower one is impeller2.
As shown in Figure 6, the pressure circumferential distribution on the outlet of the impellers is uneven. In each channel, the pressure decreases from the pressure side of one blade to the suction side of the other blade. For case0 in Figure 6a, the staggered angle is zero, and the pressure distributions of impeller1 and impeller2 outlet surfaces are symmetrical. When the staggered angle becomes nonzero, the pressure distributions for the impeller1 and impeller2 outlet surfaces become unsymmetrical, as shown in Figure 6b–d. For any position on the impeller partition on the outlet, the pressure from the upper impeller and lower impeller are different. The pressure discontinuity on the twin impellers outlet surfaces are generated, and a small disturbance is induced to make the pressure uniform. Pressure differences in the axis direction are observed as shown by the arrows. In each impeller channel, two directions of pressure differences are detected, positive Z and negative Z. These pressure differences will result in axial flow. In order to reflect the axial flows near the impeller outlet surface, two sections are specially selected to ensure that the velocity direction of flow on two sections is the same for all cases, as shown by the dotted line in Figure 6. Section1 is on the left, and section2 is on the right. To further investigate the disturbance near the impeller outlet, streamline diagrams are drawn in two sections in impellers and volutes.

Figure 7 shows the streamlines and pressure distribution near the impeller outlet on section1 and section2 for different cases. A local enlargement of the impeller outlet is shown on the right-hand side. For case0, the pressure distribution on section1 and section2 is nearly symmetrical, and the interaction streamlines in the middle of the two impellers are represented by the straight line. For case10, case20 and case30, the direction of the axial in the volute is different. A small radial flow is detected from the outlet of impeller2 to impeller1 in section1, with a noticeable downward skew of the streamlines in the middle of the two impellers. In section2, the radial flow is from the outlet of impeller1 to impeller2, with a noticeable upward skew of the streamlines in the middle of the two impellers.

The tiny perturbation between the two pressure zones mixes the fluid from different impellers. In this way, the pressure in the higher-pressure zone decreases while the pressure in the lower-pressure zone increases. This change will affect the pressure distribution in the volute and affect the pressure distribution in the impeller. The black rectangle in Figure 6 compares the pressure distribution on the outlet surface of impeller1 for different cases. The position of impeller1 remains unchanged for all cases. As shown in the Figure 6, the high-pressure zone clearly reduces from case0 to case30. As the staggered angle increases, the pressure distribution at the impeller outlet becomes more uniform. This is mainly caused by the mixture of fluid in the volute.
4.2. Axial Flow in Volute

Figure 8 shows the instantaneous axial velocities in the z direction at the middle-span plane of the volute at $t = 0.92$ s. For case0, the axial velocity in the volute is nearly zero; this phenomenon agrees well with the results obtained from the streamline diagrams in Figure 7. Obvious axial velocity can be observed for case10, case20 and case30. Positive and negative axial velocities are detected near each channel outlet, which is consistent with the results in Section 4.1. From case10 to case30, the magnitude of axial velocities gradually increases, and the maximum and minimum axial velocities in the mid-span is $1.03$ and $-1.15$ m/s for case10, $0.99$ and $-1.26$ m/s for case20, and $1.41$ and $-1.69$ m/s for case30. In comparison to the average radial velocity $3.04$ m/s at the impeller outlet, the order of magnitude of the two velocities are almost the same.

When the staggered angle is not zero, the distribution of velocity and pressure for the two impellers outlet are different, as shown in Figure 6. When the fluid from the two impellers flows into the volute, the pressure difference induces axial mixture in the volute. The pressure difference will be smaller in locations away from the impeller outlet. As a result, there are only axial velocities near the outlet of the impellers. The closer to the impeller outlet, the larger the axial velocity.

Figure 9 shows the evolution of net area-weighted average axial velocities in one revolution for different cases. The magnitude of the velocity is the largest in case30 and decreases in case20 and case10. For all the three cases, there are obviously six periods of velocity fluctuation in one impeller revolution. This demonstrates that the axial flow is closely related to the blade rotation.
Figure 8. The instantaneous axial velocities at middle-span plane of volute at $t = 0.92$ s. (a) Case0; (b) Case10; (c) Case20; (d) Case30.

Figure 9. Evolution of the area-weighted average axial velocities in one impeller revolution.
A Fast Fourier transform is conducted to investigate the frequency and amplitude of the velocity. Table 3 shows the frequencies and corresponding amplitudes of axial velocities for different cases. The blades passing frequency of the problem is \( f_{BP} = \frac{n \times z}{60} = 148 \) Hz. The frequencies of the axial velocities for case10, case20 and case30 are \( f_{BP} \) and its harmonic. There are no obvious dominate frequency and amplitude. The amplitudes of the first three frequencies are basically the same.

| Item   | Frequency (Hz) | Amplitude (m/s) | Frequency (Hz) | Amplitude (m/s) | Frequency (Hz) | Amplitude (m/s) |
|--------|----------------|-----------------|----------------|-----------------|----------------|-----------------|
| case10 | 148            | \( 2.50 \times 10^{-5} \) | 296            | \( 2.74 \times 10^{-5} \) | 444            | \( 7.53 \times 10^{-5} \) |
| case20 | 148            | \( 2.10 \times 10^{-5} \) | 296            | \( 1.67 \times 10^{-5} \) | 444            | \( 1.56 \times 10^{-5} \) |
| case30 | 148            | \( 1.24 \times 10^{-5} \) | 296            | \( 4.13 \times 10^{-5} \) | 444            | \( 1.42 \times 10^{-5} \) |

As shown in Figure 8, at one particular time, there are both positive and negative velocities on the middle-span of the volute. In order to analyze the characteristic of axial velocities in volute, the instantaneous area-weighted average velocity at the middle-span plane of volute are decomposed into positive and negative Z directions. Figure 10 shows the evolution of the decomposed velocities in the positive Z direction in one revolution: “pos” means the axial velocities in positive Z direction, and “neg” means the axial velocities in negative Z direction. For each case, the positive velocity is smaller than the negative velocity. This is the reason why the axial velocities are negative in Figure 9. The positive velocity increases from case10 to case30. The negative velocity also increases from case10 to case30. This result agrees with the conclusion of Figure 8. As with the axial velocities, the frequencies of decomposed axial velocities for case10, case20 and case30 are also \( f_{BP} \) and its harmonic.

Figure 10. Evolution of the decomposed axial velocities in one revolution for different cases.

4.3. Pressure Fluctuation in Volute

The axial flow analyzed above will affect the pressure distribution in the impeller, which is the cause of the small difference in radial force [25]. On the other hand, this axial flow will have a certain impact on the pressure fluctuation in the volute. In this article, the pressure fluctuations on monitoring points on volute mid-span are investigated. The set of the monitoring points are shown in
The pressure fluctuation amplitude of P1 is extremely large. The dominant fluctuations at $f_{BP}$ are not eliminated for case30 on P1. This is mainly because P1 is in the area of influence of axial flow mentioned in Section 4.2. This axial flow mixes the fluids from two impellers. The pressure fluctuations of the points near the impeller outlet are not only superimposed; they also have a complex mixing effect.

The spectrum analysis of pressure fluctuation on P1, P2, P3 and P4 for different cases is shown in Figure 12. The maximum amplitudes at dominant frequency decrease from P1 to P4. Compared with other points, the pressure fluctuation amplitude of P1 is extremely large. The dominant frequency of pressure fluctuation on the four monitoring points for case0, case10 and case20 is $f_{BP}$. For case30, the dominant frequency is $f_{BP}$ on P1 and $2f_{BP}$ on P2, P3 and P4. For double-suction pumps, the pressure fluctuation in the volute is the superposition of pressure fluctuations generated by the two impellers [25]. The main frequencies of pressure fluctuation induced by one impeller are multiples of $f_{BP}$, which is the result of rotor-stator interaction. This explains why the pressure fluctuation amplitudes at $f_{BP}$ are eliminated for case30 on P2, P3 and P4. However, the pressure fluctuation amplitudes at $f_{BP}$ are not eliminated for case30 on P1. This is mainly because P1 is in the area of influence of axial flow mentioned in Section 4.2. This axial flow mixes the fluids from two impellers. The pressure fluctuations of the points near the impeller outlet are not only superimposed; they also have a complex mixing effect.

Figure 11. The coordinates of Point P1, P2, P3 and P4 are $(-0.146, 0, 0)$, $(-0.204, 0, 0)$, $(-0.26, 0, 0)$, and $(-0.32, 0, 0)$, respectively.

Figure 11. Monitoring points in volute at mid-span of centrifugal pump.

Figure 12. The maximum amplitudes at dominant frequency decrease from P1 to P4. Compared with other points, the pressure fluctuation amplitude of P1 is extremely large. The dominant frequency of pressure fluctuation on the four monitoring points for case0, case10 and case20 is $f_{BP}$. For case30, the dominant frequency is $f_{BP}$ on P1 and $2f_{BP}$ on P2, P3 and P4. For double-suction pumps, the pressure fluctuation in the volute is the superposition of pressure fluctuations generated by the two impellers [25]. The main frequencies of pressure fluctuation induced by one impeller are multiples of $f_{BP}$, which is the result of rotor-stator interaction. This explains why the pressure fluctuation amplitudes at $f_{BP}$ are eliminated for case30 on P2, P3 and P4. However, the pressure fluctuation amplitudes at $f_{BP}$ are not eliminated for case30 on P1. This is mainly because P1 is in the area of influence of axial flow mentioned in Section 4.2. This axial flow mixes the fluids from two impellers. The pressure fluctuations of the points near the impeller outlet are not only superimposed; they also have a complex mixing effect.
The amplitudes of pressure fluctuation on P1 for different cases are actually larger than those which are simply superimposed. This is because P1 is in the area of influence of axial flow. The axial flow increases the strength of pressure fluctuation. This result is consistent with that of Reference [25]. However, the coefficient variation curve of P1 is different from the curve of other points. The pressure on P1 is not simply obtained by superposition. The amplitudes of pressure fluctuation on P1 for different cases are actually larger than those which are simply superimposed. This is because P1 is in the area of influence of axial flow.

Figure 13 shows the variation trend of amplitude coefficients of pressure fluctuation at f_{BP} for different cases. The amplitude coefficient of pressure fluctuation C_{pi} is defined as:

\[ C_{pi} = \frac{A_{pi}}{A_{p1}} \]  

where A_{pi} (i = 1,2,3,4) is the amplitude of pressure fluctuation at f_{BP} for case0, case10, case20 and case30 individually at a certain point.

As shown in Figure 13, the coefficient variation curves of P2, P3 and P4 agree well. This is because the pressure in this region of the volute obtained by superimposing the pressure is generated by the two impellers. When the staggered angle is not zero, the pressure is reduced due to the superposition. This result is consistent with that of Reference [25]. However, the coefficient variation curve of P1 is different from the curve of other points. The pressure on P1 is not simply obtained by superposition. The amplitudes of pressure fluctuation on P1 for different cases are actually larger than those which are simply superimposed. This is because P1 is in the area of influence of axial flow. The axial flow increases the strength of pressure fluctuation.

Then, the pressure fluctuation strength in volute is studied for different cases. The standard deviation of pressure is used to evaluate the pressure fluctuation strength. The standard deviation is a measurement used to quantify the amount of variation of a set of data. A low standard deviation indicates that the pressure variation is spread out over a wider range of value. Whereas a high standard deviation indicates that the pressure variation tends to be close to the mean value.
of pressure indicates that the pressure variation tends to be close to the mean value, whereas a high standard deviation indicates that the pressure variation is spread out over a wider range of value. Figure 14 shows the standard deviation of pressure on mid-span of volute at $t = 0.92$ s at rated condition. The standard deviation of pressure is calculated in one revolution. The standard deviation of pressure is obviously largest in case 0 near the outlet of the impellers, which shows that the pressure fluctuation strength is strongest in case 0. The shape of the red region in case 0 is like a hexagon, which indicates the blades effect of the impellers, resulting from a strong blade-tongue interaction. From case 0 to case 30, the intensity is weakened by the superposition and the blades’ effect becomes weaker. For case 30, the contour likes a circle. Pressure fluctuations are offset by the superposition of two impellers on most areas of the volute. However, strong pressure fluctuations are detected close to the impeller outlet, which is caused by the axial flow analyzed above.

Figure 14. The standard deviation of pressure on mid-span of volute at $t = 0.92$ s at $1.0 Q_n$. (a) Case 0; (b) Case 10; (c) Case 20; (d) Case 30.

5. Conclusions

In the present work, the complex flow in volute of a double-suction centrifugal pump was investigated by numerical simulation using an RNG $k$-$\varepsilon$ model with experimental validation. The detailed flow near the impeller outlet with different staggered angles was obtained, and the influence of pressure fluctuation was analyzed. The main conclusions can be drawn as follows:
For the double-suction pump with a staggered angle, the circumferential pressure distribution at the twin impeller outlet is uneven, which will induce extra axial flows in the volute. This axial flow will mix the fluid from a different impeller and affect the pressure distribution in volute. There are no axial flows in the volute for case0, but obvious axial flows for staggered cases. Thus, the proper staggered angle should be optimized in the design procedure to reduce uneven flow in the pump.

The pressure fluctuations in most regions of the volute are obtained by superimposing the pressure generated by the two impellers, except for the region of axial flows. In this region, the pressure fluctuation is enhanced close to the impeller outlet due to the mixture effect, which is not simply calculated by superposition. This pheromone needs to be focused, especially for case30 where the pressure fluctuation is relatively small in most parts of the volute. This strong pressure fluctuation may cause extra noise and vibration, which should be considered in the design procedure.

Analysis of standard deviation of pressure shows that the axial flows affect the pressure fluctuation strength distribution close to the impeller outlet, and strong pressure fluctuations are detected close to the impeller outlet for all cases. Thus, the geometrical parameter at the impeller outlet is important and should be optimized in the design procedure.

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