

Article

The Influence of Tip Clearance on the Performance of a High-Speed Inducer Centrifugal Pump under Different Flow Rates Conditions

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Abstract: The influence mechanism of the blade tip clearance (TC) of an inducer on the performance of a centrifugal pump at high speed was researched under different flow rate conditions in this work. An experiment on the pump's external performance was carried out, and numerical calculation was also performed under four different TCs. The full characteristic performance curves, static pressure and pressure pulsation distributions of the pump were obtained. Through the research and analysis, it was found that the influence of the TC on the efficiency and the head of the centrifugal pump are related to the flow rate. Under the influence of a large flow rate, the increase in the TC is helpful to improve the efficiency and the head of the pump. The increase in the TC helps to weaken the gap jet effect on the inducer. The inlet jet of the inducer, caused by TC leakage, will form a low-pressure vortex zone at the inlet of the inducer. The splitter-bladed inducer's pressure pulsation is affected by the TC. The peak pressure pulsation at the monitoring point at the short blades is larger than that at the long blades. With the increase in TC, the cavitation degree at the inlet of the long blade of the inducer is decreased, while the cavitation degree at the short blade is deepened. It is also found that the TC has little effect on the radial force of the inducer and the impeller. These results will provide the design basis for the tip clearance of an inducer.

Keywords: tip clearance; cavitation; numerical calculation; pressure pulsation

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1. Introduction

High-speed centrifugal pumps are widely used in petrochemical, cryogenic and aerospace fields for their high rotation speeds [1,2]. For a high-speed inducer centrifugal pump, cavitation will inevitably occur under high-speed operation. In order to suppress cavitation, an inducer is usually installed in front of the pump's impeller. The head generated by the inducer can reduce the pump's Net Positive Suction Head (NPSH) and this will improve the pump's anti-cavitation performance [3–10]. The design of the inducer will affect the anti-cavitation performance of the pump [11–13]. The TC value of the inducer will affect the flow performance inside the inducer. To date, the flow mechanism of an inducer under different tip clearances has been studied by many scholars.

Su et al. [13] researched the influence of different tip clearances (TCs) on the internal flow characteristics and unsteady characteristics of the inducer. They found that the TC had little influence on the efficiency and the head of the pump. The TC could improve the inlet pressure of the inducer's blades and improve its cavitation performance, but it could weaken the wall jet. Guo et al. [2] researched the cavitation performance of a high-speed centrifugal pump with a splitter-bladed inducer under different TCs. It was found that the TC had a great influence on the static pressure distribution of the splitter-bladed inducer.

Xiang et al. [14] studied the effect of the TC on the cavitation characteristics of a turbopump inducer. The results showed that a larger TC had a greater impact on the cavitation performance. Shen et al. [15] studied the effects of various tip clearance widths on the tip flow dynamics and main flow characteristics for an axial-flow pump. From the simulation results, they found that the flow structure of the tip vortex and its transportation strongly depended on the tip clearance width, especially for the extension of the tip leakage vortex, appearance of an induced vortex and the area of the tip separation vortex. Zhang et al. [16] studied the effect of tip clearance on the energy performance and flow characteristics of a multiphase pump. The entrainment effect between the tip leakage flow and main flow in the impeller was strengthened with the increase in the tip clearance size; the induced vortex area and leakage flow rate also increased. Parikh et al. [17] studied the effect of increasing the TC on the performance of the inducer. They suggested that increasing the TC appropriately could improve its cavitation performance. Karakas et al. [18] studied the influence of the inducer TC on the cavitation and non-cavitation performance of a centrifugal pump. It was noted that with the increase in TC, excessive reverse leakage and large vortex backflow appeared at the tip position. This would lead to pressure loss in the inducer, which would cause the cavitation performance to worsen. Mansour et al. [19] found that the gas accumulation resistance increases significantly on an inducer with a larger TC. Kim et al. [20] observed the effects of different inducer TCs on the turbine pump's flow mechanism. It was considered that an inducer with a large TC is easily affected by the low suction pressure under cavitation conditions, and floating chambers were observed between inducer blades. Li et al. [21] studied the cavitation performance of a three-blade inducer under different TC conditions. It was found that the synchronous rotation cavitation increased when the flow rate increased. The synchronous rotation cavitation could be suppressed under a large TC, and it disappeared with the increase in the flow rate. Kim et al. [22] conducted a numerical study to observe the vortex structure upstream of the inducer. Their results showed that the occurrence of rotating cavitation was closely related to the interaction between the tip leakage vortex and blade. Tani et al. [23] believed that the TC was the main cause of cavitation instability. Kim et al. [24] studied the influence of the TC on the cavitation performance of a turbopump inducer through numerical simulation and pointed out that a smaller TC would lead to stronger tip leakage cavitation.

It can be seen from the above literature review that many scholars have studied the inducer from many aspects, but research on the influence of the TC on the pressure fluctuation and cavitation of the inducer is still lacking. Most of the research is currently focused on the influence of the main impeller's TC on the centrifugal pump flow distribution. At present, there are few studies on the influence of the TC on the cavitation of inducers, especially for splitter-bladed inducers. Therefore, it is very important to study the influence of the TC on the flow and unsteady characteristics of splitter-blade inducers. This project takes the splitter-blade inducer as the research object, studies the influence of different TCs on the inducer and reveals the influence mechanism of the TC on the internal performance.

2. The Pump's Structure and the Experimental Device

2.1. Main Parameters of the Centrifugal Pump

A high-speed centrifugal pump with a splitter-bladed inducer is used, which is shown in Figure 1a. The pump's specific speed is 23.08. The pump rated speed $n = 6000$ r/min, rated flow rate $Q = 4$ m³/h, head $H = 100$ m. The inlet pipe diameter is designed to control the TC, and its diameter range is from 38.4 mm to 39 mm. The centrifugal impeller has eight blades, and they are designed to be straight, as shown in Figure 1b. This design will minimize the impact of complex flow on the inducer. The inducer has 4 splitter blades, and its maximum outer diameter is 38 cm, as shown in Figure 1c.

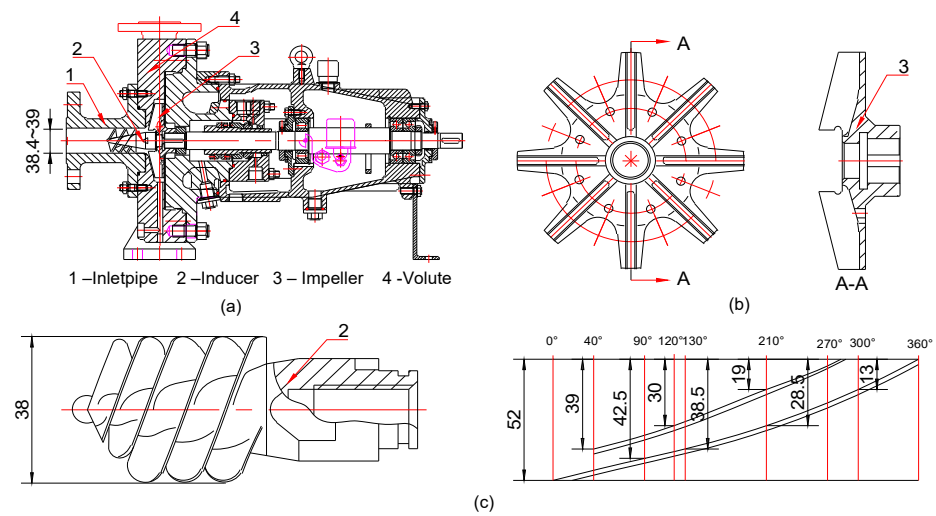


Figure 1. The high-speed inducer centrifugal pump: (a) the high-speed pump with a splitter-bladed inducer, (b) the impeller and (c) the splitter-bladed inducer.

2.2. Experimental System

The high-speed inducer centrifugal pump is tested in a closed system, as illustrated in Figure 2. An experimental pump is installed in this closed test system, and a variable-frequency motor is used to start the system. Table 1 shows the parameters for the system.

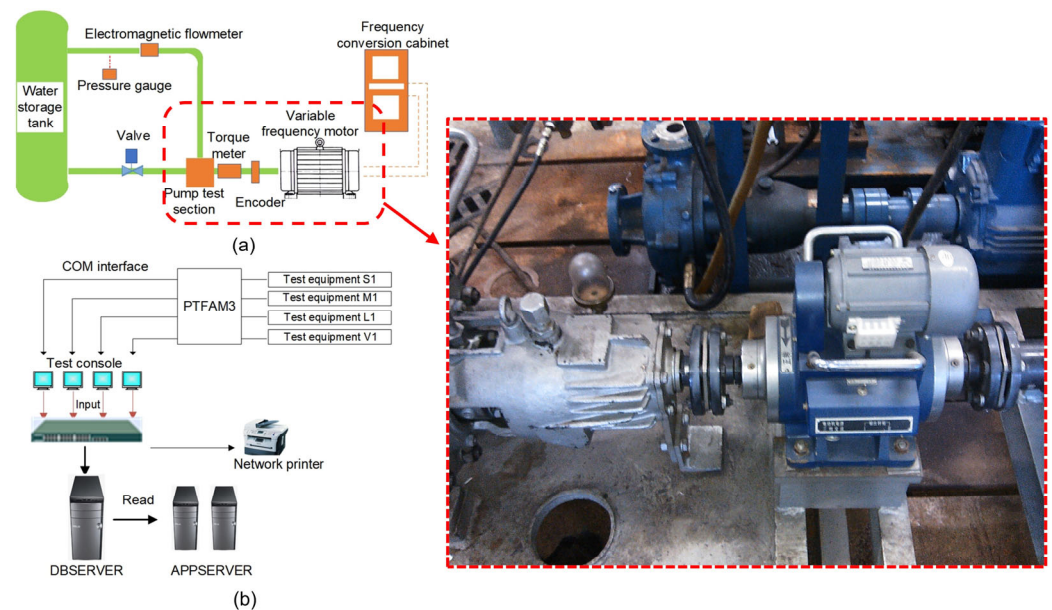


Figure 2. Experimental system: (a) closed test bench system, (b) DCS test system.

Table 1. Parameters of the high-speed centrifugal pump.

Component	Parameter
The model of the variable-frequency motor	GSB-22-06E13
The rated frequency	100 Hz
The rated output power	22 KW
The volume of the liquid storage tank	31 m ³
The torque	35 N/m
The external vacuum pump's maximum vacuum	6×10^{-2} Pa
Rotation speed n	6000 r/min
Pipe diameter D_1 (used to control TC)	38.4 ~ 39 mm
The largest diameter of the inducer	38 mm

The layout of the Distributed Control System (DCS) test system is shown in Figure 2b, which is used for collecting and monitoring data such as the pressure and the flow rate.

3. Numerical Calculation Method

3.1. Verification of Grid Independence and Computational Region

Figure 3 shows the 3D flow field. An impeller, an inducer, an impeller blade, a clearance layer and a volute are all included in the calculation of the liquid area and meshed using the ICEM software. Both the inducer and the impeller are meshed with structured grids to improve the calculation accuracy. A total of 9,148,498 calculation grids have been calculated for the whole high-speed inducer centrifugal pump, and its grid quality is good. In order to analyze the influence of the inducer tip clearance, the flow of pumps with different inducer tip sizes is studied. The tip clearance is controlled by changing the diameter of the induction wheel cover. In addition, we increase the number of nodes to refine the tip clearance area, where the tip clearance is 40 nodes and the blade width is 10 nodes, so as to more accurately predict the flow status of the tip clearance. In Figure 3b, the local grids on the inducer and impeller are shown. Its grid independence has been verified in a previous paper [1].

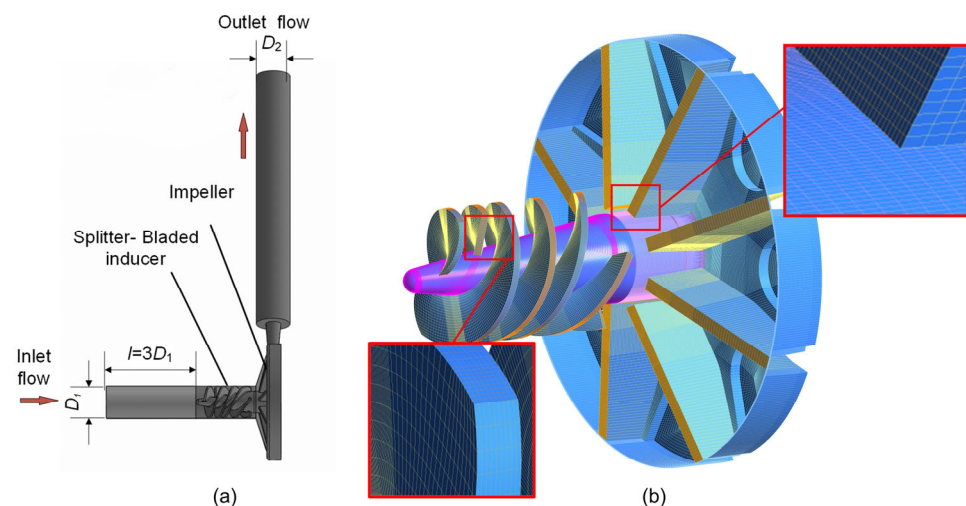


Figure 3. (a) Calculation area. (b) Split-blade inducer and local area grid of impeller.

3.2. Numerical Simulation Calculation Method

In this research, numerical work is performed using the commercial CFD code ANSYS CFX. The external characteristic and internal pressure pulsation of the centrifugal pump with a high-speed inducer are numerically calculated in this paper. During the calculation of the external characteristic performance of the high-speed inducer centrifugal pump, the steady numerical simulation method is adopted. During the calculation of the pressure pulsation of the centrifugal pump, the cavitation unsteady numerical simulation method is adopted. The boundary conditions are set as follows.

(1) The entry boundary condition is defined as the total pressure inlet according to the experiment, and the value is 15,590 Pa. The mass flow rate outlet is defined on the outlet.

(2) In steady numerical simulation, a frozen rotor mode is defined on the interface between the rotor and volute, and, on the interface between the rotor and inlet section, during unsteady cavitation calculation, the interface is changed to the transient frozen rotor mode. The blade surfaces and other walls are set as no-slip wall.

(3) The rest of the components are set to the static domain. A counter-rotating wall is defined on the volute chamber, and the volute chamber is stationary relative to the stationary domain.

(4) The time step of unsteady cavitation calculation is the time required for impeller rotation of 3 degrees, $t = 8.33333 \times 10^{-5}$ s, and the calculation convergence accuracy is set as 10^{-4} .

(5) A mixture model is used in this work. There are two phases. One is considered water and the other is considered vapor [2,3]. The vapor volume fraction at the inlet and outlet of the high-speed inducer centrifugal pump is set as zero. Water under standard conditions is defined as the liquid phase. The continuity and momentum conservation equations are as follows:

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_j}(\rho u_j) = 0 \quad (1)$$

$$\frac{\partial}{\partial t}(\rho u_j) + \frac{\partial}{\partial x_j}(\rho u_i u_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}(\mu \frac{\partial u_i}{\partial x_j}) \quad (2)$$

$$\rho = \alpha_w \rho_w + \alpha_v \rho_v \quad (3)$$

where w, v and ρ indicate the liquid water, vapor and density. The volume fraction equation for the vapor phase is

$$\frac{\partial}{\partial t}(\alpha_v \rho_v) + \frac{\partial}{\partial x_i}(\alpha_v \rho_v u) = -\frac{\partial}{\partial x_i}(\alpha_v \rho_v u_{dr,v}) \quad (4)$$

Here, u is the mass-averaged mixture velocity, and $u_{dr,v}$ is the drift velocity of the vapor phase. They are defined as follows.

$$u = \frac{\alpha_w \rho_w u_w + \alpha_v \rho_v u_v}{\rho} \quad (5)$$

$$u_{dr,v} = \frac{(\rho - \rho_v) d_v^2}{18 \mu_c f} \cdot [g - (u_v \cdot \nabla u_v)] - \frac{1}{\rho} \sum_{i=1}^n a_i \rho_i u_{wi} \quad (6)$$

Here, a_i is the volume fraction of the i phase. n is the number of phases. In this simulation case, n is set to 2 and the Reynolds number (Re) value is 29,713. The formula for the factor f is as follows:

$$f = \begin{cases} 1 + 0.05 Re^{0.687} & , Re < 1000 \\ 0.018 Re & , Re \geq 1000 \end{cases} \quad (7)$$

4. Results

4.1. Analysis of the Impact of TC on the External Performance of the Pump

The whole flow of the high-speed inducer centrifugal pumps with four TCs of 0.2 mm, 0.3 mm, 0.4 mm and 0.5 mm is numerically calculated, and the obtained external characteristics are shown in Figure 4. Through the comparison and analysis of the numerical calculation and experimentally measured values, we find that numerical calculation can predict the performance of the high-speed centrifugal pump, and the results of numerical calculation under different TCs in this paper are credible [1].

From the head flow curves ($H-Q$) in Figure 4, the following conclusions may be obtained. (1) Under small flow conditions ($Q \leq Q_d$), the head of the pump rises slowly with the increase in flow rate and reaches its peak at the design operating point $Q = 4 \text{ m}^3/\text{h}$, under the condition of four TCs. Under the same flow rate, the smaller the TC is, the higher the head is, and the head reaches the highest value when $TC = 0.2 \text{ mm}$. (2) Under large flow conditions ($Q > Q_d$), the head drops rapidly with the increase in flow rate under the condition of four TCs. When $Q_d < Q \leq 1.5 Q_d$, the larger the TC is, the smaller the head is. When $1.5 Q_d < Q \leq 2 Q_d$, the larger the TC is, the higher the head is, and the highest head is at $Q = 2 Q_d$ when $TC = 0.5 \text{ mm}$.

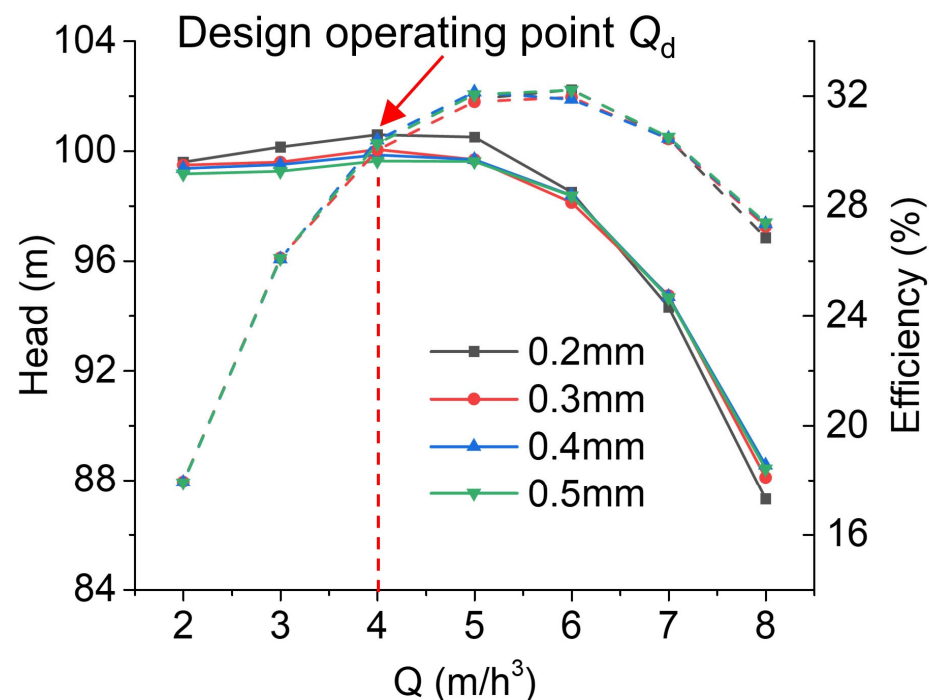


Figure 4. Comparison diagram of external characteristic curves.

From the η - Q curves in Figure 4, it can be seen that the efficiency is very similar under the four TC conditions. The following conclusions can be obtained from the analysis.

(1) Under small flow conditions $Q \leq Q_d$, the efficiency increases rapidly with the increase in the flow rate under the four TC conditions. Under the same flow rate, the smaller the TC is, the higher the efficiency is, and it reaches the highest efficiency at TC = 0.2 mm.

(2) Under large flow conditions ($Q > Q_d$), as the flow rate increases, the efficiency increases first and then decreases. When $Q = 5 \text{ m}^3/\text{h}$ to $Q = 6 \text{ m}^3/\text{h}$, the efficiency reaches the highest value. When $Q_d < Q \leq 1.5Q_d$, the larger the TC is, the higher the efficiency is. When $1.5Q_d < Q \leq 2Q_d$, the larger the TC is, the higher the efficiency is.

The reasons for the above phenomenon are analyzed as follows.

(1) Small flow condition ($Q \leq Q_d$): Compared with an ordinary centrifugal pump, the straight-blade impeller used in this study causes the outlet velocity of the fluid to maintain a relatively constant state, and the head of the pump decreases with the flow rate under the small flow condition. With the increase in the blade TC, the phenomenon of backflow and clearance leakage in the inducer is gradually aggravated, which causes a fluid energy loss and reduces the overall head and efficiency of the pump.

(2) Large flow condition ($Q > Q_d$): The large flow condition means that the pump needs to transport more fluid medium per unit time, which will lead to the acceleration of the fluid flow rate in the pump and the reduction of the impeller work, thus leading to a drop in the head. With the increase in the TC, the congested flow passage under large flow conditions can be alleviated, and the flow state can be improved. Therefore, the increase in TC causes the overall head and the efficiency of the pump to be improved.

4.2. Influence of TC on the Internal Flow Characteristics of the Pump

The pressure distribution nephogram of the inducer on its axial plane under different flow rates and clearance conditions can be seen in Figure 5. (1) The pressure distribution gradually increases from inlet to outlet along the radial and axial directions. (2) The lower pressure distribution of the inducer is related to the TC. At the inlet rim area "A", a significant low-pressure vortex appears due to the effect of the wall jet. The volume of low-pressure vortex "A" gradually decreases with the increasing TC under the three flow

conditions shown in Figure 5a–c. The reason is that the pressure difference between the two sides of the blade decreases gradually with the increase in the TC, which weakens the wall jet effect and reduces the vortex in the lower-pressure area. (3) It can be seen from Figure 5c that under the condition of a large flow rate, $Q = 6 \text{ m}^3/\text{h}$, there is a significant lower-pressure area “B” in the inducer. The B area is located at the flow passage near the inlet of the main impeller. At this position, affected by the lower pressure area of the main impeller, the lower-pressure area “B” in the inducer passage is formed.

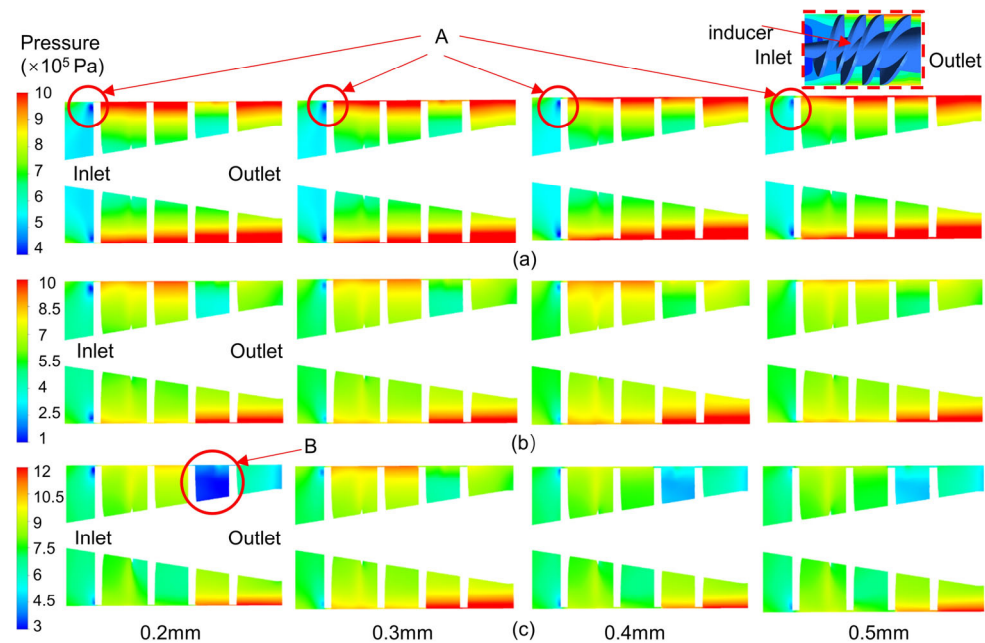


Figure 5. A nephogram showing the pressure distribution of the inducer along its axial plane at various flow rates and different TCs: (a) $Q = 2 \text{ m}^3/\text{h}$, (b) $Q = 4 \text{ m}^3/\text{h}$ and (c) $Q = 6 \text{ m}^3/\text{h}$.

Figure 6 shows the radial pressure distribution nephogram of the main impeller and volute under different flow rates and different TCs. Figure 6 makes it possible to draw the following conclusions. (1) From Figure 6a–c, it is shown that under different flow rates, the pressure from the impeller inlet to the volute outlet gradually rises, and the pressure at the impeller inlet is low. (2) With the increase in flow rate, when the liquid flows through the impeller towards the volute, the jet phenomenon occurs near the volute tongue. Thus, with the increase in the flow rate, the lower-pressure area near the volute tongue increases significantly, and the pressure accumulates near the volute tongue. (3) The pressure distribution in each channel is asymmetrical. Under the action of fluid outflow, the pressure of the flow near the volute diffusion section of the impeller will be significantly lower than that of other flow channels. The flow channel is shown as area “A” in Figure 6. The range of its low-pressure area gradually increases with the increase in the flow rate. As mentioned above, this lower-pressure zone will affect the outlet pressure of the inducer, as shown in Figure 5, where the “B” area of the inducer shows a more significant low-pressure zone. (4) There is a slight impact of the TC on the pressure distribution of the main impeller, while the impact on that of the volute tongue is great.

From the pressure analysis in Figures 5 and 6, it can be seen that cavitation easily takes place at three positions. One is at the inlet of the inducer, the second is at the outlet of the inducer and the third is at the inlet of the impeller. From the above, it can be concluded that the TC has a great influence on the pressure distribution of the inducer and volute tongue.

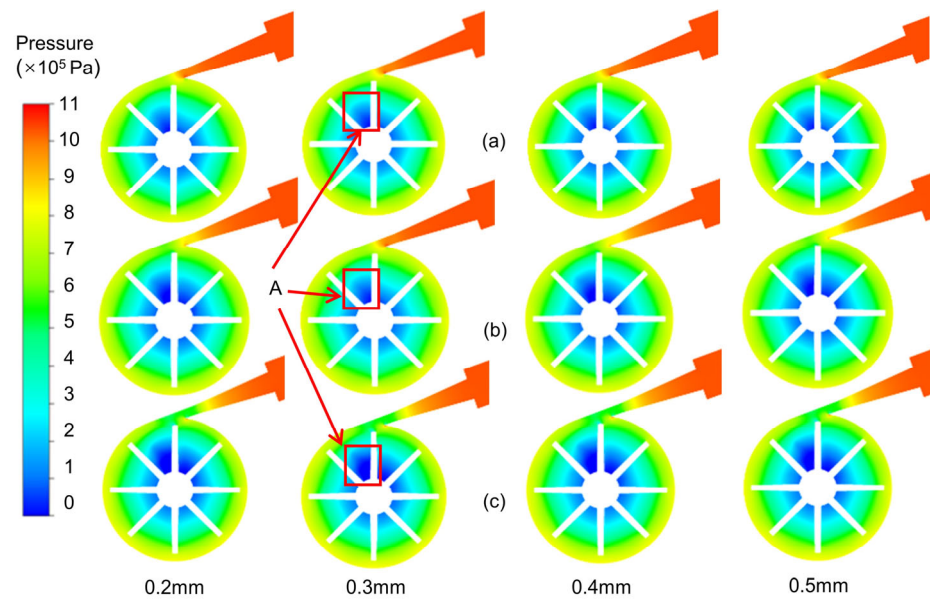


Figure 6. Radial pressure distribution nephogram of the main impeller and volute under various flow rates and TCs: (a) $Q = 2 \text{ m}^3/\text{h}$, (b) $Q = 4 \text{ m}^3/\text{h}$ and (c) $Q = 6 \text{ m}^3/\text{h}$.

4.3. Analysis of Time–Frequency Domain Results of Monitoring Points under Different TCs

In order to observe the pressure pulsation in the pump, eight monitoring points, P1~P8, are set at key positions on the inducer, which are shown in Figure 7. At the inlet edge of the long blades, three key points, P1~P3, are selected. At the entrance of the short blades, three key points, P4~P6, are selected. P7 is a key point at the center of the inducer axis, and P8 is a key point at the outlet of the inducer.

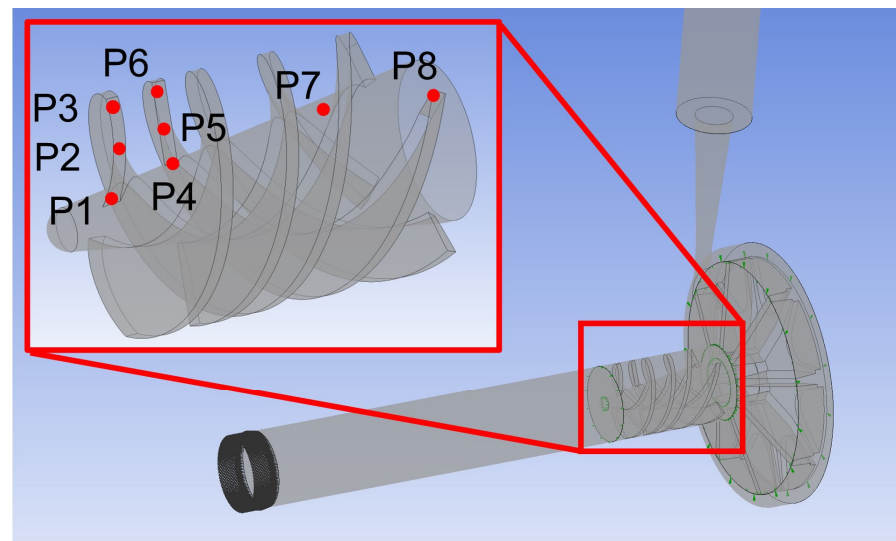


Figure 7. Schematic diagram of monitoring point positions.

The pressure distribution and pressure pulsation distribution diagrams of each monitoring point are obtained by cavitation unsteady simulation.

Figure 8 shows the static pressure changes of the monitoring points during the two-week rotation of the impeller. Figure 8 leads to the following conclusions. (1) There is a law of pressure distribution in the inducer. From Figure 8a–h, it can be seen that the closer the monitoring point to the outer edge of the inducer is, the higher the pressure value is, and the pressure in the inducer gradually rises along the axial direction.

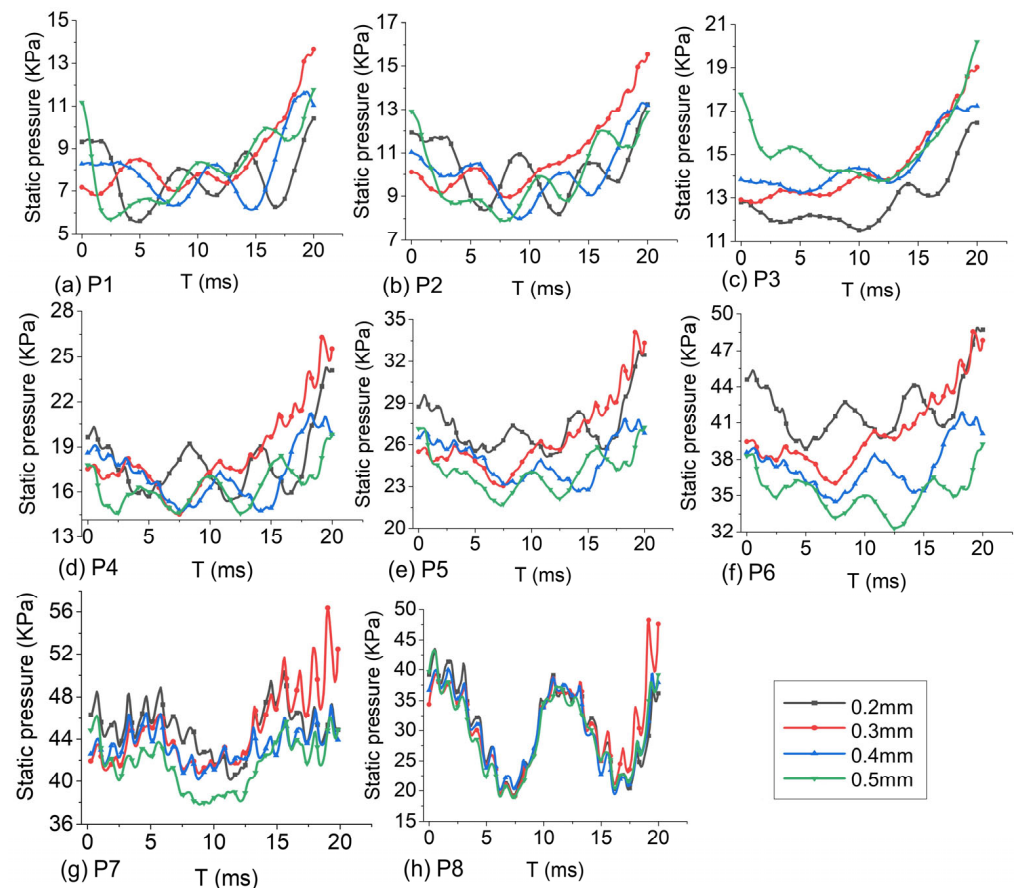


Figure 8. Static pressure change diagram of monitoring points: (a) P1, (b) P2, (c) P3, (d) P4, (e) P5, (f) P6, (g) P7 and (h) P8.

(2) The pressure at the short blade's inlet is 19–29 kPa higher than that for the long blades, and the pressure gradient is more significant. (3) From the distribution of the P1, P2 and P4 monitoring points, there is little correlation between the TC and the pressure change. The pressure fluctuation of monitoring points P1, P2 and P4 near the inlet edge of the inducer blades is quite chaotic. The reason is analyzed below. When the fluid enters the inducer from the inlet, flow separation occurs near the hub, which will affect the pressure fluctuation at the monitoring point. (4) From the pressure distribution of P3, P6, P7 and P8, there is a significant correlation between the pressure value and TC. The pressure value of the P3 monitoring point increases with the increase in the TC due to clearance leakage, but the pressure value of the P6 monitoring point decreases with the increase in the TC. The pressure fluctuations of monitoring point P7 and P8 under different TCs show a significant blade phase correlation. The pressure fluctuations at different clearances of monitoring points P7 and P8 show significant blade phase correlations. P7 is located between two blades, and its pressure fluctuation is affected by the long and short blades, showing a complex fluctuation state. At P7, the larger the TC is, the smaller the pressure is. P8 is in the middle section of the inducer outlet. TC has little impact on the pressure here, and the overall pressure fluctuation shows periodic changes.

Figure 9 shows the time domain diagram of the pressure fluctuations at each monitoring point. (1) The pressure pulsation in the inducer is intense, and the pressure pulsation intensifies along the direction of the inducer axle. The static pressure coefficient of the pressure fluctuation monitoring point on the short blade is significantly higher than that of the long blade at the inlet of the inducer. The pressure pulsation at the monitoring points at the middle section and outlet of the inducer is significantly higher than that at the inlet of the short blade. (2) The TC has a certain influence on the pressure pulsation of the inducer.

The pressure fluctuation intensity at the inlet of the long blade of the inducer is close in value under each TC, but at the short blade, the pressure fluctuation intensity decreases significantly with the increase in TC.

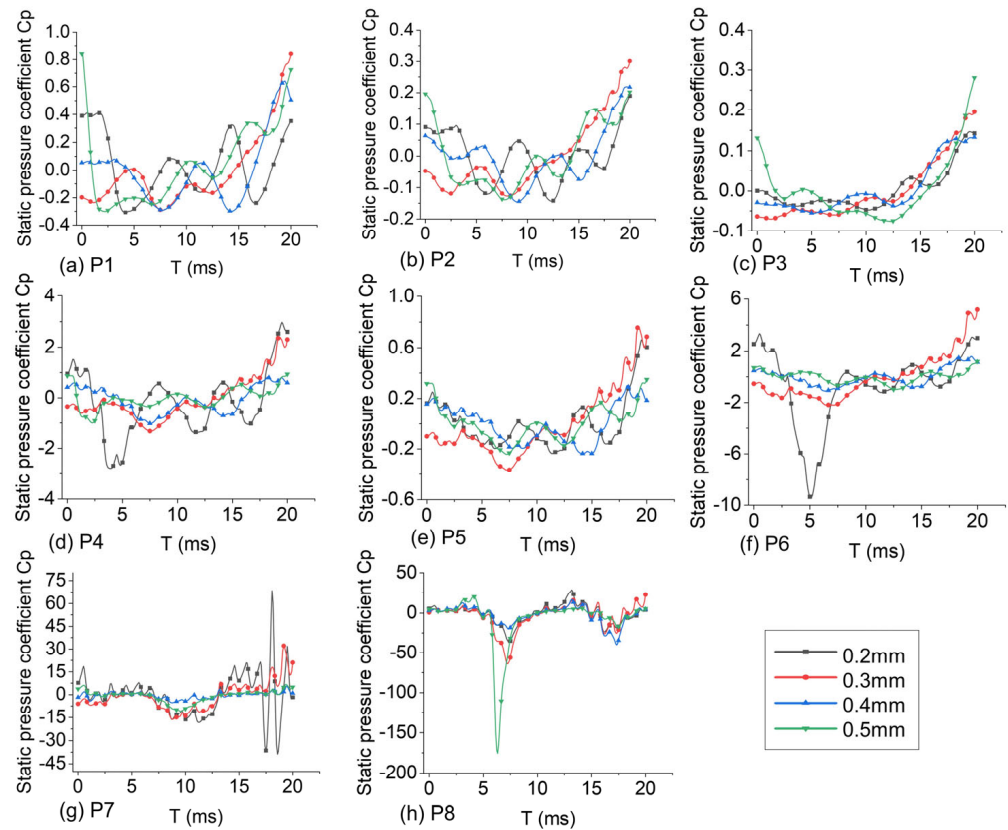


Figure 9. Time domain diagram of pressure fluctuations at monitoring points: (a) P1, (b) P2, (c) P3, (d) P4, (e) P5, (f) P6, (g) P7 and (h) P8.

By FFT transformation of the static pressure data of the monitoring points, the frequency domain diagrams of the pressure pulsation of each monitoring point can be obtained, which are shown in Figure 10.

Figure 10 makes it possible to draw the following conclusions. (1) The pressure fluctuation at the inlet of the inducer is less affected by the rotation of the inducer, and the peak in pulsation amplitude appears only at 0.5 times the rotation frequency at the hub and rim. (2) The TC has a certain influence on the pressure fluctuation. The pulsation peak appears at 0.5 times the rotation frequency of the short blade. It appears near 200 Hz. The secondary peak becomes larger with the increase in the TC. (3) There is a secondary peak near 800 Hz at the inlet of the short blade, which is caused by the rotation of the long blade. However, the amplitude of the pressure fluctuation is low at this frequency. In particular, at the entrance of the long and short blades, the amplitude of pulsation increases first and then decreases with the increase in the TC. When the TC is 0.3 mm, its amplitude is the largest. There is also a significant secondary frequency of 800 Hz, besides the main frequency of 50 Hz, at the monitoring points at the middle section and the outlet of the inducer. The peak value of pressure fluctuation at the outlet of the inducer is more significant than that at the middle of the inducer.

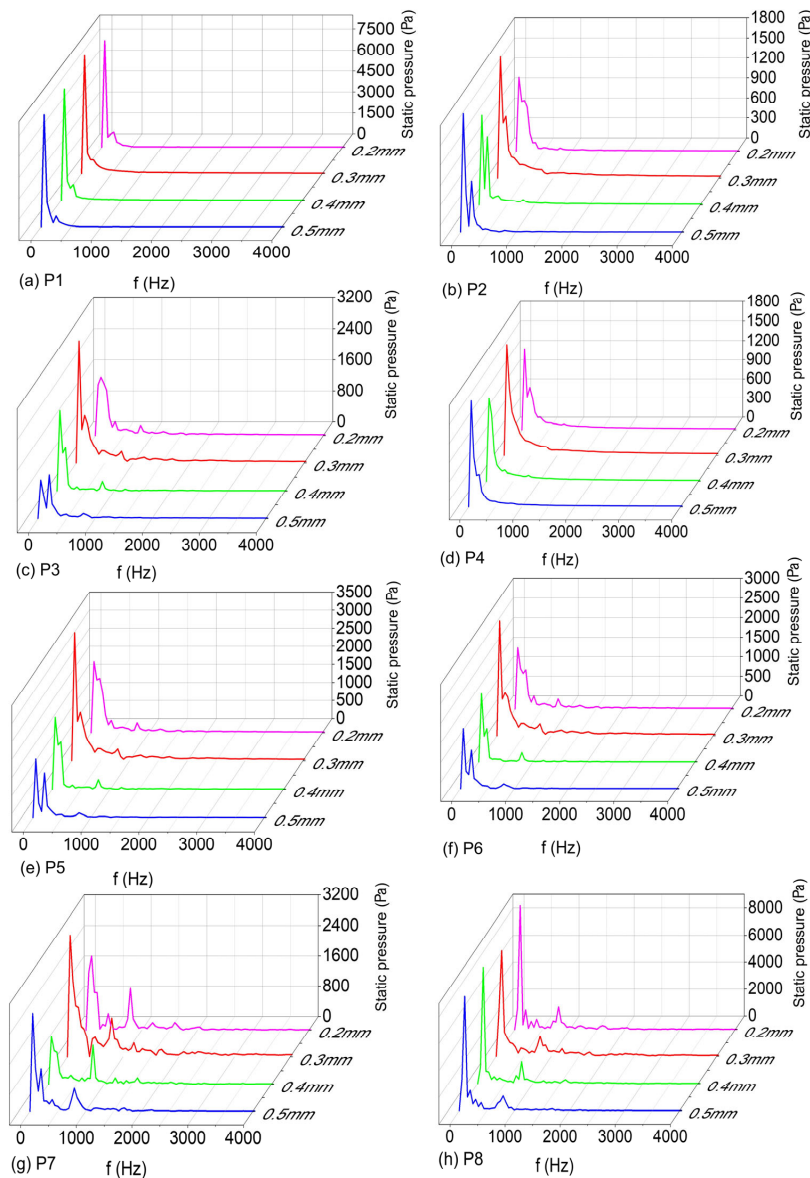


Figure 10. Frequency domain diagram of pressure fluctuations at monitoring points: (a) P1, (b) P2, (c) P3, (d) P4, (e) P5, (f) P6, (g) P7 and (h) P8.

4.4. Time–Frequency Domain Analysis at Key Sections under Different TCs

Figure 11 shows the average pressure fluctuations at the impeller inlet, the impeller outlet and the pump outlet during impeller rotation. Figure 11 makes it possible to draw the following conclusions. (1) The static pressure at each key section shows a rising law, due to the rotation of the impeller and the existence of the diffusion section of the pump body. Under different TCs, the pressure fluctuations in each section show good consistency. This shows that the static pressure of the flow field in the pump is mainly affected by the blade phase angle. (2) When the TC is 0.5 mm, the pressure fluctuation at the inlet and outlet of the impeller is the weakest due to the leakage of the clearance at the blade top. In addition, the pressure fluctuation frequency at the impeller outlet is significantly higher than that at the impeller inlet, as the number of blades of the impeller is greater than that of the inducer. (3) At the pump outlet, in one impeller rotation cycle, the static pressure fluctuation forms eight cycles, which is consistent with the number of impeller blades in the pump. This shows that the pressure pulsation at the pump outlet is still affected by the phase angle of the impeller blade. When the TC is 0.4 mm, the fluctuation amplitude of the outlet pressure is the smallest. When the TC is 0.5 mm, due to the backflow, the

outlet pressure fluctuation presents a bimodal state (as shown in Figure 11, “A”), and the amplitude of the outlet fluctuation increases.

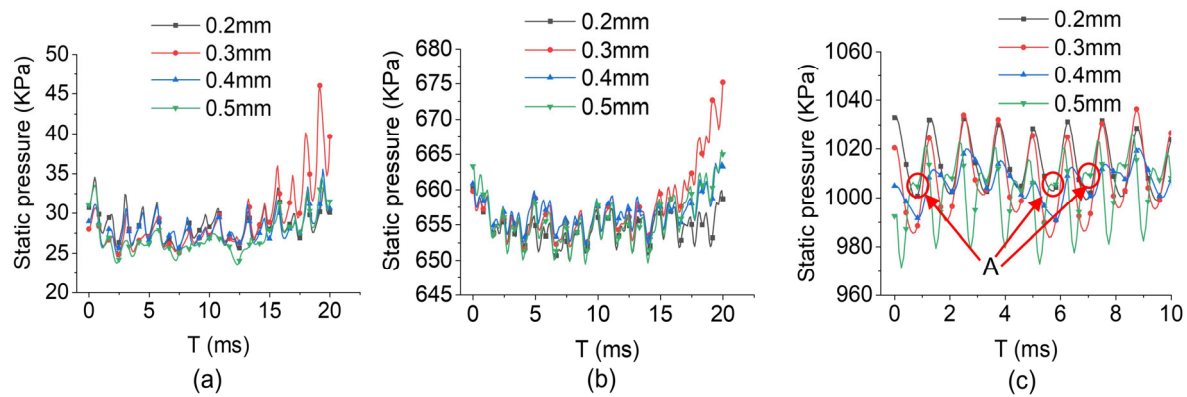


Figure 11. Pressure fluctuation diagram at key sections: (a) impeller inlet, (b) impeller outlet and (c) pump outlet.

4.5. Analysis of Radial Force under Different TCs

Figure 12 presents the distribution of the radial force on the inducer and the impeller during rotation. The following findings may be drawn from Figure 12. (1) The radial force on the flow passage parts under each TC is unbalanced. During the rotation of the inducer and the impeller, the radial force on each position in the flow passage part is not uniform. The radial force of the inducer hub and impeller presents a pattern resembling petals, and the number of petals is 8, which is consistent with the number of the impeller. (2) The influence of the TC on the radial force is not obvious. Compared with the impeller, the radial force on the inducer is weaker and more uniform. With the increase in the TC, the radial force on the inducer and the impeller decreases. The radial forces acting on the long and short blades of the inducer are elliptical. Under different TCs, the radial forces on the long and short blades do not change significantly. When the TC is 0.5 mm, the radial force on the blades is the smallest. In addition, it is found that the amplitude of the radial force on the long and short blades is staggered by nearly 90 degrees, which is mainly caused by the exit phase of the long and short blades of the inducer being staggered by 90 degrees.

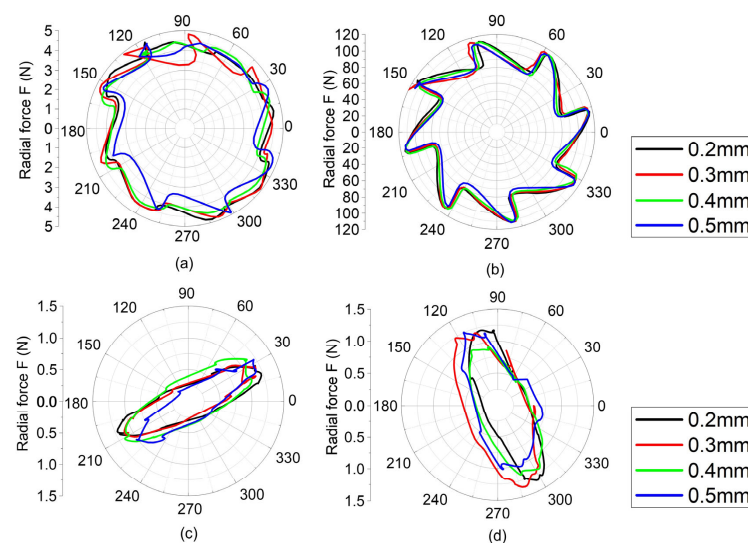


Figure 12. Radial force of different flow parts in pump under different TCs: (a) radial force of inducer hub, (b) radial force of inlet pipe, (c) impeller radial force, (d) radial force of short blade of inducer.

4.6. Analysis of Cavitation Degree of the Inducer

Figure 13 shows the cavitation distribution diagram of the inducer under different TCs and phase angles. Figure 13 makes it possible to draw the following conclusions. There are three positions where cavitation occurs on the inducer. One is at the inlet end of the long blade, the second is at the inlet end ("B") of the short blade and the third is at the outlet end ("A"). The cavitation at the inlet of the long blade is more significant, and only a small amount of cavitation occurs at the other two locations. In the process of cavitation, the gas content at the inlet of the inducer can reach 80–90%. Cavitation is mainly distributed along the outer edge of the inducer blade. Under different inducer phase angles, the degree of cavitation of the inducer is different. With the increase in the TC, the cavitation degree at the inlet section of the long blade of the inducer decreases gradually. However, the degree of cavitation at the short blade's inlet becomes increasingly severe. This is mainly due to the leakage flow in the tip clearance, which increases the pressure at the long blades' inlet, but reduces the pressure at the inlet of the short blades, resulting in the deepening of cavitation.

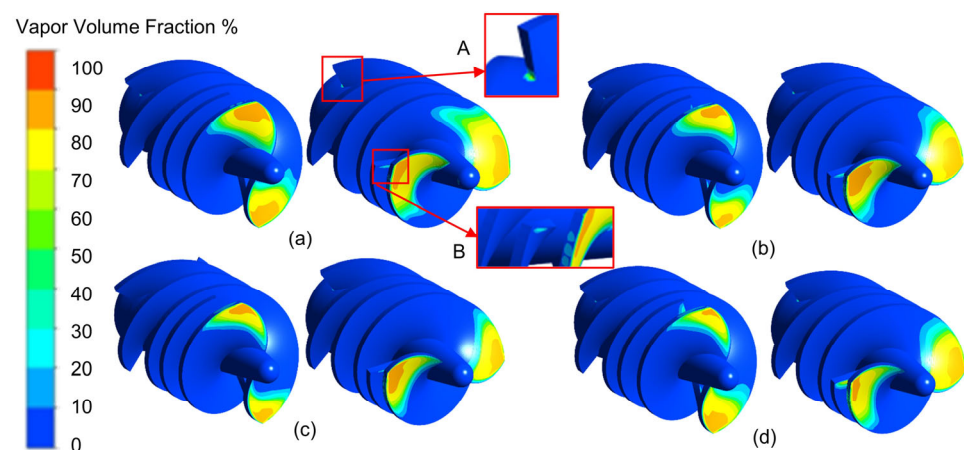


Figure 13. Cavitation distribution of inducer under different TCs: (a) TC= 0.2 mm, (b) TC= 0.3 mm, (c) TC= 0.4 mm and (d) TC= 0.5 mm.

5. Conclusions

In this paper, the hydraulic steady-state characteristics and transient characteristics of a high-speed inducer centrifugal pump are numerically simulated and experimentally studied by using a structured grid and RNG turbulence model. The changes in the performance curve and pressure pulsation in the centrifugal pump under four different TCs are compared. The following conclusions are obtained through the research.

(1) The TC has a certain influence on the head and efficiency of the high-speed induction centrifugal pump. Under the condition of small flow rates, with the increase in TC, the backflow and clearance leakage in the inducer gradually intensified, which led to the loss of fluid energy and reduced the total head and efficiency of the pump. Under the condition of large flow rates, with the increase in TC, the blocked flow passage under the condition of large flow rates could be relieved, and the flow state could be improved. Therefore, the increase in TC helps to improve the pump head and efficiency.

(2) The increase in TC helps to weaken the gap jet effect on the inducer. The inlet jet of the inducer caused by TC leakage will form a low-pressure vortex zone at the inlet of the inducer. With the increase in TC, the low-pressure vortex area will gradually decrease. The pressure distribution in the impeller is not uniform; the pressure is lower in the channel near the outlet section, and the low-pressure area will diffuse into the inducer.

(3) The splitter-bladed inducer's pressure pulsation is affected by the TC. In the middle and end of the inducer, the pressure pulsation under different TCs gradually presents consistency and periodicity. The pressure value and pressure fluctuation intensity at the

monitoring point of the long blade of the inducer are lower than those at the monitoring point of the short blade. The peak value of pressure fluctuation increases with the increase in TC.

(4) The TC has a certain influence on the cavitation of the inducer. Due to the lower-pressure area at the tip edge of the long blade, the cavitation of the inducer is primarily distributed at the tip of the long blade around the inlet, and the maximum gas phase volume fraction is approximately 80–90%. With the increase in TC, the leakage flow in the tip clearance increases the pressure at the inlet of the long blade and decreases the pressure at the inlet of the short blade, which reduces the cavitation degree at the inlet of the long blade and deepens the cavitation degree at the short blade of the inducer.

To summarize, in the design process, the positive effect of the TC, which helps to weaken the gap jet effect in the inducer, should be fully considered, and the influence of the TC on the pressure pulsation and cavitation of the splitter-bladed inducer should also be considered. The optimal TC should be selected in the design to ensure that the high-speed inducer centrifugal pump can operate under the optimal working conditions. These results will provide a theoretical basis as well as technical support for the design of a high-speed inducer centrifugal pump.

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