Effect of inducer sweepback on cavitation performance of centrifugal pump

X R Cheng¹,², H Liu¹, Y XTU¹, Y Q Wei¹ and S Y Zhang¹

¹ College of Energy and Power Engineering, Lanzhou University of Technology, Lanzhou 730050, China; ² Key Laboratory of Fluid Machinery and Systems of Gansu Province, Lanzhou 730050, China

Abstract. In order to study the effects of sweepback of inducer on cavitation performance of pump, based on the experimental results, the Reynolds N-S equation, RNG k-ε turbulent model and Schnerr&Sauer cavitation model, a three-dimensional numerical calculation is employed to study the flow characteristics of one certain LNG pump. Laws of the variation of cavitation performance, head and efficiency with the change of sweepback are studied. The numerical analysis of the eight inducer projects with sweepback angle from 120° to 290° is carried out. The results show that the cavitation bubbles appear at the suction surface near the inlet side at first. With the decrease of \( \text{NPSH}_a \), the bubbles spread to the outlet side of the inducer and the pressure surface, finally they fill the entire channel. When the inducer sweepback angle increases from 120° to 270°, \( \text{NPSH}_a \) of the pump reduces gradually. That means the anti-cavitation performance of the pump has been improved. However, \( \text{NPSH}_a \) of the pump raises gradually when the inducer sweepback angle increases from 270° to 290°. The efficiency and head of pump tend to be stable within the larger range near the optimal sweepback.

1. Introduction

Cavitation is a common phenomenon in fluid machinery. Cavitation will occur when the partial pressure of the fluid is lower than pressure of the vaporization at the local temperature [1]. The high-speed rotating blade will reduce the pressure of partial fluid. Pumps under cavitation conditions can degrade performance, produce vibration and noise, and seriously damage the over-current components [2]. One of the most effective ways to prevent the cavitation of a centrifugal pump is to install an inducer before the impeller. The inducer can increase the pressure of the fluid before entering the main impeller, improving the available net positive suction head (NPSH) [3].

A typical inducer should have a small hub ratio, small load, small attack angle, high chord-spacing ratio and other characteristics. [4] calculated the inducer with four kinds of blade wrap angle, and showed the process of bubbles from development to collapse. [5] carried out numerical simulation and experiment to study on a number of inducer with different import installation angle of blade, hub radius and flow coefficient. [6] proposed that the clocking position has effects on the dynamic and static interference between the impeller and the guide vane and the cavitation performance of the centrifugal pump. [7] studied on the effects of single and tandem inducer on head, efficiency and cavitation of centrifugal pump. [8] suggested that the inducer can inhibit the diffusion of bubbles in the main impeller and decrease slowly the inlet pressure of main impeller, but not cause a pronounced decrease in the head of the main impeller. [9, 10] proposed that the variable-pitch inducer can improve...
cavitation performance, with smaller import installation angle of the blade to obtain smaller import flow coefficient, and with larger outlet installation angle of the blade to generate enough head to meet the requirement of cavitation of inducer and performance of import pressure of centrifugal impeller.[11] found that torque and the characteristics of drop of pump head were related to cavitation of asymmetrical blade.[12, 13] put forward the design method of splitter-blade inducer, and found the variable-pitch inducer with splitter-blade have superior cavitation performance through experiment and numerical calculation. In the same flow rate, the head of pump with a pre-positioned splitter-blade inducer is slightly higher than the one with a pre-positioned equal-pitch inducer and a pre-positioned variable-pitch inducer.

At present, there are few studies on the influence of the change of the inducer import sweepback angle on the pump characteristics and the cavitation performance. In this paper, the influence of the sweepback on the pump head, efficiency and cavitation performance is obtained by using the full three-dimensional numerical calculation of the centrifugal pump with different sweepback inducers, which provides a basis for the theoretical design of the inducer.

2. Governing equation

2.1. Basic equation
Assuming that the fluid is an incompressible viscous fluid, the Reynolds N-S equation is used to calculate the fluid domain. The governing equations are as follows:

\[ \frac{\partial u_i}{\partial x_j} = 0 \]  

\[ \rho u_i \frac{\partial u_i}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left( \mu \frac{\partial u_i}{\partial x_j} + \rho u_i u_j^{\prime} \right) \]  

Here, \( u \) is velocity, \( \rho \) is density of fluid, \( p \) is pressure, \( \mu \) is turbulent viscosity, \( \rho u_i u_j^{\prime} \) is Reynolds stress.

2.2. Turbulent Model
The turbulence model uses the RNG \( k-\varepsilon \) model, in which the rotation and swirling flow are taken into account. RNG \( k-\varepsilon \) turbulence model can better deal with flow of high strain rate and larger streamline bending degree:

\[ \frac{\partial (\rho k)}{\partial t} + \frac{\partial (\rho ku_i)}{\partial x_j} = \frac{\partial}{\partial x_j} \left[ \alpha_k \mu_{ef} \frac{\partial k}{\partial x_j} \right] + G_k + \rho \varepsilon \]  

\[ \frac{\partial (\rho \varepsilon)}{\partial t} + \frac{\partial (\rho \varepsilon u_i)}{\partial x_j} = \frac{\partial}{\partial x_j} \left[ \alpha_\varepsilon \mu_{ef} \frac{\partial \varepsilon}{\partial x_j} \right] + C_{\varepsilon} \frac{\varepsilon}{k} G_k - C_{2\varepsilon} \rho \varepsilon^2 \]  

\[ \mu_{ef} = \mu + \mu_t \]
\[ \mu_i = \rho C_{\mu} \frac{k^2}{\varepsilon} \] (6)

Here, \( C_{\mu} \), \( \alpha_k \) and \( \alpha \varepsilon \) is empirical constant and they are 0.0845, 1.39 and 1.39, \( k \) is the turbulent kinetic energy generation term, \( \varepsilon \) is the turbulent dissipation rate, \( C_{1\varepsilon} \), \( C_{2\varepsilon} \) are also empirical constant [14,15].

2.3. Cavitation model

The cavitation flow is a two-phase flow with gas and liquid. The bubbles are generated in the low pressure region of the pump, developing along with the water flow and collapsing in the high pressure zone. In this paper, we use the cavitation model created by Schnerr and Sauer [16], according to the Rayleigh-Plesset model. It is an ideal cavitation model without empirical coefficients.

\[ m^+ = \frac{\rho_l \rho_v}{\rho_m} \alpha_v (1-\alpha_v) \left( \frac{2}{3} \frac{p_v - p}{\rho_l} \right)^{\frac{1}{2}} \left( p \leq p_v \right) \] (7)

\[ m^- = \frac{\rho_l \rho_v}{\rho_m} \alpha_v (1-\alpha_v) \left( \frac{2}{3} \frac{p - p_v}{\rho_l} \right)^{\frac{1}{2}} \left( p \geq p_v \right) \] (8)

Here, \( \rho_l \) is density of liquid, \( \rho_v \) is density of bubbles, \( \rho_m \) is density of two-phase fluid, \( \alpha_v \) is volume fraction of bubbles, \( r_b \) is diameter of bubbles, \( p_v \) is the vaporization pressure of a liquid at a certain temperature.

3. Physical model

3.1. Basic parameters
The research object of this paper is LNG pump. The design flow rate \( Q=550 \text{ m}^3/\text{h} \), the head \( H=170 \text{ m} \), the rotating speed \( n=3600 \text{ r/min} \). According to the design requirements, the inducer adopts conical variable-pitch structure, and the main parameters of the inducer are shown in table 1. The four solutions of the different kinds of inducer import sweepback \( \Delta \varphi \) (figure 1), involved in 120°, 170°, 220° and 270°, are numerically calculated. They have the same parameters except the sweepback. According to the parameters of table 1, the inducer is built by Pro/E, and the inducer structure is shown in figure 2.

| Parameters                | Value(mm) |
|---------------------------|-----------|
| Diameter of Flange \( D_f \) | 180       |
| Import diameter of hub \( d_{h1} \) | 52        |
| Export diameter of hub \( d_{h2} \) | 72        |
| Thickness of blade \( \delta \) | 3         |
3.2. Grids discrete
The unstructured mesh with wide adaptability is adapted to mesh the computational field. Due to the complexity of the flow path, the grid is locally encrypted, as shown in figure 3. The whole calculation fluid domain is divided into five parts: inlet section, inducer, impeller, radial guide vane, axial guide vane and outlet section. Each part is connected with interface boundary condition. Through the grid independent check, the total number of grids is determined to be about 4170000 elements, as shown in figure 4.

3.3. Boundary conditions
In this paper, CFX is used to calculate fluid domain, and the velocity and pressure coupling mode is SIMPLE. The import boundary condition is Total Pressure, the development process of the internal cavitation of the pump can be achieved by gradually reducing the total inlet pressure of the pump. The export boundary condition is Mass flow Rate. The non-slip boundary condition is used in the solid
wall, and the standard wall function is adopted in the near wall region. The fluid medium is water at room temperature (25°C), the reference pressure is 0, and the vaporization pressure is 3574 Pa. The residual type is Root Mean Square (RMS), and calculate convergence is based on two: first, all residual are less than $10^{-4}$; second, the export pressure tends to be stable.

4. Results Analysis

4.1. Parameter definition
The head will be sharply reduced when the pump cavitated. The cavitation performance of the pump is usually described by the relationship curve between NPSH$_a$ and the head. In the pump cavitation test, NPSH$_a$ is defined as the effective pressure of the pressure of pump inlet liquid above the pressure of saturated vapor:

$$\text{NPSH}_a = \frac{p_m}{\rho g} + \frac{v_{in}^2}{2g} - \frac{P_v}{\rho g} - \Delta H$$  \hspace{1cm} (9)

Here, $p_m$ is the pump inlet pressure, $v_{in}$ is average speed at the pump inlet, $\Delta H$ is the loss of head.

4.2. Comparison of numerical calculation and experiment
In order to verify the accuracy of the numerical calculation results, the model pump with $\Delta \phi = 270^\circ$ was tested validation as an example. The inducer and test equipment are shown in figure 5. The figure 6 shows the comparison of the hydraulic performance of the model pump and the results of the test. It can be seen that the calculated value of the head and the efficiency is consistent with the change of the test value, and the test value is smaller than the calculated value. The reason is the numerical calculation does not take into account the effects of front and back cavity and surface roughness. The head test values of the model pump are 231.7m, 218.8m, 203.3m, 183.4m and 158.8m respectively at 0.4, 0.6, 0.8, 1.0 and 1.2 times of the design flow conditions. The relative errors of the calculated and experimental values are 1.2%, 0.9%, 1.7%, 1.0% and 0.8% respectively. The relative error of efficiency is 1.6%, 1.4%, 1.8%, 1.7% and 1.7% respectively under different working conditions. Thus the maximum relative error of head is 1.7%, and the maximum relative error of efficiency is 1.8%.

![Figure 5. Pictures of the inducer and test equipment.](image)
The figure 7 shows the cavitation curve of experiment and numerical calculation. The required net positive suction head (NPSHr) is obtained by measuring NPSHr. NPSHr is the NPSH when the pump head drops 3%. It can be seen from figure7 that the results of cavitation numerical calculation are in good agreement with the experimental results. The NPSHr of test is 6.99m, and the NPSHr of calculation is 6.68m and the relative error is 4.4%.

According to the test results of hydraulic performance and cavitation performance, error of model pump is within the allowable range, so the numerical calculation has a certain accuracy, it can be applied to this research work.

4.3. Cavitation evolution process on inducer and impeller cascade
In order to analyze the cavitation evolution process on the inducer and impeller cascade, the bubbles distribution of inducer rim and impeller front cover at different NPSHr is extracted on turbo, as shown in figure8. According to the bubbles distribution and change of head, the cavitation evolution is divided into four processes:

The bubbles firstly appear at the entrance of the inducer, and the resulting bubbles rapidly collapse during the backward flow, no significant change in head. This moment called the initial cavitation stage.

With the decrease of NPSHr, the inducer completely cavitates, and the suction surface of the impeller entrance edge also produce bubbles. At this time, the head drops 3%, which is called the critical cavitation stage. It can be seen from the figure: the distribution of bubbles on each blade is not the same. The impeller channels corresponding to the inducer blades produce the bubbles, while the rest of the flow channels don’t produce bubbles. After the fluid outflow the tail of the inducer blades, the pressure is different between the suction surface and the pressure surface in the impeller blade, resulting in local flow disorder and cavitation unevenness. When the pump head drops 3%, the inducer has been completely cavitated and the inlet of impeller has also been cavitated, indicating that the pump head is mainly provided by the centrifugal wheel.

When NPSHr falls below the NPSHr, all the channels in the impeller produce bubbles, bubbles have occupied 4/5 of the channel in the cavitation serious area, which is called the developing cavitation stage. The figure shows that the bubbles in the inducer did not extend into the impeller channel, indicating that the bubbles collapse after flowing out of the inducer. There is no exclusion of blade to fluid between the inducer and the impeller, resulting in decrease of fluid velocity and increase of static pressure, and bubbles collapse.

When the NPSHr continues to decrease, the impeller has completely cavitated, and the degree of cavitation in each channel is almost same. At the moment, the pump does no longer work properly and cannot produce head, which is called the complete cavitation stage.
4.4. Cavitation evolution process on the surface of the inducer blade

The figure 9 and figure 10, respectively, are the bubble volume fraction distribution on suction surface and pressure surface under the different NPSH$_a$. When the cavitation generate, the bubbles only appear in the suction surface near the inlet side of blade. Because the circumference speed of the situation is maximum. The pressure of it is lowest, so it is more prone to cavitate. As the bubbles flow to the outlet along the channel, the blades continue to work on the fluid. The pressure gradually increases, and the bubbles collapse. With decrease of NPSH$_a$, the bubbles diffuse to the tail to reach more than half of the blade. At the moment, the thickness of the bubbles is gradually increased to the inlet edge of rim of the pressure face. As the NPSH$_a$ continues to decrease, bubbles on suction surface diffuse to the outlet side and extend to the hub. But the first half of the hub does not cavitate, and the bubbles on the pressure surface only extends to half of the rim. The reason of non-cavitation in the first half of the hub is that the peripheral speed of the hub is low and all the while the pressure does not decrease to less than the vaporization pressure. The increase of bubbles on the second half of the suction surface is due to the fact that the fluid pressure is still less than the vaporization pressure though the blades continue to pressurize the fluid, and the liquid is continuously vaporizing and the volume of the void is increasing.
4.5. Effect of inducer sweepback on cavitation performance of centrifugal pump

The figure 11 is the curve of cavitation characteristics with sweepback angle of 120°, 170°, 220° and 270°, respectively. With the increase of the sweepback angle, the cavitation performance of the pump is gradually increased. When fluid enters the inducer, the liquid begins to be affected by the blade and moves to the rim, the pressure gradually increase. So the cavitation performance at the outer edge of the inlet which is most susceptible to cavitation is improved. The greater the sweepback angle, the longer the pressurization process, the better the cavitation performance is. However, the head decrease with decrease of the sweepback angle. Because sweepback angle becomes smaller lead to the decrease of blade wrap angle. And the friction loss is increased, the head is reduced. Each time the sweepback angle is changed by 50°, the cavitation curve changes greatly. The cavitation performance of the sweepback angle of 220° and 270° is obviously higher than that of sweepback angle of 120° and 170°. When the sweepback angle changes from 170° to 220°, the relative amount of NPSH, is 52.8%, and the increase in the amount is not linearly changed with angle. Therefore, in order to further analyze the effect of sweepback on cavitation performance, four sweepback angles of 220°, 240°, 260° and 290° are further selected to do simulation analysis to get more accurate results.

The figure 12 shows the cavitation curves of the six schemes with sweep angles from 200° to 290°. It can be seen that the cavitation curves of the different sweepback angles are basically the same, and when NPSH, is less than NPSH, the head is drastically reduced. The NPSH, for the six sweepback angles from 200° to 290° are 7.30m, 7.17m, 7.03m, 6.95m, 6.90m and 6.91m. The NPSH, at Δφ=270° is the smallest, Δφ=290° and Δφ=260° above it, and the maximum is Δφ=200°. When the sweepback angle is changed by 20°, NPSH, does not exceed 2%. In order to facilitate the analysis of the influence of the sweepback on NPSH, the six schemes are compared with static pressure, streamline, NPSH, and external characteristics.

![Figure 11. Cavitation characteristic curves of pump under different inducer sweepback](image1)

![Figure 12. Cavitation characteristic curves of pump under different inducer sweepback](image2)

![Figure 13. Comparison of static pressure distribution under different inducer sweepback](image3)
4.5.1. The change of static pressure caused by different sweepback. Figure 13 shows the static pressure distribution of the suction surface on the inducer with sweepback angle of 200°, 220°, 240°, 270° and 290° at NPSH\(_a\)=7.30m. It can be seen from the figure that the lowest point of pressure appears in the impeller rim near the inlet edge and the highest point appears in the rim near the outlet edge. As the sweepback increases, the area of the low pressure zone increases gradually, but the pressure at 200°, 220° and 240° increase slow, and the pressure at 260°, 270° and 290° increase rapidly. With the increase of the sweepback, the blade installation angle increases, the degree of blade bending reduce, the blade exclusion reduce, the impeller flow area increase and the anti-cavitation performance improve.

4.5.2. The change of streamline caused by different sweepback. Figure 14 is streamline distribution of the suction surface on the inducer with sweepback angle of 200°, 220°, 240°, 270° and 290° at NPSH\(_a\)=7.30m. It can be seen that the streamline distributes symmetrical in the first half of the flow passage and concentrates at the rim in the second half. Because the first half of the inducer began to work on the fluid, and fluid velocity changes more uniform. While the fluid at the exit passes through the entire inducer leads to a large change in velocity. As the sweepback increases, the concentrated area of the exit streamline gradually moves from the rim to the center. And the flow distribution of the large sweepback is more uniform than the smaller one. Because the longer sweepback angle, the longer the inlet edge is, the more uniform the gradient of the velocity is, the better flow stability is, and the better cavitation performance is.

4.5.3. The change of NPSH\(_r\) caused by different sweepback. Figure 15 shows the NPSH\(_r\) of the pump under different sweepback. The NPSH\(_r\) of the pump is gradually decreasing during the inducer of the sweepback from 200° to 270°, and the change is close to the linearity from 200° to 260°, the changes relatively flat from 260° to 270°. But the NPSH\(_r\), has a slight increase when the sweepback changed from 270° to 290°, indicating that the sweepback has an optimal value. Because the work under the inlet side of inducer reduce when the sweepback decreases, making the most likely to produce the location of the cavitation cannot be improved. The sweepback continues to increase, making the inlet angle of the rim too large. Which increase attack angle, and then the impact also increased, so the cavitation performance is reduced. The optimal sweepback of this pump is 270°. According to the design requirements, cavitation is content when the sweepback from 240° to 290°. So compare the head and efficiency of the four schemes, 240°, 260°, 270° and 290°, as shown in figure 16.

4.5.4. The change of external characteristic caused by different sweepback. It can be seen from figure 16 that the efficiency is slightly reduced with the increase of the sweepback angle. The maximum efficiency is 72.88% at \(\Delta\phi=240°\), and the minimum efficiency is 72.88% at \(\Delta\phi=290°\). The relative change value is not more than 0.2%. Head istiny fluctuation, but the change of it is not obvious. The maximum head is 186.40m at \(\Delta\phi=260°\). Therefore, the effect of sweepback of the inducer on the head and efficiency of the whole pump is very little when the sweepback changes within the large range near the optimal cavitation value.
5. Conclusions

- The cavitation development process under different sweepback is roughly the same in the inducer, and the bubbles firstly appear in the rim of the inducer suction surface near the inlet side. As the NPSH decreases, the bubbles diffuse toward the outlet and the pressure surface and eventually fill the entire flow passage.
- In a certain range, with the increase of the inducer sweepback angle, efficiency, the head and cavitation performance of pump are increasing. When the sweepback reaches a certain value, the efficiency and the head are no longer increased and the cavitation performance is reduced. Thus, there is the optimal sweepback in pump.

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