Effect of inlet sweepback angle on the cavitation performance of an inducer

Xiaorui Cheng, Yibin Li, and Shuyan Zhang

Abstract

In order to study the effects of inlet sweepback angle on the cavitation performance of inducers based on the Reynolds N-S equation, RNG $k-\epsilon$ turbulent model, and Schnerr and Sauer cavitation model, a three-dimensional numerical calculation is employed to study the flow characteristics of a certain LNG pump. Laws of the variation of cavitation performance, head, and efficiency with the change of sweepback were studied. Numerical analysis of the eight inducer projects with a sweepback angle from 120° to 290° was carried out. The results show that the cavitation bubbles first appear at the suction surface near the inlet side. With the decrease of net positive suction head ($NPSH_r$), 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°, the $NPSH_r$ of the pump reduces gradually, that is to say that the anti-cavitation performance of the pump has been improved. However, the $NPSH_r$ of the pump increases gradually when the inducer sweepback angle increases from 270° to 290°. In other words, there is an optimal sweepback from 120° to 290°, and efficiency and head of pump tend to be stable 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 the pressure of vaporization at the local temperature (Guan, 2011). The high-speed rotating blade will reduce the pressure of partial fluid. Meanwhile, it will cause performance drop, produce vibration and noise, and seriously damage the over-current components under cavitation conditions (Hong, Kim, & Kim, 2015). One of the most effective ways to prevent cavitation of a centrifugal pump is to install an inducer before an impeller. The inducer can increase the pressure of the fluid before entering the main impeller, and improve the available net positive suction head ($NPSH_a$) (Torre, Pasini, & Cervone, 2011).

An inducer has great cavitation performance because of a sharper leading edge and a smaller inlet angle. Pace, Valentini, and Pasini (2015) carried out numerical simulation and experiments to study a series of inducers with different blade import installation angles of hub radius and flow coefficients. Akbarian, Najafi, Safari, and Ardabili (2018) invested a dual-fuelled constant-speed engine. A Computational Fluid Dynamics (CFD)-based numerical simulation was performed by KIVA3V, and its results showed good agreements with the experimental results under cylinder pressure. Mou, He, Zhao, and Chau (2017) attempted to ascertain wind pressure distributions on and around various squared-shaped tall buildings by the application of CFD techniques. Guo, Shi, Zhu, and Cui (2016) discovered that the excretion coefficient increases as the blade number increases and the vapor volume fraction with a three-bladed inducer is the lowest in this current work. Guo, Li, Cui, and Zhu (2013) put forward the design method of a splitter-blade inducer, and found that the variable-pitch inducer with splitter-blade has superior cavitation performance through experiments and numerical calculation. Kim and Song (2016) confirmed that cavitation numbers at the initial cavitation decrease with a high transfer parameter. It was suggested the cavitation unsteadiness is eliminated because of the thermodynamic effect, and found that the inducer with small pitch can decrease cavitation unsteadiness (Cervone & D’Agostino, 2009; Ehrlich & Murdock, 2015; Kim, Sung, Choi, & Kim, 2017). Many scholars presented the flow field in the inducer changes due to the presence of cavitation, with the bubbles appearing at the leading edge of blades (Bakir, Koudri, Noguera, & Rey, 2003; Campos-Amezcua et al., 2015).

CONTACT Xiaorui Cheng cxr168861@sina.com

© 2019 The Author(s). Published by Informa UK Limited, trading as Taylor & Francis Group
This is an Open Access article distributed under the terms of the Creative Commons Attribution License (http://creativecommons.org/licenses/by/4.0/), which permits unrestricted use, distribution, and reproduction in any medium, provided the original work is properly cited.
on pump characteristics and 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 in a 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 Averaged N-S equation is used to solve the computational fluid domain. The governing equations in steady state are as follows:

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

\[
\rho u_i \frac{\partial u_i}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left( \mu \frac{\partial u_i}{\partial x_j} + \rho \bar{u}_i \bar{u}_j \right)
\]

Here, \( u \) is velocity, \( \rho \) is density of fluid, \( p \) is pressure, \( \mu \) is turbulent viscosity, \( \rho \bar{u}_i \bar{u}_j \) is Reynolds stress.

2.2. Turbulent model

The RNG \( k-\epsilon \) model acts as a turbulence model, in which the rotation and swirling flow both are taken into account. RNG \( k-\epsilon \) turbulence model can better deal with the flow of a high strain rate and larger streamline bending degree:

\[
\frac{\partial (\rho k)}{\partial t} + \frac{\partial (\rho k u_i)}{\partial x_i} = \frac{\partial}{\partial x_j} \left[ \alpha_k \mu_{\text{eff}} \frac{\partial k}{\partial x_j} \right] + G_k + \rho \epsilon
\]

\[
\frac{\partial (\rho \epsilon)}{\partial t} + \frac{\partial (\rho \epsilon u_i)}{\partial x_i} = \frac{\partial}{\partial x_j} \left[ \alpha_\epsilon \mu_{\text{eff}} \frac{\partial \epsilon}{\partial x_j} \right]
\]

\[
+ \frac{C_{1\epsilon}}{k} \epsilon G_k - C_{2\epsilon} \rho \frac{k^2}{\epsilon}
\]

\[
\mu_{\text{eff}} = \mu + \mu_1
\]

\[
\mu_1 = \rho C_p \frac{k^2}{\epsilon}
\]

Here, \( C_\mu, \alpha_k, \) and \( \alpha_\epsilon \) are an empirical constant and they are 0.0845, 1.39, and 1.39 respectively, \( k \) is the turbulent kinetic energy generation term, \( \epsilon \) is the turbulent dissipation rate, \( C_{1\epsilon}, C_{2\epsilon} \) is also an empirical constant (Liu, Wang, Yuan, & Wang, 2013; Wang, 2011).

2.3. Cavitation model

The cavitation flow is a two-phase flow with gas and liquid. The bubbles generate in the low pressure region of the pump, developing with the water flow and collapsing in the high pressure zone. In this paper, we use the cavitation model created by Schnerr and Sauer based on the Rayleigh-Plesset model (Liu, Liu, Wang, Wu, & Zhuang, 2012). It is an ideal cavitation model without empirical coefficients.

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

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

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, and \( p_c \) is the vaporization pressure of liquid at a certain temperature. The Schnerr-Sauer cavitation model couples the gas volume fraction with the density of the bubble and directly calculates the gas-liquid transport equation. Cavity number density is the only parameter in the model that needs to be determined. A large number of studies have confirmed that the best value is around \( 10^{13} \), so we would like to ensure that the cavity number density is around \( 10^{13} \) to improve accuracy.

3. Physical model

3.1. Basic parameters

An LNG pump is a main object of research in this paper. The design flow rate is \( Q = 550 \text{ m}^3/\text{h} \), the head \( H = 170 \text{ m} \), and the rotating speed \( n = 3600 \text{ r/min} \). According to the design requirements, the inducer adopts a conical variable-pitch structure, and the main parameters of the inducer are shown in Table 1. The blade inlet edge profile of this paper refers to sweepback \( \Delta \varphi \) (Figure 1). The four solutions of the different kinds of inducer import sweepback involved \( 120^\circ, 170^\circ, 220^\circ, \) and \( 270^\circ \) and are numerically calculated. They all have the same parameters except sweepback. According to the parameters of Table 1, the inducer is built by Pro/E and the inducer structure is shown in Figure 2. The concrete variable law of variable-pitch structures is shown in Figure 3. The blade tip clearance between the inducer blade and stationary wall is 0.5 mm. At the same time,

---

**Table 1. Main dimensions of the inducer.**

| Parameters                  | Value (mm) |
|-----------------------------|------------|
| Diameter of flange \( D_h \) | 180        |
| Import diameter of hub \( d_{h1} \) | 52         |
| Export diameter of hub \( d_{h2} \) | 72         |
| Thickness of blade \( \delta \) | 3          |
the inlet angle is determined by the inducer inflow condition, and the exit angle is determined by the proposed head value. The detailed geometric parameters of each flow component are as follows: the impeller inlet diameter is 200 mm and the impeller outlet width is 24 mm. The diameter of the impeller is 365 mm. The inlet and outlet widths of the radial guide vane are 31 and 17 mm respectively. The outlet diameter of the radial guide vane is 438 mm. The inlet and outlet angle of the axial guide vane are 8° and 90°, and the subtended angle of the axial guide vane is 40°. The blade number of the axial guide vane is 16.

### 3.2. Mesh generation

The grid of the computational field is presented in Figure 4. Due to the complexity of the flow path, the grid is refined locally. The whole calculation fluid domain is divided into six parts: inlet section; inducer; impeller; radial guide vane; axial guide vane; and outlet section. Every part is connected by interface boundary conditions. A mesh independence study is performed before the numerical calculation, as shown in Table 2. The head increases first and then tends to be stable with the increase in the number of grids. The change of the head is less than 0.2% when the number of grids is more than 4167283. The total number of grids is determined to be...
Table 2. Mesh independence test.

| Number of grids | 1532843 | 2052849 | 2658293 | 3428393 | 4167283 | 4726283 |
|-----------------|---------|---------|---------|---------|---------|---------|
| Head/m          | 179.43  | 182.80  | 185.11  | 186.55  | 186.79  | 186.81  |

Figure 5. Grids of the computational domain.

4167283, as shown in Figure 5. The local flow parameters, such as velocity and pressure, have been taken into account when the local grid is refined. In view of the research work required in this paper and the amount of calculation, the local and overall encryption of the grid are not further elaborated.

3.3. Boundary conditions

In this paper, CFX14.5 is used to calculate fluid domain, and the velocity and pressure coupling mode is SIMPLE Algorithm. The inlet boundary condition is Total Pressure, and the development process of the internal cavitation in the pump can be achieved by reducing the total import pressure of the pump gradually. The outlet 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 25°C water. Its density is 997 kg/m³ and its dynamic viscosity is 0.8949 × 10⁻³ Pas. The reference pressure is 0, and the vaporization pressure is 3574 Pa. The density of vapor is 0.023 kg/m³ under the 25°C condition. The residual type is Root Mean Square (RMS) and calculated convergence is based on two aspects: one is all residual and are less than 10⁻⁴; the other is the export pressure, which tends to be stable.

4. Results analysis

4.1. Parameter definition

The head will reduce sharply when the pump has cavitation. The cavitation performance is usually described by the relationship curve between NPSHₐ and the head. In the pump cavitation test, pumps get this required energy from the system to which the pump is attached and we designate this system energy as NPSHₐ (Askew, 2006):

\[
NPSHₐ = \frac{p_{in}}{\rho g} + \frac{v_{in}^2}{2g} - \frac{p_v}{\rho g} - \Delta H
\]

Here, \( p_{in} \) is the pump import pressure, \( v_{in} \) is average speed at the inlet of pump, and \( \Delta H \) 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 \( \Delta \varphi = 270° \) was used to verify the test as an example. The inducer and test system are shown in Figure 6. Figure 7 shows the comparison of the hydraulic performance in 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 has not taken the effects of front and back cavity and surface roughness into account. The head test values of the model pump are 231.7 m, 218.8 m, 203.3 m, 183.4 m, and 158.8 m 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. Under different working conditions, the relative error of efficiency is 1.6%, 1.4%, 1.8%, 1.7%, and 1.7% respectively. Thus the maximum relative error of head is 1.7% and the maximum relative error of efficiency is 1.8%.

Figure 8 shows the cavitation curve of the experiment and numerical calculation. Technically speaking, as long as a bubble is generated by the flow, it can be considered that the pump experiences cavitation. However, the experiment found that even if more bubbles are generated, they will not affect the performance of the pump. When the cavitation in the pump develops to a considerable scale it will affect pump performance. At the same time, when the head is reduced by 3%, the corresponding NPSH$_a$ is defined as the NPSH$_r$, which is the result of a large number of practices and experiments. After the state of critical cavitation, the pump's head will drop rapidly, and vibration and noise will be generated at the same time, which is obviously unfavorable for the operation of the pump. Therefore, the pump is not allowed to operate for a long time under this condition. It can be seen that cavitation numerical calculations are in good agreement with the experimental results. The NPSH$_e$ of the test is 6.99 m and the NPSH$_c$ of the calculation is 6.68 m; the relative error is 4.4%.

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

4.3. Cavitation evolution processes on inducer and impeller cascades

In order to analyze the cavitation evolution process on the inducer and impeller cascade, the bubble distribution in the inducer rim and impeller front covering at different NPSH$_a$ is extracted on turbo, as shown in Figure 9. According to the bubble distribution and change of head, the cavitation evolution is divided into four stages: 1) The bubbles appear first at the entrance of the inducer, and the resulting bubbles collapse rapidly during the backward flow, with no significant change in head. This moment is called the initial cavitation stage; 2) The inducer has complete cavitation with the decrease in NPSH$_a$, and the suction surface of the inducer entrance also produces bubbles. At this time, the head drops 3%, which is called the critical cavitation stage. This can be seen from the figure: the distribution of bubbles is different on every impeller channel. The impeller channels produce bubbles corresponding to the inducer blades, while the rest of the flow channels don't produce bubbles. After the fluid outflow at the tail of the inducer blades, the pressure is different between the suction surface and the pressure surface on the inducer blade, which results in disorder at local flow and an unevenness of cavitation. The inducer has completed cavitation when the pump head drops 3% and so does the inlet of the impeller, which indicates that the pump head is mainly provided by the impeller; 3) When NPSH$_a$ falls below the NPSH$_r$, all the channels of the impeller produce bubbles, and bubbles occupy four-fifths of the channel, which is called the developing cavitation stage. The figure shows that the bubbles did not extend into the impeller channel in the inducer, which indicates that the bubbles collapse after flowing out from the inducer. There is no exclusion of blades to fluid between the inducer and the impeller, which results in the decrease of fluid velocity and an increase of static pressure; 4) The impeller has completed cavitation and the degree of cavitation is almost the same in each channel when the NPSH$_a$ continues decreasing. At that moment, the pump no longer works properly
and cannot produce head, which is called the complete cavitation stage.

4.4. Cavitation evolution processes on the surface of the inducer blade

Figures 10 and 11 are the bubble volume fraction distribution on suction surface and pressure surface under the different NPSH$_a$. The bubbles only appear on the suction surface near the inlet side of the blade when cavitation is generated, because the circumferential speed of the situation is at its maximum. Its pressure is at its lowest, so it is more prone to have cavitation. 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. The bubbles diffuse to the tail to reach more than half of the blade with a decrease in NPSH$_a$. At the moment, the thickness of the bubbles increases gradually to the pressure face. As the NPSH$_a$ continues to decrease, bubbles diffuse to the outlet side and extend to the hub on the suction surface. But the first half of the hub does not cavitate, and the bubbles only extend to half of the rim on the pressure surface. The reason for non-cavitation in the first half of the hub is that the peripheral speed of the hub is low and the pressure does not decrease to less than the vaporization pressure. The increase in 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, although the blades pressurize the fluid. The liquid is vaporizing continuously and the volume of the void increases.
4.5. Effect of inducer sweepback on the cavitation performance of a centrifugal pump

Figure 12 is the curve of cavitation characteristics with sweepback angles of 120°, 170°, 220°, and 270°, respectively. The cavitation performance of the pump gradually increases with an increase in the sweepback angle. The blades begin to work on liquid and the pressure increases gradually when fluid enters the inducer and continues to move towards the rim. Thus, the cavitation performance is improved at the outer edge of the inlet which is most susceptible to cavitation. The greater the sweepback angle, the longer the pressurization process, and the better the cavitation performance. However, the head decreases with a decrease in the sweepback angle, because the decrease of the sweepback angle leads to the decrease of the blade wrap angle. The friction loss is increased and the head is reduced. Each time the sweepback angle is changed by 50°, the cavitation curve changes greatly. The cavitation performances of sweepback angles of 220° and 270° are obviously higher than that of sweepback angles of 120° and 170°. The relative amount of NPSH_r is 52.8% when the sweepback angle changes from 170° to 220°, and the increase in the amounts do not change linearly with angle. Therefore, in order to further analyze the effect of sweepback on cavitation performance, four sweepback angles of 200°, 240°, 260°, and 290° are further selected to do simulation analysis to get more accurate results.

Figure 13 shows the cavitation curves of the six schemes with sweepback angles from 200° to 290°. It can be seen that the cavitation curves are basically the same under the different sweepback angles, and the head is reduced drastically when NPSH_a is less than NPSH_r. The NPSH_r for the six sweepback angles from 200° to 290° are 7.30 m, 7.17 m, 7.03 m, 6.95 m, 6.90 m, and 6.91 m.

The difference between 270° and 290° is only 0.01 m, so the difference is the variable trend. The NPSH_r at ∆φ = 270° is the smallest, ∆φ = 290° and ∆φ = 260° above it, and the NPSH_r at ∆φ = 200° is at maximum. When the sweepback angle is changed by 20°, NPSH_r does not exceed 2%. In order to analyze thoroughly the influence of sweepback on NPSH_r, the six schemes are compared with static pressure, streamline, NPSH_r, and external characteristics.

4.5.1. The change in static pressure caused by different sweepback

Figure 14 shows the static pressure distribution of the suction surface on the inducer with a sweepback angle at NPSH_a = 7.30 m. It can be seen from the figure that the lowest point of pressure is in the impeller rim near the inlet edge and the highest point is 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° increases slowly; the pressure at 260°, 270°, and 290° increases rapidly. The increase in sweepback causes the blade installation angle to increase, the degree of blade bending to reduce, the blade exclusion to reduce, the impeller flow area to increase, and the anti-cavitation performance to improve.

4.5.2. The change of streamline caused by different sweepbacks

Figure 15 is streamline distribution of the suction surface on the inducer with the sweepback angle at NPSH_a = 7.30 m. It can be seen that the streamline distributes symmetrically in the first half of the flow passage and intensively at the rim in the second half. Because the first half of the inducer began to work on the fluid, and fluid velocity changes are more uniform, the fluid at the exit passes through the entire inducer and 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 of the longer sweepback angle, the longer the inlet edge, the more uniform the gradient of the velocity, the better the flow stability, and the better the cavitation performance.

4.5.3. The change in NPSHr caused by a different sweepback

Figure 16 shows the NPSHr of the pump under a different sweepback. The NPSHr of the pump gradually decreases during the inducer of the sweepback from 200° to 270°; the change is close to the linearity from 200° to 260°. The changes are relatively flat – from 260° to 270°. The NPSHr had a slight increase when the sweepback changed from 270° to 290°, which indicates that the sweepback has an optimal value. Because the work ability of the inducer inlet side reduces when the sweepback decreases, it means the places which cavitate easily cannot be improved. The unceasing increase in sweepback makes the angle of the rim too large, which increases the attack angle; the impact is then also increased, and so the cavitation performance is reduced. The optimal sweepback of this pump is 270°. According to the design requirements, cavitation is contented when the sweepback is from 240° to 290°. The head and efficiency of the four schemes are compared in Figure 16.

4.5.4. The change in external characteristics caused by a different sweepback

It can be seen from Figure 17 that the efficiency reduces slightly with the increase in the sweepback angle. The maximum efficiency is 72.88% at Δϕ = 240°, and the minimum efficiency is 69.03% at Δϕ = 270°. The relative change value is not more than 0.2%. Head has tiny fluctuation, but the change is not obvious. The maximum head is 186.40 m at Δϕ = 260°. Therefore, the effect of the 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.
In order to verify that the cavitation performance of the pump is increased when the inducer is installed, the cavitation performance of the pump without inducer needs to be calculated and compared with the model pump with inducer at $\Delta \varphi = 270^\circ$, as shown in Figure 18. The NPSH$_r$ of a pump with inducer is 6.9 m and it is 8.0 m without inducer, which indicates the NPSH$_r$ reduces 1.1 m after installing an inducer. In Figure 18, four sets of points (NPSH$_a$ from point 1 to point 4 are 9.5 m, 8.0 m, 7.3 m, and 6.9 m) are taken to show cavitation distribution in flow channels of pump respectively.

Figure 19 shows the isosurface at 10% of the gas volume fraction. The impeller without an inducer has critical cavitation at NPSH$_a = 8.0$ m, but the impeller with the inducer does not have cavitation. The impeller with inducer produces bubbles at the entrance at NPSH$_a = 6.9$ m, but the impeller without inducer has the cavitation phenomenon completely and cannot work. It is noteworthy that the bubbles in the impeller without the inducer increase slowly and the head decreases slowly, while for the pump head with the inducer the drop is very sharp when the head reduces from the maximum value to zero. The localized cavitation of the inducer has a very small effect on pump performance (namely, head), as shown in Figure 19 (a–c). However, there is no significant drop in the head of the pump at this time. This is mainly because of the structure and characteristics of the inducer. At this point, the bubbles generated by the localized cavitation of the inducer will gather at the position of the impeller rim, but there is no bubble in the main passage of the inducer hub to the rim. Therefore, the pump can still work normally under this working condition.
The cavitating bubble would fill the whole passage only when cavitation occurs seriously; meanwhile, the performance of the pump will be greatly affected (as shown in Figure 19(d), the head of the pump drops sharply).

5. Conclusions and prospects

1. The cavitation evolution process under different sweepbacks is the same in the inducer, though one should compare the cavitation performance curves under different sweepbacks. The bubbles appear first at the suction surface rim near the inlet side. As the NPSH₄ decreases, the bubbles diffuse toward the outlet and the pressure surface and eventually fill the entire flow passage.

2. In a certain range, efficiency, the head, and the cavitation performance of the pump increase with the increase in the inducer sweepback angle, and the pressure increases gradually because the blades begin to work on liquid when fluid enters the inducer and continues to move towards the rim. So the cavitation performance is improved at the outer edge of the inlet which is most susceptible to cavitation. However, the increasing sweepback makes the inlet angle of the rim too large, which increases the attack angle as well as the impact. So the cavitation performance is reduced when the sweepback exceeds the certain range. The sweepback exists at the optimal value.

3. Changing the sweepback and the number of blades of the inducer and the impeller will change the flow state and flow angle in the flow passage. However, the blade angle of the impeller is not designed for different flow angles, so the impeller should be redesigned after getting the optimal sweepback and blade numbers. The actual operating temperature of the LNG pump is 1600°C, and the medium is liquefied natural gas, which is quite different from the 25°C water used in this paper. This makes the conclusion inaccurate, so it should be carried out with LNG as the medium under the conditions allowed by the test.

Disclosure statement

No potential conflict of interest was reported by the authors.

Funding

This work was supported by the National Natural Science Foundation of China (No. 51469013 & No. 51369015).

ORCID

Xiaorui Cheng http://orcid.org/0000-0003-4063-0501
Shuyan Zhang http://orcid.org/0000-0002-1653-6809

References

Akbarian, E., Najafi, B., Jafari, M., & Ardabili, S. F. (2018). Experimental and computational fluid dynamics-based numerical simulation of using natural gas in a dual-fueled diesel engine in a dual-fueled diesel engine. Engineering Applications of Computational Fluid Mechanics, 12(1), 517–534.

Askew, J. (2006). Calculating NPSHA in pumping – the “think method”. World Pumps, 2006, 20–25.

Bakir, F., Kouidri, S., Noguera, R., & Rey, R. (2003). Experimental analysis of an axial inducer influence of the shape of the blade leading edge on the performances in cavitation regime. Journal of Fluids Engineering, 125(2), 293–301. doi:10.11109/BALTIC.2012.6249190

Campos-Amezcua, R., Bakir, F., Campos-Amezcua, A., Kellladi, S., Palaciosgallegos, M., & Rey, R. (2015). Numerical analysis of unsteady cavitating flow in an axial inducer. Applied Thermal Engineering, 75(2), 1302–1310. doi:10.1016/j.applthermaleng.2014.07.063

Cervone, A., & D’Agostino, L. (2009). Cavitation and flow instabilities in a 4-bladed axial inducer designed by means of a reduced order analytical model. Aiaa Journal. doi:10.2514/6.2011-5781

Ehrlich, D. A., & Murdock, J. W. (2015). A dimensionless scaling parameter for thermal effects on cavitation in turbopump inducers. Journal of Fluids Engineering, 137(4), 103–111. doi:10.1115/1.4029260

Guo, X. F. (2011). Modern pump theory and design. Beijing: Aerospace Publishing House.

Guo, X. M., Li, Y., Cui, B. L., & Zhu, Z. (2013). Research on the rotation cavitation performance of high-speed rotation centrifugal pump with different pre-positioned inducers. Acta Aeronautica Et Astronautica Sinica, 34(7), 1572–1581. doi:10.7527/S1000-6893.2013.027

Guo, X. M., Shi, G. P., Zhu, Z. C., & Cui, B. L. (2016). Effects of the number of inducer blades on the anti-cavitation characteristics and external performance of a centrifugal pump. Journal of Mechanical Science and Technology, 30(7), 3173–3181. doi:10.1007/s12206-016-0510-1

Hong, S. S., Kim, D. J., & Kim, J. S. (2015). Study on inducer and impeller of a centrifugal pump for a rocket engine turbopump. International Journal of Turbo & Jet-Engines, 227(3), 311–319. doi:10.1177/0954406212449939

Kim, J., & Song, S. J. (2016). Measurement of temperature effects on cavitation in a turbopump inducer. Journal of Fluids Engineering, 138(1), 304–311. doi:10.1115/1.4030842

Kim, D. J., Sung, H. J., Choi, C. H., & Kim, J. S. (2017). Cavitation instabilities of an inducer in a cryogenic pump. Acta Astronautica, 132, 19–24. doi:10.1016/j.actaastro.2016.12.007

Liu, H., Liu, D., Wang, Y., Wu, X., & Zhuang, S. (2012). Applicative evaluation of three cavitating models on cavitating flow calculation in centrifugal pump. Transactions of the Chinese Society of Agricultural Engineering, 28(16), 54–59. doi:10.3969/j.issn.1002-6819.2012.16.009

Liu, H., Wang, Y., Yuan, S., & Wang, J. (2013). Assessment of a turbulence model for numerical predictions of sheet-cavitating flows in centrifugal pumps?. Journal of Mechanical Science and Technology, 27(9), 2743–2750. doi:10.1007/s12206-013-0720-8

Mou, B., He, B. J., Zhao, D. X., & Chau, K. W. (2017). Numerical simulation of the effects of building dimensional
variation on wind pressure distribution. *Engineering Applications of Computational Fluid Mechanics*, 11(1), 293–309. doi:10.1080/19942060.2017.1281845

Pace, G., Valentini, D., & Pasini, A. (2015). Geometry effects on flow instabilities of different three-bladed inducers. *Journal of Fluids Engineering*, 137(4), 304–316. doi:10.1115/1.4034096

Torre, L., Pasini, A., & Cervone, A. (2011). Continuous spectrum of the rotodynamic forces on a four bladed inducer. *Journal of Fluids Engineering*, 133(12), 101–111. doi:10.1115/1.4005258

Wang, F. (2011). *Computational fluid dynamics analysis–CFD software principle and application*. Beijing: Tsinghua University Press.