Influence of Forward Skew Blade Angle on Positive Slope Characteristics of Mixed Flow Pumps

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Abstract. Under the part-load conditions, the flow instability in a mixed-flow pump could lead to strong pressure fluctuations and vibrations, posing a great threat to the safety of the system. The current paper investigates effect of the forward skew blade angle on the positive slope characteristic in a high specific speed mixed flow pump using Reynolds averaged Navier-Stokes (RANS) equations coupled with the SST k-ω model. The simulation results are compared with the experimental data and show good accordance. In the present study, five types of the mixed-flow pumps with different skew blade angles are prepared based on the conventional pump. It is found that with the increasing of the skew angle, the positive slope region becomes wider and moves towards a deeper part-load region. Further investigations are carried out by calculating the impeller head. Results depict that compared with the dynamic pressure head \( H_{pd} \), the static pressure head \( H_{st} \) plays more significant role in the formulation of the positive slope characteristic (PSC). Besides, the relative static pressure head \( H_{rel} \) drops dramatically while the pump operates under the positive slope region, which matches with the pump performance well. Finally, analysis on the spanwise velocity profiles are further investigated in this paper, and results indicate that the flow accumulated near the impeller hub is one of the reasons to alleviate the head drop and shift the positive slope region to a deeper part-load condition.

1. Introduction

The flow instability such as surging, rotating cavitation, and rotating-stall are quite common in pumps, which have been investigated for decades. The positive slope characteristic is likely to be captured in the hydraulic performance curves as the pumps operate under part-load conditions, attracting great interest of the researches. In a high specific speed mixed flow pump, however, the positive slope characteristic is more likely to be observed,[1] which would induce significant pressure oscillations and threaten the safety of the system. Therefore, it is of great significance to analyze and control the positive slope characteristics in a mixed flow pump.
In the past years, many researches have been carried out by experiments and computational fluid dynamics (CFD). Hagiya et al [2] discussed the causes of positive slope characteristic (PSC) from the perspective of the angular momentum using large eddy simulation (LES) in a mixed flow pump, and found that the angular momentum becomes negative at the flow rate of the starting point of positive slope characteristics, and when the flow rate decrease, amount of work that decreases at the leading edge is larger than that which increases at the trailing edge. Nishi et al. [3] investigated the interaction of the internal flow and stall limit of two types of mixed flow pump impellers of different blade load distributions by experiments and numerical simulations. He pointed out that the instability characteristics are generated by a reverse flow resulting from wall stall in the impeller. Kurokawa et al. [4] proposed a formulation to describe a reverse flow in mixed flow pumps at low flow rate conditions, and this model could predict the Euler head and shaft power successfully. Although many investigations on the mixed flow pumps with sweepback blades have been conducted, the influence of the impeller with forward skew angle blade on the pump hydraulic performance, especially the positive slope characteristics, and the internal flow have not been studied much. Therefore, this study prepared 5 types of the mixed flow pumps with different forward skew angles and compared their hydraulic performance and the internal flow to explore the effect of the forward skew angle on the Positive Slope Characteristic. This paper is organized as follows. The design method of the mixed flow pump is introduced first. Problem description are described next. Then the results and discussion on the pump hydraulic performance, diffuser loss, impeller head and spanwise velocity profiles are illustrated. Finally, the conclusions are drawn.

2. Modification method
As shown in the Figure 1, the skew angle is defined as the angle between tip and hub of the leading edge, as seen from the impeller center. The blade shape was prepared from the blade of pump No.3, which is designed previously [5], by tilting the blade in the rotation direction without changing the cross-sectional shape of the blade in each span. Owing to the apparent PSC in the conventional pump, the current paper prepared the impeller skew angle to 14, 21, 35, and 42 degree respectively, corresponding to the Pump No.1, 2, 4, and 5. to alleviate the PSC in the mixed flow pump. The impeller of these five pumps are shown in the Figure 1. Detailed specifications on the pump No.3. is listed in the Table1.

![Figure 1. Impeller shapes.](image-url)
### Table 1. Pump specification

| Specification                                      | Value   |
|----------------------------------------------------|---------|
| Number of impeller blade                          | 5       |
| Number of diffuser blade                          | 7       |
| Impeller tip diameter at inlet (mm)               | 253.6   |
| Impeller tip diameter at outlet (mm)              | 299.4   |
| Head at the design point (m)                      | 12.82   |
| Flow rate at the design point (m$^3$/min)         | 16      |
| Specific speed at the design point (1/min, m$^3$/min, m) | 885.5   |

#### 3. Problem description

The computational domain including inlet pipe, impeller, diffuser and outlet pipe, is shown in the Figure 2. To save the computational resources, single passage is applied in current study. The corresponding meshes were generated by ANSYS TurboGrid 19.0. Figure 3 shows the mesh generation of the pump, and Table 2 shows the detailed mesh information.

![Figure 2. Computational domain](image)

**Figure 2. Computational domain**

| Flow domains    | Number of nodes |
|-----------------|-----------------|
| Inlet pipe      | 420000          |
| Impeller        | 730139          |
| Diffuser        | 787754          |
| Outlet pipe     | 1613267         |
| Total           | 3657376         |

(a) Mesh on impeller of pump No.3  
(b) Mesh on diffuser blade

**Figure 3. Mesh configuration**
In current study, steady simulations are carried out based on the Reynolds averaged Navier-Stokes (RANS) equations coupled with the SST $k-\omega$ model. Water at 25°C is selected for the flow medium. As for the boundary conditions, the total pressure is imposed at the inlet and the mass flow rate is assigned at the outlet. To reduce the computational cost, rotational periodicity is applied for the periodic span. A rotational coordinate system is set for the area of pump impeller, and a stationary system is set for other stationary components. The mixing-plane is applied on the interface.

4. Result and Discussion

4.1. Hydraulic performance curves

The figure 4 shows the simulation results of the pump performance for the 5 pumps with different skew angles as well as the Experimental data of the Pump No. 3. The head, efficiency and the shaft power are nondimensionalized by the corresponding value under the best efficiency point (bep) condition, i.e., $H_{\text{bep}}$, $\eta_{\text{bep}}$ and $P_{\text{bep}}$ in the Pump No.3. It can be noted that simulation result matches with the experimental data well.

![Figure 4. Performance curve of each pump and experimental result](image)

As shown in the Figure 4, the PSC is captured in all pumps. As the flow rate coefficient is higher than the positive slope region, the head coefficient and the shaft power coefficient decrease with the augment of the skew angle, i.e., from Pump No.1 to Pump No. 5. However, the efficiency coefficient doesn’t change much. It can be also noted in the Figure 4 (a) that with the increase of the skew angle, the positive slope region becomes wider and moves towards a deeper part-load region. The head drop ($\Delta H$) of each pump is alleviated obviously. Comparisons on the positive slope region and the head drop are listed in the Table 3.
Table 3. Flow rate region and head drop at the PSC

| Pump       | Positive slope region (Q/Q_{bep}) | Head drop (ΔH/H_{bep}) |
|------------|-----------------------------------|------------------------|
| Pump No.1  | 0.485–0.545                       | 0.240                  |
| Pump No.2  | 0.454–0.515                       | 0.211                  |
| Pump No.3  | 0.424–0.545                       | 0.181                  |
| Pump No.4  | 0.393–0.545                       | 0.132                  |
| Pump No.5  | 0.393–0.636                       | 0.113                  |

4.2. Diffuser loss

The diffuser loss defined in equation (1) is shown in the Figure 5.

\[ h_{\text{loss}} = \frac{P_{t2} - P_{t1}}{\rho g} \]  

where \( P_{t1} \) are total pressure at the inlet and outlet of diffuser shown in figure 6.

![Diffuser loss curve](image)

**Figure 5.** Diffuser loss curve

It can be noted in the figure 5 that the energy loss in the diffuser of the Pump No. 1 is slightly higher than the Pump No. 5 under the part-load conditions, which could cause the pump head difference as shown in the Figure 4 (a). A slight drop of diffuser loss is observed in the Pump No. 1, while no obvious positive slope region is observed in the Pump No. 5. Thus, it is essential to calculate the impeller head and investigate the flow pattern in the impeller.

4.3. Impeller head

As discussed above, the Pump No.1 and Pump No.5 are selected as the representatives for further analysis. The formulation of the impeller head \( H_{\text{imp}} \) is defined in the equation (2).

\[ H_{\text{imp}} = \frac{(v_2^2 - v_1^2) + (u_2^2 - u_1^2) + (w_2^2 - w_1^2)}{2g} \]  

where \( v, u, w \) are the peripheral velocity, the absolute velocity, the relative velocity, respectively. The subscript 1 represents the impeller inlet and 2 represents the impeller outlet.
Further, the impeller head can be divided into two parts, i.e., static pressure head $H_{st}$ and dynamic pressure head $H_d$.

$$H_{st} = \frac{(u_2^2 - u_1^2) + (w_1^2 - w_2^2)}{2g}$$ (3)

$$H_d = \frac{v_2^2 - v_1^2}{2g}$$ (4)

The profiles of the $H_{imp}$, $H_{st}$ and $H_d$ are presented in the Figure 6.

As indicated in the Figure 6(a), the impeller head of the Pump No. 1 is higher than the Pump No. 5 overall, especially under the high part-load conditions and large flow rate conditions. The positive slope region shifts to the lower flow rate condition with a larger skew angle. These results are accordance with the results in the Figure 4. Thus, the impeller head would reflect the pump head to some extent. In Figure 6 (b) and (c) present the static pressure head $H_a$ profiles and dynamic pressure head $H_d$ respectively. It can be also noted that the PSC can be observed in both dynamic pressure head and the static pressure head of the Pump No.1 while the PSC is only captured in the static pressure head. In addition, the static pressure head difference between these two pumps is apparent while the dynamic pressure head is almost the same. This indicates that the static pressure plays more significant role in the head difference in the impeller head and pump head.

The static pressure head $H_a$ is divided into relative static pressure head $H_{rel}$ and peripheral static pressure head $H_{per}$ further, as shown in the equation (5) ~ equation (6). The results are displayed in the Figure 7. It can be noted that PSC is observed in the relative static pressure head $H_{rel}$ profile, while it was not observed in the relative static pressure head $H_{rel}$. Therefore, it can be concluded that the relative velocity contributes to the head drop of the pump to the most extend.

$$H_{rel} = \frac{w_1^2 - w_2^2}{2g}$$ (5)
\[ H_{\text{per}} = \frac{u_2^2 - u_1^2}{2g} \quad (6) \]

**Figure 7.** Static pressure profiles

(a) Relative velocity pressure
(b) Peripheral velocity pressure

**4.4. Examination of velocity components in span direction**

Figure 8 shows the spanwise velocity profiles in the Pump No.1 and Pump No.5. Several non-dimensional parameters, i.e., absolute velocity ratio \( R_v \), relative velocity ratio \( R_w \) and angular momentum ratio \( L_u \) are defined as:

\[ R_{v_i} = \frac{v_i r}{r_2 u_2} \quad (7) \]
\[ R_{w_i} = \frac{w_i r}{r_2 u_2} \quad (8) \]
\[ L_{u_i} = \frac{u_i v_\theta v_{m_i} r}{r_2 v_{m-\text{ave2}} u_2^2} \quad (9) \]

where \( v_1, v_2; w_1, w_2; u_1, u_2; v_{m1}, v_{m2}; r \) is radius; \( r_2 \) is impeller tip radius at outlet, and \( v_{m-\text{ave2}} \) is the area averaged meridional velocity at the outlet of impeller.

These parameters mentioned above are obtained by calculating the area averaged value at each span in inlet and outlet of the impeller domain as shown in Figure 8. The angular momentum ratio is also calculated on each span.

**Figure 8.** Interfaces of impeller domain divided in span direction

(a) Impeller inlet
(b) Impeller outlet
Figure 9. Spanwise velocity profiles under different conditions
Figure 9 (a) shows that the velocity and angular momentum ratio profiles predicted by the two pumps are similar. It can be noted that a velocity hysteresis is observed in the Pump No. 1 as shown in the Figure 9 (a), the flow in the Pump No.5 blade is more likely to shift to the hub side (span direction = 0). The same results can be also observed in the Figure 9 (b). Therefore, it is considered that the flow of Pump No.5 is relatively closer to the hub side than that of Pump No.1 under the large flow rate conditions. The contour of meridional velocity is also shown in Figure 10. The stronger reverse flow is observed near the pump hub of the Pump No. 5 which matches with the result of Figure 9 (a) and Figure 9 (b).

![Figure 10. Velocity distribution on the meridional plane ($Q/Q_{bep} = 0.545$)](image)

Under hump condition, as shown in the Figure 9 (c), the maximum and minimum of the velocity is located at the same spanwise region. Therefore, as the PSC occurs, the internal flow pattern is similar in the two pumps. Figure 11 further shows that the contour of meridional velocity of the two pumps. This Figure indicates that the flow pattern is similar between the both pumps. The uniformity of the Pump No.5 is relatively higher than that of Pump No.1. Therefore, the impeller head is larger in the Pump No.5. Consequently, at the hump condition, the impeller head of pump No.5 is higher.

![Figure 11. Velocity distribution on the meridional plane ($Q/Q_{bep} = 0.485$(No.1), 0.394(No.5))](image)
5. Conclusion
In this paper, numerical investigations on the positive slope characteristic are investigated considering the forward skew blade angle in a mixed-flow pump. The main findings are listed as follows:
(1) With the increasing of the skew angle, the positive slope region becomes wider and moves towards a deeper part-load region.
(2) The loss in the impeller is the main reason inducing positive slope region in the mixed-flow pump.
(3) Compared with the dynamic pressure head $H_d$, the static pressure head $H_s$ plays more significant role in the formulation of the positive slope characteristic (PSC).
(4) Investigations on the spanwise velocity profiles indicate that the flow accumulated to the impeller hub would alleviate the head drop and shift the positive slope region to the deeper part-load conditions.

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