Suppression of the unstable head curve in mixed flow pump with adjustable vane by use of shallow grooves

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Abstract. In order to operate turbo machinery widely, a mixed flow pump with adjustable vane is often used. However, the head-capacity curve of a mixed flow pump tends to be unstable with a positive slope characteristic in the range of low capacity. The unstable head curve is one of a serious problem for the stable pumping operation. This paper studied about suppressions of the unstable head curves in a mixed flow pump with adjustable vane by use of the shallow grooves. The specific speed of the tested mixed flow pump with adjustable vane was about 700 [m³/min, min⁻¹, m]. In order to clarify the main reason of the unstable head curves, Computational Fluid Dynamics (CFD) simulation was carried out. It was confirmed that the positive slope head curve was mainly caused by the suddenly increase of the hydraulic loss due to the tip leakage flow at the middle of the impeller tip. In addition, it was also suggested that the rapidly hydraulic loss at the low capacity range was caused by a peculiar meridional shape for a mechanism of adjustable vane. Therefore, in order to improve the tip leakage flow at the low capacity range, the shallow radial grooves which were mounted on the middle of the impeller tip was tested experimentally and CFD simulation was also carried out. As a result, it was clearly confirmed that the shallow radial grooves were very effective to suppress the unstable head curve. In addition, it was also confirmed that the improvement of the tip leakage flow at the range of low capacity by the result of CFD simulation. On the other hand, it was also showed that the best efficiency becomes low compared with the original pump. It was suggested that the internal flow characteristics at the best efficiency was strongly affected by the radial groove. Therefore, in order to recovery the best efficiency, the shallow angled grooves which have a same angle of the impeller tip blade were also proposed and tested experimentally. As a result, it was confirmed that the decrease of best efficiency improves compared with the radial grooves and almost the same effect of the suppression of unstable head curve.

1. Introduction
In order to operate turbo machinery widely, a mixed flow pump with adjustable vane is often used. However, the head-capacity curve of a mixed flow pump tends to be unstable with a positive slope characteristic in the range of low capacity range. The unstable head curve is one of a serious problem for the stable pumping operation. In ordinary, the main reason of the unstable head curve is the angular momentum change caused by the reverse flow at the impeller inlet or outlet. Therefore, in order to suppress the unstable head curve, a hydraulic design of the impeller diameter and the blade angle is very important. However, a hydraulic design of a mixed flow pump is still difficult because of
the trade-off relationships between unstable head curve, the suction performance at the large capacity and the best efficiency [1].

On the other hand, J-groove [2] and the casing treatment are very effective to suppress the unstable head curve in the already operating mixed flow pump. In addition, in order to control the unstable head curve actively, some device was also proposed in the mixed flow pump [3]. However, there are few studies about the suppression of unstable head curve in a mixed flow pump with adjustable vanes.

The main purpose of this study is to suppress the unstable head curve at the low capacity range in a mixed flow pump with adjustable vane. In order to suppress the unstable head curve, the shallow grooves which are mounted on the middle of the impeller tip are proposed and studied.

2. Tested mixed flow pump with adjustable vane and numerical analysis

2.1. Tested mixed flow pump with adjustable vane

Figure 1(A) shows the hydraulic components of the tested mixed flow pump with adjustable vane. In addition, Figure 1 (B) also shows the detail of the proposed shallow radial grooves. The specific speed of tested pump was about 700 [m³/min., m, min.⁻¹], the number of impeller blades is 5, the number of bowl diffuser is 11 and the impeller type is a semi-open which has a small clearance between a impeller tip surface and a stationary wall. The Reynolds number based on a root mean square radius of an impeller outlet \( r_{2m} \) is \( Re=U_{2m} r_{2m}/\nu=2.5 \times 10^6 \).

Figure 2 shows that the configurations of proposed radial grooves (A) and angled grooves (B). In the later discussions, it is clearly suggested that the unstable head curves is caused by the hydraulic loss due to the tip leakage flow, not the angular momentum change caused by the reverse flow. This result suggests the mounted position of a groove is very important to suppress the unstable head curves. Therefore, the positions of the proposed grooves are mounted on the middle of the impeller tip at the meridional direction, and it is not extend from the impeller tip to the upstream of a suction bell as shown in Figure 2(C). In addition, in the results of the later experimental result of the radial grooves, the best efficiency becomes low compared with original pump. It is suggested that the internal flow characteristics at the best efficiency is strongly affected by the radial groove geometry. Therefore, in
order to recovery the best efficiency drops, the shallow angled grooves which has a same angle of the impeller tip blade at the adjustable angle setting is 0 degree as shown in Figure 2(B) are also proposed and tested experimentally.

Moreover, the dimensions of the tested grooves, such as the non-dimensional groove width \( W/2r_{2m} \), depth \( D/2r_{2m} \), length \( L/2r_{2m} \) and the number of grooves \( N \), are shown in Table 1. In the case of the radial groove, only the effect of groove width is studied and the other groove configurations are fixed at Groove A1 in the experiment and CFD simulation. On the other hand, there is a limitation of the groove width and length when the numbers of the grooves are fixed in the case of angled grooves. As a result, the effect of the groove length in the angled groove is mainly studied in the experiment.

2.2. Numerical analysis

In order to clarify the flow characteristics of the unstable head curves in the tested pump regardless of the grooves, Computational Fluid Dynamics (CFD) simulation was carried out. CFD simulation is conducted using commercial CFD code ANSYS CFX Ver. 14.0. [4] All the CFD simulation is performed by using Reynolds Averaged Navies- Stokes equations and k-\( \omega \) turbulence model. The boundary condition of inlet was constant total pressure, and the outlet boundary condition was constant mass flow rate. Figure 1 also shows the computational grid and the total number of grid were about 5,000,000 nodes. All computational grids were made by hexahedral grid and the 8 nodes were setting at the impeller tip clearance between an impeller tip edges and a shroud stationary wall. In the case simulation with the radial grooves, 16 nodes were setting at the direction of groove depth which was also shown in Figure 1 (B) and Figure 2(C). The interface setting between a rotating impeller and a stationary wall was set to Frozen- Rotor at the steady- state CFD simulation, and Transient Rotor-Stator interface was also used at the unsteady CFD simulation.

| Name | Groove type | N   | \( W/2r_{2m} \) | \( D/2r_{2m} \) | \( L/2r_{2m} \) | CFD or Experiment |
|------|-------------|-----|----------------|----------------|----------------|------------------|
| A1   | Radial     | 20  | 0.0769         |                | 0.0128         | CFD, Experiment  |
| A2   | Radial     |     | 0.0481         |                |                | Experiment       |
| A3   | Radial     |     | 0.0192         |                |                |                  |
| B1   | Angled     | 24  | 0.0321         | 0.0577         | 0.0833         | Experiment       |
| B2   | Angled     |     | 0.0577         |                |                |                  |
| B3   | Angled     |     |                |                |                |                  |

Figure 2. Configurations of proposed grooves.
3. Results and Discussions
In the discussions of this study, in order to easily understand the results, the angle setting of the adjustable vane is fixed. In addition, in order to show easily, the definition of non-dimensional shaft power is $P/3P_{BEP}$ at the pump performance.

3.1. Analysis of unstable head characteristics by use of CFD
In general, it is considered that the main reason of unstable head curves in a mixed flow pump is sudden angular momentum change at the impeller inlet or outlet due to the reverse flow. Therefore, In order to clear the main reason of the unstable head curves in the tested mixed flow pump with adjustable vane, CFD simulation was carried out.

Figure 3 shows the comparison of pump performance curves between the experiment and CFD. It is clearly shown that the result of CFD simulations was in good agreement with the experiment data at simulated capacity range although the angle setting of adjustable vane was slightly differed about 1 degree. In order to validate the steady state CFD simulation, unsteady CFD was also carried out and it is also shown in Figure 3. Both the predicted result of steady state and unsteady simulation is almost same result which includes the range of unstable head curve. It is confirmed that the unstable head curves of the tested pump is predictable by use of steady state CFD simulation.

It is clearly shown in Figure 3(B) that the shaft power decreases with the decrease of the capacity in the range of unstable head curve from $Q/Q_{BEP}=0.6$ to 0.4 at the both experiment and CFD. It is suggested that the main reason of the unstable head curve in the tested mixed flow pump with adjustable vane is not the angular momentum change caused by the reverse flow at the impeller.

In order to confirm the reverse flow at the impeller inlet or outlet, Figure 4 shows the circumferential averaged meridional velocity at the range of unstable head curve in the steady CFD results. It is shown that the impeller outlet reverse flow becomes large with the decrease of the capacity, however, the impeller outlet reverse flow still confirms in the $Q/Q_{BEP}=0.6$ which is before the sudden drop of the pump head curves. In addition, the impeller inlet reverse flow is clearly shown

![Figure 3. Comparisons of pump performance curves between experiment and CFD.](image-url)
in Figure 4(A) and 4(B) which is after sudden drop of the head curves, however, the impeller inlet reverse flow is not become large. It seems to be the effect of the reverse flow on the unstable head curves in tested pump is small. These results suggest that the main reason of the unstable head curves in the tested pump is not same as the ordinary mixed flow pump.

In ordinary, the flow mechanism of the unstable head curves of a mixed flow pump is associated with the sudden change of angular momentum caused by the impeller reverse flow; in other words, the sudden drop of a theoretical head $H_{th}$ is the main reason of unstable head curves. However, it is suggested that the flow mechanism of the unstable head curves of the mixed flow pump with adjustable vane is different from an ordinary mixed flow pump. Considering the sudden changing the pump head curves which is expressed by $H = H_{th} - H_{loss}$, if theoretical head $H_{th}$ increases with the decrease of the capacity, the hydraulic loss $H_{loss}$ must be suddenly increases.

In order to confirm the suddenly increase of the hydraulic loss, it is also clearly shown in Figure 4(B) (C) that the low meridional velocity regions near the impeller tip and it is sudden increases with the decrease of the capacity from $Q/Q_{BEP} = 0.6$ to 0.5. It is suggested that this low meridional velocity regions makes the hydraulic loss large, and it is the main reason of unstable head curves in tested pump.

In order to clear the reason of the low meridional velocity region as already shown in Figure 4(A) and (B), Figure 5 shows streamline of the impeller tip leakage from the impeller blade pressure side to the impeller blade suction side at the capacity range of the unstable head curve. It is also clearly shown in Figure 5(B) (C) that the strong tip leakage flow near the impeller tip is suddenly increases with the decrease of capacity from $Q/Q_{BEP} = 0.6$ to 0.5. It is suggested that this strong tip leakage flow near the impeller tip makes hydraulic loss large. It is considered that the peculiar internal flow characteristic of
the impeller as shown in Figure 5 is caused by the convex shape of the impeller meridional plane compared with ordinary concave meridional plane of a mixed flow pump impeller. As a result, it is concluded that the sudden hydraulic loss due to the strong tip leakage flow at the low capacity range is caused by a convex meridional shape for the mechanism of adjustable vane.

3.2. Improvements of unstable head characteristics by use the radial shallow grooves in the CFD

The main reason of the unstable head curves in the mixed flow pump with adjustable vane was confirmed that the increase of sudden hydraulic loss due to the strong tip leakage flow at the low capacity range by the analysis of flow characteristics of the original pump. Therefore, in order to improve the tip leakage flow at the low capacity range, the shallow radial grooves which are mounted on the middle of the impeller tip is proposed in this study. The expected effect of a shallow groove is the improvement of the strong tip leakage flow near the impeller tip by use of the flow through the inside of groove caused by differential pressure between the impeller blade pressure side to the impeller blade suction side. As a result, if expected effect of the shallow groove will be confirmed, thus it seems to be finally improves the unstable head curves in the mixed flow pump with adjustable vane.

Therefore, Figure 6 shows the effect of the radial shallow grooves on pump performance curves in the CFD simulation. The simulated groove configuration was already shown in Table.1. It is clearly confirmed that the proposed shallow groove is very effective to suppress the unstable head curves in the range of low capacity. However, the best efficiency becomes slightly low because of the decrease of pump head. It is suggested that the internal flow characteristics in the tested pump is strongly affected by the radial groove geometry. In addition, CFD result of both steady state and unsteady simulation is almost same except for the $Q/Q_{BEP}=0.4$ with a radial groove, however, the difference of pump performance in both simulations is can be acceptable for the analysis of internal flow characteristics.

In order to clarify the effect of a shallow radial groove on the tip leakage flow, Figure 7 shows the streamline of the impeller tip leakage from the impeller blade pressure side to the impeller blade.

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{figure6.png}
\caption{Effect of radial grooves on pump performance in the numerical simulation.}
\end{figure}
suction side at the low capacity range. It is clearly confirmed that the improvement of the tip leakage flow compared with original case as shown in Figure 5. In addition, it is also shown that the sudden change of streamline improves at the range of low capacity. As a result, the strong tip leakage flow becomes too small to affect the unstable head curve as shown in Figure 7(B), thus the bottom of unstable head curves at $Q/Q_{BEP}=0.5$ improves as shown in Figure 6.

3.3. Effect of shallow grooves on unstable head curves in the experimental results

In order to confirm the effect of proposed the shallow radial grooves on unstable head curves, experiment was carried out. Tested groove dimensions are already shown in Table 1. Figure 8 shows the experimental result of the effect of the radial groove width $W/2r_{2m}$ on pump performance. It is shown that the proposed shallow radial groove is very effective to suppress the unstable head curve in the low capacity range experimentally. Moreover, it is also confirmed that the unstable head curve completely suppress at the groove width increases over than $W/2r_{2m}=0.0481$. However, best efficiency drops with the increase of the groove width. It is also suggested that the internal flow characteristics at the best efficiency point is strongly affected by the radial groove geometry. It is considered that the tip leakage flow from the impeller blade pressure side to the impeller blade suction side is disturbed by the radial grooves geometry at the best efficiency. Thus, a hydraulic loss becomes large near the best efficiency and it makes pump head low compared with the original as shown in Figure 8. It is also shown that the shaft power at the best efficiency is almost same as original pump, and it suggests that theoretical head $H_{th}$ is not affected by the radial grooves.

It is concluded that the decrease of best efficiency is mainly caused by the increase of hydraulic loss based on the disturbed tip leakage flow in the case of the radial grooves. In addition, it is suggested that the trade-off relationships between the improvement of unstable head curves and the drops of best efficiency.

In order to recovery the best efficiency, the shallow angled grooves are also proposed as already shown in Table 1 and tested experimentally. One of the effects of a shallow angled groove is to improve the unstable head curve at the range of low capacity like a radial groove. The other is to reduce the leakage flow through the inside of groove caused by the differential pressure from the impeller blade pressure side to the impeller blade suction side, this is because the length of groove along the meridional direction is shorter than that of the radial groove.

Figure 9 shows the experimental result of the angled grooves. It is confirmed that the drop of best efficiency improves compared with the radial grooves and almost same effect of the suppression of unstable head curves at low capacity region. It is shown that the unstable head curves improve with the increase of angled groove length $L/2r_{2m}$. However, it is still difficult to completely suppress the unstable head curve even though the cases of most effective angled groove configuration B3. It is suggested that the optimum geometry of angled groove is important to apply to the actual industrial pump.

Figure 7. Streamline of the tip leakage flow with the radial groove A1.
Figure 8. Effect of the radial grooves on pump performance in the experiment.

Figure 9. Effect of the angled grooves on pump performance in the experiment.
4. Conclusions
In this study, in order to suppress the unstable head curve in the mixed flow pump with adjustable vane, the shallow grooves which are mounted on the middle of impeller tip is proposed and tested. The results are summarized as follows:

· The proposed shallow radial grooves are very effective to suppress the unstable head curve. It is confirmed that the improvement of the tip leakage flow at the range of low capacity by the result of CFD simulation. On the other hand it is showed that the best efficiency becomes low compared with original pump.

· The proposed shallow angled grooves are also effective to suppress the unstable head curve. It is confirmed that drop of best efficiency improves compared with the radial grooves. However, it is still difficult to completely suppress the unstable head curves in this study.

· It is confirmed that the unstable head curve of the tested pump is mainly caused by the rapidly increase of the hydraulic loss due to the tip leakage flow at the middle of the impeller tip. It is considered that rapidly hydraulic loss at the low capacity range is caused by a convex meridional shape for the mechanism of adjustable vane.

· It is confirmed that steady state CFD simulation was in a good agreement with the experimental data includes the range of the unstable head curve. It can be easily predictable that the effect of groove on the pump performance by use of CFD simulation.

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Nomenclature

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\begin{array}{lll}
D & \text{Depth of groove} & \text{[mm]} \\
H & \text{Pump head} & \text{[m]} \\
L & \text{Length of groove} & \text{[mm]} \\
N & \text{Number of grooves} & \text{[-]} \\
P & \text{Shaft power} & \text{[kW]} \\
Q & \text{Capacity} & \text{[m}^3\text{/min.]} \\
r & \text{Radius} & \text{[mm]} \\
U & \text{Peripheral velocity} & \text{[m/s]} \\
V & \text{Velocity} & \text{[m/s]} \\
W & \text{Width of groove} & \text{[mm]} \\
\theta & \text{Angle setting of groove} & \text{[degree]} \\
v & \text{Viscosity} & \text{[m}^2\text{/s]} \\
\text{BEP} & \text{Best efficiency point} \\
\text{L.E.} & \text{Blade leading edge} \\
\text{m} & \text{Meridional direction,} \\
\text{th} & \text{Root mean square} \\
\text{T.E.} & \text{Blade trading edge} \\
\text{loss} & \text{Hydraulic loss} \\
\text{2} & \text{Impeller outlet} \\
\tilde{A} & \text{Circumferential average of A} \\
\end{array}
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