Analysis of blade-passage flow of a mixed-flow pump at performance-curve instability

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Abstract. To design a high-performance mixed-flow pump, performance-curve instability must be controlled and suppressed. To do this, the essential flow physics that leads to the performance-curve instability must be clarified in addition to an accurate prediction of the performance curve. We analyzed internal flows of a test mixed-flow pump, with performance-curve instability by using Large Eddy Simulation (LES). In particular, we investigated in detail unsteady internal flows of individual blade passages near the head-drop point (60% ~ 61% of the designed flow rate). As a result, we clarified the following. (a) When the flow rate through a blade passage is low, the Euler’s head is high for that particular passage, and this condition propagates in the opposite direction of the impeller rotation for the whole impeller while the impeller makes approximately 20 revolutions. (b) The upstream flow of the stalled blade passage deviates to the downward neighboring blade passage, resulting in an increase in the load of the neighboring blade near its trailing edge. When the load of the blade increases, the separation of flow occurs from the trailing edge of the blade, and the separation point gradually moves toward the leading edge of the blade. The neighboring blade passage is then stalled similarly because the angle of attack becomes large. (c) When the flow rate in a stalled blade passage minimizes, the passage starts to recover from the stall, whereas the pressure gradient in the streamwise direction decreases and the flow rate in the passage increases.

1. Introduction
Performance-curve instability (positively sloped head-flow characteristics in a low flow rate region), which destabilizes machine operation, has long been regarded as a challenge in a turbo-machinery. Pumps require high efficiency. Moreover, cavitation performance needs to be improved, operating range expanded, and shutoff head rate reduced, which will improve the reliability of the piping connecting the pump. However, when we try to satisfy all these requirements, performance-curve instability is likely to occur. Therefore, performance-curve instability must be controlled and suppressed to design a high-performance pump.

To suppress the performance-curve instability, the mechanism of the performance-curve instability occurs needs to be clarified in detail and many studies have reported [1][2][3]. For example, Goto [1] found signs of flow separating on the shroud side of the suction surface of the impeller’s blades and low energy fluids accumulating when the performance-curve instability occurs and considered the performance-curve instability to be caused by the stall on the impeller’s blades. However, because internal flow around the flow rate at which the performance-curve instability occurs has strong three-dimensional and unsteady property, no report has yet analyzed in detail flow field change with respect to change in the flow rate and explained the mechanism by which the performance-curve instability occurs. Furthermore, there are various stall types of an impeller’s blade such as leading edge stall type, trailing edge stall type, and thin airfoil stall type. These stalls have different characteristics [4][5], and the stall type when the performance-curve instability occurs has not been clarified. Thus, the
mechanism of the performance-curve instability occurrence in pumps has not yet been completely clarified, and therefore, no pump has been designed that maximizes performance while suppressing the performance-curve instability.

To clarify the essential flow physics that leads to the performance-curve instability of a pump, the internal flow must be understood in detail. However, the internal flow cannot be completely grasped in detail through experiments, so simulation technology that can accurately predict the internal flow needs to be developed. Reynolds-Averaged Navier-Stokes Simulation (RANS) has been applied in the design of pumps along with improved computer performance and development of simulation technology [6]. However, RANS has difficulty predicting the internal flow with high accuracy because the internal flow near the flow rate at which the performance-curve instability occurs has large-scale separation, local stall, and strong unsteady property. To analyze the internal flow when the performance-curve instability occurs, Large Eddy Simulation (LES) is useful because it can predict strong unsteady flow that has large-scale separation flow with high accuracy. Yamade et al. [7] investigated a prediction method for a mixed-flow pump using LES to clarify the phenomenon of the performance-curve instability. They reported that the characteristic flow field, in which pressure at the trailing edge of the impeller’s blade increased and the flow reversed, was observed at the flow rate point where the total head locally dropped during the performance-curve instability. Paco et al. [8] analyzed a pump turbine by relatively large-scale LES, succeeded in reproducing rotating stall occurring in the guide vane passages under the low flow rate condition, and clarified the mechanism of propagating stall cells in the guide vane passages. However, little is understood about how flow in an impeller’s blade passage changes over time at a flow rate at which head-flow characteristics slope positively in a mixed-flow pump with a relatively high specific speed, which is the target of this research.

In this research, we aim to clarify in detail the occurrence mechanism of the performance-curve instability and report results of detailed analysis of blade-passages flow by LES for a test mixed-flow pump when the performance-curve instability occurs. The test mixed-flow pump has a specific speed of 1000 [min\(^{-1}\), m\(^3\)/min, m] and a performance-curve instability around 60% of the designed flow rate.

2. Computational model
The test mixed-flow pump that we used had an open impeller with 5 blades and a diffuser with 12 guide vanes. Figure 1 shows a perspective view of the computational model of the test mixed-flow pump. As mentioned above, the test pump had the open impeller’s blades and the guide vanes. The test pump had a straight suction pipe upstream of the impeller, and the suction pipe had a cross-shaped straightening plate just before a cap of the impeller. The test pump had a bend pipe and a straight discharge pipe downstream of the guide vanes. The pump was divided into six regions as shown in Figure 1. Each region was overlapped, and the overlap was as wide as some grids. We calculated the continuous flow field in the whole pump on the basis of the overset method. The velocity and pressure data in each overlapped region were exchanged during the calculation. The lengths of the suction and discharge pipes were 6.6 and 7.6 times the representative impeller diameter. The regions with the cap of the impeller and the impeller were set in a rotating frame of reference. Those regions were set rotating with the speed of the impeller around its center.

The calculated model consisted of approximately 8 million grid nodes, with the impeller constituting approximately 2 million of those. The dimensionless size in the chord direction of the representative grid was approximately 400 at the leading edge on the tip side of the impeller, that in the vertical direction on the wall was approximately 400, and that of the span direction was approximately 1000.

The LES was conducted using the open source software FrontFlow/blue [9], which is an explicit finite-element code that calculates incompressible unsteady flows in arbitrarily shaped geometries. We used the Dynamic-Smagorinsky model as the turbulence model for the LES. The calculated increments were set at 5120 steps per revolution. The inlet boundary condition in Region 1 was set as
uniform and constant velocity without prewhirl. The outlet boundary condition in Region 6 was traction free with a constant pressure of 0 Pa. The boundary condition of the stationary surfaces was set as non-slip.

We defined the flow rate at the best efficiency point as ‘100% flow rate.’ We calculated a total of 17 flow rate points of 40% to 130% flow rate. Particularly around the 60% flow rate where the performance-curve instability occurred, we carried out the LES at 8 flow rate points: 55%, 58%, 60%, 60.5%, 61%, 62%, 63%, and 65%. Unsteady internal flow was analyzed by sampling 64 data per revolution and performing statistical processing.

Figure 1. Perspective view of computational model for test pump

3. COMPARISON OF HEAD CURVES BETWEEN ANALYSES AND MEASUREMENTS

Figure 2 shows a schematic diagram of an apparatus for the performance test. And Figure 3 (a) compares the total head curves and the Euler’s head curves between analysis and measurements. The flow rate was normalized by the flow rate of the best efficiency ($Q_{BEP}$) and shown. The total head of the analysis was computed as the difference in an averaged total pressure between the inlet of the suction pipe and the outlet of the discharge pipe. The Euler’s head of the analysis was computed as the difference in an angular momentum between the inlet and the outlet of the impeller. Friction torque on the casing wall surface is subtracted, but this effect is negligibly small. The Euler’s head of the measurement was computed as the ratio of the total head and the efficiency. Figure 3 (b) shows the total head curves obtained by focusing on the flow rate range where the total head falls.

Figure 2. Schematic diagram of apparatus for performance test
The analysis and measurement results were found to agree well. Figure 3 (a) shows that the head curves had a positive slope, and the gradients of both the Euler’s head and the total head dropped at an approximately 60% flow rate. The total head drop was due to the flow in the impeller since the total head dropped as the Euler’s head dropped. As shown in figure 3 (b), we defined a 61% flow rate as (A) the start point of the performance-curve instability, a 60% flow rate as (B) the end point of the performance-curve instability, and a 60.5% flow rate as (C) the transitional point of the performance-curve instability. We particularly focused on the flow state at (C) and analyzed it in detail.

![Figure 3 (a). Comparison of measured and computed Euler’s head and total head](image)

![Figure 3 (b). Enlarged view of total head near instability points. (A), (B) and (C) respectively indicate start, end, and transitional points of positive slope](image)

4. Features of flow field around performance-curve instability flow rate point

4.1. Angular momentum in flow direction through impeller’s blade-passage

Figure 4 shows positions of cross sections evaluating angular momentum and name definitions of the evaluation sections. We defined the middle position between L.E. and mid-chord as ‘L.E.2’ and the middle position between mid-chord and T.E. as ‘T.E.2.’ We also defined the region between L.E. and L.E.2 as ‘Region 1,’ the region between L.E.2 and mid-chord as ‘Region 2,’ the region between mid-chord and T.E.2 as ‘Region 3,’ and the region between T.E.2 and T.E. as ‘Region 4.’

![Figure 4. Definition of cross-section for evaluating angular momentum](image)
Figure 5 (a) shows changes of angular momentum through each cross-section in the flow direction of the impeller. By comparing the results at 60% flow rate with the results at 61%, we found the difference in the angular momentum maximizes at mid-chord. The drop of the impeller’s blade load on the leading edge side exceeded the Euler’s head drop at 60% flow rate.

Figure 5 (b) shows differential angular momentum between each region. We found that the angular momentum drop in regions 1 and 2 (the leading edge side) exceeded the angular increment caused by the decrease in the flow rate in regions 3 and 4 (the trailing edge side) at the 60% flow rate, which was the end point of the performance-curve instability.

4.2. Impeller’s blade load in span direction

Figure 6 shows the distribution of blade load on each side in the span direction of an impeller’s blade in the test pump. As shown, the blade load only on the leading edge side of the tip side was reduced at the 60% flow rate. The impeller’s blade was found to enter a stall condition on the leading edge side of the tip side at the 60% flow rate.

\[
P_{\infty} = \frac{2(P - P_{\infty})}{\rho U^2}
\]

\(P\): Pressure, \(P_{\infty}\): Reference pressure

Figure 6. Distribution of blade load on each span of impeller’s blade near instability point
5. Time changes of Euler’s head and flow rate through each blade-passage

Figure 7 shows definitions of the impeller’s blades and the blade-passages in this research. At a point in time, the impeller’s blade in the direction of approximately 12 o’clock was defined as ‘blade 1.’ Blades 2 to 5 were defined in the clockwise direction, which was the opposite direction to the rotation of the impeller, from blade 1. In addition, we defined the blade-passage between blades 1 and 2 as ‘passage 1’ (written as ‘pass 1’ in the figures). We also defined passages 2 to 5 in a clockwise direction like blades 2 to 5.

![Diagram of blade and passage definitions](image)

**Figure 7.** Naming rule for blades and blade passages

Time changes of the Euler’s head and flow rate of each impeller’s blade-passage at the transitional point of the performance-curve instability are shown in figures 8 and 9, respectively. These values were obtained by performing moving average processing for one revolution of the impeller. The figures show the average values between all impeller’s revolution time and all blades. The equation used for averaging is shown in equation (2). Moreover, the Euler’s head and the flow rate of passages 2, 3, 4, and 5 are respectively plotted with an offset of 4, 8, 12, and 16 impeller revolutions in these figures. Both the Euler’s head and the flow rate fluctuated about ±10% of the average value. Figures 8 and 9 show that both the waveforms of the Euler’s head and the flow rate through each blade-passage were approximately matched by offsetting the time corresponding to four revolutions per blade. This shows that the phenomenon which the Euler’s head and the flow rate decrease or increase was moving between one blade-passage while the impeller rotates four times. Thus, it was moving between five blades, which was 1 impeller wrap, while the impeller rotated 20 times. That is, the condition propagated in the opposite direction to the impeller rotation for the whole impeller while the impeller made approximately 20 revolutions. The time changes of the Euler’s head and the flow rate clearly correlated, so the Euler’s head was high for that particular blade-passage when the flow rate through the blade-passage was low.

![Graph of Euler’s head and flow rate](image)

**Figure 8.** Variation of Euler’s head of each blade passage with respect to impeller’s revolution at transitional point of positive slope. Euler’s heads of passages 2, 3, 4, and 5 are plotted with offsets of 4, 8, 12, and 16 impeller resolutions, respectively.
Figure 9. Variation of flow rate through each blade passage with respect to impeller’s revolution at transitional point of positive slope. Flow rates of passages 2, 3, 4, and 5 are plotted with offsets of 4, 8, 12, and 16 impeller resolutions, respectively.

\[
\bar{x} = \frac{\sum_{j=1}^{5} \left( \sum_{n=1}^{n} x_{ij} \right)}{5n}
\]

\(x\): Variable, \(i\) and \(n\): Number of history data, \(j\): Number of blade passages

The fluctuation range of the Euler’s head at the transitional point of the performance-curve instability (0.14) was larger than the values at the start point (0.07) and the end point (0.08). The maximum value of the Euler’s head at the transitional point of the performance-curve instability (0.63) was slightly larger than the maximum value at the start point (0.61), whereas the minimum value (0.49) was slightly smaller than the minimum value at the end point (0.50). Hence, it is considered that the condition of each blade-passage at the transitional point of the performance-curve instability changed alternately between the conditions close to the start point and the end point. At the start point and the end point of the performance-curve instability, the time changes of the Euler’s head of each blade-passage were not approximately matched by offsetting the phase of the impeller. Therefore, it is considered that the phenomenon of propagating in the impeller did not occur at these flow rates.

6. Time changes of flow field in blade-passages near blade tip

The pressure distributions, which are the blade load distributions, on the each blade tip side at the transitional point of the performance-curve instability are shown in figure 10 (a) to (d) for every four impeller revolutions. In figure 10 (a), the blade loads on leading edges of blades 4 and 5 decreased, which are written as ‘stall,’ and these were found to be stall conditions. Figure 10 (b) – (d) show that the stall conditions of the impeller’s blades propagated by one pitch of the impeller’s blades in the opposite direction while the impeller rotated about four times. At the end point of the performance curve instability, the blade loads at the leading edges decreased for all blades, and all blades had stall.

Next, the relative velocity distribution and the pressure distribution in the conical cross sections on the impeller’s blade tip side at the transitional point of the performance-curve instability are respectively shown in figure 11 (a) and (b) in order to investigate the relationship between the impeller’s blade load change and the flow field change inside the blade-passages. The flow fields at the fourth and eighth revolutions of the impeller are on the left and right, respectively. Figure 11 (a) shows that the blade-passages covering the low speed and stalled region were moving in the opposite direction to the rotation of the impeller (passages 3 and 4 to passages 4 and 5). At this time, as indicated by the dotted arrow, the low speed region appeared near the trailing edge on the suction surface side of blade 5 and expanded toward the leading edge, and the entire blade-passage covered the low speed region. This, the trailing edge stall type was found. The trailing edge stall is considered to be due to the relatively high blade load on the trailing edge side. In contrast, as indicated by the one-dot chain line arrow, the entire blade-passage on the suction surface side of blade 3 covered the low speed region.
speed region at the fourth revolution of the impeller, and the low speed region had disappeared from the leading edge toward the trailing edge at the eighth revolution. Figure 11 (b) shows that the pressure gradient on the leading edge side of the blade-passage that covered the low speed region, where ‘B’ is written, was small. This is because the flow did not deflect when the impeller’s blade stalled, and pressure did not increase either. It is considered that the flow rate through the passage started to increase again as the pressure gradient (pressure increase) became small after that.

Figure 12 shows the circumferential distribution of inflow direction at the leading edges and the pressure distribution shown in figure 11 (b) on the developed cylindrical surface in order to facilitate comparison of the flow fields between each impeller’s blade-passage. At the fourth revolution of the impeller, blades 4 and 5 were stalled, passage 3 was recovering from the stall, passage 4 was completely stalled, and passage 5 started to stall.

**Figure 10.** Evolution of pressure distributions near tip of each blade for transitional point of performance-curve instability
First, we focused on blade 5 and passage 5, which was the passage on the suction surface side of blade 5. The blade load on the leading edge of blade 5 had already decreased at the fourth revolution of the impeller, but the separation region (low speed fluid region) on the suction surface
side continued to expand from the trailing edge to the leading edge. The flow rate through passage 5 decreased, and the tangential relative velocity on the leading edge side decreased. As a result, the angle of attack (AoA) of the flow into blade 1 opposed to blade 5 via passage 5 became large, and the blade load of blade 1 also increased. Then, the flow started to separate near the trailing edge on the suction surface side of blade 1, and the separation region gradually expanded to the leading edge. At the eighth revolution of the impeller, the blade load at the leading edge of blade 1 had decreased and blade 1 was stalled, but the flow rate of passage 1, which was the passage on the suction surface side of blade 1, continued to decrease. This is the mechanism by which the stall condition propagates to the blade-passage in the opposite direction of the impeller rotation.

Next, we focused on blade 3 and passage 3, which was the passage on the suction surface side of blade 3. As described above, the flow rate through passage 3 started to increase at the fourth revolution of the impeller, and passage 3 started to recover from the stall condition. At this point, the blade load at the leading edge of blade 3 had already increased. This means that the flow that had been separated at the leading edge on the suction surface side of blade 3 was attached again. However, the blade load of blade 3 was still smaller than those of blades 1 and 2. Therefore, the pressure rise of passage 3 was smaller than those of passages 1 and 2. That is, passage 3 satisfied the condition for increasing flow rate. In contrast, the condition for increasing flow rate was not satisfied in passage 4 because the flow at the leading edge of blade 4 separated on the suction surface side. Since the total flow rate from passages 1 to 5 did not change, the flow rate through passage 3 selectively increased to compensate for the flow rate between the passages whose flow rate had decreased due to the stall condition. As the flow rate of passage 3 increased, the separation point on the suction surface side of blade 3 moved to the trailing edge side. At the eighth revolution of the impeller, the flow on the suction surface side attached until the trailing edge, and then the blade load of blade 3 also increased. The AoA of blade 4 decreased because the flow on the suction surface side of blade 3 was attached until the trailing edge. As a result, the flow from around the leading edge of blade 4 to the suction surface reattached, and the blade load of the leading edge of blade 4 started to increase. Since the blade load of blade 4 was still small and the pressure rise was not large at this point, passage 4 became the passage where flow rate tended to increase most, and the flow rate through passage 4 increased. This is mechanism by which the blade recovers from the stall condition and propagates.

7. Conclusion
This paper aimed to clarify the occurrence mechanism of the performance-curve instability in a mixed-flow pump with a specific speed of 1000 [min⁻¹, m³/min, m]. We focused on the transitional condition of the performance-curve instability and analyzed in detail the change in flow field in the blade-passages by Large Eddy Simulation (LES). We obtained the following conclusions.

- Time changes of the Euler’s head and flow rate in each impeller’s blade-passage were analyzed. When the flow rate through a blade passage is low, the Euler’s head is high for that particular passage, and this condition propagates in the opposite direction of the impeller rotation for the whole impeller while the impeller makes approximately 20 revolutions.
- Time changes of blade load, relative speed, and static pressure on the tip side of the impeller were analyzed. When the load of the blade increases, flow separates from the trailing edge of the blade, and the separation point gradually moves toward the leading edge of the blade. The downward neighboring blade-passage is then stalled similarly because the angle of attack becomes large. When the flow of the neighboring blade becomes attached, the angle of attack decreases, and the flow reattaches from the leading edge. Then, the passage starts to recover from the stall because the pressure gradient through blade-passage is not large and the flow rate through the passage increases.
Nomenclature

\( D_2 \)  impeller representative diameter
\( H \)  total head
\( g \)  acceleration due to gravity
\( U_2 \)  impeller representative peripheral velocity
\( U \)  peripheral velocity
\( C_a \)  tangential absolute velocity
\( \rho \)  density of working fluid

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