Effect of blade thickness on hydraulic performance of a mixed-flow pump impeller

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Abstract. This paper presents the effect of blade thickness on the hydraulic performance of a mixed-flow pump impeller. For the numerical analysis, three-dimensional steady-state Reynolds-averaged Navier-Stokes (RANS) equations are discretized using the finite volume method with the shear stress transport (SST) turbulence model. The equations were solved using hexahedral grids to analyze the internal flow in the mixed-flow pump impeller. The blockage concept is employed to express the quantitative amount by the variation in the blade thickness. The effects of hydraulic performance on the blockage amount are systematically analyzed with the variation in the best efficiency point (BEP) in the performance curve. The detailed flow characteristics with the blockage effect are analyzed and discussed.

1. Introduction

A mixed-flow pump, which is widely used in various industries, is a high energy-consuming device. In response to environmental problems, such as resource depletion and rising oil prices, an effective operation of equipment is an essential factor for energy conservation. The mixed-flow pump is widely used because it has characteristics of both centrifugal and axial pumps. Therefore, it is essential to optimize the performance of mixed-flow pumps.

Several studies have been conducted on the optimization of mixed-flow pumps. The results help set the design parameters of pumps [1-4]. Moreover, advanced studies on the design parameters of the impeller, which directly affect the pump performance, have been actively performed. For example, Lee et al. [5] validated the outlet diameter of a mixed-flow pump impeller calculated using theoretical equations by comparing with the results of numerical analysis. Shim et al. [6] changed the meridional shape to analyze the effects of design parameters rated with impeller shroud on internal flow characteristics and hydrodynamic efficiency. Kim et al. [7] optimized the impeller by adjusting the blade angle using design of experiments (DOE) such as 2k factorial design and response surface methodology (RSM).

The blade thickness of a mixed-flow pump is selected on the basis of accumulated equations. However, studies on the effect of blade thickness on the performance is insufficient. When a manufactured pump is applied to the field, the blade thickness or the design specification can change.
depending on the material of the blade and the processing method. However, it is impossible to adjust the blade thickness of an already manufactured pump. Therefore, the initial design of the impeller blade is very important for selecting the appropriate thickness, making it necessary to predict the performance of the pump based on the various thicknesses of the blade.

In this study, a numerical analysis is conducted to investigate. The blade thickness is defined as “blockage” and is expressed as a percentage. The design parameters, except for the blade thickness, are kept constant. The trends of the total head and efficiency curves are derived using numerical analysis at the design and off-design points for each blade thickness. From the results, the causes of internal flow are discussed.

2. Mixed-flow pump unit

2.1. Design specification

The impeller of a mixed-flow pump unit optimized by Choi et al. [8] is considered for this study. Table 1 lists the design specifications. The specific speed can be obtained using equation (1) as a non-dimensionalized value with the type number format. The flow coefficient ($\phi$) and the head coefficient ($\psi$) at the design point are expressed in equations (2) and (3), respectively.

\[
N_s = \frac{\omega \sqrt{Q}}{(gH)^{3/4}}
\]

\[
\phi = \frac{C_{m2}}{u_2}
\]

\[
\psi = \frac{2gH}{u_2^2}
\]

where $\omega$, $Q$, $g$, $H$, $C_{m2}$, and $u_2$ denote the angular velocity, volumetric flow rate, acceleration due to gravity, total head, meridional component of the absolute velocity at the impeller outlet, and tangential component of the rotational speed at the impeller outlet, respectively.

Table 1. Design specifications of the mixed-flow pump.

| Pump type | Mixed-flow |
|-----------|------------|
| Specific speed ($N_{s\text{type number}}$) | 2.43 |
| Flow coefficient ($\phi$) | 0.19 |
| Head coefficient ($\psi$) | 0.51 |
| Number of blade (Z) | 5 |

2.2. Definition of blockage

The blockage is defined as the extent of blade thickness in the annular passage. In this study, the blockage is defined as the ratio of the length occupied by the blade thickness ($t'_1$) to the total circumference ($t_1$) at the impeller leading edge (LE), expressed in equation (4).

\[
\text{Blockage} = \frac{t'_1}{t_1} = \frac{t'_1}{\frac{2\pi Z}{2}} = \frac{s_1}{\frac{2\pi Z}{2}} = \frac{s_1Z}{2\pi r_1 \sin \theta_{1b}}
\]

where $\pi$, $r_1$, $Z$, $s_1$, and $\theta_{1b}$ denote the circular constant, radius, number of blades, normal thickness of the blade, and inlet blade angle, respectively. The subscript 1 represents the impeller leading edge.

As shown in Figure 1, the Pythagorean theorem is applied to the definition. The radius value at the mid-span of the impeller leading edge ($r_{1m}$) is substituted with $r_1$ and is expressed in Figure 2. Table 2 lists the percentages of blockage at each span calculated using equation (3). The inlet blade angle ($\theta_{1b}$) is assigned to the actual angle at the mid-span of the impeller leading edge.
Figure 1. Definition of blockage.

Figure 2. Meridional shape of mixed-flow pump.

The blade thickness distribution is constant from the leading edge to the trailing edge (TE). The blockage value is expressed as a percentage, and a numerical analysis is performed on a section of blockage ranging from 1 to 20%. The reference blockage for this work is 9.8%. The blockage percentage decreases from the hub to the shroud with the gradual increase in the impeller radius.

Table 2. Blockage percentage at each span.

| Span    | $\beta_{1b}$ (°) | Blockage (%) |
|---------|------------------|--------------|
| Hub     | 46.2             | 1.41         |
|         | 6.94             | 13.6         | 20.8         | 27.7         |
| **Mid** | **24.6**         | **1.00**     |
|         | **5.00**         | **9.80**     | **15.0**     | **20.0**     |
| Shroud  | 16.0             | 0.97         |
|         | 4.77             | 9.35         | 14.2         | 19.0         |

3. Numerical analysis method

The numerical analysis is performed using the commercial code ANSYS CFX 18.0. Three-dimensional steady-state Reynolds-averaged Navier–Stokes (RANS) equations are discretized using the finite volume method with the shear stress transport (SST) turbulence model. The analysis area consists of a rotating area (impeller) and a fixed area (inlet and outlet). The boundary condition between the analysis areas is employed as a frozen-rotor method. Atmospheric condition is applied to the inlet, and a mass flow condition is applied to the exit. To ensure the temporal efficiency of the numerical analysis results, a periodic condition is applied to the impeller. As shown in Figure 3, the grid system of the mixed-pump is constructed using ANSYS TurboGrid 18.0, which has hexahedral grids.
The values of $y^+$ on the impeller and the wall are kept below 10 considering the aspect ratio. For an accurate application of the low Reynolds SST model, a grid system in the $y^+ \leq 1$ range is recommended; however, in this study, an automatic wall function is employed as the wall function so that $y^+$ does not affect the analysis results [9, 10]. Figure 4 shows the results of the grid dependency test. In the first iteration of the numerical analysis, approximately 350,000 nodes were selected as the minimum grid system; this condition does not affect the performance of the pump. However, with this grid system, the convergence of the numerical analysis at the off-design point was reduced because of the increase in $y^+$. Accordingly, approximately 500,000 nodes were employed for the optimum grid system. The graph is non-dimensionalized based on the maximum value of the grid dependency test ($\eta_{\text{max}}$).

4. Results and discussion

4.1. Validation of numerical analysis
An experimental test is conducted on the reference blockage (9.8%) model, and the results are then compared to those of the numerical analysis [11]. The numerical analysis and experimental test were conducted with a diffuser attached to the impeller. The experimental test facility is in the form of a closed loop, which is shown in Figure 5, thus meeting the ISO 5198. The differential pressure is calculated by measuring the absolute static pressure at a point where the diameter is twice that of the inlet and outlet flanges. The flow rate and torque are measured with an accuracy of $\pm 0.2\%$ using a magnetic flowmeter and a torque meter, respectively. The flow rate is controlled using a gate valve.
As shown Figure 6, both the experimental test and the numerical analysis results show the head is decreased from the low flow rate to the high flow rate point, and the best efficiency point is appeared at the design point. Also, the total head and efficiency are largely consistent with the tendencies of the numerical analysis and experimental test. The numerical analysis results exceed the results of the experimental test because the leakage, clearance, roughness, and mechanical loss, which degrade the performance of the pump, were not considered in the numerical analysis. The total head and efficiency are dimensionless based on the results of the numerical analysis at the design point (Ht\text{ des}, \eta_{\text{des}}).

![Figure 5. Schematic diagram of experimental equipment.](image)

![Figure 6. Validation of CFD results with performance curves for reference blockage.](image)

Figures 7, 8, and 9 are analyzes of the numerical results of the performance curves of Figure 6. Figure 7 is the pressure contour at the impeller shroud. The pressure value is dimensionless based on the maximum pressure value at the low flow rate (Q/Q_{\text{des}} = 0.8). It is confirmed that the overall pressure distribution decreases from the low flow rate point to the high flow rate point. This shows the same tendency as the head curve of the performance curve.

![Figure 7. Pressure contour.](image)

Figure 8 is the stream line at the impeller shroud for each flow rate point. It can be seen that the stagnation point moves with the same directionality from the low flow rate point to the high flow rate point. This means the difference of the flow angle with the change of flow rate. This phenomenon is schematized as a velocity triangle. The absolute velocity of the meridional component increases as the increase of flow rate. Thus, the flow velocity is increased, and at the same time, the flow angle is increased. According to the analysis results, the flow angle at the high flow rate (Q/Q_{\text{des}} = 1.2) is very well matched with the blade angle. And also, the incidence angles are getting bigger at low flow points. At low flow rate (Q/Q_{\text{des}} = 0.8), the possibility of back flow is observed.
Figure 7. Pressure contour on the impeller shroud at the off-design point for reference blockage.

Figure 8. Streamline on the impeller shroud at the off-design point for reference blockage.

Figure 9 is the pressure contour and streamline on the meridional plane at each flow rate. It is expressed as a local value in order to examine individual characteristics at each flow rate. As shown in Figure 9(a), recirculation flow is observed near the shroud, which is a direct effect on pump performance. And also, in Figure 9(c), it can be seen that the pressure distribution is uneven at the impeller trailing edge. This is in contrast to the uniform pressure distribution of the design point. Due to these reasons, the best efficiency point appeared at the design point.
Figure 9. Pressure contour and streamline on the meridional plane at each flow rate.
4.2. Effect of blockage at the design point

Figure 10 shows the performance graph from the numerical analysis at blockages ranging from 1 to 20% with intervals of 5%, obtained by adjusting only the blade thickness in the design specification. The total head and efficiency are dimensionless based on the results of the numerical analysis at a blockage of 9.8%, which is set as the reference model \((H_{t\text{ ref}}, \eta_{\text{ ref}})\). The results show that the total head decreases with the increase in the blockage. The highest efficiency is observed at the reference blockage, and at a blockage of 5% or lower, there is a significant decrease in the performance.

![Performance curves for each blockage at the design point.](image)

To analyze the cause of the significant decrease in the performance when the blockage percentage is below 5%, a streamline analysis is performed on the suction side of the impeller blade. As shown in Figure 11, the flow can be seen to closely follow the blade surface at blockages of 9.8, 15, and 20%. However, at 1 and 5% blockages, an irregular flow, such as a separation, is observed on the suction side of the impeller, particularly at the leading edge of the impeller hub.

![Streamline on the suction side of the impeller blade at the design point.](image)
An additional flow analysis is performed to confirm the occurrence of the separation at the impeller hub. As shown in Figure 12, the streamline of the 1% blockage model is different from those of the others, and a separation is observed at the suction side of the impeller. This is the reason for the significant decrease in the performance at a blockage of 1%. Figure 13 shows the pressure distribution at the impeller shroud. As the blockage increases, the overall pressure distribution decreases, with the same tendency as that of the total head curve shown in Figure 10. The pressure values are dimensionless based on the maximum pressure of the 1% blockage model.

![Streamline on the impeller hub of LE at the design point.](image)

**Figure 12.** Streamline on the impeller hub of LE at the design point.

![Pressure contour on the impeller shroud at the design point.](image)

**Figure 13.** Pressure contour on the impeller shroud at the design point.

### 4.3. Effect of blockage at the off-design point

Figure 14 shows the analysis results at the off-design point for blockages ranging from 1 to 20% with an interval of 5%. Figures 14(a) and (b) show the total head and efficiency curves, respectively, and are non-dimensionalized as a result of the numerical analysis at the design point for the 9.8% blockage model \((H_{des}, \eta_{des})\). In Figure 14(a), the total head at the design point, indicated by the solid line, decreases with the increase in the blockage, similar to the total head curve shown in Figure 10. However, the total head at the best efficiency point (BEP), indicated with black color, increases with the increase in the blockage, and the slopes of each graph change steeply.

The BEPs for each blockage, shown in Figure 14(b), are indicated with black color. Figure 15 shows the efficiencies at the BEPs for each blockage. The BEPs based on the non-dimensional values are 0.954, 0.992, 1.000, 0.999, and 0.998, for blockages of 1, 5, 10, 15, and 20%, respectively. The BEPs for a high-blockage percentage range (9.8–20%) are largely the same. However, the BEP drastically reduces at a blockage of 1%. In addition, the BEPs tend to move to the higher flow rate point as the blockage becomes smaller and to the lower flow rate point as the blockage becomes larger. In other words, the design point changes depending on the blockage.
Figure 14. (a) Head and (b) efficiency curves with respect to different blockages.

Figure 15. Efficiencies at the BEPs for each blockage.
Figure 16(a) shows the total head and mass flow at the BEP. As the blockage decreases, a tendency of low head and high flow rate, which implies a high specific speed, is observed. Figure 16(b) shows the specific speed obtained using the total head and mass flow at each BEP. The specific speed at each BEP deviates from the original specific speed.

![Graphs showing head, mass flow rate, and specific speed](image)

**Figure 16.** (a) Head, mass flow rate, and (b) specific speed for each blockage at the BEP.

Figure 17 shows the results of the flow analysis at each BEP. For the 1% blockage, a separation is observed on the suction side of the impeller blade, even though it is the BEP for the 1% blockage model. This implies that when all the design parameters are fixed and only the blade thickness is varied, the BEP for a particular blockage may not be the actual BEP. Figure 18 shows the pressure distribution on the impeller shroud at each BEP. As the blockage increases, the overall pressure distribution at the BEP increases, contrary to the result obtained at the design point. The pressure values are dimensionless based on the maximum pressure at the BEP of the 20% blockage model.

![Streamline on the impeller hub of LE at the BEP](image)

**Figure 17.** Streamline on the impeller hub of LE at the BEP.

![Pressure contour on the impeller shroud at the BEP](image)

**Figure 18.** Pressure contour on the impeller shroud at the BEP.
5. Conclusions
In this study, the concept of blockage was defined to analyze the effect of blade thickness on the hydrodynamic performance of a mixed-flow pump, the performance characteristics of which were investigated using numerical analysis at the design and off-design points. The numerical analysis results at the design point showed that the performance of the mixed-flow pump changes depending on the blockage for the same design specifications, and there is an optimum blockage percentage for the mixed-flow pump. The flow field analysis results showed that the performance of the mixed-flow pump at a blockage of 1% significantly decreased because of the separation observed at this blockage. Moreover, the numerical analysis results at the off-design point showed that the BEPs tend to move to the lower flow rate point as the blockage becomes larger and to the higher flow rate point as the blockage becomes smaller. The total head at each BEP decreased with the decrease in the blockage percentage. As the blockage decreased, a tendency of low head and high flow rate was observed.

Acknowledgment
This work was supported by a grant (No. 10044860) from the Korea Institute of Industrial Technology (KITECH), which was funded by the Ministry of Science and ICT (MSIT). The authors gratefully acknowledge this support.

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