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High-efficiency design of a mixed-flow pump

KIM Jin-Hyuk[1], AHN Hyung-Jin[2] & KIM Kwang-Yong[1*]

[1] *Department of Mechanical Engineering, Inha University, 253, Yonghyun-Dong, Incheon, 402-751, Republic of Korea;*
[2] *Industrial Machinery Division, Iljin Electric Co., Ltd, 5-5, Hwasu-Dong, Incheon, 401-020, Republic of Korea*

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High-efficiency design of a mixed-flow pump has been carried out based on numerical analysis of a three-dimensional viscous flow. For analysis, the Reynolds-averaged Navier-Stokes equations with a shear stress transport turbulence model were discretized by finite-volume approximations. Structured grid system was constructed in the computational domain, which has O-type grids near the blade surfaces and H/J-type grids in other regions. The numerical results were validated with experimental data for the heads and hydraulic efficiencies at different flow coefficients. The hydraulic efficiency at the design flow coefficient was evaluated with variation of the geometric variables, i.e., the area of the discharge and length of the vane in the diffuser. The result has shown that the hydraulic efficiency of a mixed-flow pump at the design condition is improved by the modification of the geometry.

mixed-flow pump, numerical analysis, impeller, vane diffuser, head, hydraulic efficiency

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1 Introduction

Numerical analysis based on three-dimensional Reynolds-averaged Navier-Stokes (RANS) equations has been widely used for the flow analysis of turbomachinery with the aid of recent development of computing power. Application of this analysis has reduced the cost and the time for the design of pumps by reducing the number of experimental tests.

Miner [1] reported on CFD analysis of the first-stage rotor and stator in a two-stage mixed-flow pump. Goto [2] reported on the methodology to design low specific speed diffusers having a good efficiency based on an inverse design method and CFD simulation. Gulich [3] performed analysis of three-dimensional flow for the design of pumps. Tamaki et al. [4] have carried out the study of matching between centrifugal impellers and diffusers.

Recently, efficiency of pumps becomes increasingly important for the applications to industrial pumps. Oh and Kim [5] performed design optimization of mixed-flow pump impellers using mean streamline analysis. Zangeneh et al. [6] used a three-dimensional inverse design method to optimize radial and mixed-flow pump impellers. Goto et al. [7] reported on the suppression of meridional secondary flows on the blade suction surface of a mixed-flow pump impeller by controlling the blade pressure distribution using an inverse design method.

In this work, three-dimensional RANS analysis is performed to find the effects of two geometric variables related to the area of the discharge and length of the vane in the diffuser on the efficiency of a mixed-flow pump.

2 Specifications

In this work, flow analysis for a mixed-flow pump for the irrigation and drainage has been carried out. The mixed-

*Corresponding author (email: kykim@inha.ac.kr)

flow pump model has the specific speed, $Ns=N\ Q^{0.5}/H^{0.73}=1099.1$ at the best efficiency point (BEP). Figure 1 shows the three-dimensional geometries of the impeller and diffuser of the mixed-flow pump and the main flow area on the meridional plane. The mass flow rate and total head at the reference design are 568.15 m^3/min and 8.92 m, respectively. The major design specifications are listed in Table 1.

3 Numerical analysis

The commercial code ANSYS-CFX 11.0 [9] was used for present flow analysis in a mixed-flow pump. Blade profile creation, computational mesh generation, boundary condition definitions, flow analysis and post processing were performed by Blade-Gen, Turbo-Grid, CFX-Pre, CFX-Solver, and CFX-Post, respectively.

Steady-state simulations with water were performed. Adiabatic and hydraulically smooth walls with no slip condition were considered at solid boundaries. Periodic boundaries were set at the blade passage interfaces. The total pressure at the inlet was set to 1.0 atm, and the design mass flow rate was set at the outlet. The stage method was used for interface condition between the rotating impeller and stationary diffuser.

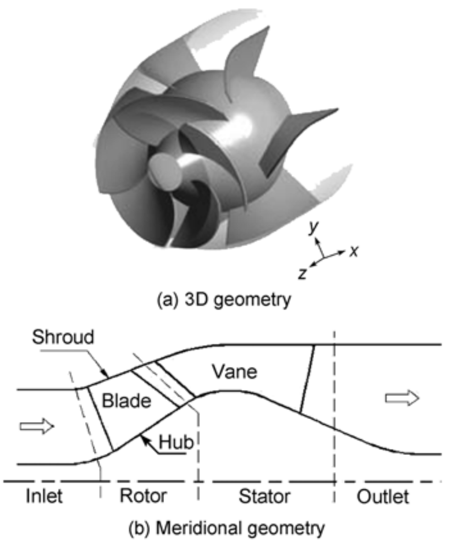

Figure 1 Shape of the mixed-flow pump.

Table 1 Design specifications

Design volume flow rate (m^3/min)	568.15
Rotational speed (r/min)	238.0
Total head (m)	8.92
Tip clearance (mm)	1.0
Number of rotor blade (& stator blade)	5 (6)
Maximum diameter of impeller (mm)	1799.0

A structured grid system was constructed in the computational domain, which has O-type grids around the blade surfaces and H/J-type grids in other regions. All of them are composed of hexahedral grid system as shown in Figure 2.

RANS equations were discretized using finite volume approximations, and the shear stress transport (SST) turbulence model (Menter et al. [8]) was used as a turbulence closure. In the SST model, a k-ω model was used in the near-wall region, and a k-ε model was used beyond the wall region. And, a blending function ensured a smooth transition between the two models.

4 Results and discussion

4.1 Validation of analysis results

Prior to the step for analysis of a mixed-flow pump, in order to determine the optimal number of grids, a preliminary grid dependency test with numbers of nodes ranging from 200000 to 800000 was carried out as shown Figure 3. As the result, 600000 was selected as the optimum number of grids.

The performance test was set up as in Figure 4. The static pressure was measured at the outlet of the mixed-flow pump to determine the total head, and the dynamic pressure was calculated by measuring the discharging mass flow rate.

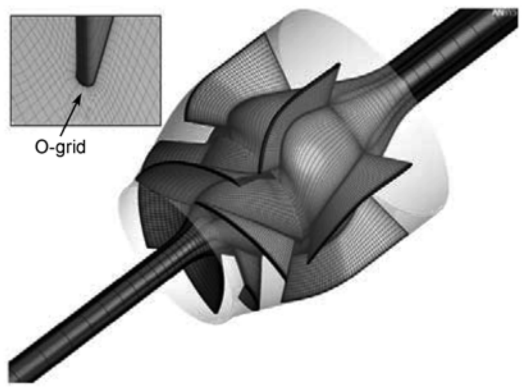

Figure 2 Computational grids.

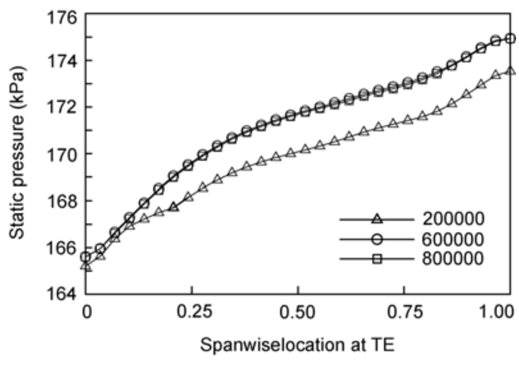

Figure 3 Grid dependency test results.

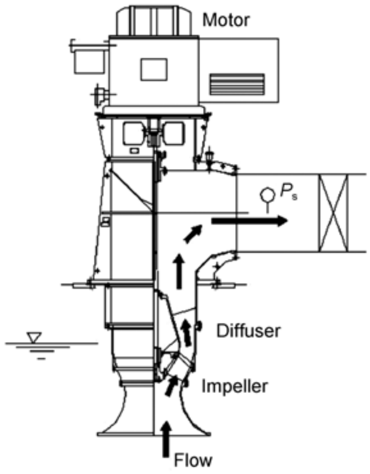

Figure 4 Overview of the test setup.

The rotating speed was measured by the optical sensor on the shaft transferring power and the power was calculated by the input power and the efficiency of the preliminarily tested motor.

For the performance presentations, the flow coefficient, the head coefficient, and efficiency were defined as follows:

$$\phi = \frac{Q}{ND^3}, \tag{1}$$

$$\psi = \frac{gH}{N^2D^2}, \tag{2}$$

$$\eta = \frac{\rho gHQ}{P}, \tag{3}$$

where Q, N, D, g, H, ρ and P indicate the volume flow rate, rotating speed, diameter, acceleration of gravity, total head, density and power, respectively.

Numerical results for the flow analysis were validated in comparison with the experimental results as shown in Figure 5. The predictions for the efficiency and head agreed well with the results of the performance test.

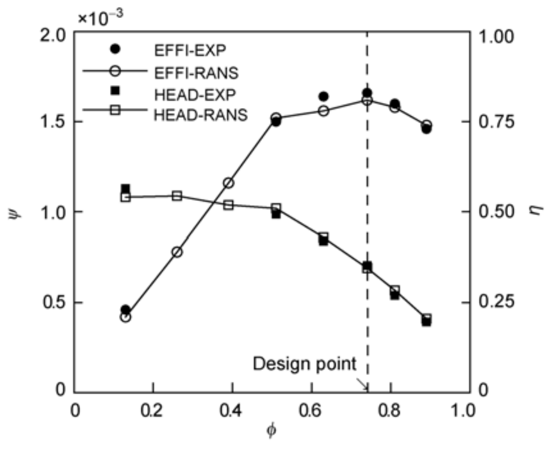

Figure 5 Validation of the flow analysis.

4.2 Results of numerical analysis at the design point

The analysis of the internal flow field at the design point indicated that some pressure losses occurred in the region of the diffuser. Figure 6 shows the total pressure and static pressure distributions at the mid-span. The total and static pressures increased well in the section of the impeller, but some pressure losses were found near the leading edge of the diffuser.

Figure 7 shows the static pressure contours at mid- span in the diffuser. A low static pressure region was found at the trailing edge of each diffuser vane, which was probably due to an undesirable flow structure and resulted in performance loss of the pump.

4.3 Modification of the geometry

Effects of two geometric parameters, i.e., the straight vane length ratio (SVLR) and the diffusion area ratio (DAR), on the efficiency (η) were tested in this work. It was assumed that the geometry change affects the flow structure and the pressure loss near the trailing edge of the diffuser vane. Figure 8 shows the straight vane length ratio and the diffusion area ratio, which are defined respectively as follows.

$$\text{Straight vane length ratio} = L_2/L_1, \tag{4}$$

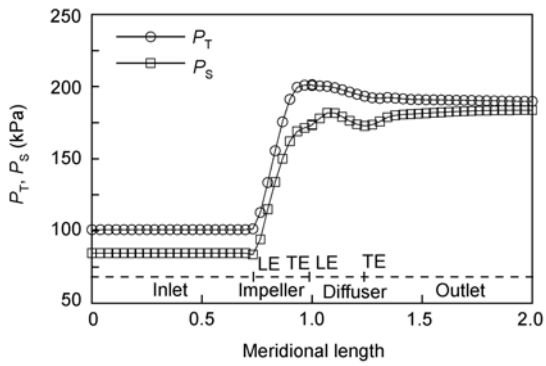

Figure 6 Total pressure and static pressure distributions at mid-span.

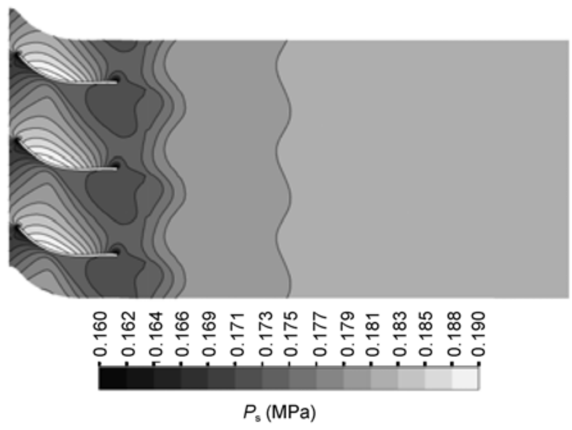

Figure 7 Total pressure and static pressure distributions at mid-span.

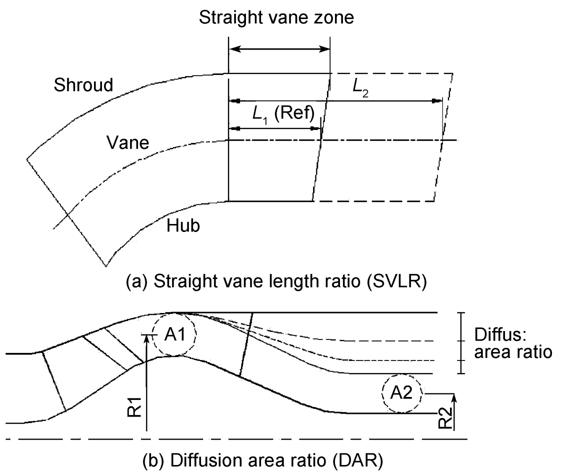

Figure 8 Definitions of SVLR and DAR.

Table 2 Efficiencies with modification of the geometry

Designs		SVLR					
		0.03	0.54	1.00	1.55	2.05	2.56
DAR	0.23	0.769	0.837	0.841	0.808	0.740	0.743
	0.35	0.803	0.866	0.872	**0.875**	**0.875**	0.874
	0.58	0.808	0.830	0.842	0.853	0.856	0.858
	1.00	0.805	0.846	**0.808**	0.802	0.811	0.796

$$\text{Diffusion area ratio} = A_2/A_1, \tag{5}$$

where L and A are vane length and area, respectively.

Table 2 shows the variation of the efficiency with the changes of the straight vane length ratio and the diffusion area ratio. Here, both the straight vane length ratio and diffusion area ratio of reference shape are 1.00. The efficiency is the highest at the diffusion area ratio 0.35 and the straight vane length ratios 1.55 and 2.05.

5 Conclusion

Numerical calculations have been performed for high-efficiency design of a mixed-flow pump through three-dimensional RANS analysis. The numerical results were validated by comparison with experimental data. The analysis results have shown that at the design condition there is a low static pressure region near the trailing edge of each diffuser vane. It is found that the efficiency of the mixed-flow pump is sensitive to the straight vane length ratio and the diffusion area ratio. With a parametric study, the optimum values of these ratios for high-efficiency design have been found.

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