

PAPER • OPEN ACCESS

Numerical Investigation on Hydrodynamic Characteristics of a Centrifugal Pump with a Double Volute at Off-Design Conditions

To cite this article: H S Shim and K Y Kim 2016 *IOP Conf. Ser.: Earth Environ. Sci.* **49** 032007

View the [article online](#) for updates and enhancements.

You may also like

- [Research of fluid-induced pressure fluctuation due to impeller-volute interaction in a centrifugal pump](#)
Q Z Liu, K Yang, D Y Li et al.
- [Volute throat area and wall modelling influence on the numerical performances of a very low specific speed pump](#)
Lilian Chabannes, David Stefan and Pavel Rudolf
- [Influence of volute section form on hydraulic performance and pressure fluctuation in double suction centrifugal pump](#)
D Zhu and R Xiao



ECS
The
Electrochemical
Society
Advancing solid state &
electrochemical science & technology

DISCOVER
how sustainability
intersects with
electrochemistry & solid
state science research

Numerical Investigation on Hydrodynamic Characteristics of a Centrifugal Pump with a Double Volute at Off-Design Conditions

H S Shim¹, K Y Kim¹

Department of Mechanical Engineering, Inha University, Republic of Korea

E-mail: kykim@inha.ac.kr

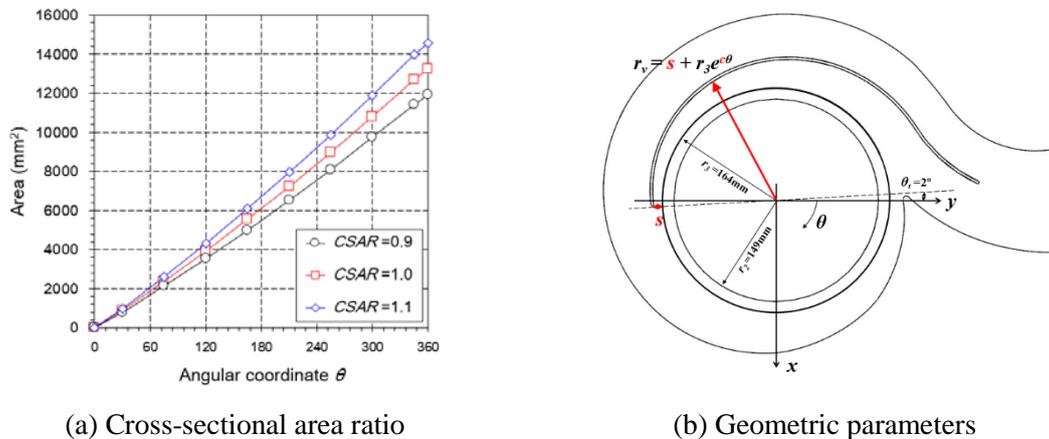
Abstract. Severe radial thrust under off-design operating conditions can be harmful factor for centrifugal pumps. In the present work, effects of geometry of a double volute casing on the hydrodynamic performance of a centrifugal pump have been investigated focusing on off-design conditions. Three-dimensional steady Reynolds-averaged Navier-Stokes analysis was carried out by using shear stress transport turbulence model. Numerical results for the hydrodynamic performance of the centrifugal pump were validated compared with experimental data. The hydraulic efficiency and radial thrust coefficient were used as performance parameters to evaluate the hydrodynamic characteristics of the centrifugal pump. The cross-sectional area ratio of the volute casing, the expansion coefficient of the rib structure, distance between the rib starting point and volute entrance, and radius of the volute entrance, were selected as geometric parameters. The results of parametric study show that performance parameters are significantly affected by both the geometric variables and operating conditions. Some configurations of the double volute casing showed outstanding performance in terms of the efficiency and radial thrust coefficient.

1. Introduction

As an important transport equipment of various fluids by the conversion of rotational kinetic energy to the hydrodynamic energy of the fluid flow, centrifugal pumps have been widely used in various applications. In last several decades, the hydraulic performance of centrifugal pumps has been significantly improved by many researchers. But, the operating stability caused by the flow complexity in centrifugal pumps, still faces a challenge [1]. As an efficient way to enhance the operating stability of centrifugal pumps, double volute casing where a rib structure is installed, is generally known to reduce the radial thrust by balancing the asymmetric pressure distribution in the volute casing of the centrifugal pump.

Several experimental and numerical studies have been performed for the double volute casing of centrifugal pumps. Benigni et al. [2] numerically investigated the hydraulic performance of a low specific speed double volute pump for Petroleum industry using the steady and unsteady RANS analysis. Baun and Flack [3] experimentally compared the performances of a centrifugal pump





(a) Cross-sectional area ratio

(b) Geometric parameters

Figure 1. Definitions of geometric parameters.

with spiral single, concentric, and double volute casings having same cross-sectional area. They confirmed that the double volute casing can significantly reduce the radial thrust of the centrifugal pump at off-design conditions, but, the hydraulic performance was deteriorated at high flow rates. They also found that the radial thrust coefficient from their research showed good agreements with the force magnitude predicted from the Hydraulic Institute correlation [4]. Kurokawa et al. [5] experimentally investigated the effects of the arrangement and length of the rib structure of the double volute casing on the performance of a centrifugal pump. It was confirmed that best efficiency point (BEP) of the conventional symmetrical design of the double volute casing moved to 90% of design flow rate of the single volute, which caused the hydraulic losses at high flow rate. But, they suggested that the hydraulic performance could be enhanced by adjusting the configurations of the rib structure.

In the present study, parametric analysis of double volute casing was performed to enhance both the hydraulic efficiency and operating stability of a centrifugal pump by using Reynolds-averaged Navier-Stokes (RANS) analysis. In order to evaluate the correlation of the geometric parameters, standard orthogonal table of L_9 (3^4) was adopted as the design of experiments (DOE) [6]. Detailed flow fields in the single and double volute casings were analysed and investigated in this work.

2. Specifications of Centrifugal Pump

Tested centrifugal pump consists of an impeller with six blades, circular cross-sectional shaped spiral volute with the rib structure inside the flow passage, suction pipe, and discharge pipe. Suction angle of the tested impeller at shroud and hub are 16.0 and 23.6 degree, respectively, and discharge angle of the tested impeller is 26 degree. The design flow rate and total head of the tested pump are $8\text{ m}^3/\text{min}$ and 34m, respectively, with the rotational speed of 1,780rpm. From these operating conditions, a specific speed of the centrifugal pump was determined as $N_s = NQ^{0.5}/H^{0.75} = 360$. The details of the double volute casing are shown in Fig. 1 (b). Tongue of the rib structure is positioned at 180 degree from the volute tongue in order to reduce the radial thrust of the centrifugal pump. The radius of the rib structure was determined by the expansion ratio of the rib structure c , distance between the rib starting point and volute entrance, s , and diameter of the impeller diameter, D_2 .

3. Numerical Methods

The flow field of the centrifugal pump was investigated by three-dimensional RANS analysis using commercial CFD code ANSYS CFX 15.0 [7]. The governing equations were discretized by finite volume method and the advection terms in the RANS equations were discretized using the high resolution scheme which has a second order accuracy. The shear stress transport (SST) turbulence model [8] was used as a turbulence closure.

Table 1. Sets of standard orthogonal table of L_9 (3^4) applied on geometric parameters of double volute casing.

Rib structure configurations	CSAR	c	s/D_2
Case 1	0.9 [-1]	0.09 [-1]	0.20 [+1]
Case 2	0.9 [-1]	0.10 [0]	0.10 [0]
Case 3	0.9 [-1]	0.11 [+1]	0 [-1]
Case 4	1.0 [0]	0.09 [-1]	0.10 [0]
Case 5	1.0 [0]	0.10 [0]	0 [-1]
Case 6	1.0 [0]	0.11 [+1]	0.20 [+1]
Case 7	1.1 [+1]	0.09 [-1]	0 [-1]
Case 8	1.1 [+1]	0.10 [0]	0.20 [+1]
Case 9	1.1 [+1]	0.11 [+1]	0.10 [0]

ANSYS ICEM 15.0 and CFX-Turbo Grid were used to construct the computational grid systems in the volute with suction/discharge pipe and the impeller passage, respectively. A hexahedral grid system in the impeller and suction pipe domain and a tetrahedral grid system in the volute domain were constructed to discretize the entire computational domain. Dense grids near the solid surfaces were employed in order to resolve high velocity gradient due to the boundary layer, and the first grid points from the wall were located in $y^+ < 2$ to apply the low-Re turbulence model. In the previous work [9], a grid dependency test with various numbers of nodes was performed to find an optimum number of nodes. As a result, the optimum numbers of nodes were determined as 1,800,000, 240,000, 550,000, and 910,000 in the entire impeller passage, suction pipe, single volute, and double volute casing, respectively.

For the boundary conditions, the total pressure and the mass flow rate were set at inlet of the suction pipe and outlet of the discharge pipe, respectively. Water of 20°C and atmospheric pressure was used as the working fluid, and the hydraulically smooth and no-slip conditions were applied at the solid surfaces in the entire computational domain. The frozen-rotor interface method was used to connect between rotating (impeller) and stationary (suction pipe and volute) domains. It is well known that the frozen-rotor method is useful when the circumferential variations are more significant than the component pitch change, for example, at interface between impeller and volute [7].

As a convergence criterion, root mean square (RMS) residuals for all the governing equations were set to fall below 10^{-4} and the mass imbalance in the entire computational domain was less than 0.001%. Also, the relative changes in the performance parameters were less than 0.5% within 200 steps. The average computing time per single analysis was in a range of 7-8 hours with approximately 1000 iterations using a personal computer with an Intel core i7 3.6GHz CPU.

4. Geometric and performance parameters

Figure 1 shows the definition of the geometric parameters used in this work; the cross-sectional area ratio of the volute casing (CSAR), the expansion coefficient of the rib structure (c), ratio of the distance between the rib starting point and volute entrance to the diameter of the impeller (s/D_2).

Originally, the possible number of combinations of the geometric parameters is 27 in finding the effects of three geometric parameters on the hydrodynamic performance of the centrifugal pump. But, in order to minimize the expenditure of experiments, standard orthogonal table of L_9 (3^4) was adopted as the DOE in this study. Sets of the orthogonal table L_9 in this work indicate the representative configurations to find the relative importance of the selected geometric parameters. Wu et al. [10] proved that the analysis using the arrangement of orthogonal table drew a satisfactory solution in terms of the

performance of the turbomachinery. Table 1 shows the sets of standard orthogonal table of L_9 applied to the geometric parameters of the double volute casing.

In order to evaluate the hydrodynamic performances of the centrifugal pump at design and off-design conditions, two performance parameters: the hydraulic efficiency (η) and radial thrust coefficient (F) were considered in this work. The hydraulic efficiency was defined as;

$$\eta = \frac{(P_{t,out} - P_{t,in}) \cdot Q}{\tau \cdot \omega} \quad (1)$$

here, P_t and Q , indicate the total pressure at inlet and outlet, and flow rate, respectively. And, τ and ω are torque and angular velocity, respectively.

The radial thrust coefficient was defined as;

$$F = \frac{-\int p e_n dA}{0.5 \rho u_2^2 \pi D_2 b_2} \quad (2)$$

where, A , p , ρ are area of exit of the impeller, static pressure, density of water, respectively. Also, u_2 , D_2 and b_2 are blade tip velocity, diameter of the impeller, and width of the impeller at exit, respectively.

In this work, in order to evaluate the hydrodynamic performance of the centrifugal pump, the hydraulic efficiency and radial thrust coefficient were evaluated in a range of flow rate, 70% to 120% of the design flow rate, which was recommended as a preferred operating region by ANSI/API standard 610 [11].

5. Results and discussion

Present numerical results of hydraulic performance for the reference shape of the volute casing (no rib structure) were validated compared with experimental data [12] and statistical correlation for the radial thrust factor obtained by the Hydraulic Institute [4], as shown in Fig. 2. The numerical results show some discrepancies in the head coefficient and radial thrust coefficient at low flow rates, but show good agreements with the experimental data in a range of $\Phi/\Phi_D=0.7$ to 1.2.

Figures 3-5 show the effects of the geometric parameters (Table 1) on the hydraulic performance and radial thrust coefficient for $\Phi/\Phi_D=0.7$, 1.0 and 1.2. Each graph is plotted by using response average related to a specific geometric parameter for three flow rates to evaluate the effects of each geometric parameter on the hydraulic efficiency and radial thrust coefficient. Normalized value was

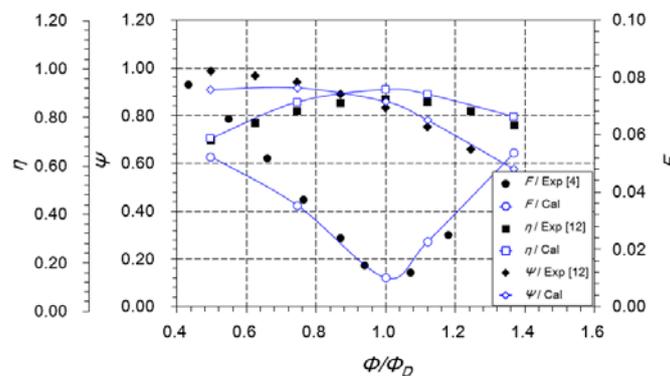
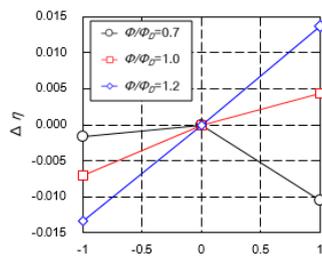
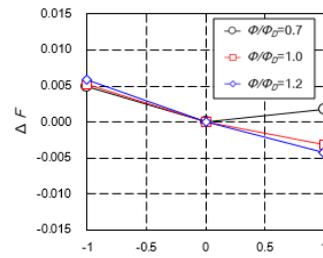


Figure 2. Validation of the numerical results [9].

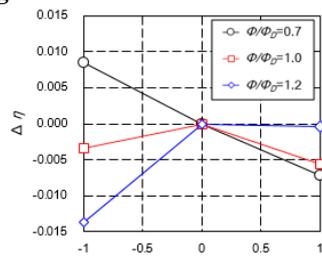


(a) Hydraulic efficiency

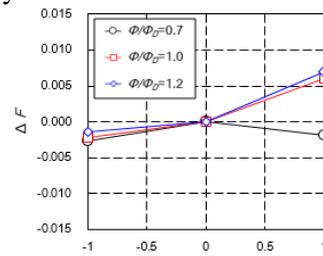


(b) Radial thrust coefficient

Figure 3. Effects of CSAR on the hydraulic efficiency and radial thrust coefficient.

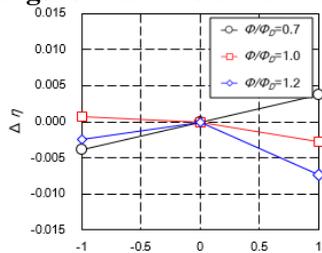


(a) Hydraulic efficiency

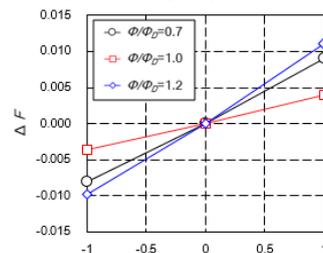


(b) Radial thrust coefficient

Figure 4. Effects of c on the hydraulic efficiency and radial thrust coefficient.



(a) Hydraulic efficiency



(b) Radial thrust coefficient

Figure 5. Effects of s/D_2 on the hydraulic efficiency and radial thrust coefficient.

Table 2. Results of the parametric study for double volute casing.

	$\Phi/\Phi_D=0.7$		$\Phi/\Phi_D=1.0$		$\Phi/\Phi_D=1.2$		Average variations (%)	
	η	F	η	F	η	F	$\Delta \eta$	ΔF
Ref	0.8229	0.0401	0.9081	0.0100	0.8660	0.0354		
Case 1	0.8347	0.0224	0.8824	0.0166	0.8245	0.0327	-1.84	4.87
Case 2	0.8246	0.0202	0.8885	0.0164	0.8437	0.0237	-1.34	-5.98
Case 3	0.8096	0.0091	0.8875	0.0160	0.8498	0.0174	-1.67	-22.71
Case 4	0.8308	0.0114	0.8959	0.0062	0.8524	0.0129	-0.60	-57.82
Case 5	0.8202	0.0032	0.8965	0.0060	0.8564	0.0073	-0.80	-70.53
Case 6	0.8228	0.0225	0.8872	0.0213	0.8493	0.0360	-1.26	23.71
Case 7	0.8203	0.0065	0.8972	0.0022	0.8546	0.0023	-0.83	-85.16
Case 8	0.8155	0.0248	0.9009	0.0092	0.8723	0.0211	-0.28	-28.76
Case 9	0.8063	0.0118	0.8945	0.0125	0.8721	0.0198	-0.81	-30.38

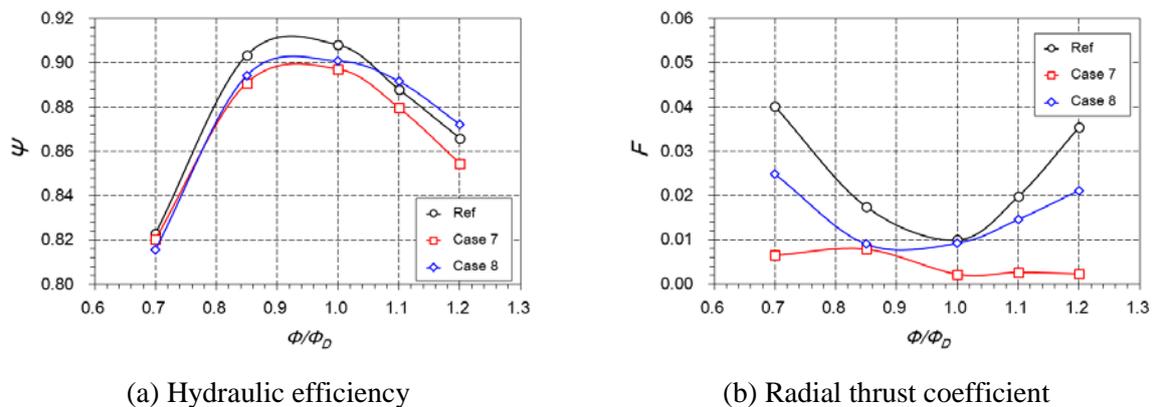


Figure 6. Performance curves.

used for each parameter as shown in Table 1. Figure 3 shows the effects of CSAR on the hydraulic efficiency and radial thrust coefficient. The variations of the hydraulic efficiency are larger than those of radial thrust coefficient. The hydraulic efficiency shows different trends depending on the flow rate. For $\Phi/\Phi_D=0.7$, the hydraulic efficiency shows a maximum value at middle of CSAR range, but, the hydraulic efficiency increases as CSAR increases at $\Phi/\Phi_D=1.0$ and 1.2 . In case of Fig. 3 (b), except for $\Phi/\Phi_D=0.7$ where a minimum value is found at middle of CSAR range, the radial thrust decreases as CSAR increases.

The effects of the c on the hydraulic efficiency and radial thrust coefficient are shown in Fig. 4. As shown in Fig. 4 (a), the hydraulic efficiency decreases as c increases for $\Phi/\Phi_D=0.7$, but shows maximum values at middle of c range for higher flow rates. Figure 4 (b) shows that the radial thrust increases with an increase of c for $\Phi/\Phi_D=1.0$ and 1.2 , but shows maximum value at the middle of c range for $\Phi/\Phi_D=0.7$.

The distance between the rib starting point and volute entrance is an important factor affecting the radial thrust coefficient (Fig. 5). As shown in Fig. 5(b), the variations of the radial thrust coefficient with s/D_2 are found to be largest among the tested geometric parameters. And, the radial thrust coefficient increases as s/D_2 increases regardless of the flow rate. However, the variations of hydraulic efficiency are relatively smaller than the other geometric parameters.

Table 2 shows the results of the parametric study for double volute casing. In each case, the hydraulic efficiency and radial thrust coefficient at $\Phi/\Phi_D=0.7$, 1.0 and 1.2 were calculated by the numerical analysis. Among these results, there are two noticeable results; one is case 7 for reducing the radial thrust coefficient and the other is case 8 for enhancing the hydraulic efficiency. Figure 6 shows the hydraulic efficiency and radial thrust performance curves for the reference shape for cases 7 and 8. Case 7 shows significant reductions in the radial thrust coefficient, especially at off-design conditions, and less than 1% reduction in the hydraulic efficient which decreases as the flow rate decreases. In case 7, the value of the radial thrust was reduced by 84% at $\Phi/\Phi_D=0.7$ and by 94% at $\Phi/\Phi_D=1.2$ compared to the reference shape. On the other hand, in case 8, the hydraulic efficiency is even better than that of the reference shape at $\Phi/\Phi_D=1.2$. At $\Phi/\Phi_D=1.2$, case 8 shows that the hydraulic efficiency was enhanced by 0.63% and the radial thrust coefficient was also decreased by 40% compared to the reference case.

Figures 7-9 represent the static pressure contours at mid-span of volute casing for different flow rates. It can be seen that the static pressure distributions of the reference shape at the impeller exit for $\Phi/\Phi_D=0.7$ and 1.2 , are more asymmetric compared to those for $\Phi/\Phi_D=1.0$, which is the reason why the larger radial thrusts of the centrifugal pump occur at off-design conditions. But, in case of cases 7 and 8, the rib structure inside volute casing, contributes to the reduction of the radial thrust by balancing the circumferential pressure distribution at the impeller exit as shown in Figs. 7 and 8.

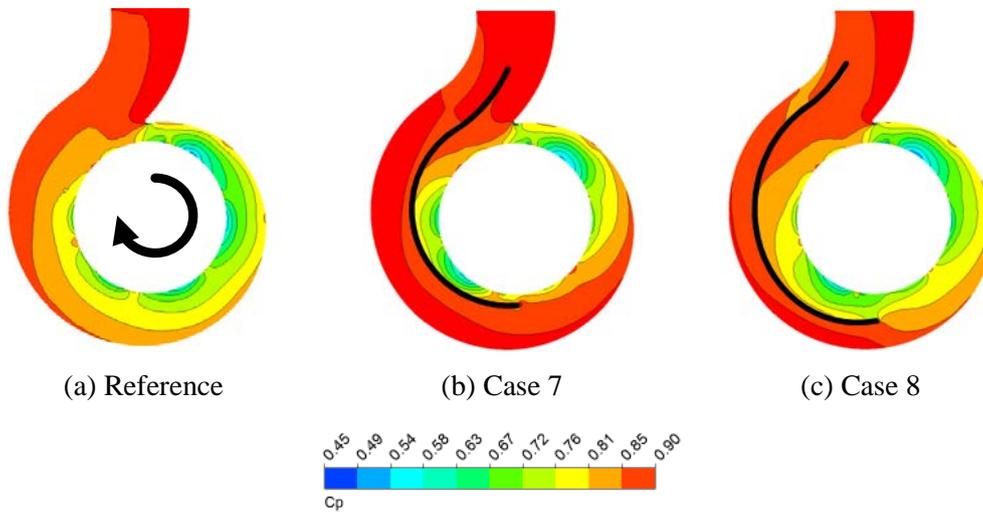


Figure 7. Static pressure contours at mid-span of volute casing at $\Phi/\Phi_D=0.7$.

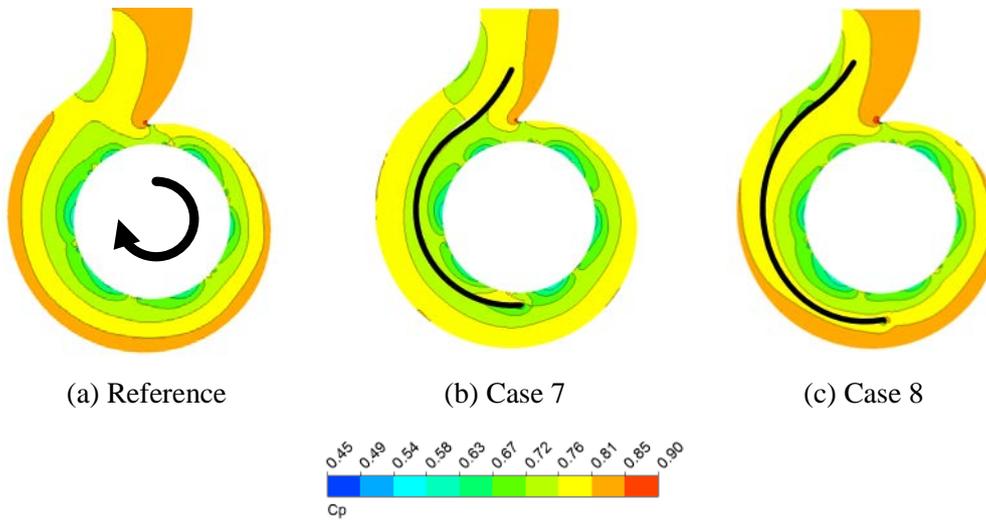


Figure 8. Static pressure contours at mid-span of volute casing at $\Phi/\Phi_D=1.0$.

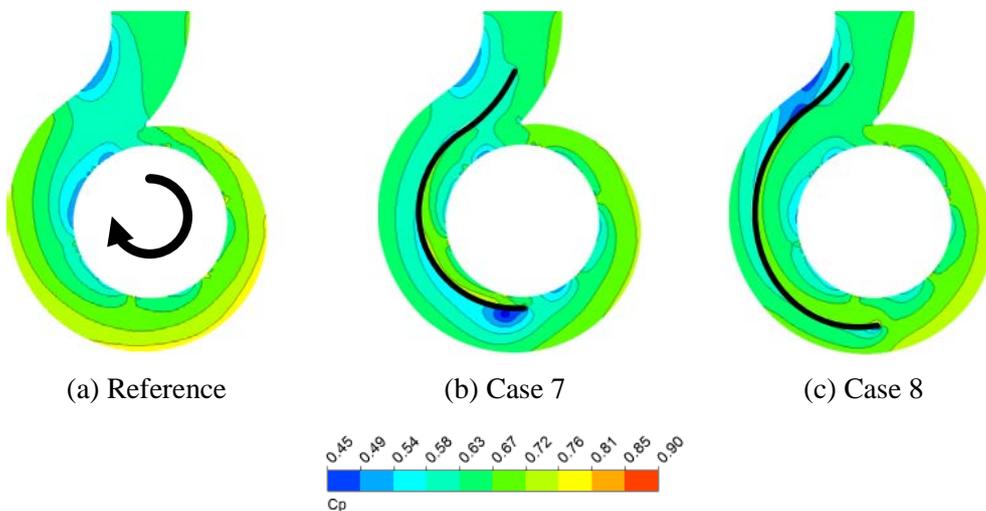


Figure 9. Static pressure contours at mid-span of volute casing at $\Phi/\Phi_D=1.2$.

6. Conclusions

In the present study, the hydraulic efficiency and radial thrust of a centrifugal pump with double volute casing were numerically evaluated using RANS analysis with SST turbulence model for different volute geometries. Tested parameters were related to the cross-sectional area of the volute casing and the geometry of the rib structure in the double volute casing. The cross-sectional area ratio of the volute casing significantly affected the hydraulic efficiency among the tested parameters; the relative variation of 2.7% at $\Phi/\Phi_D=1.2$. The expansion coefficient of the rib structure also affected the hydraulic efficiency of the centrifugal pump; the relative variation of 1.6% at $\Phi/\Phi_D=0.7$. On the other hand, the radial thrust coefficient showed strongest sensitivity to the ratio of the distance between the rib starting point and volute entrance to the diameter of the impeller among the tested parameters; the relative variations of 59% at $\Phi/\Phi_D=1.2$. From the DOE analysis, Case 7 shows the significant reduction in the radial thrust; maximum reduction of 94% at $\Phi/\Phi_D=1.2$. In Case 8, loss of the hydraulic efficiency was the lowest among the tested cases.

Acknowledgements

This research was supported by the Korea Evaluation Institute of Industrial Technology (KEIT) grant funded by the Ministry of Science, ICT, and Future Planning (No. 10044860).

References

- [1] Brennen C E 1994 *Hydrodynamics of Pumps* (New York:Cambridge University Press)
- [2] Benigni H Jaberg H Yeung H Salisbury T Berry O and Collins T 2012 *J. Fluids Eng.* **134** 024501
- [3] Baun D O and Flack R D 2003 *Int. J. Rotating Mach.* **9** 145
- [4] Hydraulic Institute 1994 *American National Standard for Centrifugal Pumps for Nomenclature, Definitions, Application and Operation* Hydraulic Institute
- [5] Kurokawa J Fujii T Matsui J and Kitahora T 1997 *Proc.of the 5th Asian Int. Conf. on Fluid Mach.* **1** 825
- [6] Montgomery D C 1997 *Design and Analysis of Experiments* (New York:John Wiley & Sons)
- [7] ANSYS CFX-15.0 2014 *ANSYS CFX-Solver Theory Guide* ANSYS Inc
- [8] Menter F R 1994 *AIAA J.* **32** 1598
- [9] Shim H S Heo M W and Kim K Y 2016 *J. Fluid Mach.(in korean)* **19** 37
- [10] Wu Y Chu W Zhang H and Li Q 2010 *J. Fluids Eng.* **132** 121103
- [11] American Petroleum Institute 2010 *Centrifugal Pumps for Petroleum, Petrochemical and Natural Gas Industries, ANSI/API Standard 610* American Petroleum Institute
- [12] Korea Ministry of Trade, Industry and Energy 2014 *Development of Design Program for Centrifugal and Mixed-flow Pump (Ns 150-1200)*.