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Shape Optimization of Impeller Blades for a Bidirectional Axial Flow Pump using Polynomial Surrogate Model

I. S. Jung, W. H. Jung, S. H. Baek, S. Kang

***Abstract*—**This paper describes the shape optimization of impeller blades for a anti-heeling bidirectional axial flow pump used in ships. In general, a bidirectional axial pump has an efficiency much lower than the classical unidirectional pump because of the symmetry of the blade type. In this paper, by focusing on a pump impeller, the shape of blades is redesigned to reach a higher efficiency in a bidirectional axial pump. The commercial code employed in this simulation is CFX v.13. CFD result of pump torque, head, and hydraulic efficiency was compared. The orthogonal array (OA) and analysis of variance (ANOVA) techniques and surrogate model based optimization using orthogonal polynomial, are employed to determine the main effects and their optimal design variables. According to the optimal design, we confirm an effective design variable in impeller blades and explain the optimal solution, the usefulness for satisfying the constraints of pump torque and head.

***Keywords*—**Bidirectional axial flow pump, Impeller blade, CFD, Analysis of variance, Polynomial surrogate model

I. Introduction

TO prevent the ship stranding by the over loading and sustain regular angle system is called “Anti-Heeling system”. For the ship balance, bidirectional axial flow the pump has transport the water in the ballast tank that installed on both sides of the ship.

Anti-heeling system has preferring the axial flow pump than general centrifugal pump because that system has a symmetric flow. Electric motor has installed inside of axial flow pump vertically, bevel gear work the impeller that installed on both side of shaft end. This type of system has a low efficiency and wasting too much space than other types [1,2]. In these days, ship building company use to internal motor bidirectional axial pump to decrease the defect of general bidirectional axial pump. Internal motor bidirectional axial pump has higher efficiency and less space than old one because that electric motor installed inside of impeller hub.

By the complex interaction of fluid dynamics variable, mechanical design and performance criteria depend on the empirical equation. Study results from Stepanoff [3] and Neumann [4] well described the relational expression of exprimental loss and general theory of pump design. The Important things that needed of bidirectional pump design is compromise between head of fluids rapid influx by impeller rotation and maximum efficiency [5,6].

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The comparison with directional axial flow pump, airfoil of bidirectional axial flow pump has a symmetric chord and same angle of inlet and outlet. Velocity triangle diagram is hard to applicable to impeller design because symmetric airfoil had no theoretical lift force. Also, guide vane construct for decrease the resistance loss is more limited than directional axial flow pump.

It is reason that decrease of efficiency and fluid dynamic performance of bidirectional axial flow pump. From this point of view, the objective of this study is impeller blades shape design optimization for improve the hydrodynamic performance (torque and head) and efficiency.

CFD (computation fluid dynamics) simulations for bidirectional axial flow pump demanded massive time for calculation because complex of geometrical, physical.

Therefore operate the surrogate model for saving time and accurate calculation by approximate design optimization [7-9]. Composition of surrogate model is using orthogonal polynomial including sensitivity information based on analysis of variance [10-13]. From procedure of design optimization, about the initial impeller blades shape, shows the improved impeller blades shape that balanced between torque, head and efficiency. Also, describe the reason of inner fluid flow improvement and effectiveness from compared with initial model and improved model.

II. Bidirectional Axial Flow Pump

A. Composition and Operating Principle of Pump

Fig. 1 shows the schematic of a bidirectional axial flow pump for ship balance. Bidirectional axial flow pump has made up of 3 parts. The motor part installed at the inside of pump. Impeller placed at the end of both axial rod ends. And 4 vanes has installed at the inside of pump.

The main operation principles is the internal electric motor rotate and the two ends of the impeller rotate in the same direction as feed the water to ballast tank for stabilize the ship posture. For fixing the motor posture and stabilize the inner flow as cylindrical direction flow change into axial direction flow, 4 guide vanes has installed inside of pump pipe.

The reason of guide vanes installed on inside of pump has 2 impeller installed on the both side of axial end. The impeller makes water rotating move and changes the fluid inlet angle. The hydrodynamic performance efficiency has dropped by that reason.

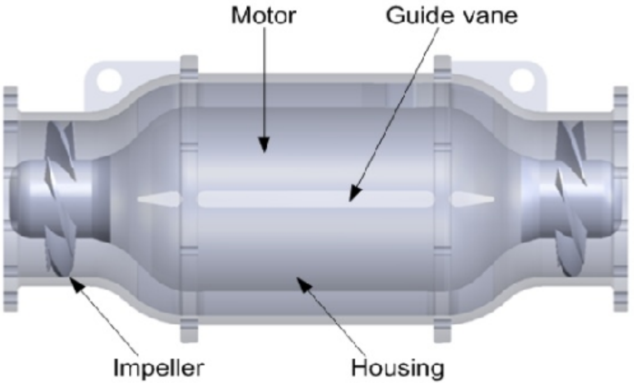
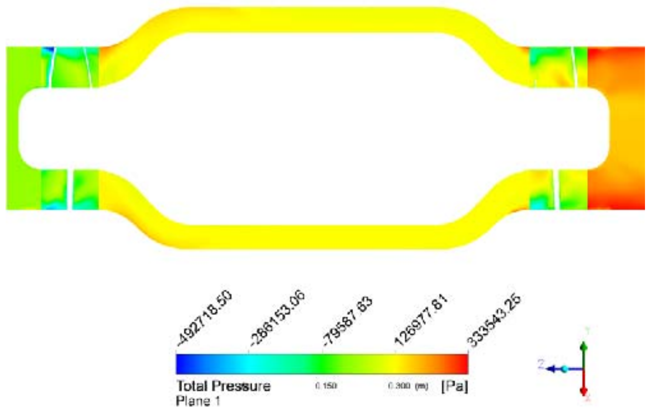
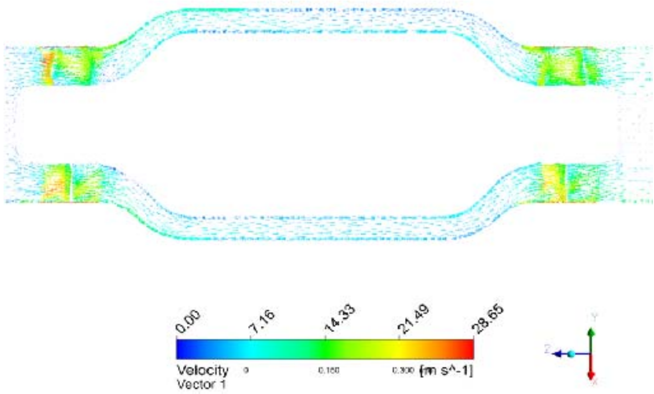


Fig. 1 Schematic of a bidirectional axial flow pump



(a) Total pressure



(b) Velocity magnitude

Fig. 2 CFD results on the surfaces and a cutting plane for the initial model

Properties of seawater as working liquid has 32 ℃ and 1.03 kg/m^3 specific density. Design objectives are as follows : Rotation speed of impellers : 1800 rpm, rated flow : 1000 m^3/h and total pump head : 15 m.

B. Flow Analysis

In order to analyze the fluid performance of the bidirectional

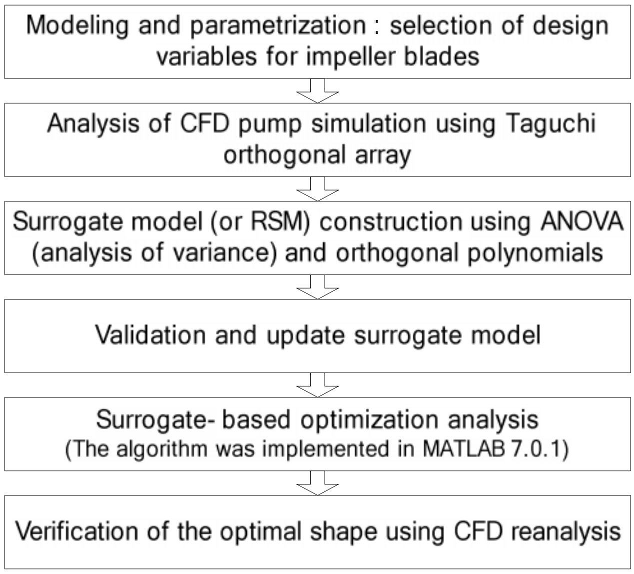


Fig. 3 Optimization procedure of impeller blades

axial flow pump, we used ANSYS CFX v.13 [14]. As the turbulence model, SST (shear stress transport) based on k-ε was used, and for convection scheme, we used high resolution based on upwind biased approach [15,16]. By using ANSYS ICEM CFD, hexahedron, tetrahedron, and prism grid types were combined and grid system was produced.

Grid system was generated by mixed as tetragonal, Hexagonal and prism mesh by using ANSYS ICEM CFD. Pump and around of impeller area was filled with tetragonal mesh, another area was filled with hexagonal and prism mesh. Boundary conditions are as follows: no-slip condition was applied all around of walls, 0 Pa applied at inlet area and 800 m^3/h ~1200 m^3/h volume flow rate applied at outlet area for estimate the performance variation of pump by variation of volume flow rate.

Fig. 2 shows the result of CFD analysis for initial model. Difference of total pressure from rated flow was 1.8bar. Torque and head was 402 N·m, 19m and efficiency was 67.4%. About the rated flow 1000 m^3/h, 19 m total pump head was excess designed. For this reason, shaft power has increased, efficiency has decreased. Therefore, design optimization has demanded for satisfy the design criteria and higher efficiency.

III. Design Optimization of Impeller Blades

Fig. 3 shows the procedure of design optimization of impeller blades for improve the bidirectional axial flow pump efficiency. There are 2 steps of main procedure for design optimization, establishing surrogate model by using ANOVA and optimization based on surrogate model. From this optimization, hydrodynamic performance and efficiency can be approximated as computation-intensive function interacted with simple analytical model (cheap-to-compute model or simulation model included gradient information) [17,18].

Therefore performing optimization is effective and simple when obtain the reliability of approximation model. Each step of under the sections of design optimization will be described in detail.

TABLE I
DESIGN VARIABLE AND THEIR LEVES

Design variable	Unit	Level 1	Level 2	Level 3	Level 4	Level 5
x_1	mm	7	7.5	8	8.5	9
x_2	mm	113	123	133	143	153
x_3	mm	160	174	188	202	216
x_4	Deg.	12	13	14	15	16
x_5	Deg.	29	31.5	34	36.5	39

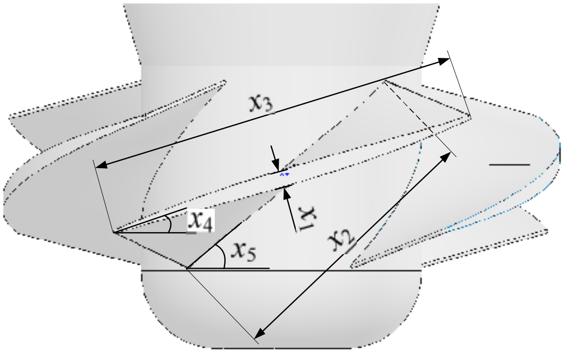


Fig. 4 Design variable of impeller blades

A. Design model and goals

The objective of design optimization for bidirectional axial flow pump is satisfy the object pump head and low shaft power. The design requirements as follows: (1) In order to satisfy the pump design specification, total head design criteria of pump is 15 m. (2) Since 200 kW motor is used, the maximum torque must be below 600 N·m.

Fig. 4 shows design variable, it is as follows: x_1(maximum thickness of blade), x_2 (hub length), x_3 (chord length), x_4 (chord angle), x_5 (hub length). A number of design variables are five. The base on initial model, range of design variable established without interference. To determine the range of the design variables of the initial model, the five impeller blades (level 3) for the lower level (level 1) and upper level (level 5) 15% each of them. Table Ⅰ shows the design variable and variable range.

B. Formulation of design optimization problem

The objective of this paper is bidirectional axial flow pump design that requires under the constraints of the head and torque to get the maximum efficiency is to determine the shape of the impeller blades. Design optimization problem of the objective function and constraints formulated as a function of the following Eq. (1).

Find x_1, x_2, x_3, x_4, x_5
to maximize $y_{efficiency}(x_i)$
subject to $y_{\text{torque}}(x_i) \le 600$ [N·m]
$y_{\text{head}}(x_i) \ge 15$ [m], $i = 1, 2, \cdots 5$ (1)

where y(x_i) is an approximate model for each response. First and second constraint was conducted the minimum head to 15m when maximum torque is 600N·m. Obtain the exact solution of approximate model by optimization module of ANSYS v. 11 and feasible direction method [19,20].

C. The surrogate model based ANOVA and orthogonal polynomial.

Design optimization method of the impeller is composed of design of experiments including the response surface methodology. The two important elements of traditional approaching method to response surface model is Taylor series that using certain design points or differential solution and orthogonal polynomial series that using integral calculus. Taylor series form is more suitable for the minor design variable response and range changing than orthogonal polynomial form. Design variables that are orthogonal to the effect of any variable on the effects of other variables will not affect. Therefore, design of experiment that had orthogonal design variable is appropriate for analyzing response range. Also, every terms of orthogonal polynomial is independent, so Sequential obtain the coefficient from lower to higher terms. This could shows the regression equation efficiently because we can normalize the base, even if doesn't know about high-order coefficient or large gap of coefficient. Response surface model for impeller blade shape using the Chebyshev orthogonal polynomial $p_n(x)$ that n degree of design variable [10,11,21]. If these are expressed in the form of a second-order polynomial, and can be expressed as Eq. (2).

$$\begin{aligned} y &= b_0 + b_1(x-\bar{x})^2 + b_2\left[(x-\bar{x})^2 - \frac{a^2-1}{12}h^2\right] \\ &\quad + b_3\left[(x-\bar{x})^3 - \frac{3a^2-7}{20}(x-\bar{x})h^2\right] \\ &\quad + b_n p_n(x) + \cdots \end{aligned} \tag{2}$$

$$p_0(x) = 1, \quad n = 0$$

$$p_1(x) = x - \bar{x}, \quad n = 1$$

$$p_2(x) = (x-\bar{x})^2 - \frac{(a^2-1)}{12}h^2, \quad n = 2$$

$$p_n(x) = p_{n-1}(x)p_1(x) - (n-1)^2\{a^2-(n-1)^2\} \; h^2 p_{n-1}(x)/[4\{4(n-1)^2-1\}], \quad n = 3,4,5\cdots$$

where $\bar{x}$ is the average of the a levels of design variable and h is the interval between the design variable levels.
Note that the degree n should be less than a. The maximum degree for each design variable is $a-1$. b_o, b_i can be expressed as regression coefficients by Eq. (3).

$$b_0 = T/lm = \bar{y} \tag{3}$$

$$b_i = \sum_{i=1}^{a} p_n(x_i)y_i \Big/ \sum_{i=1}^{a} p_n^2(x_i), \quad i = 1, 2, \cdots, a \tag{4}$$

TABLE II
LAYOUT AND CFD RESULTS FOR L25 ORTHOGONAL ARRAYS

Exp.	x_1	x_2	x_3	x_4	x_5	Torque (N·m)	Head (m)	η (%)
1	1	1	1	1	1	224.5	10.6	68.3
2	2	2	2	2	2	314.7	14.9	68.5
3	3	3	3	3	3	402.8	18.8	67.4
4	4	4	4	4	4	484.5	21.5	64.0
5	5	5	5	5	5	559.0	23.0	59.4
6	1	2	3	4	5	514.1	22.0	62.2
7	2	3	4	5	1	399.5	19.1	69.1
8	3	4	5	1	2	256.8	12.1	68.2
9	4	5	1	2	3	400.2	18.6	67.1
10	5	1	2	3	4	411.2	19.0	66.8
11	1	3	5	2	4	396.7	18.5	67.2
12	2	4	1	3	5	519.8	21.8	60.4
13	3	5	2	4	1	386.7	18.6	69.6
14	4	1	3	5	2	406.4	19.1	67.7
15	5	2	4	1	3	275.0	13.0	68.3
16	1	4	2	5	3	508.9	22.0	62.3
17	2	5	3	1	4	407.2	18.7	66.5
18	3	1	4	2	5	391.0	17.7	65.4
19	4	2	5	3	1	268.5	12.8	69.2
20	5	3	1	4	2	401.8	18.8	67.4
21	1	5	4	3	2	387.9	18.6	69.1
22	2	1	5	4	3	395.1	18.3	66.9
23	3	2	1	5	4	502.4	21.2	61.0
24	4	3	2	1	5	417.0	18.8	65.2
25	5	4	3	2	1	267.5	13.0	70.0

TABLE III
ANALYSIS OF VARIANCE FOR TORQUE

Design variable		Sum of squares	DOF	Variance	F-ratio	Effective ratio(%)
x_1	Linear	1736	1	16	2.18	1.13
x_2	Linear	12427	1	24	3.21	1.67
x_3	Linear	3978	1	213	28.35	14.71
x_4	Linear	80283	1	5	0.71	0.37
x_5	Linear	91842	1	147	19.46	10.10
	Quadratic	21	1	96	12.7	6.59
x_1x_2	Interaction	66	1	0	0.01	0.01
x_1x_3	Interaction	98	1	1	0.07	0.04
x_1x_4	Interaction	38	1	8	1.04	0.54
x_1x_5	Interaction	65	1	135	17.98	9.33
x_2x_3	Interaction	73	1	134	17.81	9.24
x_2x_4	Interaction	49	1	25	3.34	1.73
x_2x_5	Interaction	406	1	0	0.02	0.01
x_3x_4	Interaction	308	1	375	49.75	25.82
x_3x_5	Interaction	47	1	15	1.98	1.03
x_4x_5	Interaction	257	1	257	34.1	17.69
Error		60	8	60		
Total			24		190.71	100

TABLE IV
ANALYSIS OF VARIANCE FOR HEAD

Design variable		Sum of squares	DOF	Variance	F-ratio	Effective ratio(%)
x_1	Linear	2.752	1	0.046	1.04	0.39
x_2	Linear	20.752	1	0	0	0
x_3	Linear	5.139	1	1.181	26.84	10.11
x_4	Linear	123.78	1	0.059	1.33	0.50
x_5	Linear	108.84	1	1.323	30.07	11.33
	Quadratic	1.099	1	1.114	25.32	9.54
x_1x_2	Interaction	1.72	1	0.004	0.1	0.04
x_1x_3	Interaction	4.648	1	0.001	0.02	0.01
x_1x_4	Interaction	0.121	1	0.378	8.59	3.24
x_1x_5	Interaction	0.714	1	0.828	18.82	7.09
x_2x_3	Interaction	1.18	1	0.635	14.44	5.44
x_2x_4	Interaction	0.099	1	0.154	3.5	1.32
x_2x_5	Interaction	0.29	1	0.64	14.54	5.48
x_3x_4	Interaction	2.402	1	2.097	66.07	24.89
x_3x_5	Interaction	0.576	1	0.08	1.81	0.68
x_4x_5	Interaction	2.329	1	2.329	52.92	19.94
Error		0.352	8	0.352		
Total			24		265.41	100

TABLE V
ANALYSIS OF VARIANCE FOR EFFICIENCY

Design variable		Sum of squares	DOF	Variance	F-ratio	Effective ratio(%)
x_1	Linear	1.02	1	0.724	5.51	6.81
	Quadratic	0.44	1	0.201	1.53	1.89
x_2	Linear	2.38	1	0.567	4.31	5.32
	Quadratic	0.70	1	0.531	4.04	4.99
	Cubic	0.54	1	0.545	4.14	5.11
x_3	Linear	5.36	1	0.661	5.02	6.20
	Quadratic	9.48	1	1.428	10.86	13.40
x_4	Linear	35.24	1	0.083	0.63	0.78
	Quadratic	7.04	1	0.665	5.06	6.25
x_5	Linear	136.25	1	0.127	0.96	1.19
	Quadratic	2.29	1	0.126	0.96	1.19
	Cubic	0.13	1	0.132	1	1.24
x_1x_2	Interaction	0.76	1	0.202	1.54	1.90
x_1x_3	Interaction	16.14	1	3.178	24.17	29.80
x_2x_5	Interaction	0.44	1	0.083	0.63	0.78
x_3x_4	Interaction	0.78	1	0.446	3.39	4.19
x_3x_5	Interaction	1.36	1	0.608	4.62	5.71
x_4x_5	Interaction	0.33	1	0.338	2.57	3.18
Error		0.789	6	0.789		
Total			24		80.94	100

TABLE VI
HISTORY OF OPTIMAL SOLUTION FOR OBJECTIVE FUNCTION AND DESIGN VARABLES

Iteration	x_1 (mm)	x_2 (mm)	x_3 (mm)	x_4 (Deg.)	x_5 (Deg.)	Torque (N·m)	Head (m)	Efficiency (%)
1	7.7	153.0	190.8	116.0	26.4	382.9	19.1	71.5
2	7.2	153.0	206.6	14.5	29.0	359.1	17.7	70.6
3	9.0	117.4	163.5	12.0	29.0	218.0	10.5	72.7
4	7.8	152.9	192.5	15.9	23.9	320.7	16.2	74.8
5	7.8	152.9	192.2	15.9	24.0	324.0	16.3	74.6
6	7.8	153.0	192.3	16.0	24.4	338.4	17.1	74.0
7	7.7	153.0	191.0	15.8	24.8	336.4	16.8	73.6
Opt.	7.8	152.9	192.5	15.9	23.7	316.0	15.9	75.0
Reanalysis						313.7	15.5	71.2

where $p_n(x_i)$ means each level of x and y_i means the average of the experiment data at each level. The use of the orthogonal polynomial is advantageous in the ANOVA. In ANOVA of orthogonal polynomials, each effect is estimates independently. The effects can take a linear, quadratic, or higher order (n-1) functional form. Also, since every term in an orthogonal polynomial is independent of each other, the estimates of the regression coefficient can be calculated successively from low order to higher order. This enables the derivation of an efficient approximation model, since the base can be normalized even when the coefficient of the higher order is not known or the coefficient difference becomes large.

Table II shows the result of CFD analysis and design matrix of L25 orthogonal arrays including that adopted 5 design variable. About each design variable of surrogate model and order choice is considered to ANOVA. Also that use Chebyshev orthogonal polynomial coefficient [10,22]. Chebyshev orthogonal polynomial has a character that lower order term is independent of each other rank. So coefficient is estimated sequentially through lower to higher rank, even if it had undiscovered higher rank coefficient or large gab of coefficient.

Table Ⅲ to Ⅴ show the result of ANOVA of torque, head and efficiency. ANOVA, each design variables on the response sensitivity is evaluated separately with polynomial components. Interaction of design variable can validated quantitatively. For example, design variable affect to torque x_3 (shroud length), x_5 (hub angle) is 31.4%. Also, entire interaction effective is very high as 65.4% and interaction x_3x_4 affect to torque as 25.8%. head as a similar pattern likes torque.

In contrast, efficiency affect to x_2 (hub length), x_3 (shroud length) as 15.4%, 19.6%. An interaction affect to efficiency is smaller than head and torque. Describe the surrogate model Eq. (4)~(6) based ANOVA, considering the interaction and significant degree. Small error but approximation value and CFD analysis are approximate effectively. The way of obtain more accuracy of surrogate model using orthogonal polynomial is defined as R^2_{adj}.

Torque, head and efficiency show the good quality of approximation as 99.9%, 99.6% and 98.6%.

Table VI the summary of optimal solution from surrogate

$$y_{torque}=-1323.2+46.42x_1-3.022x_2-6.399x_3+14.02x_4+98.51x_5 -0.5222x_5^2-0.0125x_1x_2+0.02285x_1x_3+1.197x_1x_4 -2.282x_1x_5+0.01771x_2x_3+0.10735x_2x_4-0.00392x_2x_5 +0.29594x_3x_4-0.02704x_3x_5-1.5716x_4x_5 \quad (4)$$

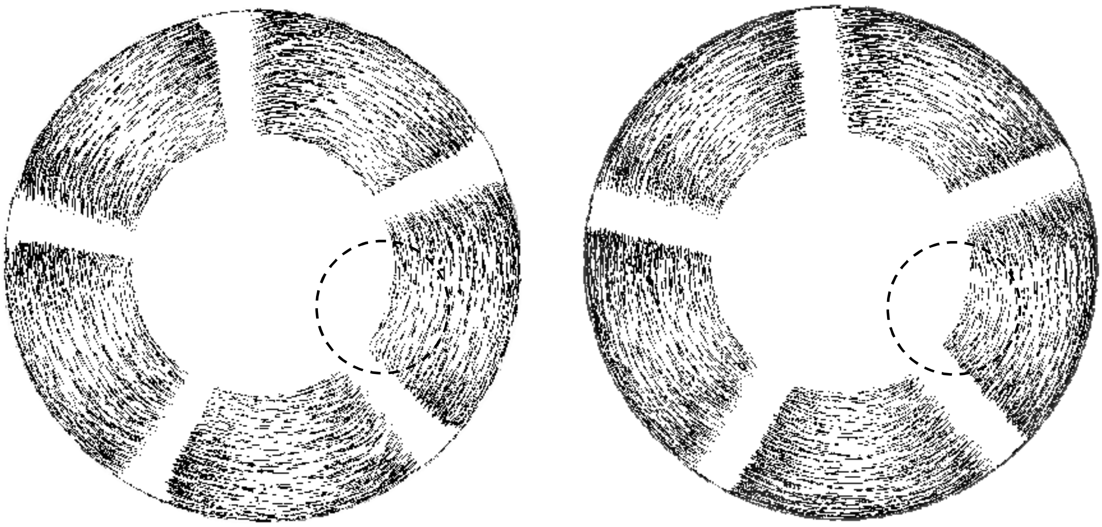
$$y_{head}=-116.9+2.447x_1+0.0065x_2-0.47603x_3-1.464x_4 +9.362x_5-0.05638x_5^2-0.002794x_1x_2-0.000853x_1x_3 +0.26326x_1x_4-0.17851x_1x_5+0.001219x_2x_3 +0.008403x_2x_4-0.007846x_2x_5+0.026073x_3x_4 -0.001976x_3x_5-0.14966x_4x_5 \quad (5)$$

$$y_{efficiency}=-176.3+19.75x_1-0.3814x_1^2+4.756x_2 -0.03432x_2^2+0.5607x_3-0.001138x_3^2+0.000087x_3^2 +2.746x_4-0.13754x_4^2-9.351x_5+0.2735x_5^2 -0.002741x_5^3-0.02704x_1x_2-0.05194x_1x_3 -0.002344x_2x_5+0.011843x_3x_4+0.004131x_3x_5 -0.05768x_4x_5 \quad (6)$$

model by using feasible direction method. Obtain the optimal solution from surrogate model, reanalysis is necessary to validate the accuracy. According to re-analysis result of optimal solution, maximum efficiency is 71.2%, torque is 313.7 N·m, head is 15.5m. optimal solutions using surrogate model are x_1=7.8 mm, x_2=152.9 mm, x_3=192.5 mm, x_4=15.9°, x_5=23.7°. Error of efficiency is 1.7%, head and torque is 2.5%, 0.7%. So this model shows high quality accuracy.

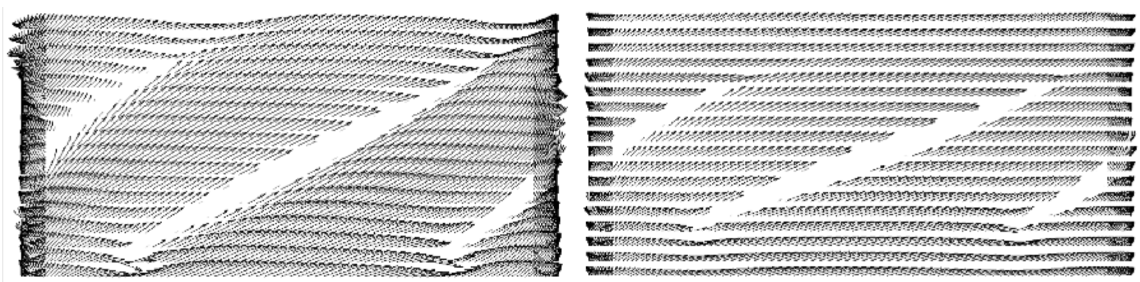
Fig. 5 shows the initial and optimal model shape. Inlet and outlet of the impeller blade angle changes compared to the initial model can be seen that a lot of changes. It also reduces the impeller blade angle of the airfoil, the chord length smaller, a lot of torque that occurs in the entire impeller considered is reduced.

(a) Initial model (b) optimal model
Fig. 5 Initial model vs. optimal model



(a) Initial model (b) Optimal model

Fig. 6 Velocity vectors at the central plane of the impeller



(a) Initial model (b) Optimal model

Fig. 7 Velocity vectors at the horizontal plane of the impeller

It is considered that torque reduced while decrease of total pump head from decrease of total pump head by shape design optimization has cut off the excess design. From the result of optimal design, pump performance curve becomes following: torque reduced as 313.7 N·m of 22%. Also Maximum head improved as 15.5 m of 17%, efficiency has rise 71.5% it is improved 5.6% than initial model.

Total pump head was excess design as 19 m. However optimal model satisfy the 15m of head and improve the efficiency. Rate flow of Initial model shows efficiency decrease but optimal model shows maximum efficiency. Fig. 6 shows the vector distribution of fluid on the horizontal plane. Initial model has fluid collapsing on leading edge and variation of velocity is large. Cylindrical direction vector is stable where optimal model. Also, collapse of leading edge and unstable fluid flow has decreased or disappeared. Fig. 7 shows the vector on horizontal plane. Unstable fluid flow occurred from hub area. However, cylindrical direction vector becomes stable or decreased the unstable fluid flow.

IV. Conclusion

This paper present the results of pump impeller shape optimization for anti-heeling system. For the effective optimal design, we made an approximate model of each response by using orthogonal polynomial in DOE sample points, which used the table of orthogonal arrays. The summary of the main results are as follows:

(1) Unstable flow occurred from hub area of initial impeller model but cylindrical direction vector is stable where optimal model. Also, collapse of leading edge and unstable fluid flow has decreased or disappeared.

(2) Compared to the initial model (torque 402 N·m, head 19 m, efficiency 67.4%), impeller blade's optimal model showed the maximum torque decreased 22% at 313.7 N·m, head improved 17% at 15.5 m, and efficiency improved 5.6% to 71.5%.

(3) The optimal solution of the impeller blade using the approximate model are : x_1=7.8 mm, x_2=152.9 mm, x_3=192.5 mm, x_4=15.9°, x_5=23.7°. In regards to the optimal solution, if it is compared to the CFD reanalysis results, the margin of error were at about 1.7% for efficiency, and 2.5% and 0.7% for maximum head and torque, respectively, and show actual dispositions very well.

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