Experimental and Numerical Evaluation and Design Optimization of Centrifugal Pump Impeller Parameters through Response Surface Methodology

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Abstract

This research discusses a research methodology dutiful to renovate the design approach of a centrifugal pump for performance enhancement through Response Surface Methodology concept and also focused on to enhance the efficiency of a centrifugal pump and to produce energy efficient pump for the requirement of Bureau of Energy Efficiency. This paper exposed to find optimum impeller geometries that affect the efficiency as well as performance characteristics of a centrifugal pump impeller. An impeller blade profile was developed by tangent circular arc method and a 3D model was generated. Computational Fluid Dynamics analysis was done to predict the performance characteristic of the impeller which is an important component of a centrifugal pump. For Numerical simulation, the standard k-ε turbulence model for steady state condition was performed with the unstructured grid.CFD results are validated with experimental performance values. Design of experiments was employed based on five factors and five levels of central composite rotatable design with full replication techniques. Second order mathematical models were formulated to articulate the relationship between the impeller parameters and its responses. The identified input parameters and mathematical models were validated statistically. Optimized impeller parameters were found from RSM optimizer and an impeller was fabricated for the optimized parameters. The experimental evaluation was conducted on the optimized impeller and their results are discussed. The newly designed centrifugal pump has been tested in the Scientific and Industrial Testing and Research Centre located in Coimbatore.

Keywords: Centrifugal pump, CFD, DOE, RSM, Impeller design, Numerical Simulation

1. INTRODUCTION

Today the global market is flooded with plenty of varieties, types and sizes of pumps. Inspire of its age, today the design of pumps has not attained saturation phase. The design of centrifugal pumps involves a lot of parameters that are interdependent and in-turn iterative makes the difficulty at hand more complexity for the human brain to manipulate. Centrifugal pumps are widely used in domestic, irrigation and process industries to handle different types of fluids for a different application. However, the innovation in Computational Fluid Dynamic (CFD) field has made it possible to simulate and visualize some of the features of flow through rotating machines that would be difficult to measure experimentally[1]. A pump consumes thousands of MW electricity and to save the electrical energy is by improving the efficiency of the pump. In this work, a centrifugal pump commonly used for an irrigation purpose was selected from the leading pump manufacturer and its performance characteristics were found through the experimental test. The efficiency of the selected pump at the guaranteed duty point is 65%.

This efficiency value is less than the minimum efficiency required by the pump standard at the guaranteed duty point [2]. Really, pump design compatible with production environment had been a most important challenge to researchers. predominantly, researchers along with engineers had been striving to improve the design of impellers with the specific attention directed to enabling the enhancement of pump performance[1]. For meeting the requirement of the pump standard and to satisfy the customer, the manufacturer needs to improve the efficiency of the pump. Centrifugal pump performance mainly depends on various parameters of the impeller and blade profile. Designing a pump for maximizing the total head and efficiency with minimizing power input at required flow rate is a challenge for the pump designer. However, during the past few years, the design and performance analysis of turbomachinery has experienced great progress due to the joint evolution of computer power and the accuracy of numerical methods and optimization techniques. The 3D inverse design of impeller blade profile, 3D CAD modeling, automatic grid generation and CFD analysis of flow passage reduces the time and manufacturing cost of the pump and produce efficient pump[2]. Numerical simulations of centrifugal pump impeller at steady and unsteady state condition were performed using k - ε, k - ω and SST turbulence models. The comparison of the predicted numerical and experimental performance characteristics shows a good agreement[3]–[5]. Effects of different impeller outlet blade angle and thickness of blade on the pump performance of the mini-centrifugal pump were studied and they were found that the head and efficiency values significantly increase with decreasing blade thickness[6]. Various studies on the effects of increasing number of blades on the impeller[7] and inlet geometry of the impeller were performed[8]. The total head and efficiency of the pump were considerably increased due to adding the number of blades. Numerical and experimental study of centrifugal pump impeller with different blade wrap angle was studied and the results show that the larger impeller wrap angle gives better efficiency and stable pump operations[9]. Among the various turbulence model available in commercial CFD packages, k-ε, k-ω and SST models were commonly used[10]. The effects of changing the total head of pump using these three models were performed and show that only 0.3% total head variation on these models[11].CFD simulation of the centrifugal pump with larger tongue clearance and the back blade was performed. Re-circulation has been occurred due to the larger tongue clearance and it reduces the pump performance[12]. Numerical study at steady state condition of the pump using RNG k-ε turbulence model was performed and found that efficiency of the pump is minimum in off-design condition compared with design condition due to flow separation of the blade pressure surface[13].

Design optimization of centrifugal pump handling multiphase fluid was performed by a different researcher. They combined the numerical study with the design of experiments (DOE) and prove that the effect of blade length and the impeller shroud outlet angle increases the total pressure (30.9kPa) and efficiency (1.9%)[14]. Optimization of mixed flow pump diffuser using radial basis neural network (RBNN) model with 3D Reynolds Average Navier-Stokes (RANS) analysis has described. The results show that the efficiency of the pump is increased by 9.75%. Numerical and experimental analysis of the centrifugal pump for optimizing pump efficiency and the total head was performed[15]. Three surrogate models Viz., Response Surface Approximation (RSA), Kriging (KRG) and RBNN with steady-state RANS turbulence model was used. They found RSA model gives the minimum prediction error compared with the RBNN model but RBNN model gives maximum efficiency. Optimization of centrifugal pump impeller was performed using DOE, RSM and NSGA II optimization techniques and the pump efficiency has improved by 6.1%[16]. The 2^k factorial design with RSM was used to improve the meridional geometry of the centrifugal pump impeller[17] and centrifugal fan diffuser [18].Some researcher coupled the orthogonal test and CFD simulation to improve the performance of the multistage pump [19]. This work reports a research project to build up a design improvement methodology pertaining to pump impellers using parameter design approach as no one applies the Off-line quality technique in pump field till date. In this research work, the optimal design of a centrifugal pump impeller to satisfy the design specifications was investigated. During the first phase of work, the baseline model of the impeller was designed using turbo machinery theory and generates a 3D CAD model for the required guaranteed duty point performance condition. CFD analysis of the baseline impeller was performed and it is used to study the effects of geometrical features of the impeller on the performance characteristic curves. The results of the CFD analysis are validated through experimental analysis on a fabricated baseline impeller. To improve the performance of the baseline impeller, design optimization was performed. Design of experiments was used to explore the effect of different combinations of geometrical features of the pump performance[20], [21]. The identified process parameters are Outer diameter (X_1), Inlet blade angle (X_2), Outlet blade angle (X_3), Number of blades (X_4) and blade thickness (X_5). Further using the second order mathematical model, the responses were analyzed. Optimization of impeller parameters was carried out by using response surface methodology (RSM). The selected responses governing the centrifugal pump are the total head (H), power input (P) and pump efficiency (η). Finally, an impeller was fabricated for the optimized parameters and its performance characteristics were verified through experimental study. This experimental study provides a new direction to the pump designer especially for new design and new relevance without immediately going for a physical model. This research attempts the pump performance enhancement has done with the objective of winning customer confidence and thereby serving the pump manufacturers to hold the customer in the competitive global markets.

2. PROBLEM FORMULATION

The pump industries in India has made tremendous progress; nevertheless, the fluid flow field that uses various hydraulic pumps continues to undergo from a major drawback of not utilizing the pumps at their full potential one. A foremost cause leading to such a situation is not to run the pumps at their optimum operating parameters. Regrettably, the manufacturing engineers in India continue to decide the operating conditions solely on the basis of handbook values of manufacturer's suggestions or employee experiences. The objective of this research study is to obtain (customized) optimum dimensions of impeller (outer diameter, inlet and outlet blade angles, number of blades and blade thickness) for performance enhancement of the pump that is to enhances total head at the desired duty point level with minimum power consumption and also the pump efficiency. In this research maximizing the total head, efficiency and minimizing power consumption are the objective functions.

3. SELECTION OF THE BASELINE IMPELLER

3.1. Test Pump with the Impeller

Last few decades several researchers have taken more effort to increase centrifugal water pump for performance enhancement through the study of different configurations of several hypothetical flow analysis method. By studying existing literature review and genera! insights obtained by the in-depth study of pump impeller for enhancing pump performance like head, efficiency, power and discharge the most influencing design parameters are impeller eye diameter, impeller width, vane outlet angle and number of vanes. Such process liquid enters the suction nozzle and then into the eye of a revolving device called an impeller. While the impeller rotates, it rotates the liquid sitting in the cavities between the centrifugal acceleration and vanes outward. As the liquid leaves, the eye of the impeller a low-pressure area is generated causing more liquid to flow towards the inlet. Since the impeller blades are curved the fluid is pushed in radial and tangential directions by centrifugal force. Hence this force acting inside the pump is the same one that remains water inside a bucket that is spinning at the end of a cord.

3.2 Experimental setup and data collection of Baseline Impeller

The problem was illustrated with a real case study. The centrifugal pump impeller is designed for the operating head range and guaranteed duty point condition. The Pump standard specifies the operating head range and minimum efficiency of the pump corresponding to the guaranteed duty point[1]. The Guaranteed duty point is the optimum condition of the pump at which the pump gives maximum efficiency corresponding to the discharge and total head. Pump designer or the manufacturer declares the efficiency of the pump for the duty point discharge and total head. In this study the pump impeller was designed for the duty point discharge (Q) =15 lps, total head (H) = 32 m, speed of rotation (N) = 2900 rpm, the minimum pump efficiency (η) = 66% and pump suction and delivery sizes are 75 mm and 65 mm respectively. The Impeller has many numbers of geometrical dimensions and these dimensions are calculated using turbo machinery equations [22] and are shown in table 1.

Table 1 Dimensions of the Baseline Impeller

Impeller Parameters	Values	Unit
N_S	96.35	rpm
D_n	77.82	mm
D_1	65	mm
B_1	18	mm
B_2	11.2	mm
X_1	178	mm
X_2	18	^{o}
X_3	16	^{o}
X_4	6	-
X_5	4	mm

3.3 3D Modeling of Baseline Impeller

The 3D modeling of the pump impeller was created as per dimensions are shown in table 1 using a standard CAD modeling package. The geometry of the blade profile depends on outer and inner diameters of the impeller and corresponding blade angles. The blade profile is constructed using tangent circular arc method[23],[24]. Figure 1and 2 shows the geometrical configuration of blade profile and 3D model of the baseline impeller.

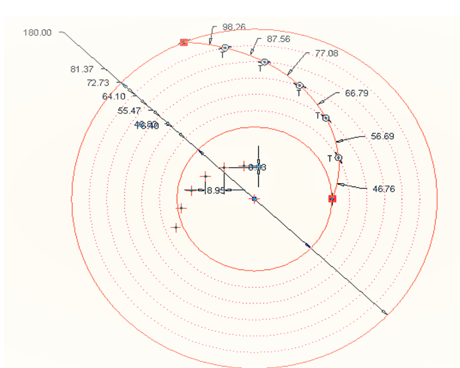

Figure.1. Blade Profile of the Baseline Impeller

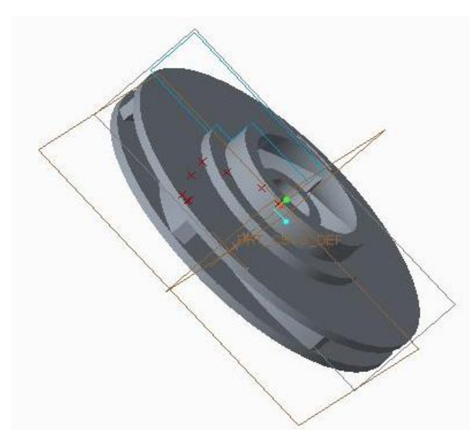

Figure.2. 3D Model of the Baseline Impeller

3.4 Summary of the Experimental Investigation

In many experimental situations in perform, more than one response will be calculated for the different combination of values for which a set of design variables may be counted. Where there are several responses and optimum condition for one response is not very possible the same as an optimum condition for the other response. Therefore it requires discussing how to find the overall optimum values for the several responses when employing based on parameter designs. In this study, responses total head, power and efficiency are counted and indentified that each response has different optimal levels and significant factors.

3.5. Performance Study of the Baseline Impeller

3.5.1Numerical Study of the baseline Impeller

Numerical studies of the impeller with volute casing were performed by using ANSYS code that solves the basic governing equations. The fluid domains of the baseline impeller with volute casing were modeled. Various turbulence models available on commercial CFD code were studied[10]. The standard k-ε, k-ω and Shear Stress Transport (SST) turbulence model are used to simulate the fluid domain. Among these three models, the standard k-ε model offers a good relationship between numerical effort and computational accuracy when compared to the other two models. Hence the standard k-ε model was selected for further analysis. Unstructured grids with the tetrahedral element are generated for the computation domain of the impeller and volute casing and it is shown in Figure 3. The results of the numerical study were validated with work performed by K.W.Cheah et al [25],[26].Table 2 present the boundary conditions for numerical simulation.

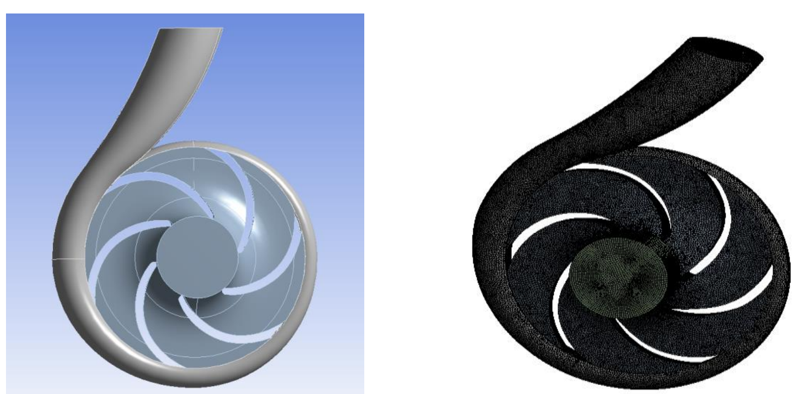

Figure. 3. Computational Domain of the Impeller

Table 2 Boundary Conditions for Numerical Simulation

Analysis type	Steady State Incompressible
Fluid	Clear cold Water
CFD Model	$k-\varepsilon$ Turbulence Model
Turbulence Intensity	5% (medium intensity)
Inlet	Total Pressure(100000 Pa)
Outlet	Mass flow rate
Domain	Periodic Boundary
Rotational Speed	2900 rpm

3.5.2 Grid Dependency Test

The number of grids and selection of suitable turbulence model plays a vital role in having close matching with experimental data. An investigation is required before carrying out the rest of the simulations. Grids dependency test of the impeller and volute casing was performed for pump total head at a different number of grids ranging from 0.151 to 5.67 million for the duty point discharge 15 lps and are shown in figure 4. From the figure, it was observed that the total head values are increases when the number grids increase up to the grids values of 1.69 million and a further increase of the grids, there is no significant improvement on pump total head values. Hence the grid value of 1.69 million was used for the entire simulation work. The selected convergence criteria were a maximum residual of 10^{-4} and mass imbalance of less than 10^{-2}[3],[27]. The performance values of the baseline impeller were confirmed by numerical analysis and it was compared with the experimental results.

Table 3 Grid Information

Gird Size	No of Gird (x 10^5)	Total Head, m
3.5	1.51	18.81
3.0	2.21	24.57
2.5	3.81	27.12
2.0	7.25	28.41
1.5	16.95	33.11
1.0	56.70	33.13

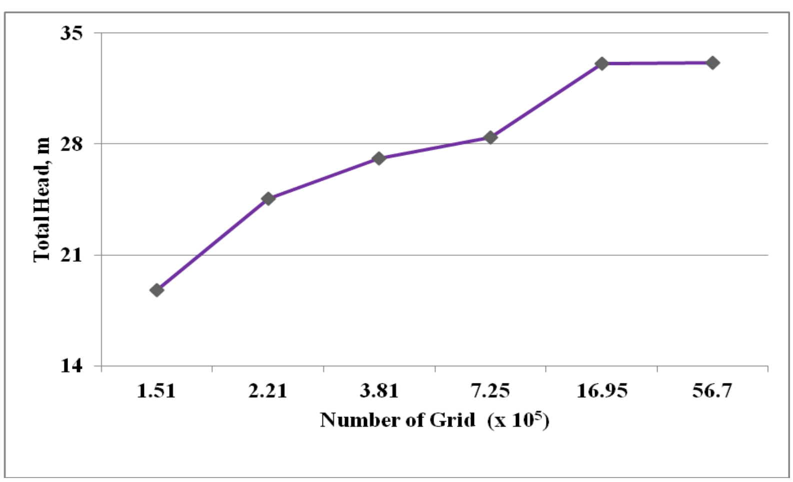

Figure 4 Grid Dependency Test

3.5.3 Numerical Simulation of the Baseline Impeller

The numerical analysis of the baseline impeller with volute casing was performed using k-ε, k-ω and SST turbulence models. Figure 5 shows the pressure (Pa) and velocity (m/s) distribution of baseline impeller with the volute casing.

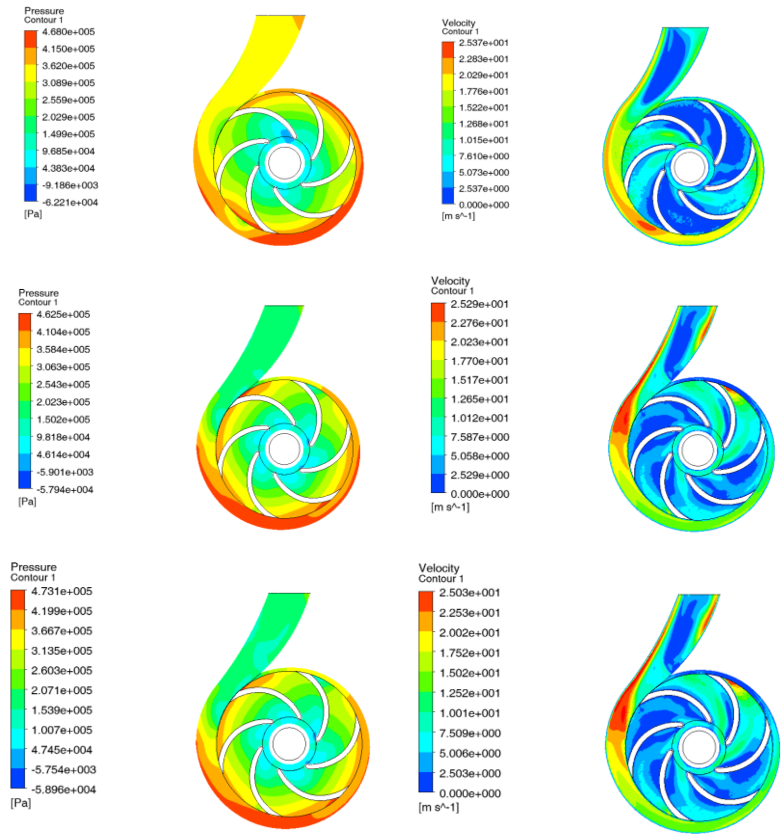

Figure 5 Pressure (Pa) and Velocity (m/s) distribution of a k – ε, k – ω and SST Model

3.5.4 Experimental Study of the Baseline Impeller

The baseline impeller shown in figure 6 was manufactured with grey cast iron material for the dimensions given in table 1 and tested in accordance with ISO 9906 standard[28]. The experimental test setup for testing of centrifugal pump set is shown in figure 7. The testing setup consists of centrifugal pump with electric motor, suction and delivery pipes with foot valve, sump, liquid collecting tank, pressure transmitter at inlet for measuring suction pressure, pressure transmitter at outlet for measuring delivery pressure of the pump, flow control valves at inlet and outlet to control flow of fluid, electromagnetic flow meter, three phase power meter, control panel and power supply., pressure and vacuum transmitter of 0.2 class, electromagnetic flow meter of 0.5 class and power meter of 0.2 class is used for measuring pressure, flow rate and input power. All measuring instruments readings are recorded simultaneously at regular intervals from fully open to shutoff condition of the delivery control valve. Power input to the pump is measured in kW and the output power has calculated the product of total head and discharge through the pump. Performance characteristics curves of the pump are shown in figure 8.

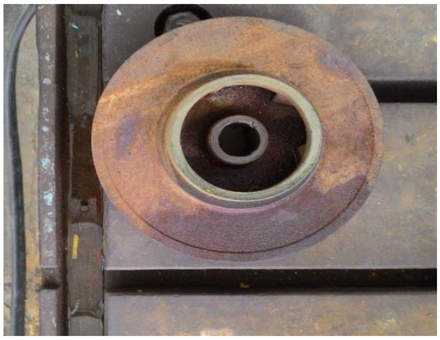

Figure 6 Baseline Impeller

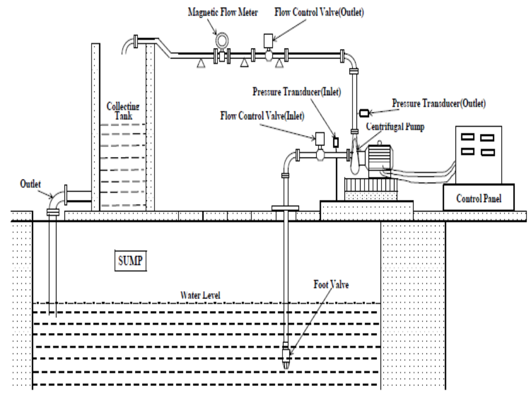

Figure 7 Experimental test Setup of the Centrifugal Pump as per ISO 9906

3.5.5 Comparative Study of Experimental and Numerical Performance Characteristics

The comparative study of experimental and numerical performance characteristics was done for the baseline impeller and is shown in Figure 8. Numerical results are matching closely with experimental results. At the duty point, the numerical performance values are the discharge(Q) = 16.4lps, total head(H) = 35.23 m, pump efficiency(η) = 67.1% and power input(P) = 10.62 kW and the corresponding experimental performance values are 16 lps, 34.1 m, 64.2% and 10.3 kW respectively. Table 3 presents the CFD simulation and experimental results of the baseline impeller. The experimental results confirm the increase of power input and pump efficiency when the discharge and total head increase and satisfied the requirements of the pump standard[1]. However, the performance characteristic of the baseline impeller needs to improve for maximizing the efficiency of the requirements of BEE. Hence the dimensions of the baseline impeller required to modify.

Table 3 Experimental and CFD Simulation Results of the Baseline Impeller

Q, lps	Experimental Result			CFD Simulation			The deviation between Experimental and Numerical Results, %		
	H, m	P, kW	η ,%	H, m	P, kW	η, %	H	P	η
22	10.69	9.2	31.31	11.19	9.39	32.24	4.47	2.02	2.88
21.95	15.89	10.02	42.98	16.63	10.22	43.74	4.45	1.96	1.74
21.5	20.59	10.73	50.55	21.56	10.95	51.92	4.50	2.01	2.64
21.2	25.22	11.13	58.86	26.44	11.36	60.49	4.61	2.02	2.69
19.75	29.45	11.11	64.13	30.93	11.34	65.95	4.78	2.03	2.76
16.5	33.75	10.41	65.57	35.23	10.62	67.11	4.20	1.98	2.28
13.65	35.88	9.72	61.74	37.53	9.92	63.23	4.40	2.02	2.36
9.85	38	8.64	53.06	39.82	8.82	54.39	4.57	2.04	2.45
6.5	39.15	7.65	40.74	40.95	7.81	41.76	4.40	2.05	2.44
0	-	-	-	41.25	5.72	0	-	-	-

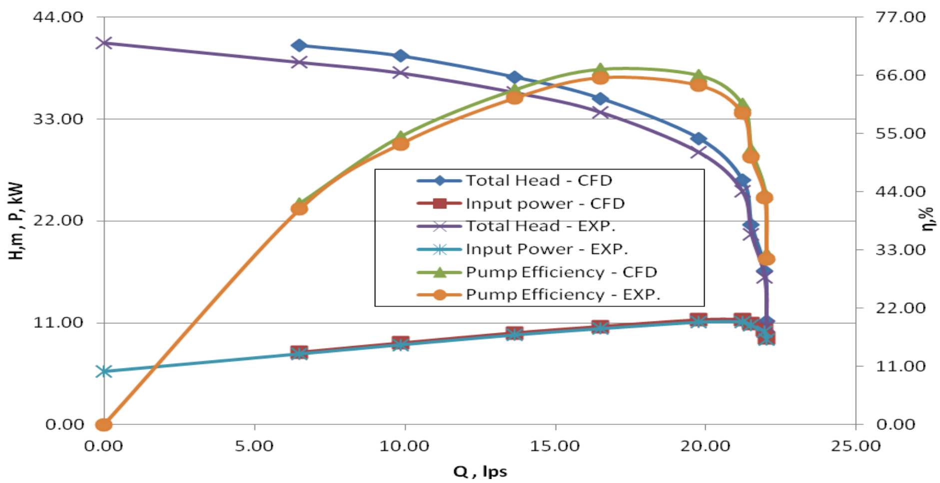

Figure 8 CFD and Experimental Performance curves of a Baseline Impeller

4. Optimization of the Pump Impeller

In order to maximize the efficiency of the impeller, an optimization process was combined with response surface methodology and numerical analysis was performed. From the RSM, an optimum set of impeller parameter was found. Finally and impeller was fabricated for the optimum parameters and an experiment was conducted. The general optimization process is shown in the flowchart in figure 9.

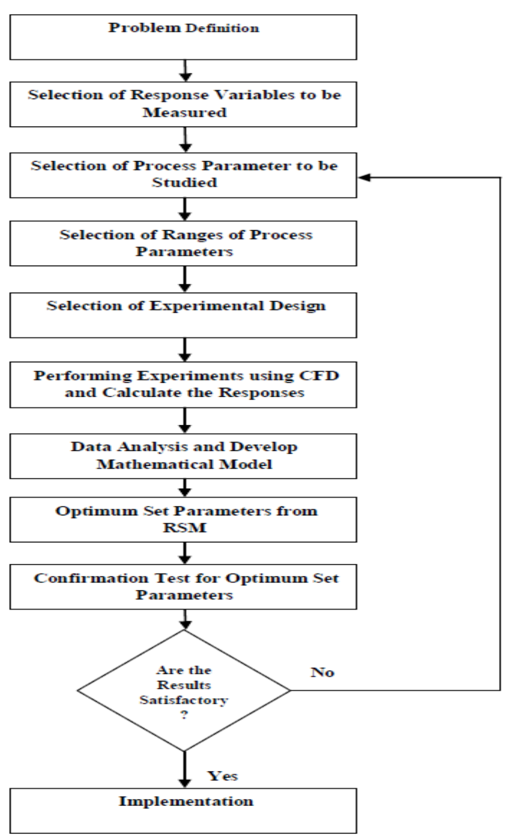

Figure 9 Optimization Process Flowchart

4.1 Results and discussion of the experimental investigations and its mathematical model

Response surface methodology was utilized to develop a second-order mathematical model to relate impeller geometry with performance characteristics. For optimum design of the centrifugal pump, the maximum and minimum values of impeller parameters were chosen based on the data used by pump designer. The maximum and minimum limits were coded as +2 and –2 respectively and in-between levels were calculated using the equation (1). Table 4 lists the details of impeller parameter levels and coding. The intermediate coded values are calculated as

$$X_i = \frac{2(2X - (X_{max} + X_{min}))}{(X_{max} - X_{min})} \qquad (1)$$

Where X_i is the coded value of a process variable *(X)* between X_{max} and X_{min}

Table 4 Impeller Parameters and their Levels

Sl No	Impeller Design Parameters	Units	Levels				
			(-2)	(-1)	(0)	(+1)	(+2)
1.	Outer Diameter	Mm	176	178	180	182	184
2.	Blade Angle at Inlet	º	18	20	22	24	26
3.	Blade Angle at Outlet	º	22	24	26	28	30
4.	No of Blades	--	4	5	6	7	8
5.	Blade Thickness	Mm	2	3	4	5	6

In this investigation, experimental runs were formulated using five factors each with five levels of central composite rotatable design (CCD). The experimental runs consist of 16 (2^4) factorial points with ten-star points and six center points.[29].Table 5 presents the process parameter combinations for each experimental run along with recorded responses from CFD simulation method. All the experiment sets are performed for performance evaluation of the pump using numerical simulation. The relationship between the responses and Impeller design parameters were expressed in the form of a mathematical function given by the equation (2).

$$Y = f(X_1, X_2, X_3, X_4, X_5) \qquad (2)$$

Where

Y – *R*esponse Variable

X_1 - Outer Diameter (coded)

X_2 - Blade Angle at Inlet (coded)

X_3 - Blade Angle at Outlet (coded)

X_4 - No of Blades (coded)

X_5 - Blade Thickness (coded)

The statistical software Minitab is used to calculate the coefficients of linear, quadratic and interactive terms. The second order mathematical model with all the coefficients is given by equation (3).

$$\begin{aligned} Y &= \beta_0 + \beta_1 X_1 + \beta_2 X_2 + \beta_3 X_3 + \beta_4 X_4 + \beta_5 X_5 \\ &+ \beta_{11} X_1^2 + \beta_{22} X_2^2 + \beta_{33} X_3^2 + \beta_{44} X_4^2 + \beta_{55} X_5^2 \\ &+ \beta_{12} X_1 X_2 + \beta_{13} X_1 X_3 + \beta_{14} X_1 X_4 + \beta_{15} X_1 X_5 \\ &+ \beta_{23} X_2 X_3 + \beta_{24} X_2 X_4 + \beta_{25} X_2 X_5 + \beta_{34} X_3 X_4 \\ &+ \beta_{35} X_3 X_5 + \beta_{45} X_4 X_5 \end{aligned} \qquad (3)$$

Where Y is a response parameter, X_i is the independent variables, β refers to the regression coefficients. The full model equation for the responses total head, power input and pump efficiency is given by the equations (4), (5) and (6) respectively.

$$\begin{aligned} H &= 31.6926 + 0.8363 X_1 - 0.2538 X_2 + \\ &0.0813 X_3 + 0.5287 X_4 - 0.1404 X_5 + \\ &0.2024 X_1^2 + 0.0499 X_2^2 + 0.0111 X_3^2 + \\ &0.2486 X_4^2 + 0.0624 X_5^2 + 0.0131 X_1 X_2 \\ &-0.2631 X_1 X_3 - 0.0844 X_1 X_4 \\ &-0.01019 X_1 X_5 + 0.0994 X_2 X_3 + \\ &0.1581 X_2 X_4 - 0.4369 X_2 X_5 - \\ &0.1906 X_3 X_4 + 0.1969 X_3 X_5 \\ &+0.0831 X_4 X_5 \end{aligned} \qquad (4)$$

Table 5 Impeller Design Parameters (coded) with Numerical results of Responses

Exp. No	Impeller Design Parameters (Coded)					Responses(CFD Simulation)		
	X_1	X_2	X_3	X_4	X_5	H, m	P, kW	η ,%
1	1	1	-1	1	-1	34.53	9.95	63.79
2	-1	1	1	1	-1	32.40	9.48	62.83
3	-1	-1	1	1	1	32.90	9.19	65.81
4	0	0	0	0	0	31.70	9.14	64.18
5	-2	0	0	0	0	30.78	9.09	62.25
6	0	0	0	-2	0	31.62	8.78	66.20
7	1	-1	1	-1	1	33.27	9.49	64.44
8	0	0	0	0	2	31.65	8.92	65.22
9	-1	-1	-1	1	-1	31.83	9.34	62.65
10	1	1	1	1	1	32.79	9.48	63.58
11	0	-2	0	0	0	32.39	9.39	63.41
12	2	0	0	0	0	34.20	9.98	61.15
13	-1	-1	1	-1	-1	31.18	8.92	64.26
14	1	-1	-1	-1	-1	33.23	9.95	60.84
15	0	0	0	0	0	31.70	9.14	64.18
16	0	0	0	0	0	31.70	9.14	64.18
17	1	-1	-1	1	1	34.18	9.43	66.63
18	0	0	0	2	0	33.73	9.03	68.66
19	0	0	2	0	0	31.89	9.02	64.99
20	0	0	0	0	0	31.70	9.14	64.18
21	0	0	0	0	-2	32.21	9.38	63.12
22	-1	1	-1	1	1	31.11	8.34	68.57
23	1	-1	1	1	-1	32.68	9.26	64.87
24	0	0	-2	0	0	31.56	9.15	63.40
25	0	0	0	0	0	31.70	9.14	64.18
26	0	2	0	0	0	31.37	9.12	63.79
27	-1	-1	-1	-1	1	30.94	8.99	63.26
28	1	1	1	-1	-1	32.93	9.18	65.94
29	-1	1	1	-1	1	30.68	8.69	64.90
30	-1	1	-1	-1	-1	30.53	8.71	64.43
31	1	1	-1	-1	1	31.19	9.45	60.67
32	0	0	0	0	0	31.70	9.14	64.18

$$
\begin{aligned}
P &= 9.13216 + 0.26292X_1 \\
&- 0.07625X_2 - 0.03042X_3 + 0.06625X_4 \\
&- 0.11042X_5 + 0.10659X_1^2 + 0.03659X_2^2 \\
&- 0.00591X_3^2 - 0.05091X_4^2 + 0.01034X_5^2 \\
&+ 0.07187X_1X_2 - 0.14187X_1X_3 - 0.06187X_1X_4 \\
&+ 0.04688X_1X_5 + 0.07687X_2X_3 + 0.08438X_2X_4 \\
&- 0.06188X_2X_5 + 0.07313X_3X_4 + 0.10938X_3X_5 \\
&- 0.09062X_4X_5
\end{aligned}
\tag{5}
$$

$$
\begin{aligned}
\eta &= 64.1822 - 0.3396X_1 + 0.1129X_2 \\
&+ 0.3738X_3 + 0.6212X_4 + 0.5187X_5 \\
&- 0.6222X_1^2 - 0.1472X_2^2 + 0.0016X_3^2 \\
&+ 0.8103X_4^2 - 0.0047X_5^2 - 0.4719X_1X_2 \\
&+ 0.5006X_1X_3 + 0.2481X_1X_4 - 0.5306X_1X_5 \\
&- 0.3881X_2X_3 - 0.2706X_2X_4 - 0.4244X_2X_5 \\
&- 0.9306X_3X_4 - 0.4119X_3X_5 + 0.7906X_4X_5
\end{aligned}
\tag{6}
$$

4.2 Analysis of Experimental Results

Table 6 presents the estimated regression coefficients and p- values for response parameters. Also, present the R^2-value and adj.R^2- value for all responses. The linear effects squared effects and interaction effects between parameters of p-values less than 0.05(α = 0.05) are a significant contribution to the model. p-values more than 0.05, the impeller parameters have no significant contribution to the model. The R^2-values of full model and adj.R^2-value of the reduced model has more than 95%. Hence the models predict the responses to the observations with reasonable accuracy.

Model Adequacy test was conducted for responses based on the F-ratio and R-ratio. The calculated F-ratio of the responses is less than the critical value at 95% confidence level and the calculated R-ratio of the responses is more than the critical value at 95% confidence level, then the model is adequate within the specified confidence level [30]. Calculated values of F-ratio and R-ratio for all responses are meeting the above criterion. Details of the model adequacy test are present in table 7.

Table 6 Estimated Regression Coefficients and p-values of responses

Variables Terms	H		P		η	
	Coefficients	p – values	Coefficients	p – values	Coefficients	p – values
constant	31.692	0.000	9.1321	0.000	64.182	0.000
X_1	0.8363	0.000	0.2629	0.000	-0.3396	0.000
X_2	-0.2538	0.000	-0.0762	0.000	0.1129	0.000
X_3	0.0813	0.000	-0.0304	0.015	0.3738	0.000
X_4	0.5287	0.000	0.0662	0.000	0.6212	0.000
X_5	-0.1404	0.000	-0.1104	0.000	0.5187	0.000
X_1*X_1	0.2024	0.000	0.1065	0.000	-0.6222	0.000
X_2*X_2	0.0499	0.000	0.0365	0.003	-0.1472	0.000
X_3*X_3	0.0111	0.033	-0.0059	0.547	0.0016	0.911
X_4*X_4	0.2486	0.000	-0.0509	0.000	0.8103	0.000
X_5*X_5	0.0624	0.000	0.0103	0.300	-0.0047	0.744
X_1*X_2	0.0131	0.058	0.0718	0.000	-0.4719	0.000
X_1*X_3	-0.2631	0.000	-0.1418	0.000	0.5006	0.000

X_1*X_4	-0.0844	0.000	-0.06187	0.001	0.2481	0.000
X_1*X_5	-0.1019	0.000	0.04688	0.004	-0.5306	0.000
X_2*X_3	0.0997	0.000	0.07687	0.000	-0.3881	0.000
X_2*X_4	0.1581	0.000	0.08438	0.000	-0.2706	0.000
X_2*X_5	-0.4369	0.000	-0.06188	0.001	-0.4244	0.000
X_3*X_4	-0.1906	0.000	0.07313	0.000	-0.9306	0.000
X_3*X_5	0.1969	0.000	0.10938	0.000	-0.4119	0.000
X_4*X_5	0.0831	0.000	-0.09062	0.000	0.7906	0.000
R^2	99.98%	-	99.25%	-	99.94%	-
R^2(adj)	99.95%	-	97.90%	-	99.83%	-

Table 7 Analysis of variance for Testing Adequacy of Models

Responses	First order terms		Second order terms		Lack of fit		Error term		F – ratio*	R – ratio**	Adequacy of the model
	SS	DOF	SS	DOF	SS	DOF	SS	DOF			
H	25.67	5	2.83	15	0.01	6	0.01	5	0.83	71.25	Adequate
P	2.21	5	0.47	15	0.03	6	0.03	5	0.83	22.33	Adequate
η	22.14	5	34.09	15	0.06	6	0.06	5	0.83	23.42	Adequate

Note: SS – Sum of Squares
DOF – Degree of Freedom
F-ratio = MS of lack of fit / MS of error term
R-ratio = (MS of first-order +MS of second order term) / MS of error terms
**Critical value of R-ratio (20, 5, 0.05) = 4.56
*Critical value of F-ratio F (6, 5, 0.05) = 4.95

4.3 Modeling and Optimization of Impeller

Parameters

The main objective of any pump impeller design is to maximize the total head and pump efficiency and minimize the power consumption. These responses were interrelated with each other and best combinations of geometrical values were determined using the response surface methodology. Response surface methodology is used to identify the input parameters combinations to optimize the response variables. Total head and pump efficiency limits are selected between 30.53 m to 34.53 m and 60.67% to 68.66% whereas power input limits are selected between 8.34 kW to 9.98 kW for setting optimum point. The optimum setting of impeller parameters for the responses is shown in figure 10. From the RSM analysis, the optimum set parameters of the impeller are outer diameter (X_1) = 180 mm, inlet blade angle (X_2) = 18^o, outlet blade angle (X_3) = 22^o, number of blades (X_4) = 8 and blade thickness (X_5) = 6 mm. And corresponding performance values are total head (H) =36.12 m, input power (P) = 8.34 kW and pump efficiency (η) = 77.89%.

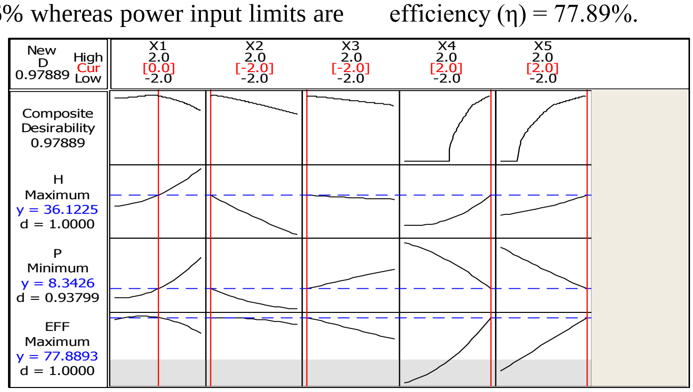

Figure10 RSM Optimization Plot

4.3.1 Experimental Study of the Optimum Impeller

The impeller for the optimized parameters from RSM was fabricated. Performance test on the impeller was conducted as per ISO 9906 standard. The experimental performance characteristic curves are shown in figure 11. The pump performance values are presented in table 8. From the performance curves, the actual pump total head (H) = 37.02 m, input power (P) = 10.14 kW and pump efficiency (η) = 77.14% were obtained corresponding to the guaranteed duty point.

Table 8 Experimental Results of the Optimized Impeller

Q, lps	H, m	P, kW	η, %
21.85	12.89	9.43	36.60
21.80	22.17	10.75	55.09
21.73	27.48	11.47	63.83
20.41	31.73	11.12	71.37
17.24	37.02	10.14	77.14
15.23	38.35	9.71	73.72
11.41	39.70	8.76	63.37
5.11	40.12	6.38	39.38
0.00	41.53	4.72	0.00

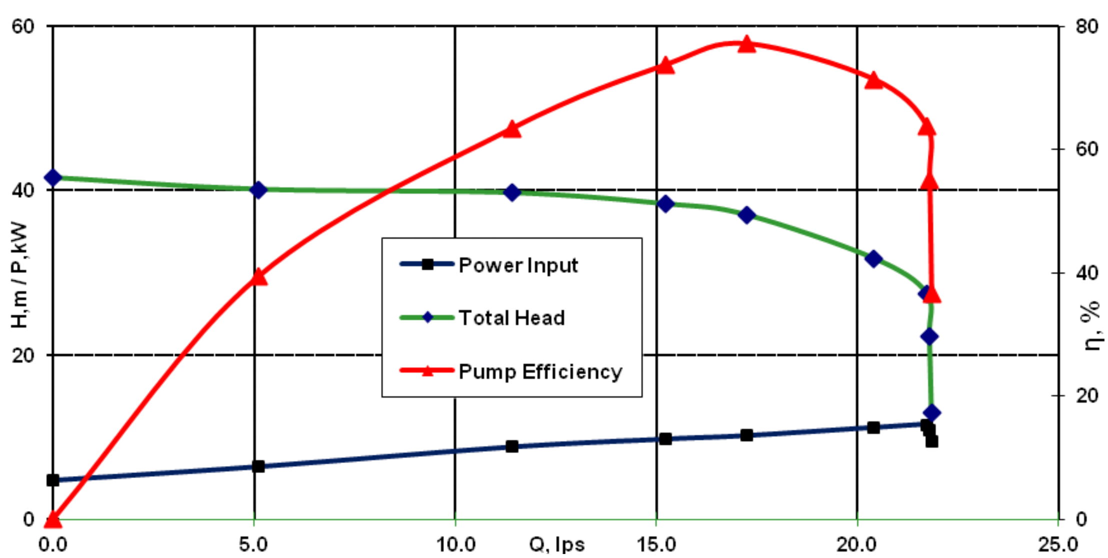

Figure11 Experimental Performance curves of the Optimized Impeller

4.4 Comparison of RSM Predicted Performance and Experimental Performance of the Baseline and Optimized Impeller

Table 9 presents the RSM predicted performance values and experimental performance of the baseline and optimized impeller. The results show that the total head and pump efficiency of

the optimized impeller compared with baseline impeller were increased by 2.92 m and 12.94 % and the power input has decreased by 0.16 kW. While comparing with RSM predicted performance results, total head increased by 0.9 m and power input and efficiency values are decreased by 1.8 kW and 0.75 % respectively. This shows that the RSM is a good optimization tool for improving the pump performance of the centrifugal pump.

Table 9 Comparison of RSM Predicted Performance and Experimental Performance of the Baseline and Optimized Impeller

Response Parameters	RSM Predicted Performance	Experimental Performance	
		Baseline Impeller	Optimized Impeller
H, m	36.12	34.1	37.02
P, kW	8.34	10.3	10.14
η, %	77.89	64.2	77.14

5. CONCLUSION

The results of this experimental analysis have been compared with validated results of the pump test conducted in the research laboratory and identified to be in good agreement. The centrifugal pump impeller was developed by tangent circular arc method and it was studied numerically by using CFD simulation and results are validated experimentally. This is the exclusive benefit of CFD simulation wherein one can visualize the performance of the pump through the design stage itself without even building the physical prototype methods. The numerical results show good agreement with the experimental results. Design of experiments was used to formulate the various combinations of impeller .parameters and develop the second order mathematical model for the total head, power input and pumps efficiency. A mathematical model for the response parameters is statistically validated. Finally, design optimization of the impeller was performed by using response surface methodology. These test runs are performed to identify the optimized parameters which result in saving a huge amount of man-hours and cost.

Following conclusions has been drawn from this work.

(1) Numerical and experimental results of the baseline impeller are closely matched with each other within the pump operating head range. The deviation between CFD and experimental performances are within the range of 4.0 to 5.3%. And also microanalysis of flow properties on any point of the interest in the hydraulic passages of the pump could be visualized only by CFD analysis.

(2) The effects of various impeller geometrical parameters which are greatly influenced by the pump performance characteristics are an outer diameter, number of blades and blade thickness. The contribution of outer diameter on pump performance has more influenced compared to the inlet blade angle and outlet blade angles.

(3) Comparing the performance values of the developed baseline impeller and optimized impeller, the pump performance characteristics total head, and pump efficiency at the guaranteed duty point are increased by 8.56 % and 16.77 % and the power input is decreased by 1.56% respectively.

4. The efficiency requirements of the BEE for the selected model pump has been achieved through this research work and this help to the manufacturer to produce energy efficient pump.

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