Desalination and Water Treatment
www.deswater.com
doi:10.5004/dwt.2018.23055

132 (2018) 79–88
November

Study the effects of surface coating on impeller and volute casing to improve the efficiency of the water pump

K. Ulaganathan[a,*], P. Karuppuswamy[b]

[a]*Department of Mechanical Engineering, Kumaraguru College of Technology, Coimbatore – 641049, India, email: ulaganathan.k.mec@kct.ac.in (K. Ulaganathan)*
[b]*Department of Mechanical Engineering, Sri Ramakrishna Engineering College, Coimbatore – 641020, India, email: pkaruppuswamy_cbe@rediffmail.com (P. Karuppuswamy)*

Received 5 January 2018; Accepted 1 September 2018

ABSTRACT

The focus on the improvement of the efficiency of the centrifugal pump handling water has become a significant research area in the global arena. The prime objective of the research work is aimed to bring in the detailed effects of surface coating on the impeller and volute casing to improve the efficiency of the centrifugal pump handling water. Initially, the performance characteristics of a six model centrifugal pump of same design specification are measured. This six model centrifugal pump of same design specification is assembled with cast iron impeller and volute casing of surface roughness value 4.39 μm R_a. After that, three sets of impeller and casings are coated with the epoxy material, and another three sets of impeller and casing are coated with polyurethane material of different coating thicknesses. The experimental analysis measures the performance characteristics of these six pumps with surface coating conditions. The experimental performance values are shown with a significant improvement in the pump efficiency up to 4.75%, and the input power is decreased up to 0.75 kW. The computational fluid dynamics tool in Ansys is used to simulate the pump with the uncoated condition and surface coating conditions of different surface roughness values. For numerical simulations, the k-ω SST turbulence model with an unstructured tetrahedral mesh of impeller and volute casing is used. The pressure and velocity distribution of numerical simulations are analyzed, and the experimental results are validated with the numerical model. Cost analysis was carried out using the payback period, and the net present value methods and its values show the effect of coating materials on the pumps in cost saving.

Keywords: Centrifugal pump; Impeller; Volute casing; Surface roughness; Coating thickness; Experimental study; CFD

1. Introduction

The primary function of centrifugal pumps is to handle fluid from lower level to higher level. The purpose and the use of these pumps are more extensive, which includes domestic, agricultural, surface water treatment and its distribution, wastewater treatment plant, oil and chemical industries. These pumps consume around 60–75% of the electric energy which is produced around the globe. Owing to the increase in the cost of electricity and the mandatory usage of the pumps, improving the efficiency of the pump has become inevitable and the need for the hour for the manufacturers of these pumps. Using numerical methods and optimization techniques a number of researches were carried out in the past to improve the performance of the pump. Three-dimensional inverse design coupled with computational fluid dynamics (CFD) is performed for designing and optimization of a low specific speed (0.109 nondimensional) centrifugal pump diffuser [1]. Using the same, a significant improvement in the performance of the pump is achieved. Internal flow analysis of different centrifugal volute casings is numerically examined [2]. The impeller has discretized with structured mesh,

*Corresponding author.

and the volute has discretized with unstructured tetrahedral mesh. The k-ε turbulence model is employed for this study. The geometry of the volute casing and type of fluid through the pump has a reasonable effect on the pressure, velocity and shear stress of the fluid. Experimental and numerical analysis of a centrifugal pump impeller using the k-ε turbulence model is performed [3]. It is found that the differences between the numerical and experimental results are between 3.5 and 4.9%. Study on the effects of impeller blade geometry with different outlet blade angle on the performance of the centrifugal pump is performed. For numerical simulations, k-ε turbulence model with unstructured mesh is employed. Optimum total head and pump efficiency are obtained at 30° outlet blade angle [4]. A combined study of CFD simulation and inverse design method of the high specific speed helical axial pump is carried out for the two-phase gas-liquid mixture. The steady-state Reynolds averaged Navier-Stoke (RANS) equation for the incompressible flow is employed and the numerical results are compared with experimental results [5]. Experimental and numerical investigation of the axial flow pump is carried out. The most commonly used turbulence modes such as the standard k-ε and Renormalization Group (RNG) k-ε turbulence models are used for numerical simulations. The pump performance and cavitations characteristics of the numerical and experimental analysis are well matched at the operation range of the pump [6].

The effects of surface roughness, Reynolds number, and viscosity of fluid flowing through the pump on performance is analyzed. Maximum efficiency of the pump is in good agreement with a good surface finish and lower viscosity of the fluid [7]. The effect of scale and surface roughness values on the performance of the jet pump is investigated. The k-ε, SST k-ω, RSM and transition SST models are employed numerically [8]. Among the four turbulence models, the transition SST model results are in good agreement with the experimental data. The efficiency of the pump is increased by improving the surface finish of the pump components. The influence of surface roughness parameter on the performance of low specific speed centrifugal pump is done with the application of numerical and experimental analysis [9]. The total head and cavitations characteristic of the pump has improved at the minimum surface roughness value of the pump parts. The effects of inlet and outlet blade angle of a centrifugal pump impeller handling fluid with various viscosities and surface roughness values are performed in the numerical and experimental analysis [10]. The total head of the pump has improved with an increase of impeller outlet blade angle, but significant effects are nor seen in the head with an increased inlet blade angle. Input power and total head are increased with the increase of surface roughness value and simultaneously pump efficiency is decreased. Effects of surface roughness and surface coating on the mixed flow submersible pump impeller and diffuser are analyzed numerically and experimentally [11]. The pump efficiency has increased up to 4.5% and input power is decreased 0.3 to 0.5 kW of the pump operating range due to the application of different coating materials applied on the surface of the impeller and diffuser. Applying the surface coating on the pump components is one of the methods to improve the efficiency of the pump. The influence of surface roughness values and application of coating materials with different coating thickness on the centrifugal pump was not performed earlier. This research work is divided into three stages. In the first stage, six model pumps are assembled with same designed configurations and a performance test is done on these pumps. In the second stage, these six pumps are disassembled and the coating is applied to the impeller and volute casing. Out of the six set of impeller and casing, three sets are coated with the epoxy coating material and another three sets are coated with polyurethane coating material of different coating thickness. In the third stage, surface roughness and coating thickness values of the coated impeller and volute casing are measured and the pump performance study is done experimentally and numerically for different surface roughness values.

2. Selection of model centrifugal pump

In this study, six numbers model centrifugal pump of same design specification is assembled with cast iron impeller and volute casing of surface roughness value 4.39 μm R_a. The MCP are selected from M/s. Coimbatore Engineering Corporation (CEC) which is one of the leading pump manufacturers in Coimbatore. The cost of the cast iron materials is comparatively lesser than the cost of the steel or phosphor bronze material. Hence the cast iron material is commonly used and preferred for casting impeller and volute casing for manufacturing agricultural pumps. The specification of the MCP is presented in Table 1.

A set of three impellers and volute casing are coated with epoxy coating materials of different coating thickness and the surface roughness values and remaining set is coated with polyurethane. The surface roughness value of impeller and volute casing is measured using Mitutoyo make surface roughness tester (Model: ST-210) and the coating thickness is measured by using Times Group make coating thickness gauge (Model: TT-210). Table 2 clearly

Table 1
Specification of model centrifugal pump

Parameters	Value
Discharge at duty point (Q), m^3/s	0.015
Total head at duty point (H), m	32
Speed of pump (N), rpm	2900
Specific speed ($N_s = N \cdot Q^{1/2}/H^{3/4}$), rpm	100
Inner diameter of impeller (D_1), mm	75
Width of impeller at inlet (B_1), mm	18
Outer diameter of impeller (D_2), mm	180
Width of impeller at outlet (B_2), mm	11.5
Blade thickness (t), mm	4
Number of blades	6
Volute inlet mean diameter (D_3), mm	75
Volute mean diameter at outlet (D_4), mm	65
Prime mover rating (P), kW	7.5
Best efficiency point (BEP), %	66
Surface roughness value of impeller and volute casing (before coating) (R_a), μm	4.39
Pump efficiency (η), %	–

illustrates the details of coating thickness, surface roughness values of impellers and volute casings. Fig. 1 and Fig. 2 show the epoxy coating and polyurethane coating applied on the impeller and volute casing for various coating thicknesses. Similarly, Fig. 3 and Fig. 4 show the measurement of coating thickness on polyurethane coated and surface roughness on uncoated impellers.

Table 2
Surface coating of impeller and volute casings

Types of coating	Epoxy coating		Polyurethane coating	
	Surface roughness, µm	Coating thickness, µm	Surface roughness, µm	Coating thickness, µm
Impeller and volute 1	1.53	61.2	–	–
Impeller and volute 2	1.22	121	–	–
Impeller and volute 3	0.78	172	–	–
Impeller and volute 4	–	–	0.30	34.4
Impeller and volute 5	–	–	0.14	71.5
Impeller and volute 6	–	–	0.11	128

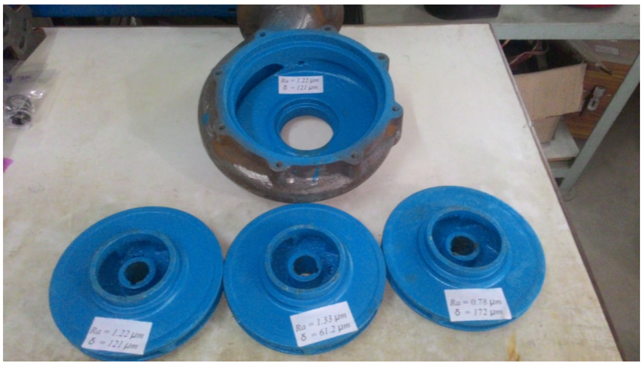

Fig. 1. Epoxy coating.

Fig. 2. Polyurethane coating.

3. Performance analysis of model centrifugal pump

3.1. Experimental study

The performance testing is done on the six sets of the pump before and after applying the coating. The same prime mover (Three phase induction motor) is used for testing six sets of pump configurations. The testing setup for testing MCP is fabricated as per ISO 9906 [12] standard and is shown in Fig. 5. The pump test setup consists of MCP

Fig. 3. Coating thickness measurement.

Fig. 4. Surface roughness measurement.

Fig. 5. Centrifugal pump performance test setup as per ISO 9906.

coupled with a three-phase induction motor, inlet and outlet pipes with control valves, foot valve at inlet pipe, sump and volumetric tank. Vacuum and Pressure transducers are mounted at the inlet and outlet pipes in order to measure the inlet and the outlet pressure. An electromagnetic flow meter and power meters are connected to measure the amount of fluid through the pump and input power. To reduce the uncertainty in the measurement, the measuring devices with better measurement capability is used. Table 3 presents the uncertainty of the measuring devices used in this work. Testing of the pump has been conducted from fully open to shutoff condition of the outlet control valve at an equal interval. Test readings of all the measuring instruments are recorded simultaneously to reduce the error. The output power of the pump is calculated using the total head and flow rate through the pump for all sets of readings and input power is directly measured using a power meter and finally, the efficiency of the pump is calculated. The performance values are presented in Table 4 and the corresponding curves are shown in Fig.6.

3.1.1. Experimental performance

Tables 4, 5 and 6 give the details of experimentally measured pump performance before and after coating. Table 4 presents the performance values of the uncoated pump. The variability in the results of the six uncoated pumps is negligible and hence a test result of the one pump is presented. The performance of the epoxy coated pumps with different coating thickness is presented in Table 5a, 5b and 5c. The performance of the polyurethane coated pumps with different coating thickness is presented in Tables 6a, 6b and 6c and the corresponding performance curves are shown in Figs. 6 and 7. The details of the effect of surface roughness and coating materials on the performance in the operating region of the pump are presented in Figs. 8a–c and 9a–c.

3.1.2. Cost analysis

Cost analysis was performed for the application of a surface coating of the MCP. For cost analysis, Payback Period (PBP) and Net Present Value (NPV) were used. The PB is the simple and common method used to estimate the period in years to recover the initial investment for the coating process. Eq. (1) is used to calculate the PBP.

$$PBP = \frac{C_i}{C_e} \tag{1}$$

where C_i - initial cost invested for the coating process and C_e - expected cost recovered per year.

The NPV method is used for estimating the net present value of the money invested in a long-term process. It helps the manufacturer and user to know how much cost

Table 4
Experimental performance values of model pump without coating (R_a = 4.39 μm)

Q × 10^{-3}, m^3/s	H, m	P, kW	η, %
22.06	11.19	9.39	32.24
21.93	16.63	10.22	43.74
21.53	21.56	10.96	51.92
21.21	26.44	11.36	60.49
19.73	30.93	11.34	65.95
16.51	35.23	10.62	67.10
13.63	37.57	9.92	63.23
9.83	39.82	8.82	54.39
6.50	40.95	7.81	41.76
0.00	41.26	5.72	0.00

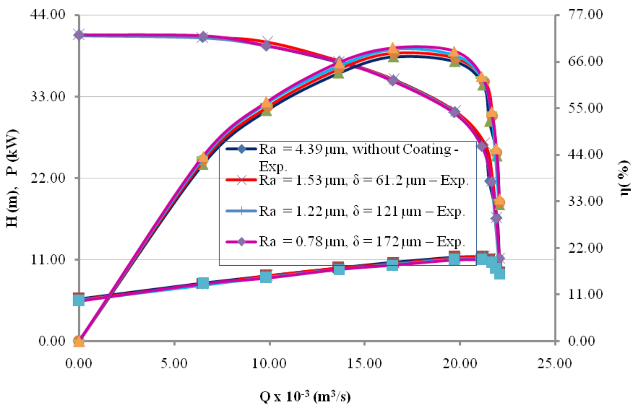

Fig. 6. Experimental performance curves of an uncoated and epoxy coated pump.

Table 3
Details of measurement uncertainty

Name of the measuring instruments	Make	Parameter to be measured	Class of accuracy	Measurement uncertainty at 95% confidence level (% full-scale reading)
Magnetic flow meter	YOKOGAWA	Flow rate	0.5 class	± 0.23
Pressure transducer (inlet)		Inlet pressure	0.2 class	± 1.65
Pressure transducer (outlet)		Outlet pressure	0.2 class	± 1.65
Three phase AC power meter		Voltage	0.2 class	± 0.075
		Current		±0.40
		Power input		±0.53
		Frequency		±0.60

Table 5a
Experimental performance values of model pump with epoxy coating (R_a = 1.53 μm and δ = 61.2 μm)

Q × 10^{-3}, m^3/s	H, m	P, kW	η, %
22.07	11.20	9.24	32.68
21.92	16.61	10.05	44.34
21.70	21.57	10.80	53.03
21.29	26.64	11.34	61.35
19.75	31.00	11.23	66.75
16.52	35.27	10.49	67.91
13.66	37.77	9.85	64.26
9.89	40.34	8.84	55.29
6.50	40.96	7.66	42.52
0.00	41.31	5.58	0.00

Table 5b
Experimental performance values of model pump with epoxy coating (R_a = 1.22 μm and δ = 121 μm)

Q × 10^{-3}, m^3/s	H, m	P, kW	η, %
22.02	11.14	9.08	33.18
21.84	16.49	9.85	44.94
21.64	21.46	10.63	53.73
21.13	26.25	11.00	62.04
19.70	30.86	11.06	67.43
16.50	35.16	10.35	68.85
13.65	37.71	9.73	64.89
9.83	39.79	8.57	56.01
6.50	40.90	7.56	43.18
0.00	41.23	5.47	0.00

Table 5c
Experimental performance values of model pump with epoxy coating (R_a = 0.78 μm and δ = 172 μm)

Q × 10^{-3}, m^3/s	H, m	P, kW	η, %
22.07	11.20	9.09	33.55
21.89	16.56	9.85	45.37
21.65	21.48	10.58	54.16
21.14	26.28	10.96	62.50
19.68	30.79	10.96	68.48
16.47	35.06	10.25	69.19
13.65	37.71	9.67	65.70
9.83	39.84	8.53	56.48
6.52	41.16	7.75	43.59
0.00	41.33	5.43	0.00

Table 6a
Experimental performance of model pump with polyurethane coating (R_a = 0.30 μm and δ = 34.4 μm)

Q × 10^{-3}, m^3/s	H, m	P, kW	η, %
22.01	11.14	8.99	33.44
21.88	16.55	9.82	45.24
21.63	21.46	10.55	53.83
21.15	26.30	10.95	62.17
19.70	30.83	10.96	68.00
16.49	35.14	10.26	69.22
13.63	37.59	9.60	65.22
9.84	39.89	8.52	56.50
6.50	40.95	7.49	43.56
0.00	41.28	5.40	0.00

Table 6b
Experimental performance of model pump with polyurethane coating (R_a = 0.14 μm and δ = 71.5 μm)

Q × 10^{-3}, m^3/s	H, m	P, kW	η, %
22.03	11.14	8.89	34.00
21.90	16.57	9.73	45.86
21.63	21.84	10.46	54.62
21.16	26.30	10.84	63.11
19.72	30.87	10.87	68.87
16.51	35.17	10.17	70.12
13.65	37.55	9.48	66.34
9.84	39.83	8.39	57.38
6.51	40.90	7.37	44.33
0.00	41.24	5.29	0.00

Table 6c
Experimental performance of model pump with polyurethane coating (R_a = 0.11 μm and δ = 128 μm)

Q × 10^{-3}, m^3/s	H, m	P, kW	η, %
22.02	11.16	8.67	34.91
21.89	16.59	9.51	47.09
21.61	21.48	10.22	55.98
21.16	26.34	10.63	64.53
19.69	30.84	10.62	70.62
16.48	35.11	9.90	71.89
13.63	37.54	9.24	68.11
9.82	39.79	8.14	59.06
6.49	40.90	7.13	45.65
0.00	41.31	5.06	0.00

be saved by using the application of the coating process. Eq. (2) is used to calculate the NPV.

$$NPV = C_s \cdot \frac{\left[(1+i)^n - 1\right]}{(1+i)^n \cdot i} \quad (2)$$

where C_s – annual cost saving, i - annual interest rate and n - number of years

In this analysis, a cost of electricity for water distribution applications was taken as 0.109 USD/kWh in Tamil Nadu, India. The initial investment cost of epoxy and poly-

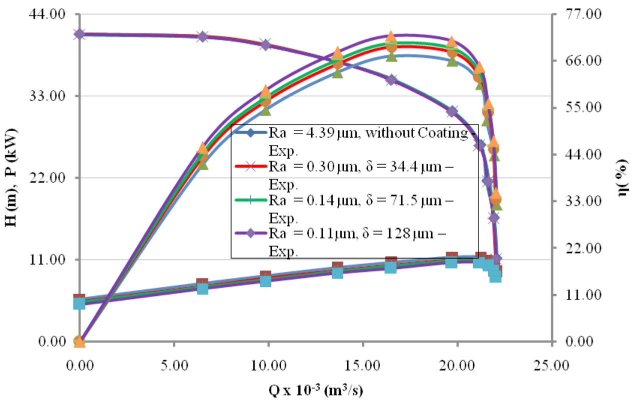

Fig. 7. Experimental performance curves of an uncoated and polyurethane coated pump.

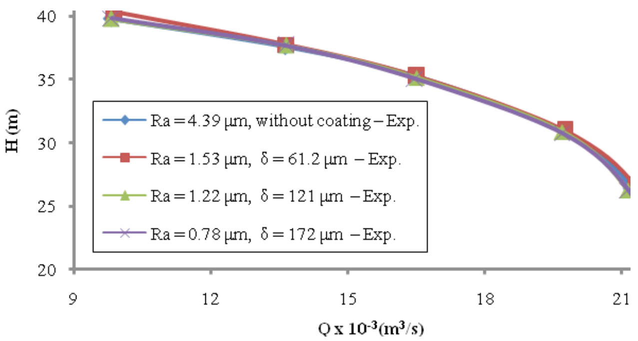

Fig. 8. (a). Q–H curve in the operating region (epoxy coated).

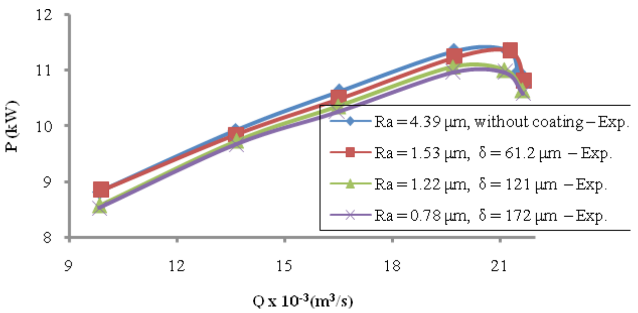

Fig. 8. (b). Q–P curve in the operating region (epoxy coated).

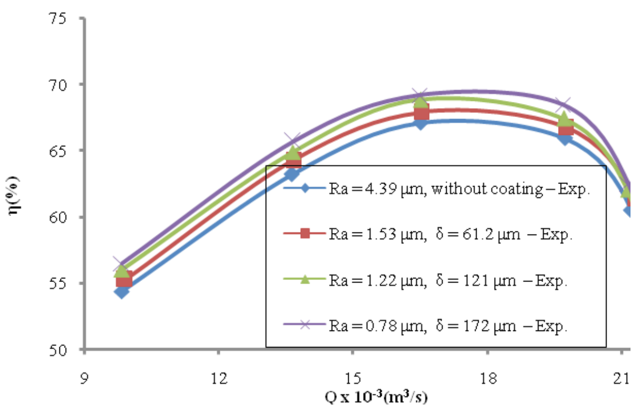

Fig. 8. (c). Q–η curve in the operating region (epoxy coated).

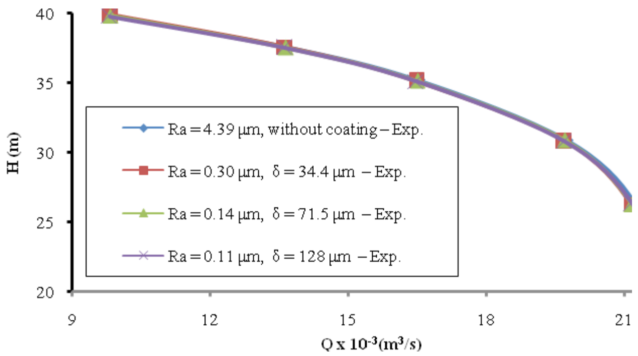

Fig. 9. (a). Q–H curve in the operating region (polyurethane coated).

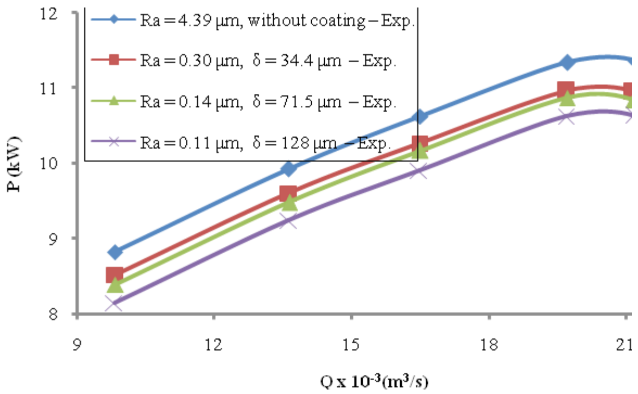

Fig. 9. (b). Q–P curve in the operating region (polyurethane coated).

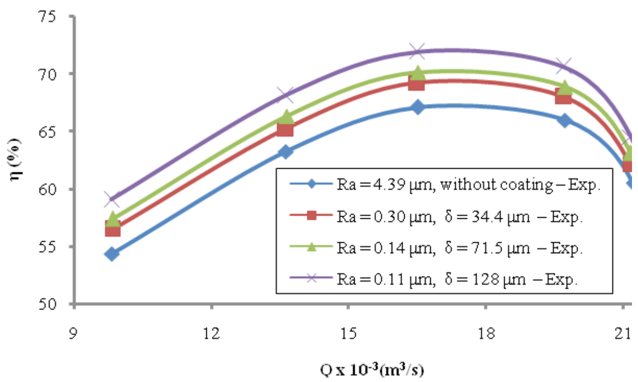

Fig. 9. (c). Q – η curve in the operating region (polyurethane coated).

urethane coating was approximately 375 and 625 USD for the flow passages of the MCP including the cost of material and labor. Operating time of the pump was assumed as 20 h/d for the estimation of the electric power consumption and saving.

3.2. Numerical study

The numerical simulation of MCP is done based on the solutions of the governing equations of fluid dynamics. The governing equations selected are the conservation of mass

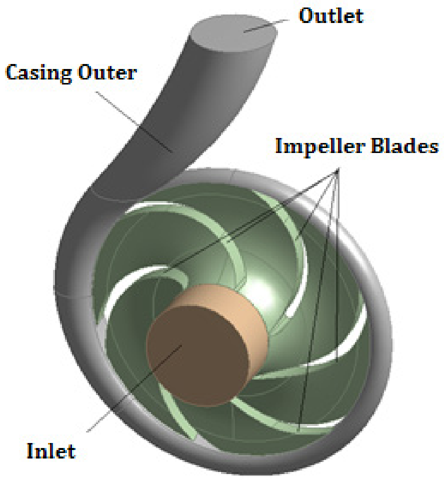

Fig. 10. Computational fluid domain of MCP.

and momentum equations. The fluid domain of the MCP shown in Fig. 10 is modeled using 3D CAD software and it is discretized with unstructured tetrahedral mesh. The unsteady RANS model is used to simulate the unsteady flow in the pump. In order to capture the wall roughness and skin friction effect, k-ω SST turbulence model is used [11]. To specify the relative motion between the impeller and volute casing, a mesh interface is provided and mesh motion is selected for the transient simulation. The condition of the inlet boundary and outlet boundary are selected at the pressure at the inlet and a mass flow rate at the outlet. Mesh motion with 2900 rpm is applied to the impeller cell zone condition. Table 7 shows the detailed boundary conditions of the numerical simulation. Based on the number of blades in the impeller and speed of rotation, the time step for the transient simulation is set as 0.00345 s. The unsteady simulation is computed for 10 impeller rotations to achieve solution convergence.

3.2.1. Mesh dependency study

The accuracy of the numerical results is entirely based on the mesh size and turbulence model selected. The entire flow domain is discretized start from the coarse mesh of size 3.5 to fine mesh of size 1.0 and the number of a mesh in the flow domain is between 0.136 and 5.64 million. Mesh dependency test for the total head of the pump is performed for each mesh size and mesh numbers for the duty point mass flow rate. From the analysis, an optimum mesh size of 1.5 and mesh number 1.69 million is selected for the further analysis. For the better quality of mesh, skewness of mesh is also considered [13]. The average mesh skewness is maintained as 0.21.The details of mesh and mesh dependency tests are presented in Table 8 and Fig. 11 respectively. In order to capture the wall roughness effect, a boundary layer mesh is created over the impeller blades and volute surface. Mesh inflation of maximum 5 layers with a smooth transition and uniform growth rate is selected for the boundary layer mesh. Fig.12 shows the unstructured mesh of the flow domain and a close-up view near to impeller blade with boundary layer mesh.

Table 7
Boundary conditions for CFD analysis

Analysis type	Unsteady-state incompressible
Fluid	Water
CFD model	k–ω SST model
Turbulence intensity	5% (medium intensity)
Inlet	Total pressure (100000 Pa)
Outlet	Mass flow rate
Rotational speed	2900 rpm
Convergence criteria	10^{-4}
Mass imbalance	10^{-2}

Table 8
Computation mesh details

Mesh size	No of mesh ($\times 10^5$)	Total head, m
3.5	1.36	18.72
3.0	2.18	24.56
2.5	3.72	27.01
2.0	7.18	28.34
1.5	16.90	33.09
1.0	56.40	33.12

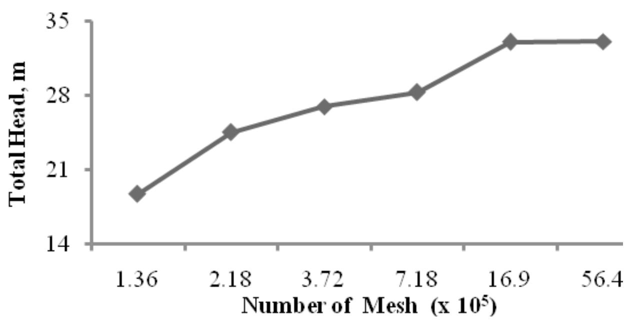

Fig. 11. Mesh dependency test.

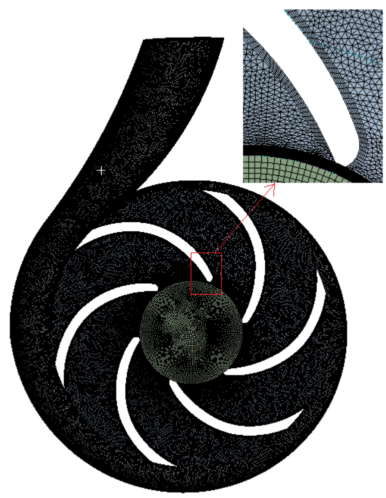

Fig. 12. Meshed flow volume with close-up view of blade.

3.2.2. Numerical performance

The numerical performance values of an MCP are measured by simulating three sets of the pump. In which, one set is in uncoated form and the other two are coated with epoxy and polyurethane coatings. The simulations are performed for all mass flow rates. Figs. 13a and 13b shows the pressure and velocity distributions of the uncoated pump and coated pump with polyurethane coating. The wall shear stress is based on the fluid properties of density and viscosity and physical properties such as cross-section and surface roughness. Also the ratio of wall shear stress and reference dynamic pressure gives the non-dimensional parameter called skin friction coefficient. The skin friction coefficient increases with an increase in wall roughness and wall shear stress. Figs. 14a and 14b show the effects of skin friction coefficient

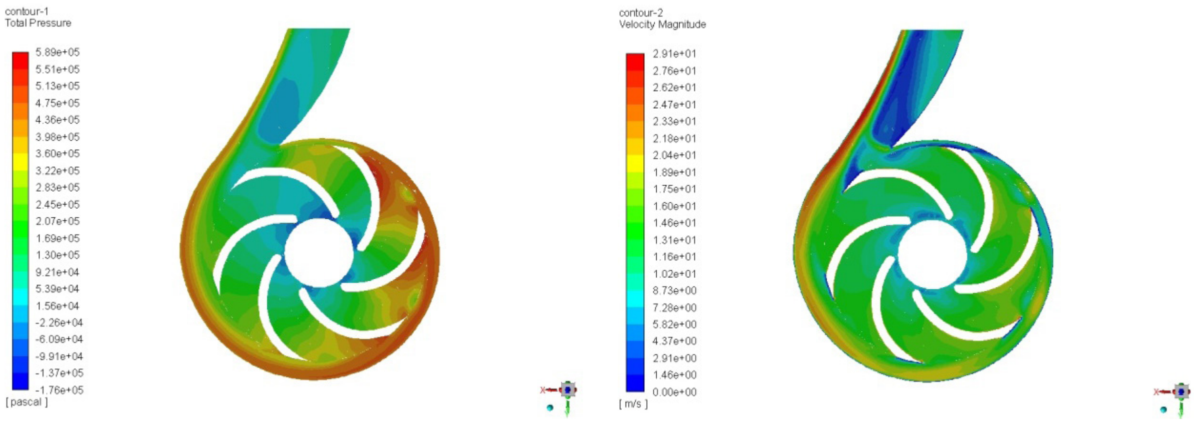

Fig. 13. (a). Pressure (Pa) and velocity distribution (m/s) of the uncoated pump (R_a = 4.39 µm).

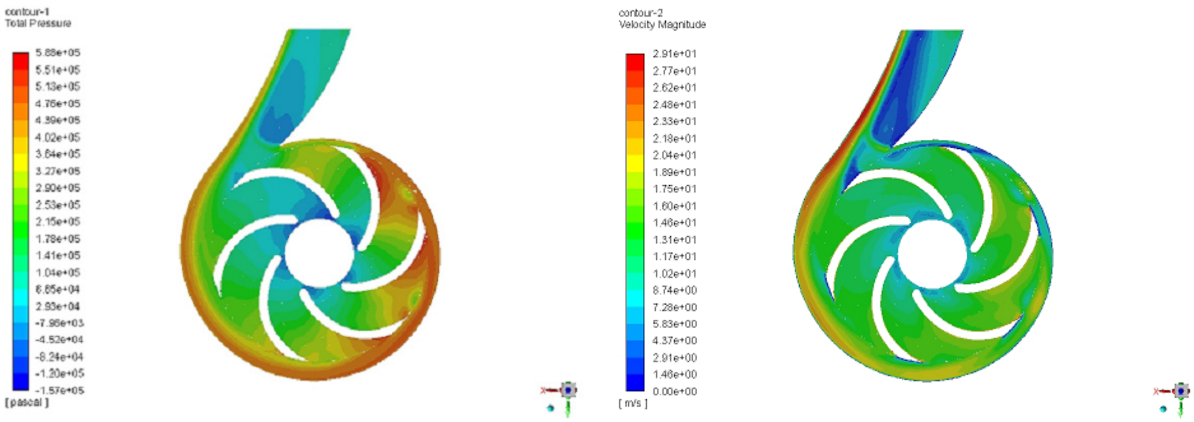

Fig. 13. (b). Pressure (Pa) and velocity distribution (m/s) of the polyurethane coated pump (R_a = 0.11 µm).

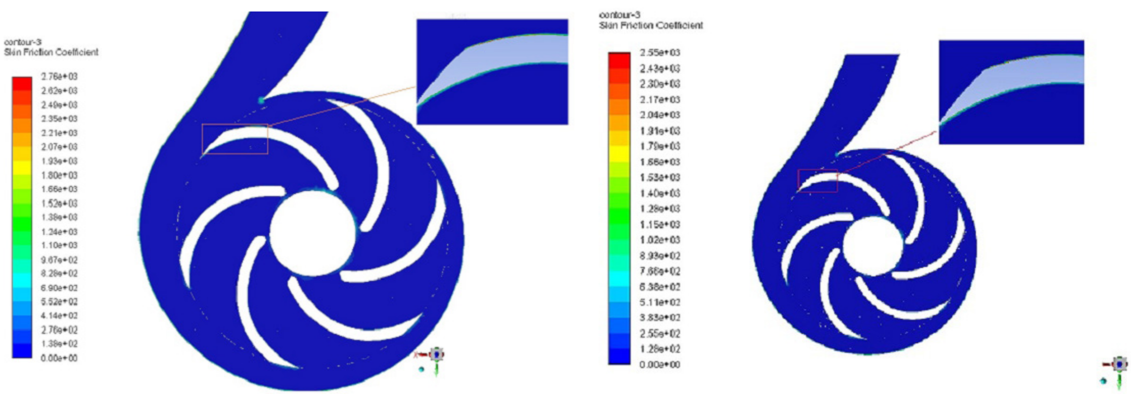

Fig. 14. (a). Effect of skin friction (uncoated) (b). Effect of skin friction (coated).

over the impeller blades of the uncoated pump (R_a = 4.39 µm) and polyurethane coated pump (R_a = 0.11 µm)and its value is decreased from 0.027 to 0.023. Tables 9a, 9b and 9c present the performance values of uncoated and coated pumps with epoxy and polyurethane coatings respectively and the numerical results are compared with the experimental results. Fig. 15 shows the numerical and experimental performance curves of uncoated and coated pumps.

Table 9a
Numerical performance of a model pump without coating

$Q \times 10^{-3}$, m^3/s	H, m	P, kW	η, %
22	11.73	7.23	34.99
21	23.01	9.67	48.99
20	27.52	9.46	57.04
18	32.11	8.62	65.74
17	33.41	8.22	67.74
15	35.62	7.75	67.59
11	38.63	7.12	58.51
7	40.52	6.19	44.92
5	40.83	5.42	36.93

Table 9b
Numerical performance values of model pump with epoxy coating (R_a = 0.78 µm and δ = 172 µm)

$Q \times 10^{-3}$, m^3/s	H, m	P, kW	η, %
22	11.78	7.02	36.19
21	23.02	9.41	50.37
20	27.56	9.21	58.67
18	32.21	8.39	67.75
17	33.42	8.01	69.54
15	35.75	7.52	69.91
11	38.64	6.92	60.22
7	40.53	6.03	46.13
5	40.86	5.29	37.86

Table 9c
Numerical performance values of model pump with polyurethane coating (R_a = 0.11 µm and δ = 128 µm)

$Q \times 10^{-3}$, m^3/s	H, m	P, kW	η, %
22	11.84	6.72	38.00
21	23.12	9.02	52.77
20	27.59	8.86	61.06
18	32.3	8.08	70.54
17	33.43	7.72	72.17
15	35.79	7.27	72.40
11	38.69	6.61	63.12
7	40.56	5.81	47.91
5	40.92	5.08	39.49

4. Results and discussion

4.1. Comparisons of experimental test results

Table 4 presents the experimental test results of MCP before coating and Tables 5a–5c and 6a–6c present the performance values of pumps with epoxy and polyurethane coating with different coating thickness and surface roughness values and the corresponding performance curves are shown in Figs. 6 and 7. The enlarged view of the performance curves in the operating range of the pumps is also presented in Figs. 8a–c and 9a–c. From the performance curves, it is found that the pump total head and discharge rate have not improved significantly. The power input and pump efficiency are improved by applying coating materials on the flow passages of the impeller and volute casing. While comparing the power input and efficiency of the coated against the uncoated pump components, the input power is decreased and efficiency is increased. At the duty point, the power input of the coated pumps are decreased by 0.12, 0.25 and 0.35 kW and efficiency values are increased by 0.8, 1.6 and 2.2% by applying an epoxy coating on the pump components. Power input values of the coated pumps are decreased by 0.35, 0.47 and 0.75 kW and efficiency values are increased by 2.1, 3.05 and 4.75% by applying the polyurethane coating. For the entire flow range, the power input 0.13 to 0.38 kW has been reduced by applying the epoxy coating and 0.36–0.74 kW is reduced when using polyurethane coating. At the pump duty point, the efficiency of the uncoated pump is 65.7%, whereas the epoxy coated pump is 67.9% and polyurethane coated pump is 70.45%. It is shown that the efficiency of the pump has been improved by applying a polyurethane coating with minimum coating thickness. At the duty point, considering the good surface finish of the pump components, electricity saved by the MCP with epoxy and polyurethane coating are 0.35 kW and 0.75 kW. Using Eq. (1), annual cost saving was calculated as 275, and 585 USD and PBP was found as 0.73 and 0.94 y. The NPV was calculated at the annual interest rate of 10% and 5 y life of the coating materials (recommended by the coating materials supplier) using Eq. (2), are 1043 and 2218 USD. This is really higher than the initial investment of 375 and 625 USD.

4.2. Comparisons of numerical simulation results

Tables 9a–c present the numerical performance of uncoated and coated pumps and the corresponding perfor-

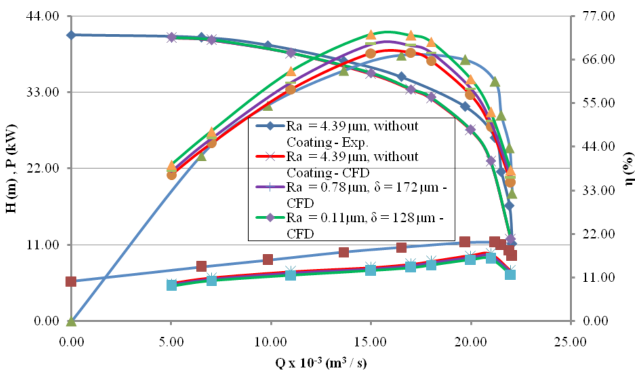

Fig. 15. Numerical and experimental performance curves of an uncoated and coated pump.

mance curves are shown in Fig.15. From the Fig.15, it was observed that the difference in numerical performance and experimental performance values are within the acceptable limit [3]. The differences in corresponding performance value at the duty point are 2.7% variation on the total head and 2.9% on pump efficiency. While comparing the numerical performance values of the coated pump with uncoated pumps, there are no significant changes in head and discharge values. Significant changes are found in the input power and pump efficiency values. At the duty point conditions, the input power is reduced in the range of 0.23–0.48 kW and the efficiency was increased in the range of 2.32–4.81%. The above result shows that the application of the surface coating on the pump components has improved the surface finish and finally improves the overall performance characteristics of the pump.

5. Conclusion

The following are conclusion arrived from this study.

The surface finishing of the pump components has significantly improved by the application of the surface coating. When compared to the coating thickness of the epoxy and polyurethane coating, polyurethane has resulted in better surface finishing than that of the epoxy of minimum coating thickness. Therefore the polyurethane materials are preferred coating material.

1. The total head and discharge characteristics of the all the six sets of pump models are identical; this is due to the improvement in the surface finishing of the impeller and volute casing. Therefore there are no significant changes in these performance values.
2. The input power of the model pump is decreased by 0.35 kW and pump efficiency is increased by 2.2% in the epoxy coated pumps.
3. The input power of the model pump is decreased by 0.75 kW and pump efficiency are increased by 4.75% for the polyurethane coated pumps.
4. By applying the epoxy coating on the pump components, the improvement on the surface finish is by 0.78 μm for the coating thickness of 172 μm. Whereas for the polyurethane coating, the surface finish of 0.11μm is obtained with a coating thickness of 128 μm.
5. An epoxy and polyurethane coatings on pump components results in the efficiency of the pump is improved by 67.9 and 70.45% at the same time the efficiency of the uncoated pump is 65.7%. This clearly shows that the efficiency of the pump is improved with the application of surface coating.
6. The cost analysis of the MCP shows significant cost saving by the application of coating materials and safeguards the pump from corrosion and cavitations.

Acknowledgment

Authors thank the management of Scientific and Industrial Testing and Research Centre(Si'Tarc), A M Selvaraj and D Anguraj for their support for conducting an experimental study.

References

[1] A. Goto, M. Zangeneh, Hydrodynamic design of pump diffuser using inverse design method and CFD, J. Fluids Eng., 124 (2002) 319–328.
[2] A. Allali, S. Belbachir, A. Lousdad, L. Merahi, Numerical approach based design of centrifugal pump volute, Mechanika., 21 (2015) 301–306.
[3] M. Nataraj, R. Ragoth Singh, Analyzing pump impeller for performance evaluation using RSM and CFD, Desal. Water Treat., 52 (2014) 6822–6831.
[4] N. Mohammadi, M. Fakharzadeh, Analysis of effect of impeller geometry including blade outlet angle on the performance of multi-pressure pumps : Simulation and experiment, Mechanika., 23 (2017) 107–119.
[5] S. Cao, G. Peng, Z. Yu, Hydrodynamic design of rotodynamic pump impeller for multiphase pumping by combined approach of inverse design and CFD analysis, J. Fluids. Eng., 127 (2005) 330–338.
[6] H. Ding, F.C. Visser, Y. Jiang, M. Furmanczyk, Demonstration and validation of a 3D CFD simulation tool predicting pump performance and cavitation for industrial applications, J, Fluids Eng., 133 (2011) 011101-1–14.
[7] J.F. Gülich, Effect of Reynolds number and surface roughness on the efficiency of centrifugal pumps, J. Fluids Eng., 125 (2003) 670–679.
[8] K. Alda, R. Yap, Investigation of effects of scale and surface roughness on efficiency of water jet pumps using CFD, Eng. Appl. Comp. Fluid. Mech., 8 (2014) 14–25.
[9] P. Limbach, Numerical and experimental analysis of cavitating flow in a low specific speed centrifugal pump with different surface roughness, J. Fluids Eng., 139 (2017) 101201–101208.
[10] S. Ahmed, I. Bellary, A. Samad, Pumping crude oil by centrifugal impeller having different blade angles and surface roughness, J. Petrol. Explor. Prod. Technol., 6 (2016) 117–127.
[11] C. Murugesan, R. Rudramoorthy, Experimental and numerical study of efficiency improvement by surface coating on the impellers and diffusers of mixed flow submersible borewell pumps, J. Scient. Ind. Res., 75 (2016) 300–305.
[12] BS EN ISO 9906: 2000, Rotodynamic Pumps — Hydraulic Performance Acceptance Tests — Grades 1 and 2, 2000.
[13] ANSYS CFX Theory Guide, ANSYS Inc., 2013.