Performance enhancement of centrifugal pump by minimizing casing losses using coating

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Abstract

Present study focuses on the experimental and numerical investigation of the effect of surface roughness of casing and adhesive coating on the efficiency of centrifugal pump. It consists of two parts; in the first part, effect of coating centrifugal pump casing by ultra-smooth ceramic based coating on the efficiency of the centrifugal pump is investigated experimentally. In the second part, effect of change in surface roughness of casing is investigated numerically by using the CFD tool. The surface roughness of casing is modeled in terms of effective sand grain size parameters and investigated by solving Reynolds-averaged Navier–Stokes equation at volume mesh. Effects of three surface roughnesses, i.e., 100 μm , 200 μm and 300 μm , are investigated numerically. It is observed that surface roughness of casing has considerable effect on pump performance, and pump head decreases by 9.06–14.73% by change in surface roughness from 100 to 300 μm for different mass flow rates values. For same flow rate, centrifugal pump with coated casing produces more head than that of pump without coating. Coating casing with the ultra-smooth adhesive material results in maximum efficiency enhancement of 9.6%.

Keywords Centrifugal pump · CFD · Coating · Surface roughness

1 Introduction

The primary function of centrifugal pumps is to handle fluid from a lower level to a 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 20% of the electric energy which is produced around the globe [1]. Owing to the increase in the cost of electricity and the mandatory usage of the pumps, efficiency enhancement initiatives are considered to be the key solution in reducing the energy utilization and eventually resisting the global environmental impacts.

Many researchers have worked on efficiency enhancement of the centrifugal pump by investigating effects of design variables on pump performance. Various design

parameters such as the number of impeller blades, axial and radial clearance, surface roughness of various parts and the specific speed at which pump works have a considerable effect on the performance of centrifugal pump [1, 2]. As hydraulic losses are the major contributors to performance deterioration of the centrifugal pump, many recent investigations focus on hydraulic losses. Two prime mechanisms of hydraulic losses are skin friction due to the surface condition of various components and interfacial surface energy at the solid–liquid interface. Recent investigations have shown that surface roughness of various parts of the centrifugal pump has considerable effects on the efficiency and performance curves of a centrifugal pump. After demonstrating the effect of smoothing several parts of the centrifugal pump at a low and high specific speed, it is observed that efficiency improvement of 5–18.5% can be achieved with partial smoothing of

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various parts. Smoothing the impeller, front shroud and back shroud is found very effective for efficiency enhancement [3]. The influence of volute surface roughness on the performance of centrifugal pump was investigated earlier, and it was observed that volute surface roughness increased from 0.08 to 1.2 mm, and pump maximum efficiency decreased by 7.5%. It was also noticed that no considerable effect of volute surface roughness is produced on the impeller flow field [4]. Upon investigating the effect of Reynolds number and surface roughness, it is found that the lower the hydraulic efficiency, the higher is the gain from surface improvement. The higher the Reynolds number, the more is the gain from surface roughness. It was also observed that surface improvement of small pumps brings higher efficiency gain than that of large pumps [5].

CFD has emerged as a very effective tool in handling various turbomachinery problems. Various researchers have modeled the centrifugal pump with different surface roughness in CFD to investigate the effect of surface roughness on pump performance. In the CFD analysis, there are two ways to analyze surface roughness. The first one is by taking into account total surface shape, but the very fine mesh is required for such an approach. This is not an appropriate method for most of the industrial applications. The second method is by defining surface roughness in terms of some special parameters. Sand grain equivalent parameter is used to define surface roughness in most of the commercially used software. Sand grain equivalent parameter is a function of the amplitude of roughness, frequency of roughness and shape of the surface. From CFD analysis, it was observed that with an increase in surface roughness till critical surface roughness value, the efficiency of the centrifugal pump decreases. The above critical surface roughness value efficiency increases as compared to efficiency at the critical value. The effect of surface roughness was maximum at a higher mass flow rate [6]. Out of different turbulence models, SST k - ω model is widely used for the analysis of centrifugal pumps due to its capability to capture flow physics near the solid surface. In core region of flow, it gives features of k - ϵ model, which is added advantage of SST k - ω in pump analysis. Realizable wall function gives better results in the boundary layer region on penalty of higher computational time and storage space.

Considering the effect of surface roughness of various parts of the centrifugal pump on efficiency and recent advances in polymer technology, surface coating by polymer coating materials to reduce surface roughness is an effective way of enhancing the energy efficiency of the centrifugal pump. The surface coating improves the efficiency of the pump by smoothing the surface and decreasing adhesive coefficient between surface and fluid. Experimental and numerical investigation shows

that the effect of geometric modification in impeller and polyurethane coating on volute and impeller of old pump results in efficiency regain of about 15% and head increase by 18%. Payback period of the coating was observed to be less than a year [7]. Nanoparticle coating on pump impeller and casing are found to be very effective against corrosion. The thickness of nanoparticle coating and working atmosphere of the pump governs corrosion rate of impeller and casing [8]. Coating thickness on impeller affects the flow field of impeller considerably, while coating thickness on the casing has a negligible effect on impeller flow field [9]. Recently developed hydrophobic coating materials are found to be very effective as it decreases surface roughness as well as interfacial forces between solid-liquid interface. Hydrophobization of pump components results in an efficiency enhancement of 0.5–7.5% at different operating conditions [10]. Drag forces at solid-liquid interaction are a function of surface wettability of fluid. The extent of surface wettability is mainly depended on solid-liquid interfacial energy. Numerical investigation on the pump with different values of adhesive coefficient shows that adhesive coefficient reduction significantly affects the hydraulic efficiency of the centrifugal pump, by hydraulic lowering losses. Adhesive coating shows a positive effect on cavitation [11]. From the investigation of the effect of surface roughness on contact angle and sliding angle of a water droplet with the surface, it is observed that in the highly hydrophobic region, the sliding angles of water droplets decrease with increasing contact angle, which is considered to be dependent on the surface roughness [12]. Effect of Belzona 1341 supermetalgilde polymer coating shows a maximum 6% increase in peak efficiency. Also, there was little change to the pump characteristics with the peak efficiency duty coinciding with that of the uncoated pump [13].

From literatures, it is observed that effect of surface roughness of pump is extensively investigated both numerically and experimentally by different researchers. However, contribution of surface roughness of different parts like impeller, shrouds, casing, etc., is not investigated separately. Effect of surface smoothing of entire pump is very costly approach considering efforts and coating material required to apply coating on entire pump. Coating thickness on impeller has a significant effect on pressure and flow field inside pump. Surface coating on entire pump, i.e., impeller, shrouds and casing, may result in greater increase in total head developed by pump, but it associates with higher coating cost and increase in weight of the pump which ultimately results in higher power consumption. Considering all above factors effect of surface roughness of casing only is investigated in the present work.

In this work, the effect of ultra-smooth ceramic-based coated casing on efficiency of a centrifugal pump is analyzed experimentally. Effect of surface roughness of casing is investigated numerically by using ANSYS 19.2.

2 Numerical approach

A 3D flow simulation was performed on geometric model of the pump using Reynolds-averaged Navier–Stokes (RANS) equation. For current study, full computational domain considered consists of three separate cell zones, i.e., impeller fluid, casing fluid and inlet pipe. Tetrahedral grids are generated on impeller and casing geometric model separately by using ANSYS 19.2 meshing. Figures 1 and 2 show volume mesh of impeller and casing, respectively.

In the numerical analysis of complex geometries, computational time and accuracy of results are prime concern. Computational time can be decreased with generation of suitable grid without hampering accuracy of obtained results. In the present study, four different grids are generated and most suitable grid considering element counts and accuracy of results is chosen. Numerical results and element numbers for different grid sizes are shown in Table 1. Numerical results for grid 3 and grid 4 were very close and were in acceptable accuracy range when compared with experimental results. Considering lesser number of elements and accuracy of results, grid 3 was selected for further analysis.

The 3D flow dynamics in centrifugal pump are described by continuity equation and Reynolds-averaged Navier–Stokes equations, which gives a detailed mathematical description of the velocity and pressure fields:

Fig. 1 Volume mesh on impeller

Fig. 2 Volume mesh on the casing

Table 1 Grid independence test

Computational domain	Element numbers (millions)	Head (m)	Deviation of head from experimental value (%)	Efficiency (%)	Deviation of efficiency from experimental (%)
Mesh 1	3.57	15.82	10.55	57.24	13.2
Mesh 2	3.86	15.03	5.03	54.38	7.61
Mesh 3	4.1	14.62	2.16	52.9	4.69
Mesh 4	4.35	14.59	1.95	52.79	4.47
Experimental		14.31		50.53	

$$\frac{\partial \bar{u}}{\partial x} + \frac{\partial \bar{v}}{\partial y} + \frac{\partial \bar{w}}{\partial z} = 0$$

$$\begin{aligned} & \rho \left(\frac{\partial V_x}{\partial \tau} + V_x \frac{\partial V_x}{\partial x} + V_y \frac{\partial V_x}{\partial y} + V_z \frac{\partial V_x}{\partial z} \right) \\ &= -\frac{\partial P}{\partial x} + \mu \frac{\partial^2 V_x}{\partial x^2} + \mu \frac{\partial^2 V_x}{\partial y^2} + \mu \frac{\partial^2 V_x}{\partial z^2} \\ & - \rho \frac{\partial v'_x v'_x}{\partial x} - \rho \frac{\partial v'_x v'_y}{\partial y} - \rho \frac{\partial v'_x v'_z}{\partial z} + S_x \end{aligned}$$

$$\begin{aligned} & \rho \left(\frac{\partial V_y}{\partial \tau} + V_x \frac{\partial V_y}{\partial x} + V_y \frac{\partial V_y}{\partial y} + V_z \frac{\partial V_y}{\partial z} \right) \\ &= -\frac{\partial P}{\partial y} + \mu \frac{\partial^2 V_y}{\partial x^2} + \mu \frac{\partial^2 V_y}{\partial y^2} + \mu \frac{\partial^2 V_y}{\partial z^2} \\ & - \rho \frac{\partial v'_y v'_x}{\partial x} - \rho \frac{\partial v'_y v'_y}{\partial y} - \rho \frac{\partial v'_y v'_z}{\partial z} + S_y \end{aligned}$$

$$\begin{aligned} & \rho \left(\frac{\partial V_z}{\partial \tau} + V_x \frac{\partial V_z}{\partial x} + V_y \frac{\partial V_z}{\partial y} + V_z \frac{\partial V_z}{\partial z} \right) \\ &= -\frac{\partial P}{\partial z} + \mu \frac{\partial^2 V_z}{\partial x^2} + \mu \frac{\partial^2 V_z}{\partial y^2} + \mu \frac{\partial^2 V_z}{\partial z^2} \\ & - \rho \frac{\partial v'_z v'_x}{\partial x} - \rho \frac{\partial v'_z v'_y}{\partial y} - \rho \frac{\partial v'_z v'_z}{\partial z} + S_z \end{aligned}$$

To solve the closure problem of RANS equation, SST k- ω turbulence model was used. The SST k- ω model is one of the most commonly used turbulence models. It is a two-equation model, which means it includes two extra transport equations to represent the turbulent properties of the flow. This allows a two-equation model to account for history effects like convection and diffusion of turbulent energy. The first transported variable is turbulent kinetic energy (k), which determines the energy in the turbulence. The second transported variable in this case is the specific dissipation (ω); it is the variable that determines the scale of the turbulence.

The most widely used SST k- ω model was selected with a scalable wall treatment. This is the most

widely used combination which helps with convergence issues. No enhanced wall treatment is provided; by selecting this option, more accurate results would have been obtained because it could have captured the boundary layer physics more properly than the realizable wall functions, but at the cost of more time and memory. One more reason for not using enhanced wall treatment is that if we opt for enhanced wall treatment by default, it disables the option of providing surface roughness of individual surface. As analyzing the effect of surface roughness is the prime objective of this numerical investigation, realizable wall function is used. The surface roughness of casing is modeled by using equivalent sand grain size on casing surface. Impeller and casing are coexisted in computational domain as rotating and stationary frame of references.

Multiple reference frame method is used for this investigation, which is a steady-state approximation in which individual cell zones move at different rotational/translational speeds. This approach is appropriate when the flow at the boundary between the zones is nearly uniform.

A numerical investigation is carried out at a constant speed of 1350 rpm by varying mass flow rate at outlet and sand grain surface roughness value. Investigation is carried out at three different surface roughnesses of 100 μm , 200 μm and 300 μm . The boundary conditions used are shown in Table 2.

Pressure contours for different mass flow rates and surface roughness are shown in Figs. 3, 4, 5, 6 and 7.

Figures 3, 4, 5, 6 and 7 show variation of pressure in centrifugal pump for different surface roughnesses at

Table 2 Boundary conditions

Parameters	Description
Cell zones	Impeller, casing and inlet pipe
Mesh type	Unstructured
Turbulence model	SST k- ω
Inlet boundary condition	Pressure = 101,325 Pa
Outlet boundary condition	Mass flow outlet (kg/s)

Fig. 3 Pressure contour for 6.6 kg/s and **a** 100 μm , **b** 200 μm , **c** 300 μm surface roughness

Fig. 4 Pressure contour for 5.95 kg/s and **a** 100 μm , **b** 200 μm , **c** 300 μm surface roughness

identical boundary conditions. Area average pressure at outlet for pump with casing surface roughness of 100 μm for all mass flow rates is higher than that of for 200 μm and 300 μm , and this is due to the fact that at lower surface roughness pump casing losses are minimum which results in higher value of head produced. As surface roughness increases, casing frictional losses increase which results in decrease in total head produced by pump. Variation of pressure field for different

mass flow rate values can be clearly observed from the pressure contours. Maximum pressure difference of 21,090 Pa is recorded for the mass flow rate of 6.6 kg/s, which is equivalent to 2.14 m of water head.

Effect of surface roughness on head produced by pump is shown in Table 3.

These results are plotted in terms of performance curve as shown in Fig. 8.

Fig. 5 Pressure contour for 5.3 kg/s and **a** 100 μm , **b** 200 μm , **c** 300 μm surface roughness

Fig. 6 Pressure contour for 4.65 kg/s and **a** 100 μm , **b** 200 μm , **c** 300 μm surface roughness

From head versus flow rate curve, it is observed that head produced by centrifugal pump with lower surface roughness value is higher than that of with higher surface roughness value, and this is due to higher amount of hydraulic losses due to vortices and cross-currents in flow over rough surface. Change in surface roughness from 100 to 300 μm results in head difference of 1.53 to 2.15 m of water for different mass flow rate values. Effect of surface roughness is found more severe at higher mass flow rate

values. Maximum difference is pressure head observed 2.14 m at mass flow rate of 6.6 kg/s. Validation of numerical results is done by comparing numerical results of actual surface roughness at working condition with experimental results. Actual surface roughness of centrifugal pump measured with surface profilometer is 100 μm . Comparison of experimental and numerical results in terms of performance curves is shown in Fig. 9.

Fig. 7 Pressure contour for 4 kg/s and **a** 100 μm , **b** 200 μm , **c** 300 μm surface roughness

Table 3 Effect of surface roughness on performance parameter

Mass flow rate (kg/s)	Head produced (m of water)		
	100 μm	200 μm	300 μm
4	16.88	16.16	15.35
4.65	15.66	14.85	14.06
5.3	14.98	14.37	13.20
5.95	14.70	13.68	13.07
6.6	14.59	13.39	12.44

Fig. 8 Head versus flow rate curve

As shown in performance curve in Fig. 9, numerical results are found to be in good agreement with experimental results. Percentage errors of 2.55–9.89% are recorded for different mass flow rates.

3 Experimental investigation for analyzing effect of coating

3.1 Experimental setup

Experimental setup consisting of two centrifugal pumps of identical rating, one without coating and with coating on casing, is developed for investigation of coating effect on efficiency. Layout of experimental setup is shown in Fig. 10.

Pump 2 is coated with ultra-smooth ceramic-filled adhesive material, while pump 1 is uncoated. The used adhesive material was an ultra-smooth, ceramic reinforced epoxy that provides a high gloss, low friction coating designed to protect against turbulence and abrasion under typical dry service temperatures of -29 – 120 $^{\circ}\text{C}$.

Readings are taken by varying head and recording corresponding change in volume flow rate of fluid and energy extracted by pump for both coated and uncoated pump. Head of the pump is measured from the Bourdon pressure gauge installed at the delivery side of both pumps. Volume flow rate of water is measured by using flow meter

Fig. 9 Experimental and numerical performance curves for 100 μm surface roughness

Fig. 10 Layout of experimental setup

installed in delivery line. Energy extracted by pump can be observed from the set electronic panel.

3.2 Experimental results

From experimental results, the relationship of head with volume flow rate of water and head with efficiency

obtained for both coated and uncoated pump is shown in Figs. 11 and 12, respectively.

From Figs. 11 and 12, it is observed that for all values of flow rates, head produced by coated pump is more than that of uncoated pump. Because of use of coating on casing of centrifugal pump, volume flow rate of water at constant head increases as compared to uncoated pump. Polymer coating produces an ultra-smooth surface that

Fig. 11 Head versus discharge pump characteristics with coating and without coating

Fig. 12 Efficiency versus discharge pump characteristics with coating and without coating

Fig. 13 Comparative power consumption of pump with and without coating

reduces the boundary layer of the pumped fluid and reduces internal turbulence, thus increasing volume flow rate and hydraulic efficiency of centrifugal pump. Surface coating of casing with ultra-smooth ceramic-filled coating

results in efficiency enhancement of 2.11–9.02% for different volume flow rates. Power consumption of both coated and uncoated pump is shown in Fig. 13.

As referred from Fig. 13, power consumption of both coated and uncoated pump is considerably similar. Variation of 1.4–5.45% in power consumption is observed between coated and uncoated pump for different flow rate values.

4 Summary and conclusions

The study reveals effect of surface roughness of casing on centrifugal pump performance. Based on this numerical and experimental work, the following conclusions can be drawn:

1. Surface roughness of casing has considerable effect on pump performance,
2. Surface roughness of casing does not affect flow field inside impeller zone considerably, but it affects pump performance by increasing pressure drop at surface which ultimately results in efficiency drop.
3. Change in surface roughness from 100 to 300 μm results in head difference of 1.53–2.15 m of water for different mass flow rate values.
4. Effect of casing surface roughness is more severe at higher mass flow rate values.
5. Surface coating with adhesive material improves efficiency of centrifugal pump by decreasing surface roughness and solid–liquid interfacial energy.
6. Surface coating of casing with ultra-smooth ceramic-filled coating results in efficiency enhancement of 2.11–9.02% for different volume flow rates.

Compliance with ethical standards

Conflict of interest The author(s) declare that they have no competing interests.

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