Performance Analysis of Flameproof Submersible Pump for Mining Application

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Abstract

This paper seeks to predict and compare the performance of flameproof pumps same design parameters but with different operation stages. Firstly, two pump cases A and B were modelled. Secondly, the shear stress transport (SST) turbulence model was used to solve the 3D Navier Stokes equation. The performance characteristics were obtained and compared with experimental data. The pressure contours are plotted at the design point to delineate its internal flow characteristics. Pump case A was found to require more power for its operation and would hence consume more energy than pump case B. Further analysis shows that pump B is more efficient in operation, but with complex configuration. Lastly, the numerical simulations were validated with experimental data.

Keywords

Flameproof Submersible Pump, Performance, Simulation, Fluid-Structure Interaction, Optimization

1. Introduction

Pumps used on mine site are flameproof, submersible, light weight, low noise, stable in operation, multi-stage and operates with a head capacity of 450 meters [1]. The high head requirement is significant so as to discharge the sewage within the mine to the surface without a need for additional pumps nor reservoirs along the flow path [2].

Africa is blessed with large deposits of mineral resources but for her to attain a robust economic growth, special attention must be given to these natural resources. These large mineral resources when explored would help to drive industrial growth. Mining pumps are necessary on mine sites since they are needed to move fluid from low depth to a higher surface [3]. To effectively improve mining operations, the performance of mining pump must be addressed.

In recent times, several scholars presented numerical and experimental studies on high-speed pumps. Draghici et al. [4] outlined experimental test on centrifugal pump at different speeds. Georgescu et al. [5] examined the characteristic pump curves with two speeds based on EPANET. Rakibuzzaman et al. [6] examined performance of a multistage pump characteristics using inverter controlled speed variable. Sha et al. [7] obtained the internal with centrifugal flow pattern at variable speeds both numerically and via experiments.

Furthermore, the use of computational fluid dynamics (CFD) to optimize and design pump has become a useful analytical tool in evaluating the flow characteristics inside the pump. This method can also be used to obtain fluid flow performance within the pump's flow channels. A number of researchers explored the merits of CFD in pump performance analysis [8-11].

The increasing demand for draining pump used on mine is attributed to the increase and the expansion of mining sites [12]. The flameproof submersible pumps (FSP) and its motor are closely integrated into one unit and commonly drive

operation. FSP should function freely with low noise. However, FSP are failure prone as a result of its design and the harsh working conditions. The average operation time before first failure of FSP is approximately 3,800 hours, hence detrimental to its performance [8].

In view of the above challenges, a thorough assessment of this class of pump is essential. Pump BQS50-450/2-200 and BQS50-450/6-160 referred to as Case A and Case B were selected for this study. The case studies were modelled, simulated and their results verified experimentally.

2. Theoretical Framework

2.1 Governing Equations

The flow interaction within the flame proof pumps was studied by means of numerical computations. The Numerical calculations were executed based on the continuity and Reynolds averaged Navier-Stokes equations for incompressible as presented in equations 2 and 3.

$$\frac{du_i}{dx_i} = 0 \quad (1)$$

$$\rho \left[\frac{\partial \bar{u}_i}{\partial t} + \frac{\partial (\bar{u}_j \bar{u}_i)}{\partial x_j} \right] = \frac{\partial}{\partial x_j} \left[\mu \left(\frac{\partial \bar{u}_i}{\partial x_j} \right) - \rho \overline{u_j u_i} \right] - \frac{\partial \bar{p}}{\partial x_i} \quad (2)$$

2.2 Turbulence Model

Due to the intricate nature of the pumps under investigation, a suitable turbulence model (shear stress transport SST) with a unique feature of blending the transformed k - ω and k - ε equations was selected. SST automatically alternates between both models to address flow problems both along the wall or free regions. Menter [13] and Wilcox [14] proposed SST as an enhancement of k - ω turbulence model. Menter [13] established the function used by Wilcox [14] to enhance SST's sensitivity to free flow streams. Menter [13], further modified k - ω model by the introducing F_1 and $(1-F_1)$ in k - ε equation. the suitability of SST in resolving turbulence in pumps is well documented [15-17]

k - ω equations:

$$\frac{\partial(\rho k)}{\partial t} + \frac{\partial}{\partial x_j}(\rho U_j k) = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_{k1}} \right) \frac{\partial k}{\partial x_j} \right] + P_k - \beta' \rho k \omega \quad (4)$$

$$\frac{\partial(\rho \omega)}{\partial t} + \frac{\partial}{\partial x_j}(\rho U_j \omega) = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_{\omega 1}} \right) \frac{\partial \omega}{\partial x_j} \right] + \alpha_1 \frac{\omega}{k} P_k - \beta_1 \rho \omega^2 \quad (5)$$

Transformed k - ε model:

$$\frac{\partial(\rho k)}{\partial t} + \frac{\partial}{\partial x_j}(\rho U_j k) = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_{k2}} \right) \frac{\partial k}{\partial x_j} \right] + P_k - \beta' \rho k \omega \quad (6)$$

$$\frac{\partial(\rho \omega)}{\partial t} + \frac{\partial}{\partial x_j}(\rho U_j \omega) = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_{\omega 2}} \right) \frac{\partial \omega}{\partial x_j} \right] + 2(1-F_1) \rho \frac{1}{\sigma_{\omega 2} \omega} \frac{\partial k}{\partial x_j} \frac{\partial \omega}{\partial x_j} + \alpha_2 \frac{\omega}{k} P_k - \beta_2 \rho \omega^2 \quad (7)$$

The blending functions F_1 and F_2 are:

$$F_1 = \tanh(\Gamma_1^4) \quad (8)$$

$$F_2 = \tanh(\Gamma_2^2) \quad (9)$$

The turbulent viscosity, μ_t is calculated using:

$$\mu_t = \frac{\alpha_1 k \rho}{\max(\alpha_1 \omega, SF_2)} \quad (10)$$

The k - ω equation for the inner boundary layer prediction is applied using the following coefficients: $\sigma_{k1} = 1.176$, $\sigma_{\omega1} = 2.0$, $\beta1 = 0.075$, $\alpha_1 = 5/9$, $\beta' = 0.09$ [18]. Meanwhile, the k - ε equation for the free-stream flows is solved using the following coefficients: $\sigma_{k2} = 1.0$, $\sigma_{\omega2} = 1/0.856$, $\beta_2 = 0.828$.

2.3 Pump Performance Evaluation

To effectively establish performance for pumps, estimating its overall head as presented in equation (11) is a fundamental requirement [19].

$$H = \frac{P_1 - P_2}{\rho g} \quad (11)$$

where $P(m)$ denotes the pressure head, $\rho(kg/m^3)$ denote fluid density, $g (m/s^2)$ denote the gravitational acceleration subscript 1 and 2 denote inlet and outlet of pump. Efficiency as outlined in equation (12) is another important variable for determining pump performance.

$$\eta = \frac{\rho g Q H}{P} \quad (12)$$

where the P is the power input and is given as:

$$P = T \omega \quad (13)$$

3. Methodology

3.1 Computational Fluid Dynamics

The theoretical equations presented in Section 2 were solved using computational fluid dynamics (CFD). CFD steps are outlined in Figure 1.

Figure 1. Steps involved in Numerical Simulations.

3.2 Pump Model

Two in-built fireproof multi-stage submersible pump BQS50-450/2-200 and BQS50-450/6-160 which would be classified as pump case A and case B respectively are presented in Figure 2 and Table 1. Pump case A and B share similar design specification with head of 450 m, flowrate of 13.9 l/s and rotation speed 2,950. The fluid path of pump cases A and B were modelled using CREO Parametric 7.0.

1. Deep groove ball bearings; 2. Housing; 3. Motor rotor; 4. Angular contact ball bearings;
5. Mechanical seal; 6. Final stage impeller; 7. Guide vane; 8. Head impeller.

Figure 2. Pump Models: (a) Case A; and (b) Case B.

Table 1. Main geometric parameters of the pump

Case	D_j (mm)	D_2 (mm)	b_2 (mm)	β_2 (°)	Φ (°)	z	D_3 (mm)
A	115	393	14	11	132	7	410
B	105	249	22	9	173	4	270

3.3 Mesh Generation

The meshing of the fluid path was executed with ANSYS ICEM by means of hexahedral structural grid using multi-blocks. The intricate pump sections were refined with more grids so as to ensure numerical accuracy. Grid independent test was conducted in line with Celik *et al.* [18] grid C with 1883755 was selected using pump case A. Pump case B was grided with same technique. See Figures 3 and 4 and Table 2 for detailed grid report for each pump component.

Figure 3. Computational Mesh for Pump Case A: (a) impeller (b) volute (c) guide vane w (d) mouth ring gap/cavity.

Figure 4. Computational Mesh for Pump Case B: (a) impeller (b) volute (c) guide vane w (d) mouth ring gap/cavity.

Table 2. Number of main water grids for pumps

Model	Impeller	Guide vane	Volute	Gaps/cavities	Others	total
Case A	959,768	152,564	113,019	193,769	706,084	1,975,291
Case B	539,038	958,390	90,980	17,192	436,191	1,879,272

3.4 Physics Definition

CFX a sub-program under ANSYS was employed to evaluate the flow equations inside the pump. The computational set up was defined as shown in Table 3.

Table 3. Computational Physics Definition

Domain Location	Boundary Type	Mass and Momentum
Inlet Pipe	Inlet	Total Pressure
Outlet pipe	Outlet	Opening
Domain Surface	Wall	No-slip
Turbulence Model		
Model Applied	Standard SST k- ω	
Interface Configuration		
Steady State	Frozen rotor	
Numerical Calculation Solver Control		
No of iterations	1000	
Timestep	1.15×10^{-4} s	
RMS residual target	10^{-5}	
Mesh connection	General Grid Interface (GGI)	

4. Result and Discussion

4.1 Performance Characteristics Curve

External characteristics of the pump cases A and B were analysed and compared using equations 11 and 12. These

characteristics comprehensively and intuitively express the pump performance in form of curves shown in Figure 5. The complex nature of flow in pump is unsteady and requires experiments to authenticate pump performance. As a result, CFD results for pump cases A and B was compared with experimental data provided by pump manufacturer in Figure 5. It was observed from Figure 5 as the flow rises, head decreases but shaft power increases. The data evaluated from CFD is higher than experimental data. The experimental process is prone to friction and volume losses which are not considered in CFD.

Figure 5. Performance Curves Pump Cases: (a) case A (b) case B.

The result depicts that both pumps exhibit similar performance trend. However, Pump case A performs at a higher head than Case B. Therefore, case A is endorsed for high head. On the contrary, the power requirement for pump case B is low, hence more energy would be saved using Pump case B. However, its structural make up is complex to either build or maintain.

4.2 Internal Flow Characteristics

Figure 6 and 7, depicts pressure contours at design flow. It was observed that the lowest pressure was recorded at the inlet region for both pump cases. Gradual upward trend in pressure within pump's flow channel was observed as maximum value was registered at pump outlet. The fluid pressure rises gradually and is equally distributed along the flow circumference. The impeller flow passage, as presented in Figures 6 and 7, accounts for low pressure at the mid-section when viewed radially outward. The impeller's pressure (see Figure 7) increases radially from its mid-section to its maximum value at its edges. The distribution in pressure at the design condition is circumferentially symmetry as presented in Figure 6 and Figure 7. A low-pressure zone was observed at the inlet which means that this zone is prone to cavitation.

Figure 6. Pressure Contour plots: (a) 3D of case A (b) 3D of case B (c) Case A at plane $x=0$ (d) Case B at plane $x=0$.

Figure 7. Pressure Contour plots: (a) blades of Case A (b) blades of case B (c) Impeller of Case A (d) Impeller of Case B.

5. Conclusion

Two pump models for mining operation were modelled using CREO parametric software. Their fluid path was meshed using ANSYS ICEM CFD. The pump cases were separately imported into ANSYS CFX and the Shear Stress Transport turbulent model was used to solve the continuity and momentum equation in the flow channels of the pump. After post processing, the performance of both pumps were analysed and compared with experimental values. The pressure distribution and external characteristics of the two pumps which shows an increasing trend of pressure from the pump inlet to outlet was observed in both pump cases. The analysis shows that pump B is more efficient in operation, but the structure is more complex as compared with pump A. Furthermore, the design, manufacturing and cost implication of pump case B is relatively high.

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