Design and implementation of a crossflow turbine for Pico hydropower electricity generation

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ABSTRACT

This study covered the design, implementation and performance evaluation of a crossflow turbine at various nozzle positions. The chosen blade material was analyzed using ANSYS for stress and deformation degree under the impact of hydraulic jets to ascertain its suitability while in operation. The shaft was analyzed under static and dynamic conditions using ANSYS in order to ensure a non-plastic deformation of the shaft at both conditions. The outcome of this analysis was employed in the harmonic response analysis of the runner shaft. Convergent tests were done for both the blade and runner shaft analysis. An experiment was designed for the evaluation of the crossflow turbine performance using optimal (custom) design tool of response surface methodology and 69 simulations/runs were obtained. The factors considered in the experimental design arc: nozzle distance from the shaft, nozzle height and attack angle. The crossflow turbine was constructed using computed design values for all the machine's parts. The runner blades were positioned specifically at 28° outer blade angle and 90° inner blade angle. The turbine was tested under a water head and flow rate of 6.4m and 0.0042m³/s respectively. The shaft power and efficiency were evaluated using their respective formula. The responses were optimized in order to get the optimum position of the nozzle that would give the best performance of the responses using the two factor interaction (2F1) mathematical models in coded factors, developed for each of the response. The results obtained, proved that low carbon steel material was suitable for the turbine blading and the shaft is safe at both static and dynamic conditions since the induced stresses and deformations never exceeded the permissible range. Also, each of these considered nozzle positions had a significant effect on the responses with the nozzle height and attack angle having a combined effect on the performance of the turbine. The best turbine performance was obtained at lower angle of attack, nozzle distance very close to the runner shaft and at a nozzle height that will actualize greater energy impartation to the upper and lower blade profiles. The developed mathematical models for each response has higher correlation value, suggesting that the models are suitable for predicting the responses at the considered factor levels. An optimal nozzle distance, height and attack angle of 102mm, 413mm and 5° respectively, were obtained. At this nozzle position, the alternator gave an output of 35watts and 6V. When two voltage transformers were employed, it gave 200Volts AC. The turbine can be commercialized on large scale for greater output power using the determined optimal nozzle positions.

1. Introduction

The use of conventional energy sources (fossil fuels) for electricity generation and other applications, has in no small measure caused serious pollution hazards like greenhouse effect, ozone layer depletion and other health related effects. This source of energy is geometrically going to extinction, hence the need channel to interest cleaner and renewable energy sources. Renewable energy sources such as solar, wind, hydro, geothermal, biomass, ocean wave, tides etc., replenish depleted portions in a finite amount of time, through natural reproduction. Although wind and solar energy have gained popularity in the last decades, hydroelectric energy still remains the world's main source of renewable energy, with a global installed capacity of 1000GW, accounting for over 16% of the world's net electricity production and more than 65% of the global power generation capacity from renewable sources [1].

The crossflow turbine provides shaft power from the kinetic energy of a moving fluid. The fluid is supplied to the runner through a rectangular nozzle positioned at a specific angle. The supplied jets on striking the blades, transfers 72% and 28% of its energy to the upper and lower blade profiles at specific angles respectively [2]. Hence, generating double...
torque in the process. Improving the efficiency of the crossflow turbine for increased hydropower electricity output, is an area that has gained the interest of lots of researchers. Olgun [3] modified the runner of the crossflow turbine adding interior guide tubes, designed to collect and guide the cross flow towards the second stage of the runner. Three different guide tubes were tested, and the efficiencies for different positions of interior tubes and openings of the nozzle, were determined. From the investigation, a 5% decrease in efficiency for turbine with the interior guide tube was discovered and a maximum efficiency of 64% was obtained. Maciej and Janusz [4] carried out a computational fluid dynamic analysis on a reaction crossflow turbine, by finite volume approach using Fluent 5.0 software. A certain degree of turbulence was discovered at the outflow system which can be reduced by using a properly designed draft tube, nozzle and guide vane. Zar et al [5] designed a crossflow turbine, with a 260–300kW AC power, 6m head and volumetric discharge of 0.091m³/s. The performance test was conducted by varying the flow rate through the aid of a gate valve as well as the guide vane. A maximum efficiency of 86% was obtained at the middle position of nozzle guide vane with flow rate 0.091m³/s and 260W power. Structural steel was selected for blading after the von-mises stress and deformation analyses using ANSYS. Vincenzo et al [6] optimized the design parameters of the crossflow turbine system using two experimental approaches. The first studied the inlet and outer blade angles, the outer runner diameter, and the shape of the nozzle using a simple hydrodynamic analysis. In the second, the inner runner diameter, the number of blades and their shape were analyzed using computational fluid dynamics (CFD) simulations. The performed analysis gave an optimal runner inner to outer diameter ratio of 0.65, inlet and outlet angles of 38.9° and 90° respectively, spread ratio of 1.5, 22° attack angle and 29.8mm blade radius. The simulation was carried out at a discharge value of 601/s, revolution speed of 757rpm and peripheral velocity of 13.7 m/s, giving an efficiency greater than 80%. Ram and David [7] designed a crossflow turbine and analyzed the system using RANS simulation software, for optimal design parameters. The result gave an 88–90% efficiency range at 0.68 runner inner to outer diameter ratio, inner and outer blade angles of 38.9° and 90° respectively, 22° attack angle, 61.5° central blade angle and 1.5 flow spread ratio of.

This research work is focused on the design and implementation of a crossflow turbine plant for Pico hydropower electricity generation. The nozzle height, nozzle distance from the runner shaft and attack angle were optimized for increased energy generation, with the outer and inner blade angles of the turbine set at 28° and 90° respectively. The effect of these design parameters on the efficiency, shaft power and runner shaft speed of the crossflow turbine were also studied.

2. Materials and method

2.1. Design of the crossflow turbine

The parameters employed in the crossflow turbine design are stated in Table 1.

2.1.1. Runner wheel design

The runner of the crossflow turbine consists of the blades positioned at specific angles and the shaft for power transmission to the alternator unit. The ratio of the inner to the outer diameters of the wheel is an imperative parameter of the crossflow turbine design.

2.1.1.1. The outer diameter of the runner wheel (D₁). Mockmore and Merryfield [2] derived an equation relating the outer diameter, flow head and the revolitional speed of turbine as:

\[ D_1 = 39.85 \sqrt{\frac{H}{N}} \]  \hspace{1cm} (1)

2.1.1.2. Inner diameter of the runner wheel (D₂). According to Mockmore and Merryfield [2], the ratio of D₂ to D₁ is 0.66.

\[ \frac{D_2}{D_1} = 0.66 \] \hspace{1cm} (2)

2.1.1.3. Radial rim width (a). This is the difference between the radius of the outer and of the inner wheel diameters of the turbine. The radial rim width represents the width of the blade. Mockmore and Merryfield [2] gave the computational formula as:

\[ a = 0.17D_1 = r_1 - r_2 \] \hspace{1cm} (3)

2.1.2. Nozzle design

The geometry of the nozzle and runner strongly affects the efficiency of the machine. From literature, the inlet angle or attack angle (α) of water jets to the outer blades varies.

2.1.2.1. Absolute velocity of water before striking the blades. The absolute velocity of the water from the nozzle is given by Eq. (4). The velocity coefficient Cᵥ accounts for losses through the nozzle and its value is less than unity (about 0.98).

\[ V_1 = Cᵥ \sqrt{2gH} \] \hspace{1cm} (4)

2.1.2.2. Nozzle width/Throat (B). From the researches of Aziz and Desai [8] and Nakase et al [9], the flow stream spread \( \frac{W}{B} \) was observed to have a positive effect on the turbine performance, with an increase in efficiency up to a spread value of 1.5.

\[ \frac{W}{B} = 1.5 \] \hspace{1cm} (5)

2.1.2.3. Distance of jet from centre of shaft (y₁) \[ y_1 = (0.1986 - 0.945k)D_1 \] \hspace{1cm} (6)

\[ k = 0.087 \]

2.1.2.4. Distance of jet from inner periphery of wheel (y₂) \[ y_2 = (0.134 - 0.945k)D_1 \] \hspace{1cm} (7)

2.1.2.5. Thickness of jets

Jet thickness, \( s_n = \frac{\text{Area of jet}}{\text{Nozzle width}} = \frac{Q/V_1}{B} \) \hspace{1cm} (8)

2.1.3. Blade design

The outer and inner blade angles, blade thickness, radius of the blade, number of blades, pitch circle diameter, blade spacing, inlet discharge angle and central angle of blade were the blade design parameters to be considered.

2.1.3.1. The outer and inner blade angles. Mockmore and Merryfield [2], gave the relationship between the attack angle (α) and outer blade angle (β₁) as:

\[ \tan \beta_1 = 2 \tan \alpha \] \hspace{1cm} (9)

| Table 1. Fixed design parameters used in the crossflow turbine design. |
|-------------------------------------------------|---------|
| Design Parameters                             | Specifications |
| Pressure head                                 | 5.2m    |
| Volumetric discharge                          | 0.0015m³/s |
| Turbine shaft speed                           | 395rpm  |
| Water pressure                                | 50.83kPa |
| Penstock diameter                             | 0.025m  |
The positions of the outer and inner blade angles on the outer and inside diameters of the runner wheel are vividly shown in Figure 1. Design variables of the crossflow turbine such as: outer diameter \( D_1 \), inner diameter \( D_2 \), attack angle \( \alpha \), jet thickness \( S_0 \), outer blade angle \( \beta_1 \), inner blade angle \( \beta_2 \), central angle of the blade \( \delta \) and blade radius \( P_b \) are all shown on Figure 1. According to the works of Aziz and Desai [8], Andrade et al [10], Mockmore and Merryﬁeld [2], Aziz and Totapally [11], the inner blade angle \( \beta_2 \) was suggested to be \( \frac{\pi}{2} = 90^\circ \) providing a radial direction to the relative outlet velocity inside the runner at the inner diameter \( D_2 \), to the fluid in the inner part of the runner. Also only the energy of the rotating system, can be recovered during the next blade crossing.

2.1.3.2. Inlet discharge angle \( \lambda \). Experimental studies of Fiuzat and Akerkar [12], as well as that of Nakase et al [9] showed that the inlet discharge angle influences crossﬂow efﬁciency and a value of \( 90^\circ \) is more efﬁcient than \( 120^\circ \) suggested by others.

2.1.3.3. Central angle of the blade \( \delta \). This is the angle between the outer and inner sides/periphery of the blade. This angle aids in the actualization of the crossﬂow process. The blade position on the runner rims depends principally on it. From the formula developed by Mockmore and Merryﬁeld [2], shown in Eq. (10):

\[
\tan \left( \frac{\delta}{2} \right) = \frac{\cos(\beta_1)}{\sin(\beta_1 + \frac{\pi}{2})}
\]  

(10)

2.1.3.4. 3d radius of the blade \( P_b \). It is derived from the equation of Mockmore and Merryﬁeld [2]:

\[
P_b = 0.326r_1\]

(11)

2.1.3.5. Pitch circle diameter \( P \). This is the diameter of the circular shape from which the blades were cut-off from.

\[
P = 2P_b
\]  

(12)

2.1.3.6. Thickness of the blade. The blades were cut-out from a hollow low carbon steel pipe of thickness 5mm and diameter 75mm which corresponds to the pitch circle diameter. The cutting was done using the calculated blade angles.

2.1.3.7. Positioning of the blades. The leading edge of the blade was positioned at an outer blade angle of 28° to the tangent at the contact point while the trailing edge of the blade is radial. This is clearly illustrated in Figure 1b.

2.1.3.8. Blade spacing \( s \). Proper blade spacing allows water to strike on the blades optimally for maximum thrust production. The blade spacing depends on the number of blades used on the runner. Mockmore and Merryﬁeld [2] suggested the formula for finding the blade spacing as:

\[
s = \frac{k \times D_1}{\sin(\beta_1)} (k \text{ ranges from } 0.075 - 0.10)
\]  

(13)

2.1.3.9. Number of blades \( n \). The number of blades employed in the crossﬂow turbine design has an effect on the efﬁciency of the turbine. Relatively low number of blades, results to results to high energy losses due to low water impingement area on the runner, and vice versa. Hence, a good number of blades should be used in other to raise the thrust motion of the runner and also the efﬁciency of the plant. Mockmore and Merryﬁeld [2] suggested the formula for ﬁnding the number of blades as:

\[
n = \frac{\pi D_1}{s}
\]  

(14)

2.1.4. Central angle determination of the jet path inside the wheel \( \phi \). This is an imperative parameter of the crossﬂow turbine design. The angle helps in the actualization of the crossflow action of water in operation. Mockmore and Merryﬁeld (2) gave the computational formulae of \( \phi \) to be:

\[
\tan \left( \frac{\phi}{2} \right) = \frac{\pi \sin(\beta_1)}{\frac{\pi}{2} r_1}
\]  

(15)

2.1.5. Theoretical shaft power of the turbine \( P_s \). According to Mockmore and Merryﬁeld [2], the theoretical shaft power is given as;
\[ P_t = \left( \frac{u_0 U_1}{g} \right) \left( V_1 \cos \alpha_1 \right) \left( 1 - \frac{\varphi \cos \beta_1}{\cos \delta_1} \right) = \omega T \]  
\( \varphi = 0.98 \) (an empirical coefficient due to friction effect with value less than unity).

2.1.6. Revolutational and angular speed of the shaft

\[ U_1 = \frac{\pi d_1 N}{60} \]  
\( \omega = \frac{2\pi N}{60} \)

2.1.7. Power transmission to the alternator unit using an open belt drive system

This is obtained from speed ratio formula,

\[ N_2 = \frac{d_1}{d_2} \]  
\( N_1 \) is speed of the driven; \( d_1 \) is diameter of the driven pulley, \( d_2 \) is diameter of the driver pulley.

2.1.8. Theoretical efficiency of the turbine

\[ \eta_t = \frac{\text{Shaft power}}{\text{Water power}} \]  
The summary of the computed design variables employed in the design implementation of the crossflow turbine are shown in Table 2.

| S/N | Parameters | Specifications |
|-----|------------|----------------|
| 1   | The outer diameter of the runner wheel \( (D_1) \) | 230mm          |
| 2   | The inner diameter of the runner wheel \( (D_2) \) | 151.8mm        |
| 3   | Radial rim width \( (\alpha) \) | 39.1mm         |
| 4   | Outer blade angle \( (\beta_1) \) | 28\(^\circ\)     |
| 5   | Inner blade angle \( (\beta_2) \) | 90\(^\circ\)    |
| 6   | Inlet discharge angle \( (\delta) \) | 90\(^\circ\)    |
| 7   | Central angle of the blade \( (\delta) \) | 76\(^\circ\)    |
| 8   | Radius of the blade \( (r) \) | 37.49mm        |
| 9   | Blade thickness \( (t) \) | 5mm            |
| 10  | Pitch circle diameter \( (p) \) | 75mm           |
| 11  | Absolute Jet velocity \( (V_1) \) | 9.9 m/s        |
| 12  | Nozzle width \( (B) \) | 76.67mm        |
| 13  | Distance of jet from shaft’s centre \( (y_1) \) | 26.77mm        |
| 14  | Distance of jet from inner periphery of wheel \( (y_2) \) | 11.313mm       |
| 15  | Blade spacing \( (s) \) | 36.74mm        |
| 16  | Number of blades \( (n) \) | 20             |
| 17  | Central angle of jet’s path inside the wheel \( (\delta) \) | 101.3\(^\circ\) |
| 18  | Peripheral/tangential speed of the wheel \( (U_1) \) | 4.76 m/s      |
| 19  | Angular speed of the shaft \( (\omega) \) | 41.4 rad/s     |
| 20  | UCP Pillow block bearing | 25mm bore diameter |
| 21  | Shaft diameter | 25mm           |
| 22  | Jet thickness \( (a) \) | 2mm            |
| 23  | Cross-sectional area of the penstock | 804.2mm\(^2\) |
| 24  | Length of runner \( (w) \) | 115mm          |
| 25  | Diameter of the driver pulley \( (d_1) \) | 230mm          |
| 26  | Diameter of the driven pulley \( (d_2) \) | 64mm           |
| 27  | Theoretical Rotational speed of the alternator \( (N_2) \) | 1420 rpm       |
| 28  | Design Volumetric discharge | 0.0015m\(^3\)/s |

2.2. Material selection

Material selection is a step in the process of designing any physical object. In the context of product design, the main goal of material selection is to minimize cost while meeting product performance goals. The systematic selection of the best material for a given application begins with the properties and cost of the materials. On this note, the material type used for the implementation of the designed crossflow turbine plant was selected on the basis of availability, applicability, machinability and cost. This is because, the crossflow plant will not function optimally once the deformation and stress factors on the blade exceed the designed range. On this basis, the blade material was selected on the basis of high strength and low cost using the Granita material selector software 2009 version. The selected material was further analyzed for deformation and stress using ANSYS workbench 15.0 software. Also, ANSYS workbench 15.0 was used to carry out static and dynamic analysis on the runner shaft in order to ensure its safety during operation. AutoCAD 2017 was employed in modelling the crossflow turbine blade used in the analysis.

2.2.1. Blade material selection

The selection of candidate materials suitable for a particular application are based on the performance and desirability factors of the materials. According to Ashby [13], the performance matrix of a material is related as:

\[ P = \left( \begin{array}{c} \text{functional requirement, } f \\ \text{Geometric parameters, } G \\ \text{material properties, } m \end{array} \right) \]

where \( P \), the performance matrix describes some aspects of the performance of the component. Optimum design is the selection of a material and geometry that maximizes or minimizes \( P \), according to its desirability or otherwise. The blade material is needed to be of high strength and minimal cost. The blade material must be ductile in other to withstand the hydraulic load impact of 50.83KPa. Hence,

\[ \sigma_y = \frac{E e}{A} \]  
\( A = \frac{F}{\sigma_y} \)

The mass of the blade is given as:

\[ m = \rho At \]

Substituting Eq. ,

\[ m = F \frac{\rho}{\sigma_y} \]

where; \( \sigma_y \) is the elastic yield strength, \( E \) is the young’s modulus, \( e \) is the elongation, \( m \) is the mass of the blade, \( \rho \) is the density, \( A \) is the area, \( t \) is the material thickness and \( F \) is load.

Therefore, the mass is minimized by selecting material with the highest value of:

\[ M_1 = \frac{\sigma_y}{\rho} \]  
For cost minimization, \( M_2 = \frac{\sigma_y}{C_{\gamma\rho}} \)

The selected blade material was further analyzed with ANSYS to ensure the stress and deformation degree do not exceed the permissible range. The extreme left and right sides of the chosen blade material were fixed since they are welded to the two runner plates and rotate as one assembly.
2.2.2. Stress and deformation analysis of the runner shaft

The runner shaft was analyzed statically and dynamically under loads for its safety while in operation and plastic deformation that will affect measured experimental parameters. The induced stress on the shaft obtained from the static analysis using ANSYS was also validated using analytical method- Euler/Bernoulli’s beam equation.

2.3. Material properties of the shaft

The selected material is 304 annealed austenitic stainless steel with the following properties: Young’s modulus (190GPa), Density (7750 kg/m³), Tensile yield strength (290MPa), Ultimate tensile strength (580MPa), Compressive yield strength (207MPa) and Compressive ultimate strength (310MPa).

2.4. Static analysis of the runner shaft

Static load analysis was conducted to determine the level of deformation and induced stress on the runner shaft. This is to ensure the safety of the shaft under static load. The boundary conditions are: weight of the driver pulley (6.87N), weight of the runner (113.8N), weight of the flywheel (58.86N), self-weight of the shaft (considered automatically by ANSYS as 23.54N) and fixed supports at the two bearings.

2.5. Dynamic analysis of the runner shaft

A dynamic analysis is one in which the results of a modal analysis are used with a known spectrum to calculate displacements and stresses in the model. The modal analysis was done and the results were employed in the harmonic response analysis of the shaft. The boundary conditions employed are: weight of the driver pulley (6.87N), weight of the runner (113.8N), weight of the flywheel (58.86N), self-weight of the shaft (considered automatically by ANSYS) (23.54N), moment (1641.3 N-mm), vibrational frequency (0.002171Hz), acceleration (9810 mm/s²) and fixed supports at the two bearings.

The loadings on the shaft in all the analysis were converted to pressure loading by dividing the force with the area of the load section being considered. This was done in order to avoid stress singularity on the runner shaft. The materials used in constructing the designed crossflow turbine are stated in Table 3.

2.6. Virtual CAD model of the crossflow turbine plant

The 3D CAD model showing the components of the crossflow turbine designed with Autodesk Inventor software 2014 version is shown in Figure 2.

2.7. Crossflow turbine design implementation

The turbine was constructed using the calculated design parameters of the crossflow plant at the technical workshop of Eagle and Hetch (W.A.) Company, Enugu, Nigeria. The design was meticulously followed in developing the functional turbine. Figure 3 shows the constructed parts and other needed components of the turbine and the assembled turbine.

2.8. Experimental testing procedures of the fabricated crossflow turbine

The aim of this study is to obtain the optimal nozzle position, considering nozzle positional factors of: nozzle height, nozzle tip distance from the runner shaft and the attack angle/inlet angle of water jets

| S/N | Components           | Material Type               |
|-----|----------------------|-----------------------------|
| 1   | Runner blades        | Low carbon steel            |
| 2   | Runner wheel         | Low carbon steel            |
| 3   | Machine frame        | Low carbon steel            |
| 4   | Rotor/Shaft of the runner | 304 annealed austenitic stainless steel |
| 5   | Nozzle               | Low carbon steel            |
| 6   | Penstock             | Galvanized steel pipe       |
| 7   | Water tank           | Low carbon steel            |

Table 3. The Major Crossflow Turbine Components and the Material Type Used for it.
2.8.2. Description of the test site/Nozzle's positional varying technique

The turbine was tested under a water head of 6.4m and volumetric flow rate of 0.064m³/s. The experimental design was used to vary the positions of the nozzle. The nozzle was positioned at a particular height, tip distance from the runner shaft and attack angle according to the developed experimental runs. The turbine was operated at this nozzle position and a digital tachometer was used to measure the rpm of the runner shaft. The measured rpm was used in computing the runner shaft power and efficiency of the crossflow turbine for that particular nozzle position using Eqs. (16) and (20) respectively. The performance of the turbine was calculated for each of the 69 runs following the described method. At the end of the experiment, optimization was done in order to get the best nozzle positional factor combination that will yield higher response values.

3. Results and discussions

3.1. Material selection result for the runner blade using Granta software

The selection of blade material was on the bases of strength and cost, for the crossflow turbine runner blade that will withstand the pressure load of water jets striking on it with little or no induced stress and deformation. The candidate materials for the blade based on strength were sourced from the material families such as composites, foams, non-technical ceramics, technical ceramics, natural materials, polymers, metals and alloys. The selection results based on the performance indices of the material shows that only few elements from the family of metals and their alloys were chosen for the minimal mass and high strength blade design. Table 4 gives the summary of the candidate materials selected for the runner blade of the crossflow turbine.

Low carbon steel was employed for the turbine blading owing to the factors of easy accessibility, cost and it is still among the suitable materials for the design implementation.

3.2. Blade Material's stress and deformation analysis using ANSYS 15.0

The deformation and induced von mises stress levels on the runner blade under the hydraulic load effect of 50.83kPa are shown on Figures 4 and 5 respectively.

The level of deformation produced on the turbine blade under the impact of a water pressure of 50.83kPa is shown in Figure 4. The inner view of the blade in which the water jet strikes revealed that the maximum deformation (0.0000043249m) on the blade occurred at the central blade axis. This is so because the water jets profile from the nozzle points centrally on the blade before its dissemination towards the ends of the blade. A zero deformation was observed at the ends of the blade due to the reduction of the water pressure as it extends to the outer periphery of the blade. In all, the maximum deformation produced on the blade do not exceed the permissible deformation boundary of 26–47% for low carbon steel materials.
Table 4. Candidate materials for high strength and minimum blade mass design.

| S/N | Material Type          | Density (Kg/m³) (ρ) | Price (USD/Kg) (Cₚ) | Yield Strength (MPa) (σₚ) | M₁ = σₚ/ρ (× 10⁶) | M₂ = σₚ/Cₚρ |
|-----|------------------------|---------------------|---------------------|--------------------------|-------------------|----------------|
| 1   | Stainless steel        | 7950                | 6.85                | 585                      | 0.0745            | 10875.91       |
| 2   | Low carbon steel       | 7950                | 0.67                | 322.5                    | 0.0411            | 61343.3        |
| 3   | Low alloy steel        | 7950                | 0.85                | 950                      | 0.1210            | 142352.9       |
| 4   | Nickel-based superalloys | 8200              | 29.70               | 1100                     | 0.1341            | 4516.7         |
| 5   | Nickel-chromium alloy  | 8400                | 30.70               | 412.5                    | 0.0491            | 1599.3         |
| 6   | Tungsten carbide       | 15600               | 23.90               | 442.5                    | 0.0284            | 1188.3         |
| 7   | Bronze                 | 8750                | 3.74                | 300                      | 0.0343            | 9171.1         |

Figure 4. Total deformation produced on the runner blade.

Figure 5. Equivalent (Von-mises stress) induced on the blade.
The von-mises stress values for the blade under the hydraulic load effect is shown in Figure 5. It is evident that the maximum stress occurred at the tip ends of the blade with a value of 12.9MPa while the minimum stress value of 85078Pa was observed along the central blade axis. The maximum stress occurrence was as a result of stress concentration at the blade tip ends. Low carbon steel material is thus safe for the design since the permissible yield strength value of 322.5MPa for low carbon steel material was not exceeded.

3.3. Static analysis result for the runner shaft

The results for the static analysis carried out on the runner shaft of the crossflow turbine to ascertain its safety while in operation are shown on Figures 6 and 7. The maximum stress effect was observed at points immediately before the deflected position of the runner shaft as shown in Figure 6. This observation is as a result of load concentration at those points on the shaft. The load severity acts on the position of the shaft where the flywheel and runner wheel/blade assembly are situated, which thus induces the maximum stress effect at shaft’s end after the flywheel and runner assembly positions. A maximum stress of 4.8091MPa was noticed at these positions as seen in the ANSYS result. Also, the maximum stress on the shaft evaluated using the Euler-Bernoulli’s/classical beam theory gave a value of 5.748MPa with a percentage error deviation of 16.33% from the ANSYS result. The induced stress on the shaft propagates along the shaft axis with high values within the load concentrated region and minimum value at shaft ends. This stress values suggests a non-plastic deformation of the shaft under static condition since the maximum induced stress of 4.8091MPa does not exceed the compressive yield strength value of 207MPa for stainless steel materials.

In addition to the static shaft analysis, Figure 7 shows the deformation degree produced on the shaft under static conditions. The highest deformation was observed at the shaft axis where the runner assembly was mounted while a zero deformation occurred at the ends of the shaft. These observations are as a result of load impact on the shaft. The shaft position where the maximum deformation occurred was the region carrying the highest load while the other without any deformation was not loaded. The maximum deformation value of 0.0086255mm is not significant enough to cause shaft failure. Hence, the shaft is safe under static conditions of operation.

3.4. Dynamic analysis result of the runner shaft

The result of the dynamic analysis carried out on the runner shaft of the turbine to ensure its safety while operating are shown on Figures 8 and 9. The stress induced on the runner shaft while in operation is shown in Figure 8. Maximum stress of value 4.5609MPa occurred at points immediately before the deflected position of the shaft where the load...
concentration is higher. While the minimum induced stress was observed at the free-load ends of the shaft. As the maximum stress induced on the shaft while rotating does not exceed the compressive yield strength of 207MPa for stainless steel material hence, the runner shaft is safe and will not undergo plastic deformation.

Figure 9 shows that the maximum deformation on the runner shaft occurred at the shaft axis where the flywheel and the runner wheel/blades assembly were mounted, constituting the highest loading on the shaft body. The maximum deformation value of 0.0081785mm on the shaft proves that the shaft is ductile and will not undergo plastic deformation.

3.5. Performance evaluation of the crossflow turbine

The turbine was designed for a head of 17ft (5.2m approximately) and a volumetric discharge of 0.0015m$^3$/s but was tested with a head of 6.4m and a flowrate of 0.0042. The performance of the turbine was measured at varying positions of the nozzle as developed in the experimental design. The runner shaft RPM was measured at various nozzle positions and the crossflow turbine’s efficiency and shaft power were computed using their respective formula. The responses were optimized to obtain the optimal nozzle position for better performance of the turbine when the blades are positioned at an outer and inner angles of 28°/C14 and 90°/C14 respectively. The optimization graphs are shown in Figures10, 11, 12, and 13.

Figure 10a shows that the highest desirability value that will yield an optimum efficiency, shaft power and runner shaft speed can be obtained at a nozzle distance of 102mm. Figures 11a and 12a indicated that the highest desirability value that will yield an optimum efficiency, shaft power and runner shaft speed can be obtained at nozzle height of 413mm and an attack angle of 5°/C14 respectively. Figure 13a reveals that the highest desirability value that will yield an optimum efficiency, shaft power and runner shaft speed can be obtained at a nozzle height of 413mm and the corresponding attack angle. Furthermore, Figure 12 shows a remarkable drop in the values of desirability and responses at an attack angle of 30°. This occurred because the water jets from the nozzle does not participate actively in the second stage blades, which is the lower blade profile. The
responses; efficiency, shaft power and runner shaft speed decrease in value as the nozzle positional factors shift from these optimal position of the nozzle.

From Table 5, numerical optimization performed using response surface methodology, indicated that the plant will function optimally at a nozzle distance of 102mm, nozzle height of 413mm and at a very low angle of attack of 5° for runner blades positioned specifically at 28° outer blade angle and 90° inner blade angle. The mathematical models developed, predicted efficiency, shaft power and runner shaft speed values at the stated optimal nozzle position to be 96.233%, 253.739watts and 464.178rpm respectively.

4. Conclusion

This study was centered on the design and implementation of a crossflow turbine for small scale utilization and the effects of some design factors: the nozzle height, nozzle distance from the runner shaft and the attack angle on the turbine performance characteristics; efficiency, shaft power and runner shaft speed determined. These factors have revealed a high level of significant effect on the operational responses of the turbine. Each of these factors has its individual effect on the responses and a combined effect of the nozzle height and attack angle was very significant in the performance of the turbine. The turbine performed optimally at lower angle of attack, nozzle distance very close to the runner shaft and at a nozzle height that will actualize greater energy impartation to the upper and lower blade profiles. These operational conditions thus contributed to the minimization of energy losses while imparting the runner blades of the turbine. The optimal factor position of nozzle distance from runner shaft, nozzle height and attack angle obtained using response surface optimization technique are 102mm, 413mm and 5° respectively. The optimal values of efficiency, shaft power and runner shaft speed were predicted to be 96.233%, 253.739watts and 464.178RPM respectively at the stated nozzle position. At this nozzle position, the alternator gave an output power of 35watts and 6V AC.
Figure 11. (a) Plot of desirability values against nozzle height (b) plot of efficiency against nozzle height. (c) plot of shaft power against nozzle height. (d) plot of runner shaft speed against nozzle height.

Figure 12. (a) Plot of desirability values against attack angle (b) plot of efficiency against attack angle. (c) plot of shaft power against attack angle. (d) plot of runner shaft speed against attack angle.
When two voltage transformers were employed, it gave a voltage of 200 Volts AC. A car alternator that was modified to output AC by removing its rectifier circuit and exciting the field with two (2) 1.5V batteries was employed for the output measurement. Hence, these factors are quite significant for optimal response actualization of the crossflow turbine.

Declarations

Author contribution statement

Achebe C. H.: Conceived and designed the experiments; Wrote the paper.
Okafor O. C.: Performed the experiments; Analyzed and interpreted the data; Contributed reagents, materials, analysis tools or data; Wrote the paper.
Obika E. N.: Analyzed and interpreted the data; Wrote the paper.

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Competing interest statement

The authors declare no conflict of interest.

Additional information

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