



DESIGN AND SIMULATION OF A PICO HYDROELECTRIC TURBINE SYSTEM

Mogaji, T. S, Fasasi, S. T, Ogundairo, A. O. and Oluwagbemi, I. A.

Department of Mechanical Engineering, School of Engineering and Engineering Technology, Federal University of Technology, P. M. B. 704, Akure, Ondo State, Nigeria.

Correspondence: tsmogaji@futa.edu.ng

Mogaji, T. S, Fasasi, S. T, Ogundairo, A. O. and Oluwagbemi, I. A. (2022): Design and Simulation of a Pico Hydroelectric Turbine System. *Journal of Engineering and Engineering Technology* /16(1), 132-139

Received: 15-01-22

Accepted: 29-02-22

Abstract

Hydroelectric power is both an efficient and reliable form of a clean source of renewable energy. It can be an excellent method of harnessing renewable energy from small rivers and streams. A Pico hydroelectric power system of a cross-flow turbine type is designed and simulated in this study. Model parameters such as runner diameter, runner length, blade spacing, radius of blade curvature, shaft diameter, water jet thickness, turbine speed, turbine power and the number of blades that play major role to achieve the turbine maximum efficiency were determined theoretically. The designed Hydro-electric power system's maximum efficiency was estimated to be 87% with a head of 6390 mm at a flow rate of $0.089 \text{ m}^3/\text{s}$. Numerically, analysis of the designed power generation system was carried out using (structural analysis and design) STAAD pro connect software and ANSYS tool. The performance evaluation results using the STAAD pro software for the system water reservoir sitting stanchion simulation indicated that the stanchion will not fail while carrying the load. Similarly, the numerical analysis of the designed cross-flow turbine type in this work using the ANSYS tool revealed that the shaft will be able to withstand deformation due to torsion when subjected to a maximum torque of 51.46 Nm. Hence the designed hydroelectric power system will be efficient to produce a 5 kVA electrical power

Keywords: Hydropower, turbine system, cross-flow, design, numerical simulation.

1. Introduction

The need for a reliable, portable and easily accessible hydropower system is of great importance for every home, office and industry. These characteristics are very essential for human comfort so as to improve electricity supply in these areas of necessity. Hence the aid of portable hydropower is required in this area where dam, ocean, river flow, canals is not accessible by the use of storage tank as a source of water supply. Fraenkel *et al.* (1991) define hydropower or hydroelectricity as the conversion of energy from flowing water into electricity. It is considered a renewable energy source because the water cycle is constantly renewed. According to Adejumbi, (2011), water in nature is considered a source of power when it can perform useful work, particularly turn water wheels and generate electricity at a rate such that the development of power can be accomplished in a more efficient and economical way Hydropower is the largest contributor of all renewable energy sources and accounts for 6.7% of worldwide electricity production (Bilal, 2013). Anyaka, (2013) pointed out that a hydroelectric power system operates to convert the potential energy of water to mechanical energy for electricity generation through a turbine. Hence it is very expedient to consider and examine the suitable type of turbine while designing the hydroelectric power system. Adejumbi, (2011) also pointed out that a

micro-hydroelectric power plant needs both the flow of water and the head (drop in height) to efficiently produce water. According to David, (2013) considering the various types of turbines, the cross-flow turbine happens to be the suitable type of turbine for a micro-hydroelectric power system because it is characterized by low head and high pressure. The cross-flow turbine comprises three major components which include the Runner, Nozzle and buckets. The runner is a circular rotor having two side walls in which the blades are attached along the periphery towards the center through which water flows. The nozzle, on the other hand, increases the speed of water flow in a cross flow turbine, while the buckets decrease water flow resistance in cross-flow turbines. The type of turbine used determines the efficiency of the hydropower system. The cross-flow hydraulic turbine is gaining popularity in low head and small water flow rate establishments, because of its simple structure and ease of manufacturing in the site of the power plant (Mrudang and Nirav, 2016). According to Abubeker, (2018), in the cross-flow turbine, water flows through the cross-flow turbine blades twice. In the first stage, water flows from the outside of the blades to the inside; in the second stage, the water flows from the inside back out.

Gutu and Auch, (2017) posited that in designing and developing efficient and productive hydropower

system, the Cross-flow turbine depends on several parameters including runner diameter, runner speed, turbine power, water jet thickness, blade spacing, number of blade curvature, and radius of blade curvature, angle of attack and angle of exit. Akerkar, (1989) conducted an experimental study of the performance of the cross-flow turbine, and concluded that the flow pattern inside the cross-flow turbine

runner is concave when viewed at either the slant or horizontal position. He also reveals that the horizontal position is less efficient than the slant position. The work of Walseth, (2009), investigates the Flow through runner of a Cross-Flow Turbine. He conducted two types of researches to evaluate the flow pattern and the transfer of torque through the runner. In the experiment, the efficiency was first measured which result in 78.65% at 5-meter head. A report from the study of IRENA, (2012) revealed that, electricity generated from hydropower can be placed on the grid faster than any other energy source. Hydropower schemes often have significant flexibility in their design and can be designed to meet base-load demands with relatively high-capacity factors, or have higher installed capacities and a lower capacity factor, but meet a much larger share of peak demand additionally. Small hydropower systems can be used in isolated areas off-grid or mini-grids.

In this study, a small-scale hydropower system capable of 5kW electricity generation is designed considering cross-flow turbine type. The system efficiency was investigated numerically while taking into consideration important factors such as stanchion for the sitting water reservoir tank, water head and flow rate in order to maximize the system power output.

Materials and Methods

The materials and methods adopted for the design of hydropower systems of cross-flow turbine type were considered in this section. The materials required for the fabrication of the turbine are commercially available and affordable.

Design Calculation

Determination of Net Head (H_n)

Net head is the gross head minus all hydraulic losses except those chargeable to the turbine. As claimed by Razak *et al.* (2010), it is the pressure at the bottom of the pipeline when water is flowing to the turbine and will always be less than the gross head measured due to energy losses within the pipeline. The Net head is obtained using Equation 1 extracted from the study of Abubeker, (2018).

$$H_n = 0.94H_g \quad (1)$$

is the gross head which is the vertical distance between water surface level at the intake and turbine, H_g is the system net head

Determination of the System Water Tank Flow Rate (Q)

The volumetric flow rate is the volume of fluid that passes per unit time, estimated as stated in Equation 2.

$$Q = V_r A_r$$

$$Q = V_r A_r \quad (2)$$

where V_r is water flow velocity and A_r is the cross-sectional area. Since, the cross-sectional area of the turbine inlet is equal to the cross-sectional area of the inlet pipe. Considering 0.1m pipe diameter in this study, the cross-sectional area of the inlet pipe is obtained as 0.008 m². The higher the speed of the turbine, the better the efficiency will be for its power output. This behaviour is known to be related to the water flow rate from the storage tank. The water flow rate is directly proportional to the water velocity. Thus, assuming there is the pressure difference between pressure inlet and outlet, the water flow velocity is obtained as follows using relations extracted from Abubeker, (2018) which is stated as Equation 3.

$$V_r = (2 \times g \times H_g)^{1/2} \quad (3)$$

Thus, water flow velocity (V_r) is estimated as $4.2945H_g^{1/2}$

Therefore, the water flow rate of the tank using Equation 2 is estimated as $33.73 \times 10^{-3} H_g^{1/2} \text{ (m}^3/\text{s)}$

Determination of Turbine Power (P_t)

The electrical power of the turbine in this study is the work done by the turbine per unit time calculated using Equation 4 adopted from the study of Abubeker, (2018).

$$P_t = \eta \rho g Q H_g \quad (4)$$

where P_t is the expected turbine power output of the system, that is 5000 (W); η is the efficiency of the turbine; ρ is the density of water (kg/m³); g is the acceleration due to gravity (m/s²); Q is the volumetric flow rate of the system (m³/s); H_g is the gross head of the system (m). The term gross head H_g in Equation 5 is obtained as follow:

$$H_g = \frac{P_t}{\rho g Q \eta} \quad (5)$$

$$H_g = \frac{5000}{0.94 \times 1000 \times 9.81 \times 0.872 \times 33.73 \times 10^{-3}}$$

$$H_g = 6.936 \text{ (m)}$$

Taking $H_g = 6.936 \text{ (m)}$ from Equation 1, the system net head (H_n) Similarly, the system water flow velocity (V) to achieve the expected turbine power output (P_t) as 5kVA is estimated using Equation 6 as follows:

$$\begin{aligned} \text{Velocity (V)} &= 4.2945 H_g^{1/2} = \\ 4.2945(\sqrt{6.936}) & \quad (6) \\ V &= 11.31 \text{ (m/s)} \end{aligned}$$

Thus, to obtain the targeted 5kVA turbine power output (P_t) in this study, the system flow rate (Q) is obtained as

using Equation 7:

$$Q = A_r \times 4.2945 Hg^{1/2} \quad (7)$$

Determination of Turbine Efficiency

In order to achieve the reliability of the designed turbine, its maximum efficiency is obtained in this study. The maximum turbine efficiency is calculated by adopting Equation 8 from Abubeker, (2018).

$$\eta_t = 0.5 \times C^2 \times (1 + \psi) \times (\cos \alpha)^2 \quad (8)$$

where: ψ Blade roughness coefficient (0.98); C is Nozzle roughness coefficient (0.98); α is attack angle. For maximum turbine efficiency, the attack angle (α) should be kept as small as possible. The angle of attack is the angle between the reference line and the oncoming flow. The angle of attack (α) considered in this study is taken as 16° as adopted from the study by Chatta *et al.*, (2014). By substitution, the system maximum turbine efficiency is obtained as:

$$\eta_t = 0.5 \times 0.98^2 \times (1 + 0.98) \times (\cos 16) ^2$$

$$\eta_t = 87.2\%$$

Determination of the system Turbine Speed (N)

The system turbine speed is obtained by implementing an established relation from Mrudang, and Nirav, (2016) as the correlation between specific speed (N_s) and net head of a hydropower system as expressed in Equation 9.

$$N_s = 513.25 \times \frac{H_n^{0.745}}{\sqrt{P_t}} \text{ (rpm)} \quad (9)$$

where P_t is the turbine power of the system (5kVA); H_n is the net head of the system (m); N_s is turbine speed

$$N_s = 513.25 \times \frac{6.52^{0.745}}{\sqrt{5000}} \text{ (rpm)} \cong 928 \text{ rpm}$$

Determination of the turbine Runner Outer Diameter (D_o)

The outer diameter selection depends upon the flow conditions. As reported by Abubeker, (2018), a larger diameter is selected for more flow while a smaller diameter is selected for less flow. Taking this condition into consideration the turbine Runner Outer Diameter is obtained using Equation 10 as follows:

Where D_o is the outer diameter of the runner. Using the obtained values for H_n and N the outer diameter (D_o) of the

$$D_o = 40 \sqrt{\frac{H_n}{N}} \quad (10)$$

Estimation of Radial Rim Width (w)

Radial rim width according to Mrudang, and Nirav, (2016) is defined as the difference between the outer and the inner radius of the runner of the turbine. It is expressed mathematically in Equation 11 as follows:

$$w = 0.174 D_o \quad (11)$$

where: w is the radial rim width. Thus, the system radial rim width is estimated as 19.14 (mm)

Estimation of Blade Spacing (t_b)

The tangential blade spacing is the distance between two successive blades of a turbine. It is obtained by adopting Equation 12 extracted from the study of Alie, (2016). This also confirms the blade spacing is equal to the radial rim width of the turbine.

$$t_b = 0.174 D_o \text{ (mm)} \quad (12)$$

where: D_o is the outer diameter of the runner

Determination of the Turbine Runner Inner Diameter (D_i)

The inner diameter of the turbine runner is obtained by adopting Equation 13 extracted from the study of Alie, (2016) as follows:

$$D_i = D_o - 2a \text{ (mm)}$$

where: D_o is the outer diameter of the runner; a is equivalent to the radial rim width (w). Thus, the inner diameter of the turbine runner is obtained as 71.72 (mm)

Determination of the Effective Number of Runner Blades (n) for the Turbine System

Pulsating power and reduced turbine efficiency are caused by excess in the number of blades while if it is less, it would result in incomplete utilization of water. The effective number of turbine blades of the system is important in order to aid complete utilization of the provided driving force from the water flow rate, and consequently improve the turbine efficiency. In this study the optimum number of blades required for the system effectiveness is obtained using Equation 14 as follows;

$$n = \pi \frac{D_o}{t_b} \quad (14)$$

where D_o is the outer diameter of the runner; t_b is the turbine blade spacing

$$n = \pi \frac{110}{19} = 18 \text{ blades}$$

Determination of Turbine Runner Blade Length (L)

The length of the runner along the axis of the system turbine is calculated using the relation given in Equation 15 as follows:

$$L = \frac{210 \times Q}{D_o \sqrt{H_n}} \quad (15)$$

where: Q is the volumetric flow rate of water; D_o is the diameter of the runner, H_n is the system net head. Thus, the turbine Runner Blade Length (L) is calculated to be 66.541(m)

Determination of Turbine Blade Radius of Curvature (r_b)

According to Alie, (2016), the turbine blade radius of curvature is defined as the radius of the curvature of the curved blades of the turbine. The radius is very important for the efficient working of the turbine. It is calculated as follows using Equation 16:

$$r_b = 0.163 \times D_o \text{ (mm)} \quad (16)$$

$$r_b = 0.163 \times 110 = 17.93 \text{ (mm)}$$

Determination of the Turbine Shaft Diameter (D_s)

For effective service delivery of the turbine considered in this study, a shaft of diameter capable of carrying the load of the turbine and the optimum size for better flow of water across turbine is designed for using Equation 17 as follows:

$$D_s = 0.22 \times D_o \text{ (mm)} \quad (17)$$

$$D_s = 0.22 \times 110 = 24.2 \text{ (mm)}$$

Determination of the Turbine Torque

The torque of the turbine is required to achieve the expected system power output. If the torque is not sufficient to turn the alternator no power can be produced. In this study, the turbine torque is obtained as 51.46 (Nm) using Equation 18 adopted from the study of Alie, (2016) as follows:

$$T = \frac{P_t}{2\pi N/60} \text{ (N.m)} \quad (18)$$

where: P_t is the turbine power of the system (kVA); N_s is the turbine speed (rpm)

Computer Aided Design (CAD) of the Hydropower Plant System

By implementing the attained design calculation, the drafting of parts was done on a computer workstation using SolidWorks 2016 platform. The parts would be assembled in an assembly environment where constraints and mates are imposed to make the assembled model behave as it were in the real machine statically and dynamically. The design drawings for the Pico hydroelectric system as created in SolidWorks 2016 are presented in Figures 1 to 2.

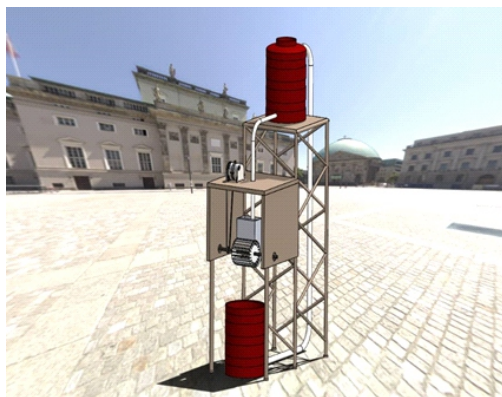


Figure 1: The hydropower plant system front view

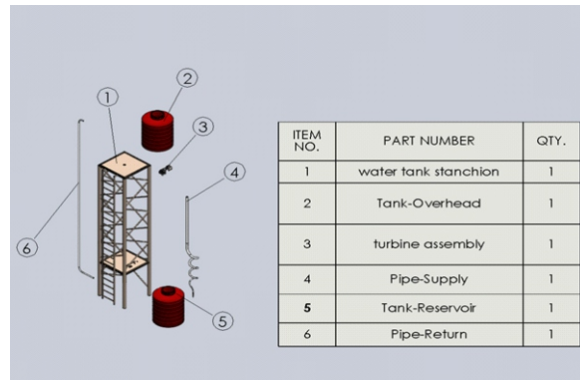


Figure 2: The Exploded view of the hydropower plant system

Operating Principle of the Designed Hydropower Plant System

Water is the working fluid to be used in generating power in the designed hydropower plant system. The water is delivered from a height under the influence of gravitational force. Water is fed into the reservoir tank (5) via the return pipe (6) as shown in Figure 2. A D-C powered submersible pump is positioned in the reservoir tank (5). The pump transfers the water to the overhead tank (2) by pressurizing the water through the return pipe delivery (6). The valve at the outlet of the overhead tank (2) is ensured to be shut for it to be filled. Once it has been ascertained that the overhead tank (2) is filled the valve is opened to release the water under free fall due to gravity. As the waterfall from the height, it falls with high pressure flows through the turbine (3) system. Thus, the kinetic energy of the water is converted into mechanical energy as rotation which in turn turns the turbine blades and this turns the turbine shaft connected to the alternator. The continuous rotating motion of the turbine shaft connected to the alternator causes the current to flow in the opposite direction to the motion of the armature. The continuous motion of the amateur creates a continuous interaction between the coil and the magnetic field, thus leads to a continuous generation of electricity.

Simulation Procedure

In order to ascertain the manufacturability, functionality, integrity and efficiency of the designed turbine runner fixed to the shaft in this work, being one of the major components of the system, connected to the power generating section of the designed hydropower plant when fabricated, simulation of the designed components: turbine runner fixed to the shaft and the stanchion for the sitting water reservoir tank was carried out respectively using ANSYS and (structural analysis and design) STAAD pro software A 3D model of the turbine runner fixed to the shaft was developed using SolidWorks software and imported into ANSYS (Figure 3) for finite element simulation. The simulation was done to determine the operational strength of the runner. Mild steel was selected as the

material for both the shaft and the runner while both were assumed to be firmly welded together. The mechanical property of mild steel is presented in Table 1. The estimated maximum torque (51.46 Nm) expected to be transmitted by the shaft with the view of generating 5 kVA power is considered in the simulation process as shown in Figure 4.

Table 1: Mechanical Properties of Mild Steel Selected for the Shaft.

Property	Value
Elastic Modulus	$2.0 \times 10^{11} \text{ Nm}^{-2}$
Poisson's Ratio	0.32
Mass Density	7800 kg.m^{-3}
Tensile Strength	$4.825 \times 10^8 \text{ Nm}^{-2}$
Yield Strength	$2.482 \times 10^8 \text{ Nm}^{-2}$

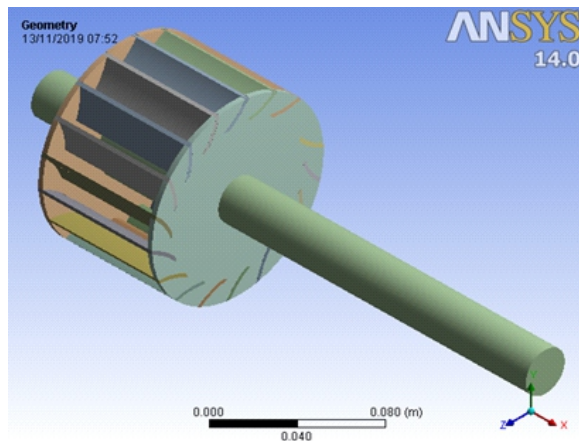


Figure 3: 3D view of the runner-shaft model

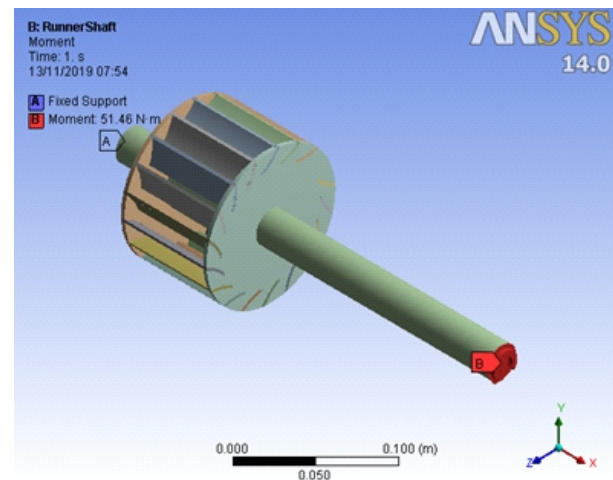


Figure 4: Analysis settings of the runner-shaft

Results and Discussion

Mesh generation and Mesh Refinement

Mesh generation is an important aspect of any finite element simulation, therefore a torsional simulation study was carried out with higher accuracy for validation of computational procedure to analyse the behaviour of the rotating runner in ANSYS-Meshing environment as depicted in Figure 5. During meshing, the geometric model is discretized to a number of small elements which are called cells by partial differential equation and algebraic methods. The grid generation started with a coarse mesh and followed by the refined mesh size type. Thus, the developed turbine runner fixed to the shaft 3D model was discretized into a mesh having 888722 elements and 2326867 nodes, as shown in Figure 5.

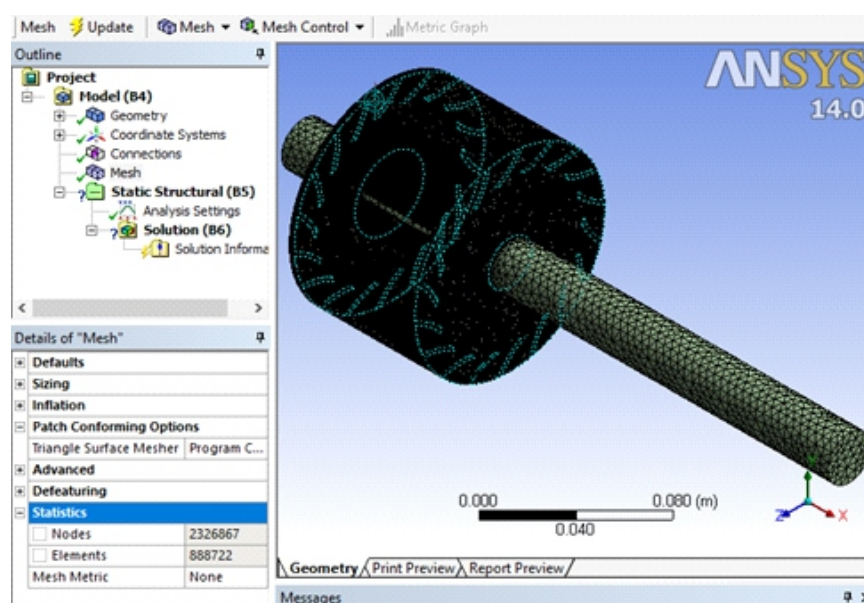


Figure 5: Meshed model of the runner-shaft

Numerical Results and Analysis

For the analysis set, one end of the shaft was assigned to be fixed while the other end was subjected to a maximum estimated torque of 51.46 Nm, as can be seen in Figure 5. The shaft is assumed to rotate freely hence the torque experienced by the shaft results from the impact of the water falling under pressure on the runner

blades. As shown in Figure 6, it was observed that a maximum stress distribution of $0.671 \times 10^8 \text{ N/m}^2$ was obtained which is below the yield strength of $2.482 \times 10^8 \text{ N/m}^2$ of the selected material see Table 1. This implies that the shaft will be able to withstand deformation due to torsion when subjected to a maximum torque of 51.46 Nm

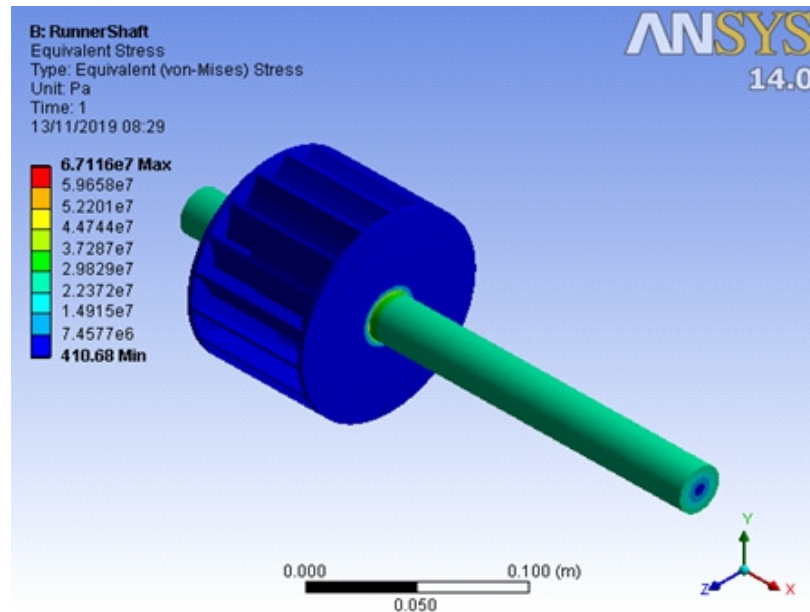


Figure 6: Maximum stress on the shaft

Furthermore, In order for the shaft to be considered safe for operation, the factor of safety of the turbine runner shaft is estimated and must be greater than 1. The Factor of safety is the ratio of the yield strength or maximum allowable strength of the selected material to the maximum stress on the shaft after loading. As depicted

in Figure 7, a minimum factor safety up to 3.72 is attained in this work. This result indicates that the shaft is fit to be used for the selected material and maximum torque. Figure 7 shows the simulation results and contours for the obtained factor of safety in this work.

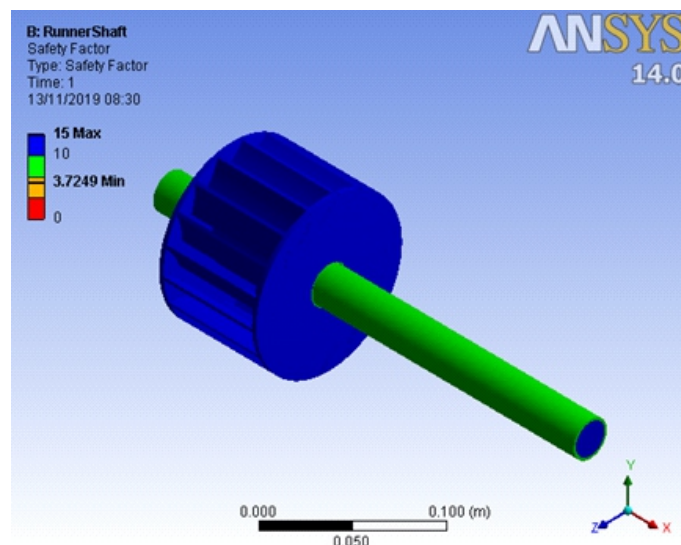


Figure 7: Factor of safety on the shaft

The obtained simulation results of the stanchion shown in Figures 8 and 9 revealed a minimal deformation on the stanchion horizontal beam and the resultant load was found concentrated and distributed on the vertical

members. These behaviours indicate that the stanchion will not fail while carrying the water reservoir tank load of the designed power generation system.

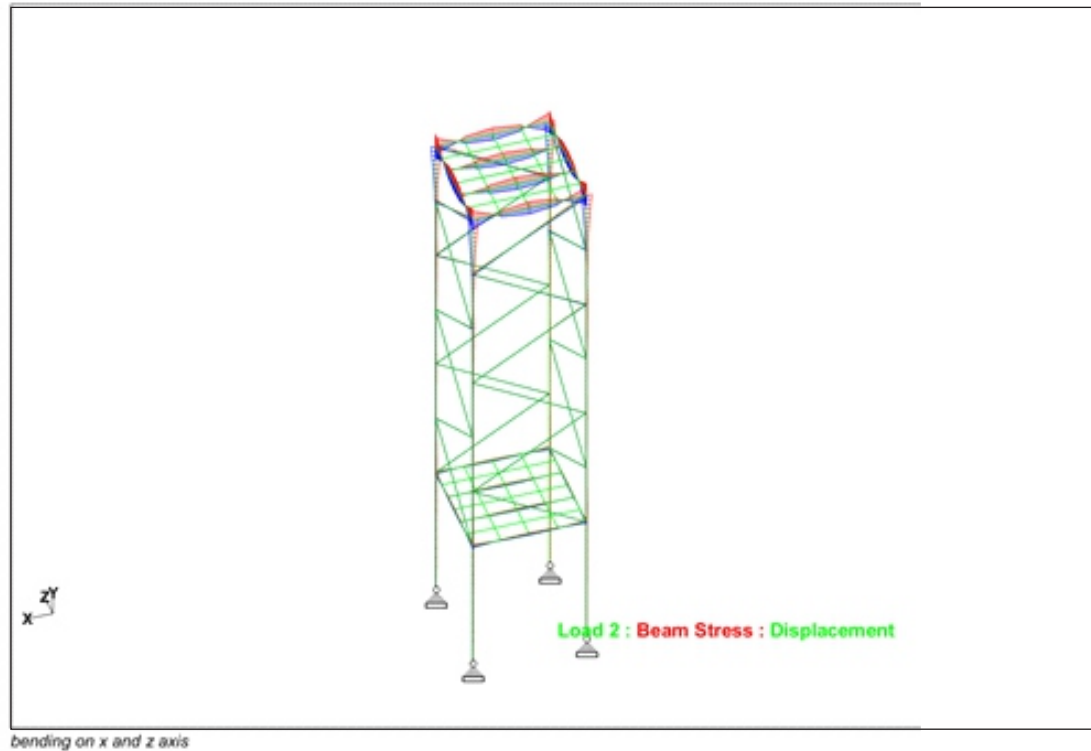


Figure8: The beam stress is uniformly distributed

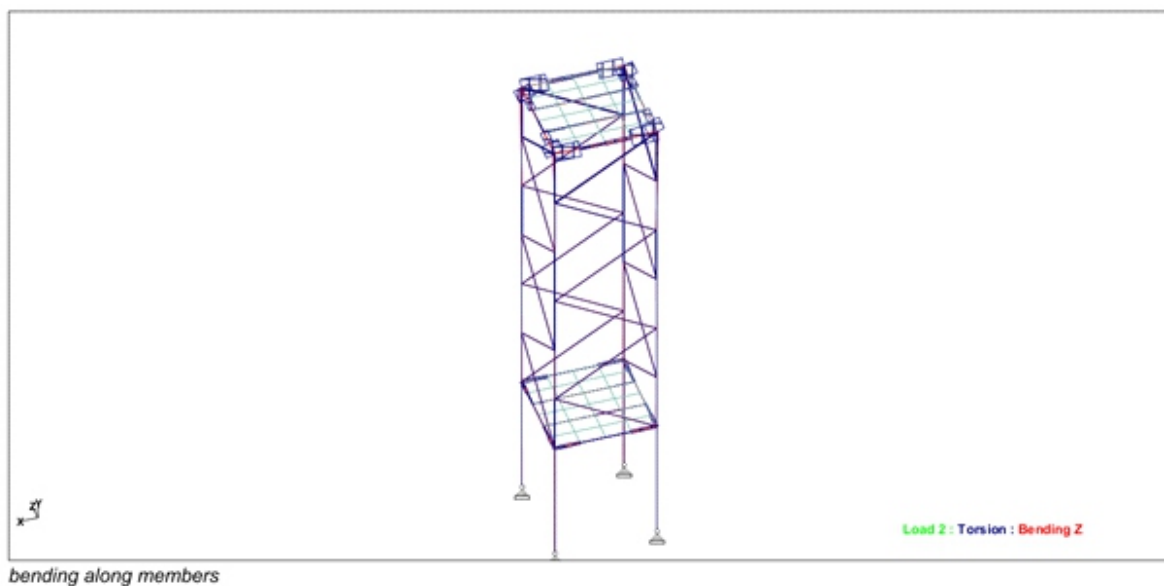


Figure 9: The load is concentrated on the four vertical members

Conclusion

A Pico hydroelectric power system of a cross-flow turbine type has been designed and simulated in this study. The model parameters such as runner diameter, runner length, blade spacing, radius of blade curvature, shaft diameter, water jet thickness, turbine speed, turbine power and the number of blades which play a major role to achieve the Turbine maximum efficiency were determined. The designed power system's maximum efficiency was found to be 87% with a head of 6390 mm at a flow rate of 0.089 m³/s. Numerically, analysis of the designed turbine runner shaft was carried out using ANSYS tool where maximum stress distribution of 0.671×10^8 N/m² and minimum factor safety of 3.72 were observed in this work. The performance evaluation results using the STAAD pro software for the water reservoir sitting stanchion simulation indicated that the stanchion will not fail while carrying the water reservoir tank load of the designed power generation system. Similarly, the numerical analysis of the designed cross-flow turbine type in this work using ANSYS tool revealed that the shaft will be able to withstand deformation due to torsion when subjected to a maximum torque of 51.46 Nm. Hence the designed hydroelectric power system will be efficient to produce a 5kVA electrical power

Acknowledgment

The Authors (TSM), (STF), (OAO) and (OIA) acknowledge the technical support of Idowu, E. T. and Alade, B. O for successful presentation of this work in 2021 SEET Annual Conference

References

- Abubeker, N. G. (2018). Design and Simulation of cross flow turbine on Huluka river in Ethiopia. *International Journal of Advance Engineering and Research Development*, 5 (3):1007-1017.
- Adejumobi, I. A. and Adebisi, O. I. (2011). Exploring Small Hydropower Potentials for Domestic and Information Communication Technology Infrastructural Application in Rural Communities in Nigeria. Proceeding of the 12th Biennial International Conference of Botswana Institution of Engineers, Gaborone, Botswana, 19-26.
- Akerkar, B. P. (1989). A Study of the Performance of the Crossflow Turbine, M. S. Thesis, Clemson University, Clemson, SC.
- Alie, W. D. (2016). Design and Analysis of Small Hydro Power for Rural Electrification. *Global Journal of Researches in Engineering: Electrical and Electronics Engineering*, 16(6): 25-45.
- Anyaka, B. O. (2013), Small Hydropower Projects for Rural Electrification in Nigeria: A Developer's Perspective, *International Journal of Innovative Technology and Exploring Engineering* 3 (5): 29-34.
- Bilal, A. N. (2013). Design of Micro - Hydro - Electric Power Station, *International Journal of Engineering and Advanced Technology*, 2(5): 39-47
- Chattha, J. A., Khan M.S., Iftekhhar H. and Shanid S. (2014). Standardization of Cross Flow Turbine for Typical Micro-hydro Site Conditions in Pakistan, *Proceedings of the ASME 2014 power conference*.
- David K.O. (2013). Review of Small Hydropower Technology, *Renewable Sustainable Energy Review*. 26: 515 – 520.
- Walseth, E. C. (2009). Investigation of the Flow through the Runner of a Cross-Flow Turbine, Master of Science in Product Design and Manufacturing, Norwegian University of Science and Technology, Department of Energy and Process Engineering. Unpublished.
- Fraenkel, P., Paish, O., Bokalders, V., Harvey, A., and Brown, A., (1991). Edwards R. Micro-Hydro Power: a guide for development workers. London. Russell Press Ltd., Nottingham.
- Gutu, B. O. and Auch, V., R. (2017). Design and Computational Fluid Dynamic Simulation Study of High Efficiency Cross Flow Hydropower Turbine. *International Journal of Science, Technology and Society*. 5(4): 120-125.
- IRENA (2012) Renewable Energy Technologies: Cost Analysis Series Volume, Power Sector Issue 3/5 https://www.irena.org/documentdownloads/publications/re_technologies_cost_analysis-hydropower.pdf
- Mrudang P. and Nirav, O. (2016). Design and Analysis of High Efficiency Cross- Flow Turbine for Hydro-Power Plant, *International Journal of Engineering and Advanced Technology*, 5 (4): 187-193
- Razak, J. A. (2010). Application of Cross-Flow Turbine in Off-grid Pico-hydro Renewable Energy Systems, Proceeding of the American-Math 10 Conference on Applied Mathematics, 519-526.