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Vibration Analysis using Finite Element Analysis (FEA): An Evaluation of Pico-Tubular Bulb Type Turbine Blades Fabricated in Composite Materials

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Abstract: ***Tubular turbines have been widely employed and evolved fast when its introduction in the 1930s due to their strong technical and economic qualities and application. Because its performance and structure differ from those of ordinary vertical shaft units, local and international academics worked extensively on research techniques and technological means using numerical simulation and model testing. The transmissions of a high quantity of power, which may cause unwanted vibrations that reduce efficiency, increase wear, and, in the worst-case scenario, cause serious damage. In this paper, the material propose in order to substantiate that the random excitations and excess vibration of the pico-turbine can be prevented is the use of CFRP (carbon fiber reinforced polymer) and PLA (polylactic acid). In this paper, ANSYS® Mechanical modal simulation is used to evaluate the structures' robustness behavior of the composite materials that were used as the main material in the fabrication of turbine blades for bulb-type turbine application. The use CFD simulation in SOLIDWORKS® is needed to examine the pressure fluctuation caused by unsteady flow that can contribute in the unwanted pulsation and to conform the modal simulation results. To validate the results, pressure pulsation experimentation is conducted to evaluate the fluctuation of the pressure affecting the blades or in the rotating region and it is analyzed through frequency response domain. Hence, in this paper, it is proven that the vibration behavior of the material is acceptable since the resulting natural frequency provides resulting stress, strain, and deformation that is allowable and below its ultimate tensile strength.***

Keywords: ***Carbon Fiber, Composite Materials, Natural Frequency, Pressure Pulsation.***

I. INTRODUCTION

Tubular hydropower station development, design, operation technology, and expertise are becoming increasingly numerous. Bulb tubular turbines have contributed for over 30 percent of hydropower since their introduction in the late 1970s. However, when the operation of bulb tubular turbine units increases, the problem of failure is increasingly revealed by the unit's operation. Also consumes hydraulic resources and diminishes the unit's working efficiency, resulting in a reduction in economic advantages, and it poses significant safety risks during operation. Several studies on the energy characteristics, cavitation characteristics, vibration characteristics, and optimal design of tubular turbines have been conducted, with a focus on the analysis of special flow phenomena (e.g., clearance flow and cavitation characteristics) of tubular turbines and the improvement of energy characteristics. The hydraulic vibration of the turbine, in particular, and the fatigue damage to the unit's structural components induced by the hydraulic vibration, have gotten a lot of attention. Now, fluid-structure coupling is mostly used to handle the problem of stress distribution, vibration deformation, and material fatigue damage in overcurrent components, and promising research findings have been obtained. The transmissions of a high quantity of power, which may cause unwanted vibrations that reduce efficiency, increase wear, and, in the worst-case scenario, cause serious damage. The worst situation that is being investigated here that will affect the prototype is the sudden surge of water. If surge occurs, the speed of the rotation increases and possible foreign objects might be inserted to the flow streamline of the turbine that creates impact and trigger its oscillation. This is why manufacturers use lighter and less stiff material to reduce the cost, to increase the flow rate, and reduce the size, which creates a high stress in the material [1]. This study will conduct vibration analysis, modal testing, CFD simulation, pressure pulsation test, and design and fabricate a 4-blade Pico axial hydro turbine using composite materials. It will use ANSYS® Mechanical Modal simulation to evaluate the structures' robustness, modal testing to examine the natural frequency of the composite material, CFD simulation to examine the pressure fluctuation caused by unsteady flow, and pressure pulsation test to validate the SOLIDWORKS® CFD simulation. Finally, it will design and fabricate a 4-blade Pico axial hydro turbine using composite materials and compare it to previous study.

II. NUMERICAL BACKGROUND

A. Finite Element Method

The finite element method (FEM) is a numerical technique for solving problems, which are described by partial differential equations or can be formulated as functional minimization [2][17][18][19].

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Several ways may be employed to convert the physical formulation of the issue to its discrete finite element equivalent. If the physical formulation of the issue is a differential equation, the Galerkin technique is the most prevalent finite element formulation approach. If the physical issue can be expressed as a functional minimization, the variational formulation of finite element equations is commonly utilized. So, FEA is a computational approach that aids in achieving good results with all complicated situations that cannot be addressed analytically. There is a large variety of sophisticated commercial code available that aids in the approximation near solution in 1D, 2D, and 3D. The whole continuum is split into a finite number of tiny components of geometrically simple shape in this FEA approach.

$$\{F\} = [K]\{u\} \tag{1}$$

The stiffness (K) of a structure is determined by its geometry and material qualities. The user must supply the load (F) value. The only thing that is uncertain is displacement (u)..

B. Theory of Vibration

Natural frequencies can be controlled and kept apart from applied load frequencies, known as excitation frequencies, by the design of a component. Modal analysis is the process of determining the natural frequencies of a structure. Excitation frequencies in a rotating structure are often thought to be engine order frequencies that result from the rotational speed. Because stiffness and mass are continuous in finite element models, square matrices are used to represent stiffness, mass, and damping. They can still have line element springs and dampers as well as point masses as special circumstances. Dampers release energy, whereas springs and weights do not. If you have a finite element system with multiple degrees of freedom, the above single DOF system generalizes to a displacement vector, X(t), interacting with a square mass matrix, M, stiffness matrix, K, damping matrix C, and externally applied force vector, F(t), but the overall form remains the same [3]. Plus the initial conditions on the displacement, X(0), and velocity, v(0) = dX / dt(0). Integrating these equations in time gives a time history solution. The solution concepts are basically the same, they just have to be done using matrix algebra.

$$M\, d^2X/dt^2 + C\frac{dX}{dt} + KX(t) = F(t) \tag{2}$$

$$M\,\frac{d^2x}{dt^2} + KX(t) = 0 \tag{1}$$

C. Computational Fluid Dynamics Numerical Analysis

SOLIDWORKS Flow Simulation can simulate laminar and turbulent flows. Laminar flow occurs at low Reynolds numbers, which are defined as the product of typical velocity and length scales divided by the kinematic viscosity. When the Reynolds number surpasses a certain critical threshold, the flow becomes turbulent. To forecast turbulent flows, the Favre-averaged Navier-Stokes equations are employed, which consider the time-averaged effects of flow turbulence on flow parameters while directly accounting for large-scale, time-dependent events. Extra terms called as Reynolds stresses occur in the equations as a result of this approach, requiring further information. To close this system of equations, SOLIDWORKS® Flow Simulation employs transport equations for the turbulent kinetic energy and its dissipation rate, using the k-ε model [4]. Lam and Bremhorst's (1981) modified k-turbulence model with damping functions characterizes laminar, turbulent, and transitional flows of homogeneous fluids using the turbulence conservation rules listed below.

$$\frac{\partial \rho k}{\partial t} + \frac{\partial p k u_i}{\partial x_i} = \frac{\partial}{\partial x_i}\left(\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_i}\right) + \tau_{ij}^{R}\frac{\partial u_i}{\partial x_j} - \rho\varepsilon + \mu_t p_B \tag{2}$$

$$\frac{\partial \rho \varepsilon}{\partial t} + \frac{\partial p \varepsilon u_i}{\partial x_i} = \frac{\partial}{\partial_{x_i}}\left(\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_i}\right) + C_{\varepsilon_1}\left(\frac{\varepsilon}{k}\right)\left(f_1 \tau_{ij}^{R}\frac{\partial u_i}{\partial x_j} + C_B \mu_t p_B\right) - f_2 c_{\varepsilon^2}\frac{\rho\varepsilon^2}{k} \tag{5}$$

$$\tau_{ij} = \mu s_{ij}, \tau_{ij}^{R} = \mu_t s_{ij} - \frac{2}{3}\rho k\,\delta_{ij}, s_{ij} = \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3}\delta_{ij}\frac{\partial u_k}{\partial x_k} \tag{6}$$

Where Cu =0.09, Ce1=1.44, Ce2= 1.92, sigma k = 1, sigma e = 1.3, sigma b = 0.9 and Cb = 1 if the turbulent viscosity is determined from:

$$\mu_t = f_\mu \cdot \frac{C_\mu p k^2}{\varepsilon} \tag{3}$$

Lam and Bremhorst's damping function fμ is determined from:

$$f_\mu = (1 - e^{-0.025Ry})^2 * \left(1 + \frac{20.5}{R_t}\right) \tag{8}$$

Where:

$$R_y = \frac{\rho k^2}{\mu} \tag{9}$$

$$R_y = \frac{\rho k^2}{\mu} \tag{10}$$

III. SIMULATION SETUP

ANSYS® is used in the simulation of Modal and Random vibration analysis. In the setting up of the modal analysis, the CAD (Computer Aided Design) file is imported to the ANSYS® Workbench, specifically in the Modal section. After reviewing the imported file, material selection follows. In the first modal analysis, structural steel is used since it is the most common material used in every steel works. The illustration of the setup in ANSYS® workbench is shown in figure 1(a) & 1(b). By clicking the model in the modal section, meshing is defined. Mesh is set up in mechanical as physics preferences, linear as element order, and 10 mm as element size, shown in figure 2. Now, modal analysis is the prerequisite of random vibration analysis. The modes used in the modal analysis are set to be 25 nodes. A mode is a point along a standing wave where the wave has minimum amplitude.

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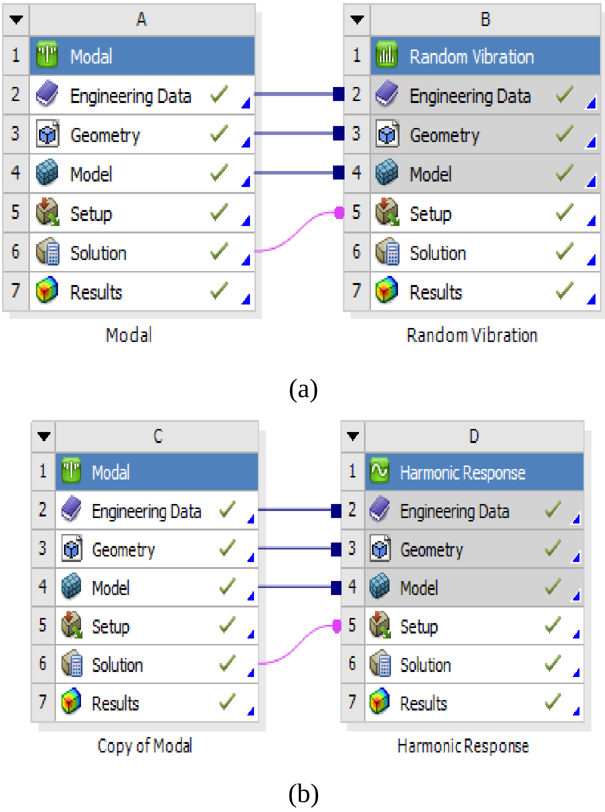

Fig. 1. ANSYS Workbench Simulation Set Up For (A) Random Vibration And (B) Harmonic Response in ANSYS Mechanical

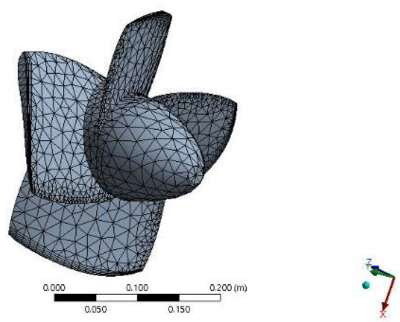

Fig. 2. Meshed Geometry

Modal analysis is the fundamental dynamic analysis type, providing the natural frequencies at which a structure will resonate [5]. These natural frequencies are critical in a variety of technical applications.

In setting up the modal analysis simulation, the number of modes is set to 25 as shown in the figure 3. Each of the modes correspond a natural frequency value of every specified material that can be transferred to the random vibration simulation as prerequisite data. Through this natural frequency data, the possible data that is needed for the research will be acquired [21]. The frequency time content of time history or spectrum is captured along with the statistics and it is used as the load in the random vibration analysis. Further, Table 8 shows the corresponding deformation values at its specific Modes and it will serve as basis for random vibration simulation.

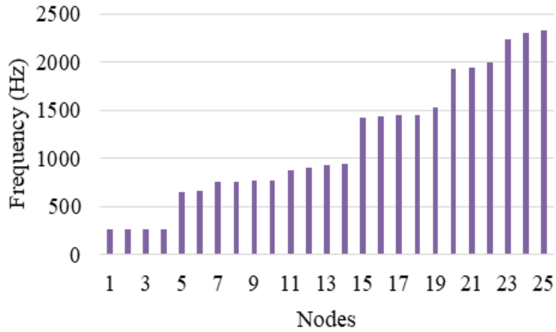

Fig. 3. Frequency Values of Composite Material at Specific Modes

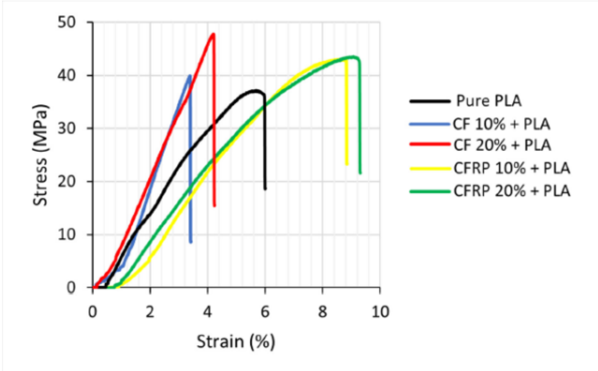

Fig. 4. Stress-Strain Curve for The Recycled Pla and Four Recycled Composites [6]

Table-I. Boundary Conditions for CFD Simulations

Variables	Fine Weather	Surge 1	Surge 2	Surge 3
Q	0.21	0.25	0.3	0.4

The materials used in the simulation are composite materials, CFRP & PLA. PLA as a leading candidate is a thermoplastic, high-strength, high-modulus polymer that can be made from annually renewable resources to yield different components for use in either the industrial packaging field or the biocompatible/bio absorbable medical device market. While, Carbon fiber reinforced polymers or plastics (CFRP) are gaining popularity owing to their wide range of applications in industries such as aviation, military, vehicle manufacturing, transportation, architecture, sports industries, and medicine [6]. In order to increase the strength of the material yet lightweight as possible, carbon fiber is coated after the PLA. Carbon fiber is one of the suggested materials. Carbon fibers have high strength (3–7 GPa), high modulus (200–500 GPa), compressive strength (1–3 GPa), shear modulus (10–15 GPa), and low density (1.75–2.00 g/cm3) [7][20]. As part of the study, figure 4 which comes from the study of AL Zahmi, et al., 2022, composite materials' ultimate strength ranges within 40 to 50 MPa.

Computational Fluid Dynamics (CFD) domain is the area of space where the CFD simulation's solution is calculated. To solve the discretized equations of fluid flows, the computational domain must be discretized into a computational grid (or mesh).

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The size of a computing domain is determined by the kind of problem: external aerodynamic or interior flow. The computational domain, also known as the outer domain in external aerodynamics, represents the region around the geometry of interest where the flow solution is required. Its shape and size are mostly determined by the geometry's aerodynamic properties. Figure 5 illustrates the computational domain of the simulation with table 1 as boundary conditions.

IV. EXPERIMENTAL SETUP

This is the experimental setup design, figure 6. The basis of this experimental is due to the claim that vibrations in hydraulic turbine are due to the extreme force fluctuation cause by cavitation [8]. The said authors also claim that pressure difference by large cavities causes hydrodynamic pulsations and cavitation-induced vibration in hydro-turbines. Hence, it be inferred that direct vibration cause by fluid force to the blade is like to occur in Kaplan turbines which is used in this study since cavitation is less likely to occur because they can regulate the angle of their runner blades depending on the head and the flow rate, ensuring a high performance in the whole operating range [9]. In addition, the type of cavitation experiences in Kaplan is Tip Vortex cavitation. Also, another design of the Kaplan turbine that lessens the tip vortex cavitation is by equipping anti-cavitation lips [10]. The red box represents the pressure transducer location as shown in figure 6(a).

Table-II. Parameter Comparison Between the Model and Prototype Using Theory of Similitude

	D(in)	D Ratio	Q(m3/s)	Q Ratio
Model	2.3	5.161	0.004123	60.63
Prototype	11.88		0.25	

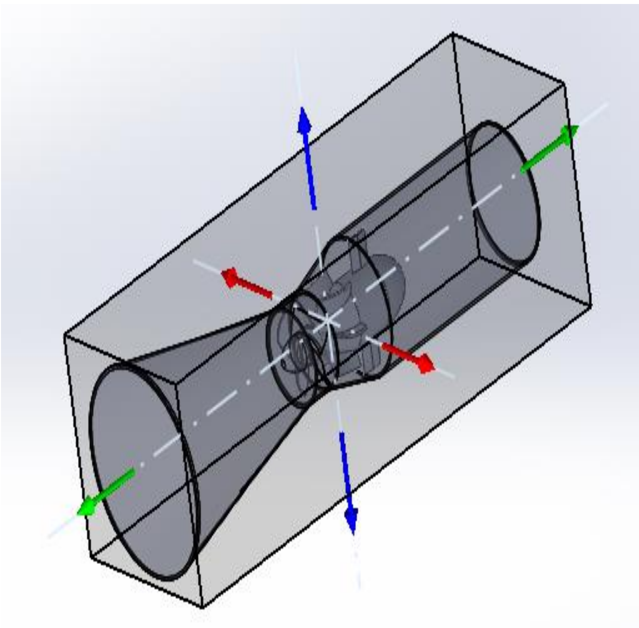

Fig. 5. CFD Analysis Computational Domain

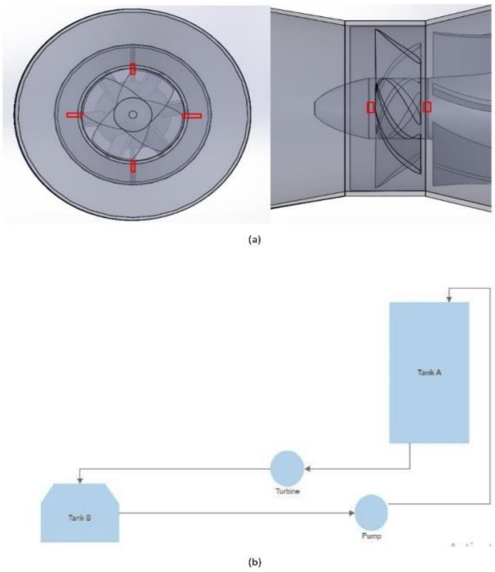

Fig. 6. Experimental Setup Diagram: (A) Placements of the Pressure Transducer (B) Schematic Diagram of the Test Rig

In this section, theory of similitude is highly recommended due to the fact that the limitation of material is beyond reachable due to standardize commercialization. By using geometric similarity, the geometric factor is set to 5 and the calculated discharge is 0.0041 cubic meter per second as shown in table 2.

V. RESULTS AND DISCUSSIONS

A. Simulation Results

Pressure pulsation in piping induces excitation forces to the turbine blades causing it to vibrate at a certain frequency that be tolerable to material or not. This excitation force cause by pressure fluctuation can cause the machine to failure. It can cause cavitation, reduced efficiency and fatigue pipe failure. Hence, the reason why pressure pulsation is analyzed in this study since the runner of the tubular turbine system one of the significant parts and it the region which changes its pressure, velocity, and vorticity once the water pass through. Inside the section where the runner is located, pressure coefficients developed are non-uniform and increases in the tip of the blade but decreases to the connection of the blade to the hub as shown in figure 8. The velocity difference in this area shows that unsteady flow shock occurs between the pressure side and the suction side. Figure 9 shows and illustrate the abrupt change pressure from the pressure side of the blades to the suction side. The maximum pressure value of the pressure side of the blade is estimated to be at 180kPa while the possible minimum pressure value occurs on the same side is estimated to be at 85kpa. On the suction side of the turbine, the estimated maximum pressure is 142kpa and the minimum pressure is 65kpa. In short, the pressure fluctuation occurs in the space between the draft tube of the rotating region and the tip of the blades, as shown in figure 9(a) and (b), and minimizes as it runs towards the hub.

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Hence, pressure fluctuation needs a constant monitoring to identify the behavior of the runner in normal operation.

The behavior of the pressure pulsation increases as the flow rate also increases. Further, it also shows that vortices predicted that vortices occur in the pressure side of the blades creating pressure pulsations which leads to deformation of the blades. One of the reasons why the pressure of the runner fluctuates it is because of the output power [11]. Large vibration amplitude occurs when the frequency of the pressure pulsation matches the mechanical natural frequency of the material. The inlet pressure as shown in the figure 9(a) illustrates how pressure changes as it hits the runner. This proves that the non-uniformity of pressure value caused the blades and the draft tube to oscillate at the same frequency.

Now, figure 10 shows the behavior of the pressure of the water in the model. The pressure is slightly lower the prototype since the whole setup is being reduced to a factor of five. The velocity is inversely proportional to the area of the tube being travelled upon that is why it is slightly faster than the water in the prototype. Hence, making the pressure value slightly lower. However, comparing to the area where the pressure fluctuates due to non-uniformity of the flow, it is still the same. The maximum and minimum pressure of the inlet in the model is calculated, and it is estimated to be 103kPa and 92kPa, respectively. It means that the blades will experience that specified value at some point in time during fluctuation in the operation.

Since the maximum and minimum pressure, and force due to liquid pressure are already calculated, as shown in table 3, it will be imported to the random vibration analysis back to ANSYS® to statistically calculate the numerical results of the vibration behavior of the blades using composite materials.

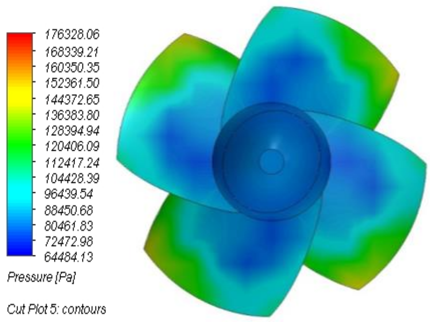

Fig. 7. Pressure Pulsation Surface Plot Contour of the Blades Under Case 1

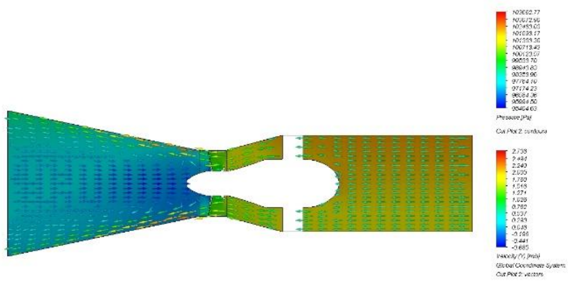

Fig. 8. Pressure Pulsation in The Bulb and Blades Flow Passage (Case 1)

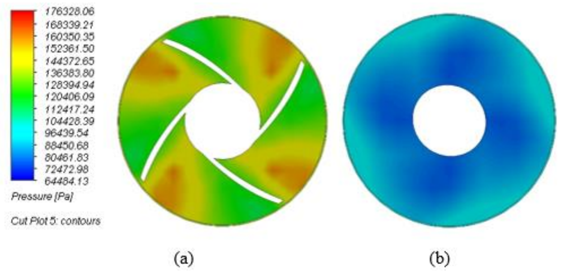

Fig. 9. Pressure Contour (a) inlet, (b) Outlet (Case1)

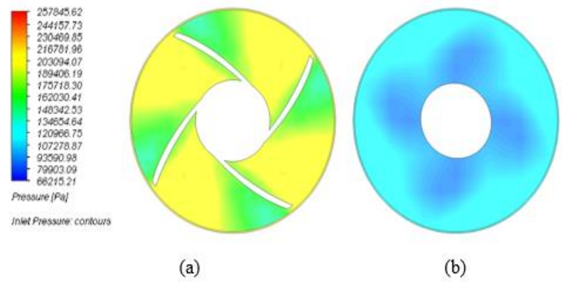

Fig. 10. Pressure Contour (a) Inlet, (b) Outlet (Case 2)

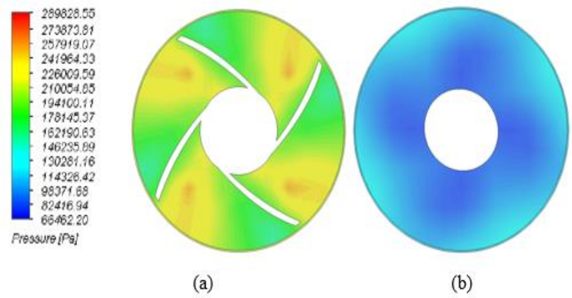

Fig. 11. Pressure Contour (a) inlet, (b) Outlet (Case 3)

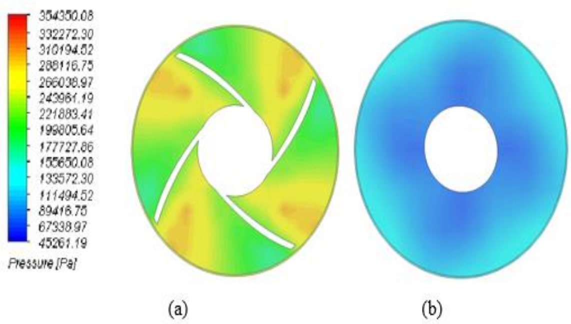

Fig. 12. Pressure Contour (a) inlet, (b) Outlet (Case 4)

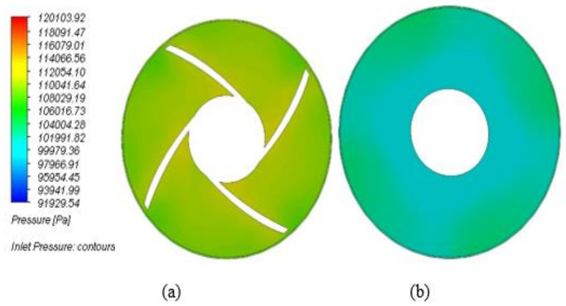

Fig. 13. Pressure Contour (a) inlet, (b) Outlet, of the Model

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Table-III. Pressure and Force Values of Both Prototype and Model.

PROTOTYPE			
	Pressure (kPa)		Force(kN)
	Min	Max	
Case 1	37.13142	128.5606	0.301
Case 2	46.91538	140.9238	0.796842
Case 3	63.49948	156.4472	1.443642
Case 4	79.78909	236.2953	8.578566
MODEL			
1/5Dp	94.62470	120.10392	1.483N

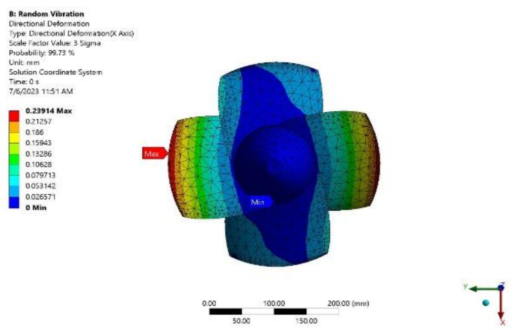

Fig. 14. Total Deformation of the Runner Blade (Composite)

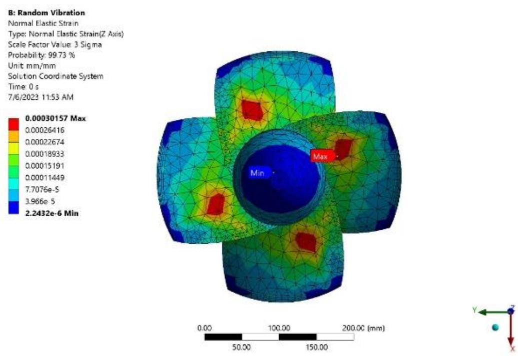

Fig. 15. Normal Elastic Strain of the Runner Blades (Composite)

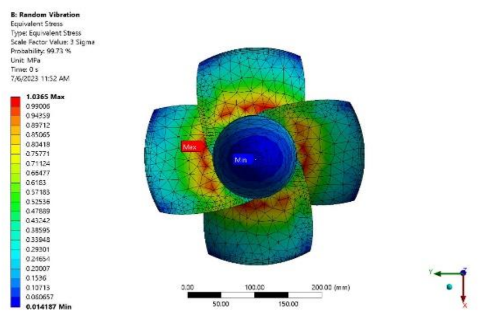

Fig. 16. Stress of the Runner Blades (Composite)

Using the data from CFD, figure 14 shows the deformation results out of the blade in random vibration analysis using composite material under random forces affecting on it. The calculation is statistical therefore the maximum deformation value is estimated 0.25019mm with the probability of 99.73%. The axis selected is based on the whereabouts of the manifestation of stress, strain, and deformation in its maximum. Next, figure 15 shows and illustrates the location of the area where most likely strain is occurring. It is affecting the connections between the hub and the runner blades. Since stress correlates the strain values then the location of the stress is also in the connections as shown in figure 16. The maximum value of strain simulated is approximately 0.00068424 and the maximum stress value is 3.3002Mpa. Since random vibration follows a Gaussian distribution aka Normal distribution, only the 3σ is used for the analysis in the meantime. 3σ is equivalent to the 99.73% of the time. Taking a note is needed that in random vibration analysis, since the input excitations are statistical in nature so also are the output responses such as displacement and stresses. A study can be compared to this same analysis where a plain steel sheet with the same thickness of the blade at 2mm has maximum deformation 109mm at 333Hz [12].

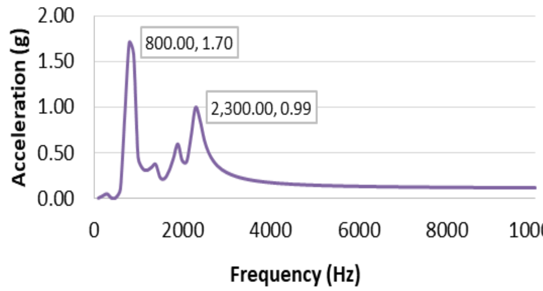

Fig. 17. Frequency Response in Acceleration at X-axis

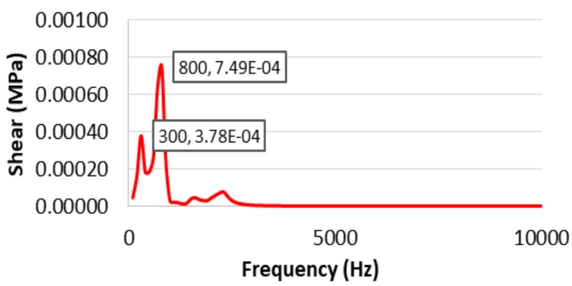

Fig. 18. Frequency Response in Shear Stress at X-axis

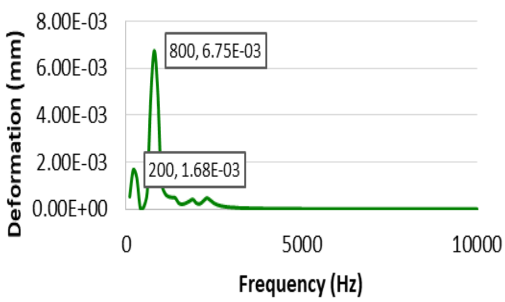

Fig. 19. Frequency Response in Directional Deformation in X-axis

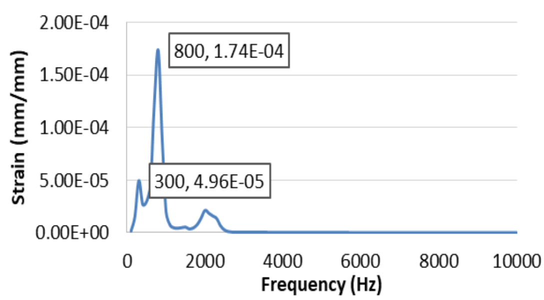

Fig. 20. Frequency Response in Directional Strain at X-axis

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In machines, monitoring start-up is needed to ensure the efficiency of the machine since it involves the possible effect of the impact load, or pressure where it can be a problem or not. In figures' 17 to 20, it describes the graph behavior of the four (4) variables like acceleration, stress, strain, and deformation in relevance to the value of its frequency. Due to high damping of the material used in the simulation, the graph immediately gives a downward slope signifying that the material is effective in reducing the behavior of the acceleration where it can contribute an unwanted value of stress, and strain of the machine. Other studies show that one factor that will increase the value of excitation is also the increase of fluid flow from the inlet pipe to the runner [13]. In this study, the maximum frequencies in different flow rate values vary. Therefore, the influence of the flowrate values greatly affects the stress, and strain. Further, blade vibration monitoring is very important for any power plant [14]. Also, in this study that HCF or high cycle fatigue is the cause of blade failure that's why vibration monitoring is need to ensure the efficiency of the blade.

Since the study is conducting vibration analysis, damping ratio is needed to examine to support the claim of the study. As shown in the figure 21, the damping ratio of the composite material, CFRP (carbon fiber reinforced polymer) and PLA (polylactic acid) is 0.0856, which is 43.4896% higher than PLA, and 62.1904% higher than CFRP. The damping ratio of the material is calculated using the logarithmic decrement equation, eq. 10, and the damping ratio equation itself, eq, 11.

$$\sigma = ln\left(\frac{P_1}{P_2}\right) = \frac{1}{n} In \frac{x(t)}{x(t+nT)} \tag{4}$$

$$\zeta = \frac{1}{\sqrt{1+(\frac{2\pi}{\sigma})^2}} \tag{5}$$

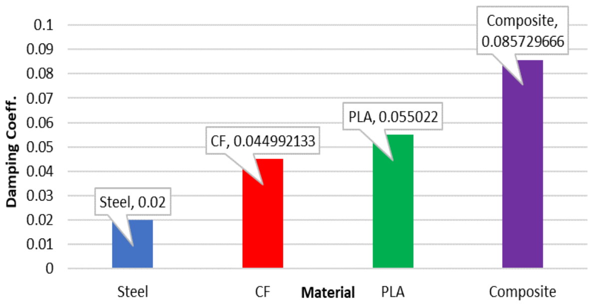

Fig. 21. Damping Coefficient of Every Material

B. Experimental Results

In this section, it is an experimental measurement of the pulsation and oscillations, through velocity, created by the fluid affecting the system especially the turbine blades. However, the data measured here is to support the claim in the modal analysis of the use of CFRP (carbon fiber reinforced polymer) reinforced with PLA (Polylactic Acid) in the application for tubular turbine blades fabrication.

In this testing, it conducts an evaluation of pressure difference in order to measure the pressure pulsation. Pressure pulsation detection is conducted on the important section of the main overflow component in the process of calculating the unsteady value of the tubular turbine to quantitatively analyze the unstable state inside each overflow component of the bulb tubular turbine, and the arrangement of the measuring points [15]. Table 4 is generated with the use Froude similarity criterion. The test rig is made in order to find the pressure fluctuation pulsation is shown in figure 43.

Table-IV. Geometrical Parameters of the Model and Prototype

Parameter	Prototype	Model
Runner Diameter	0.302 m	0.05842 m
Head	2 m	0.35m
H/D	6.62	5.99

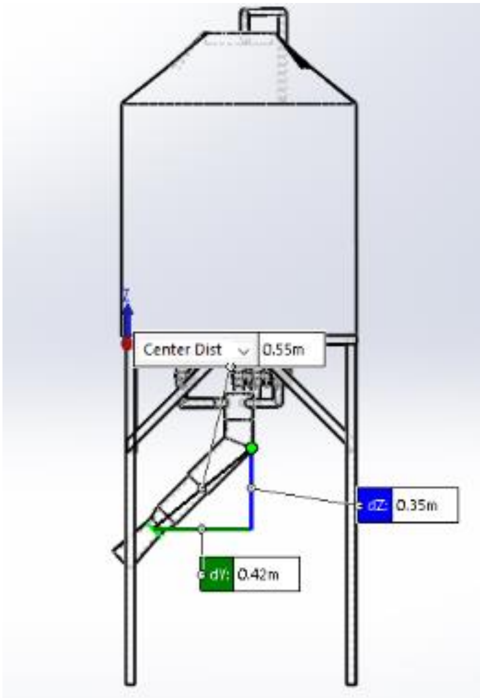

Fig. 21. CFD Validation Test Rig

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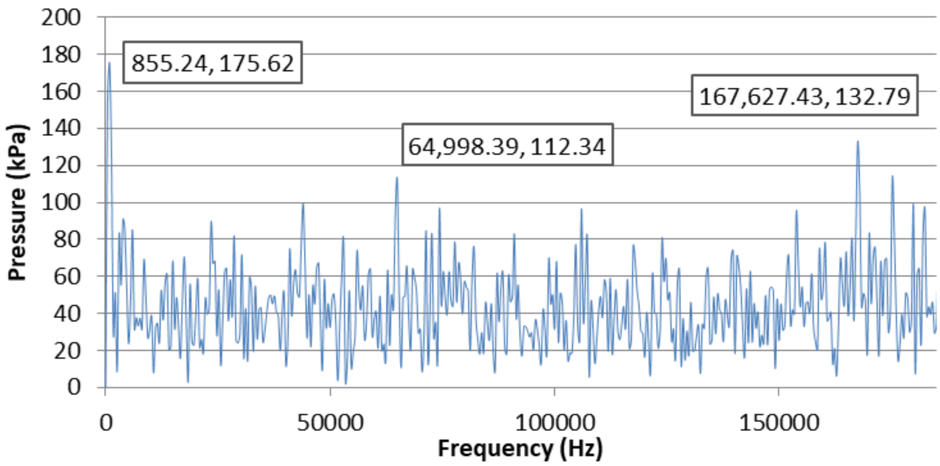

Fig. 22. Frequency Response of Pressure Pulsation at H/D = 5.99 (inlet)

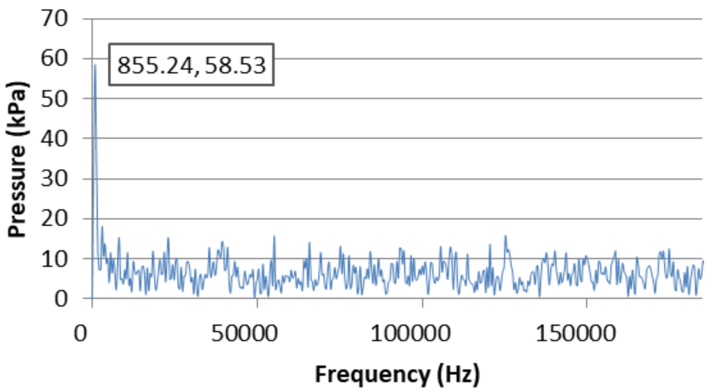

Fig. 23. Frequency Response of Pressure Pulsation at H/D = 5.99 (outlet)

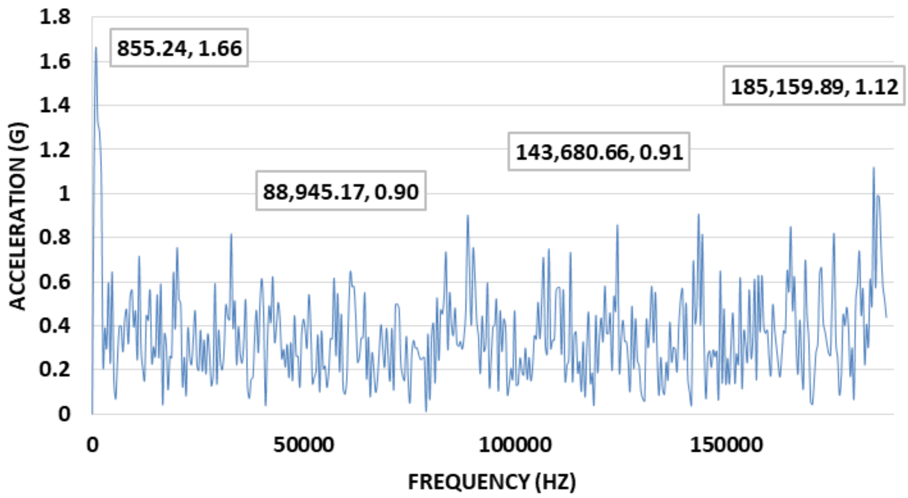

Fig. 24. Calculated Acceleration Frequency Response Affecting the Blade through Liquid Pressure

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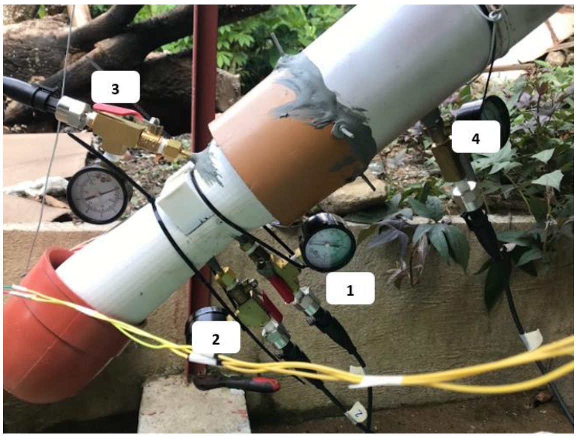

Fig. 25. Pressure Transducer and Identification in the Test Rig

Fig. 26. Experimentation of Pressure Pulsation at H/D=5.99

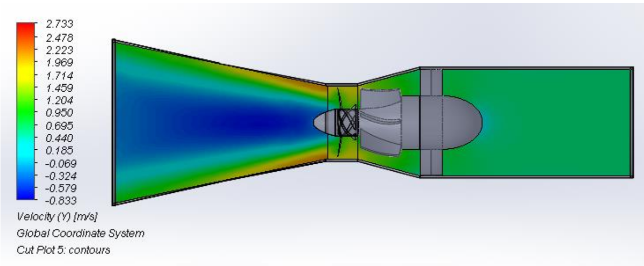

Fig. 27. Velocity Contour of the Model, H/D=5.99.

Table-V. Validation of Results.

	P (kPa)		Acc (g)	Natural Frequency (Hz)
Simulated	120.10		1.7048	800
Experimental	112.34	132.79	1.66	855
% diff.	6.68%	10.03%	2.66286%	6.65%

Figure 23 and figure 24 illustrate the frequency domain of the pressure pulsation in the Pressure transducer 1 and 2, respectively as shown in Figure 26. The highest peak of the graphs shows the natural frequency of the force due to liquid pressure manifesting towards the runner as the fluid passes by the rotating region. In figure 23, two peaks were observed. In this case, crucial monitoring is needed to avoid failure. The rotating region will experience harmonics and load input will be monitored through adjustment. It is because the aim of vibration monitoring, the succeeding peaks of the frequency graph must not be higher than the natural frequency.

The time domain data of pressure difference is converted to frequency domain in the vibration toolbox by Tom Irvine. As shown in figure 23, the highest peak of the graph is labeled in 855Hz at a value of 175 Kpa. According to the study, periodic pressure fluctuations will appear on the blade when runner rotates. The smaller the H/D ratio is, the submerged depth of the runner increases, and the amplitude of the water pressure fluctuation on the blade increases significantly [16]. Since in the model has H/D value of 5.99 therefore it has higher pressure fluctuation than that of the prototype machine. Thus, the contribution of pressure fluctuation to the composite material fabricated turbine blades is bearable. It means that if the turbine runner experienced a frequency within the range of 700Hz to 900Hz then it is in safe mode since the stress, strain, and deformation did not exceed its ultimate strength standard value. If the frequency of the hydraulic vibration due pressure fluctuation 855 Hz then it can be inferred that the vibration frequency response affecting the blade while in operation is in its allowable condition. The experimentation is shown in the

figure 26. The pattern of the water at it rushes to the outlet shows that evidence of pressure pulsation occurs.

Figure 27 can be compared to the simulation contour in figure 28 where it shows that the water flows in a rotating direction. Evidence shows that the velocity in the side of the draft tube has much larger value compared to the middle.

Pressure validation is being conducted once the experimental investigation is done. The model simulation is being validated through pressure pulsation experimentation where pressure transducers are used to detect and record pressure fluctuation at the designated point. It is found out that as the operation succeeded, fluctuation of pressures is present even in scaled prototype. Table 3 shows the maximum and minimum values of pressure and force in each case. It also shows the calculated maximum and minimum pressure and force due to liquid pressure of the model. In table 5, percentage difference between the two calculations is compared. The simulation value of the model has an estimated percentage difference from 6% to 10%. The force calculated in the CFD simulation is being imported to the random vibration analysis calculating a natural frequency of 800Hz. While the experimented data generated by the vibration toolbox by Tom Irvine, calculates 855Hz of natural frequency resulting to a 6.65% of percentage difference. Hence, since the pressure generated in the data is considered to be a normal stress, and the value is way lower than the ultimate tensile strength, then it can be concluded that the designed is considerable for actual operations. The acceleration is calculated based on the Bernoulli's principle of fluid mechanics. The value is shown in figure 17. It also evident that in high frequency the predicted acceleration that can affect the rotating region due to fluids dynamic pressure is fluctuating.

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However, it is still structurally acceptable with constant monitoring.

VI. CONCLUSION

In view of the material innovation in substituting stainless steel to composite materials for improving vibration characteristics, pressure pulsation investigation is conducted to elaborate the vibration analysis of the turbine blades fabricated in composite materials, CFRP (carbon reinforced polymer) and PLA (Polylactic Acid). The results are the following:

- Through variations of stress, strain, and deformation values in comparing stainless steel and composite materials, it can be concluded that composite material turbine blades designing pico-tubular bulb type turbine is possible. Due to the fact that at specified parameters (Head, Flow rate, etc.) in the original design, it can generate a pressure affecting the runner's structure resulting to an allowable stress value as discussed.
- Further, analyzing the composite blades frequency response in pressure fluctuation as discussed, validated also in pressure pulsation investigation, proves that PLA (Polylactic Acid) reinforced in CFRP (carbon fiber reinforced polymer) can withstand such conditions and suitable as an alternative material for hydropower generation in pico-tubular turbine.
- The damping ratio of the composite material, PLA reinforced with CFRP, is 0.0856, which is 43.4896% higher than pure PLA, and pure 62.1904% higher than CFRP. Hence, it has greater characteristics to dissipate vibration energy.

Due to limitations of the capacity of the research, recommendations are listed for further study in relation to the concept. This paper is open to further improvement such as:

- Conduct an FSI (Fluid Structure Interaction) analysis for cavitation – induced vibration for the evaluation of the composite blades.
- Effects of cavitation in a fabricated composite turbine blades.

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Availability of Data and Material	Not relevant.
Authors Contributions	All authors have equal participation in this article.

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