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# DESIGN AND PERFORMANCE ANALYSIS OF A NOVEL GASEOUS ORGANIC FLUID-POWERED PELTON TURBINE FOR WASTE HEAT RECOVERY AND SUSTAINABLE ENERGY GENERATION

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**Abstract: Diminishing reserves of fossil fuels and the surging global energy demand have revitalized the focus on harnessing greater energy potential from the emissions emanating from internal combustion engines. Organic Rankine Cycle utilizes organic fluids, emerged as a highly efficient method for capturing low-grade energy from exhaust gases. This process involves the organic fluid absorbing heat from the exhaust gases and subsequently vaporizing. The resultant gaseous fluid is then directed towards a turbine to extract mechanical energy.**

**Objectives of research are to prepare design and conduct an analysis of a Pelton turbine that can effectively generate mechanical work utilizing gaseous toluene as its operational fluid. It is noteworthy that, thus far, Pelton turbines have been exclusively developed for water as their working medium. Hence, this research represents a distinctive endeavor aimed at the development of Pelton turbines employing gaseous fluids.**

**This work entails an exhaustive analytical design procedure encompassing the determination of critical parameters and dimensions. Subsequently, CAD models are created based on the analytical design, followed by a comprehensive finite element analysis of pivotal components. Furthermore, experimental validation of the proposed system is also carried out.**

**Keywords:** *Fossil fuels, Organic Rankine Cycle, Gaseous fluid, Pelton turbine, Sustainable energy*

## INTRODUCTION

With the depletion of petrochemicals and apprehensions on environmental unbalance, efforts are made to enhance the efficiency of Internal Combustion (I.C.) engines. One promising technique for achieving this goal is waste heat recovery through the utilization of the ORC (Organic Rankine Cycle). This approach serves a dual purpose: it improves engine efficiency while simultaneously minimizing exhaust gas temperature. The ORC operates at par with conventional Rankine cycle, however employs organic fluids in place of steam. Henceforth, the heat energy at a low temperature could be transformed into work. Turbo expander in ORC system acts as power-generating component during the cyclic operations.

The turbo expander essentially functions as a turbine, and there are two primary categories of turbines based on the energy transfer mechanism: impulse turbines and reaction turbines. In impulse turbines, the entire pressure drop occurs within the nozzle, whereas in reaction turbines, the pressure drop takes place in both the nozzle and the vanes and runner. However, reaction turbines are more expensive to construct and exhibit lower efficiency compared to impulse turbines. Additionally, impulse turbines require smaller dimensions to achieve the same power output. Considering these factors, we have chosen the tangent flow impulse turbine, specifically the Pelton wheel turbine, as the preferred turbine type for the turbo expander.

Until now, Pelton wheel turbines have been exclusively designed for water as the working fluid. What sets this work apart is the endeavor to design a Pelton wheel for organic fluids. Specifically, the organic fluids will exist in gaseous form after absorbing heat from exhaust gases, adding a layer of complexity to the design process. The selection of the organic fluid becomes crucial, as it must remain stable with the materials used for its components.

Design of Pelton turbine involves two interconnected aspects: thermodynamic and structural. Thermodynamic data and working fluid selection are obtained through an extensive literature review. This work aims to provide a systematic approach to Pelton wheel design, encompassing force and strength calculations for critical components. Furthermore, a comprehensive finite element analysis of the bucket is conducted as part of this research.

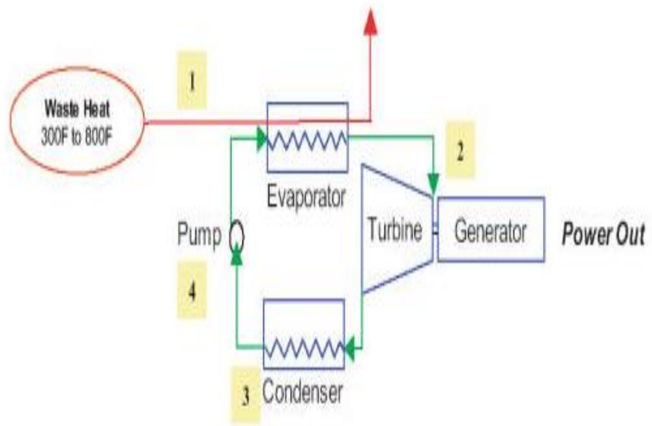


Fig. 1.Organic Rankine cycle

**Literature review:**

**The Pelton turbine is a popular research topic due to the increasing global demand for energy and the growing focus on renewable energy sources. Even a small increase in efficiency, such as 0.1%, can lead to a significant increase in electricity production.**

**Reference [2] provides the fundamental formulas for designing hydraulic Pelton turbines. J.S. Anagnostopoulos et al. [3] developed a numerical method using a Lagrangian approach to study the complex dynamics in the turbine. Vesley and Staubli's work in [4] and [5] investigated the effects of velocity of jet and its quality on the efficiency. Parkinson [6] formulated an unsteady investigation of a the runner along with simulations. Bryan's experimental work in [7] examined the impact of the jet-to-runner speed ratio on Pelton turbine efficiency. S. Yadav [8] proposed a bucket design modification to improve efficiency. V. Sharma et al. [9] studied the stress distribution in a bucket for hydraulic applications, while Solemslie et al. [10] used a parametric design approach to investigate how bucket parameters affect flow in a hydraulic turbine and its efficiency. Chen et al. [11] discussed the criteria for selecting working fluids in ORC (Organic Rankine Cycle) and screened 35 fluids for their suitability in ORC turbines.**

**The review of existing literature shows that while significant research has been conducted on the design of Pelton turbines for hydraulic applications, the use of Pelton turbines with gaseous fluids is still an underexplored area. Additionally, most turbine designs in ORC applications have been of the reaction type. As a result, the design of a turbine for ORC applications is a novel research frontier.**

**Design Methodology:**

Design methodology of Pelton Wheel are as shown in Fig. 2.

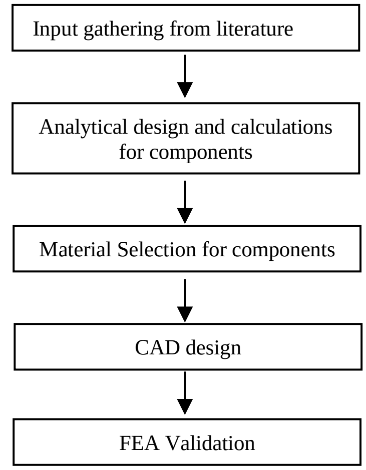


Fig. 1: Design methodology

The design methodology for a Pelton wheel for an ORC application involves a systematic approach with several key steps. These steps are crucial to ensure a comprehensive and effective design. Figure 2 illustrates the important stages in this methodology, which can be summarized as follows:

Literature Review and Input Gathering: The design process commences with an extensive literature review to gather relevant information and data. This includes studies on Pelton Wheel design principles, material properties, ORC applications, and performance considerations.

Identification of Design Requirements: Based on the literature review, the specific design requirements and constraints are identified. These may include operating conditions, working fluid properties, power output goals, and material limitations.

Thermodynamic Analysis: Conduct a thermodynamic analysis to determine the energy input and output requirements of the ORC system. This analysis helps in understanding the energy conversion process and guides the design parameters.

Structural Analysis: Perform a structural analysis to assess the mechanical stresses and strains that the Pelton Wheel components will experience during operation. This step ensures that the design is robust and can withstand the expected loads.

Material Selection: Select suitable materials for the Pelton Wheel components, considering factors such as strength, corrosion resistance, and cost-effectiveness. The choice of materials is critical for the wheel's performance and durability.

Design and Modeling: Create detailed 3D CAD models of the Pelton wheel components, including the wheel itself, buckets, shaft, and nozzle. These models serve as the basis for further analysis and validation.

FEA to simulate behavior of Pelton wheel under various operating conditions. This step helps in optimizing the design and ensures that stress levels remain within safe limits.

Prototype Development: Based on the validated design, develop a prototype of the Pelton Wheel for practical testing and evaluation. This step allows for real-world performance verification.

Testing and Validation: Conduct rigorous testing of the prototype to validate its performance, efficiency, and durability. Compare the results with design expectations and make necessary adjustments.

Optimization: Refine the design and optimize key parameters to enhance efficiency and performance further. This iterative process may involve adjusting bucket shapes, nozzle configurations, or material choices. Final Design: Once the design is thoroughly tested and optimized, finalize the design of the Pelton Wheel for ORC application, incorporating all the improvements and modifications.

Documentation and Reporting: Document the entire design process, including calculations, analyses, test results, and the final design specifications. Prepare comprehensive reports for future reference and publication.

This structured methodology ensures that the Pelton wheel for ORC application is designed systematically, meeting performance requirements and safety standards while allowing for continuous improvement through analysis and testing.

**Design Inputs:**

Toluene was chosen as the working fluid for this application following an extensive literature review, primarily due to its advantageous properties, including its classification as an isentropic fluid, allowing for heat retention during expansion, and its relatively high critical temperature of 318.75°C, permitting use at elevated temperatures without condensation. Additionally, Toluene's favorable vapor density reduces the size and material requirements of the power plant, while its resistance to condensation on turbine blades and corrosion enhances its suitability. In terms of design considerations, the project entails heating the working fluid (Toluene) to 300°C and pressurizing it to 10 bar using exhaust gases. Adiabatic flow conditions are assumed for the nozzle, with a standard inlet nozzle diameter of 12 mm and an inlet angle of 0°. Furthermore, the maximum flow rate through the turbine is constrained to address material-related limitations. These meticulous design considerations ensure that both the chosen working fluid and design parameters align seamlessly with the specific requirements of the system.

**Analytical Design:**

Nozzle

Elemental equations from fluid machinery design are used to determine critical dimensions of the nozzle. We shall use the following notations to designate the parameters of the nozzle:

Let 1 and 2 suffixes stand for inlet and outlet to the nozzles respectively

d1 = Inlet diameter of the nozzle = 12 mm (standard nozzle)

d2 = Outlet diameter of the nozzle

D = Nozzle outer body diameter

t = Thickness of the nozzle body

P1, P2 = Pressures at the nozzle inlet and outlet (P1=10 bar)

ν1, ν2 = Velocity of the fluid

ρ1, ρ2 = Density of the fluid

L = Length of the nozzle

A1, A2 = Area of the nozzle at the inlet and outlet

m = 0.5 kg/s – mass per unit time

$$A_1 = \frac{\pi}{4} d_1^{\,2} = 1.13 \text{ x } 10^{-4} \text{ m}^2$$

At a temperature of 300 °C, density of Toluene is $\rho_1$=477.28 Kg/m$^3$

Mass flow rate through a nozzle where minimum pressure is equal to critical pressure can be expressed as:

$$\dot{m} = A_2 \times \sqrt{nP_1\rho_1 \times (\frac{2}{n+1})^{\frac{n+1}{n-1}}}$$

In the above relation, polytrophic index of Toluene ($n$) is taken as 1.2 and then area $A_2$ at throat of nozzle is found as 35.29 mm$^2$. Diameter of nozzle at throat;

$$d_2 = \sqrt{\frac{4A_2}{\pi}} = 7 \text{ mm}$$

Velocity at nozzle inlet can be found as follows:

$$\dot{m} = \rho_1 A_1 v_1$$

$$\therefore \quad v_1 = \frac{\dot{m}}{\rho_1 A_1} = 9.27 m/s$$

Pressure of fluid at nozzle outlet is found by applying critical Pressure ratio principle.

$$\frac{P_2}{P_1} = (\frac{2}{n+1})^{\frac{n}{n-1}} = 5.64 \text{ bar}$$

Temperature at outlet of nozzle can be found from elementary adiabatic process relation as follows

$$T_2 = T_1(\frac{P_1}{P_2})^{\frac{1-n}{n}} = 520.8 \text{ K}$$

Let us denote coefficient of volumetric expansion for Toluene by $\beta$. As for most of the liquids, let us assume $\beta$ as 0.001 /K. Using $\beta$, value of density of fluid at nozzle outlet can be found as follows:

$$\rho_2 = \frac{\rho_1}{1 + \beta(T_2 - T_1)} = 503.5 \text{ kg/m}^3$$

Now, velocity at the exit of nozzle can be found as follows:

$$\nu_2 = \frac{\dot{m}}{\rho_2 A_2} = 28.13 \text{ m/s}$$

Empirical parameters of nozzle:

$D$=$2.5$ $x$ $d$= 30 mm

Length of nozzle, $L$=$8$ $x$ $d$≈ 100 mm

Thickness of nozzle, $t$=$(D – d_1)/2$= 9 mm

Wheel

Design of wheel:

Jet ratio = mean or pitch diameter of wheel/ jet diameter = 12 (assumed); usually (10 to 14).

Jet diameter, $d_j$ = 7 mm

Mean diameter, i.e. $D_m$,

$D_m$ = $Jet$ $Ratio$ $x$ $d_j$ =84 mm

Shaft diameter, $d_s$ = $0.3$ $x$ $D_m$ =25 mm

Shaft collar diameter, $d_c$ = $1.25$ $x$ $d_s$ ≈32 mm

Buckets, $n$, approximately = $15 + (D_m/2d_j)$ = 21 numbers

Further design of wheel depends on analysis of velocity diagram.

Following are the notations used in velocity diagram

$V_1$, $V_2$ are Absolute velocities

$V_{r1}$, $V_{r2}$ are Relative velocities of fluid

$V_{w1}$, $V_{w2}$= Tangential velocities

$u_1=u_2$= Bucket speed

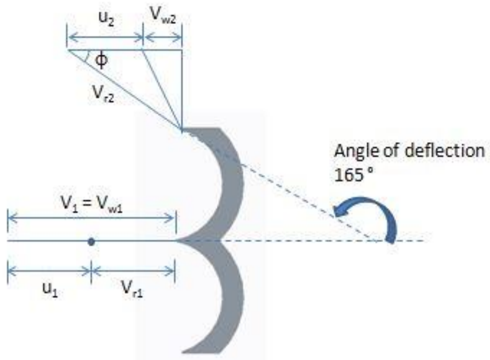


Fig. 3.Velocity diagram

$V_1$ = 28.13 m/s (velocity from nozzle)

$V_{w1}=V_1$ =28.13 m/s

$u = 0.45\ V_1$ =12.66 m/s

Angle of deflection of Pelton bucket should ideally be 180° for maximum change of momentum. However, if angle of deflection is 180°, fluid leaving a bucket hits following bucket. Hence, angle of deflection is limited to 165 °.

Relative velocity,$V_{r1}$ is determined from velocity diagram as

$V_{r1}= V_1 - u$ = 15.47 m/s

In practice, blade surfaces offer friction to fluid passage and hence relative velocity at outlet falls by 15% to relative velocity.

$V_{r2} = 0.85V_{r1} = 13.15$ m/s

$\Phi$ = 180 -165 =15°

Outlet tangential velocity,

$V_{w2}=V_{r2}(\cos\varphi) - u$ = 0.043 m/s

Power required to wheel by fluid = $\rho A V_1\ (V_{w1}+V_{w2}) \times u$ = 178.43 W

*Kinetic Energy* = $\frac{1}{2} \times (\rho A V_1) \times V_1^2$ =197.9 W

Hydraulic Efficiency of turbine, $\eta_{Hyd}$ = *Power required / Kinetic Energy* = 90.14 %

Speed, $N = \dfrac{60 \times u}{\pi \times D_m}$ = 2878 RPM

### Bucket

The design is governed by empirical relations in mm,

Axial width, $B = 3.4 \times d_j \approx 24$

Radial length, $L = 3 \times d_j$ = 21

Depth, $T = 1.2 \times d_j \approx 9$

Thickness, $t = 0.5 \times d_j$ = 3.5

Force acting on the nozzle, $F_X$, is calculated as follows:

$F_X = \rho A V_1 \times (V_{w1} - V_{w2})$ = 14.04 N

## MATERIAL SELECTION AND CAD DESIGN:

The selection of AISI 1018 as the material for the bucket, wheel, shaft, and nozzle, based on cost-effectiveness and adequate material strength, is a sound decision. Prioritizing cost considerations over the relatively higher frictional differences compared to materials like Aluminum and copper is justified, especially when cost-effectiveness is a key factor. The cost advantage of steel often outweighs any potential frictional differences it may exhibit. To potentially address friction issues, future research can explore the application of appropriate coatings on the steel surfaces of the bucket, which could help reduce friction and enhance the overall performance of the components. The material properties of AISI 1018, including a density of 7870 kg/m³, modulus of elasticity (E) of 205 GPa, yield point tensile strength of 370 MPa, ultimate tensile strength of 440 MPa, and a Poisson's ratio of 0.29, provide a clear understanding of the material's characteristics and its suitability for the application. CAD modeling of the bucket, wheel, nozzle, and shaft using ProE software, based on critical dimensions derived from previous calculations, is a crucial step in the design process. Visual representations of the CAD design will aid in the further analysis and refinement of these components, ensuring they meet the required specifications and performance criteria.

Fig. 4.Bucket model

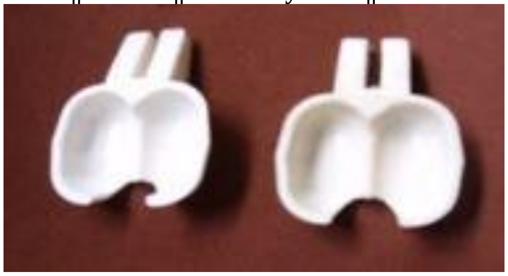

Fig. 5.Manufactured Buckets

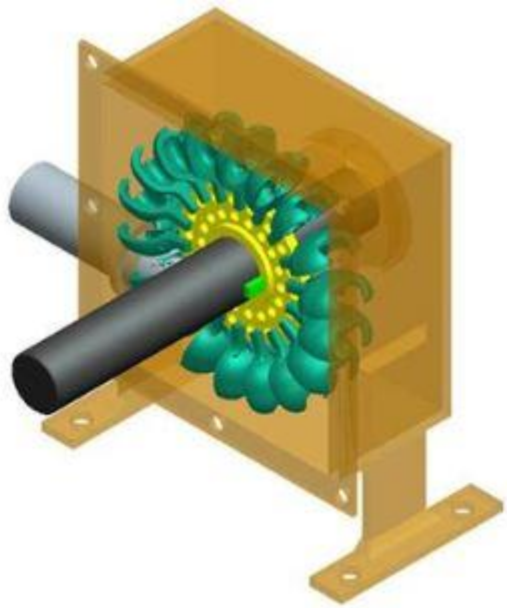

Fig. 6.Assembly model

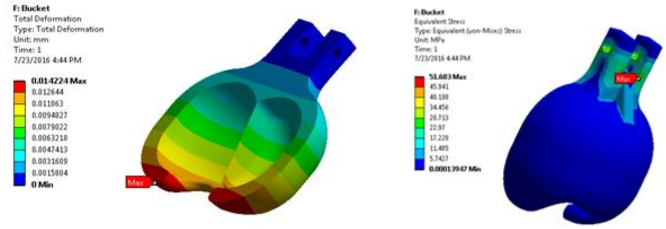

Fig. 7.Deflection and stress inBucket

FEA results for the Bucket are quite reassuring. The calculated deflection of the bucket is only 0.014 mm, which is minimal and indicates that there is very little deformation under the applied load. Furthermore, the Von-Mises stress is measured at 51.6 MPa, which is comfortably below the material's yield point stress. Based on these FEA findings, it can confidently be concluded that the bucket is structurally sound and safe when subjected to Jet Impact loading. The negligible deflection and stress levels well within the material's yield limit affirm its capability to withstand the specified loading conditions without compromising its structural integrity.

### Modal Analysis:

Starting six natural frequencies are determined as shown below..


Fig. 8.First Six Natural Frequencies of Bucket

The first modal frequency of the bucket, which is 2016 Hz, significantly exceeds the turbine's operating frequency of 48 Hz (equivalent to 2878 RPM). This substantial difference in frequencies confirms that there is no risk of resonance occurring. Therefore, based on FEA, it is concluded that the bucket is secure and not susceptible to resonance failure under the given operating conditions.

In the Nozzle Analysis, several critical parameters and conditions have been considered. The applied loads include an internal pressure of 1 MPa and external atmospheric pressure of 0.1 MPa on the body of the Nozzle outside the Casing. Additionally, there's external pressure inside the Casing, with a magnitude of 0.564 MPa acting on the body of the Nozzle inside the Casing. To ensure stability and accuracy, a fixity condition has been enforced at the cylindrical area where the Nozzle interfaces with the Casing. Moreover, the mating face with the Casing is supported using a 'Compression Only Support' condition, which helps simulate the realistic behavior of the Nozzle-Casing interface. Meshing has been carried out using Tetrahedral Elements with a size of 3 mm, enabling a detailed and accurate analysis of the Nozzle under the specified loads and boundary conditions. This comprehensive analysis approach provides valuable insights into the structural integrity and performance of the Nozzle Pressure Vessel.

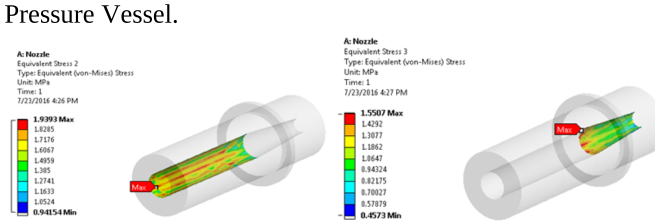

Fig. 9.Von-Mises Stress Induced in Nozzle

**VALIDATION:**

FEA is needed for solving complex differential equations, especially when dealing with intricate part geometries. It involves meshing the geometry, breaking it down into smaller elements, and calculating the displacement field for each element using matrices, which are then assembled to analyze the entire structure. In this context, Ansys R 16.1 Workbench serves as the platform for conducting the FEA. The first step in FEA is to establish the prerequisites by identifying potential failure modes and the loads imposed on each component. Special considerations are given to performance criteria for various components. For example, the bucket must withstand the impact force generated by the jet, and its natural frequency should exceed the excitation frequency. Meanwhile, the nozzle must endure the internal pressure resulting from the high-pressure gas. In the case of the Static Structural Analysis for the casing, a load of 28.67 Newtons is applied to the Bucket Splitter, and a fixity condition is applied to the screwing location. Meshing is performed using tetrahedral elements to ensure accurate results. This comprehensive approach allows for a thorough assessment of the structural behavior of the casing under the specified loads and boundary conditions, aiding in the design and optimization of the component.

These analysis parameters and conditions are crucial for evaluating the structural behavior of the Nozzle under the specified loading conditions and mechanical constraints.

### Casing Structural Analysis

For the Casing Structural Analysis, the following details and conditions have been considered:

Loads:

The system consists of several components, including a nozzle having mass of 0.48 Kg, a wheel assembly with a bucket composite mass of 0.75 Kg, a shaft weighing 0.25 Kg, and a key with a mass of 0.009 Kg. It operates under an internal pressure of 0.564 MPa. Fixity conditions are applied at the Casing Legs and Screwing Location, where a fixed boundary.

In this analysis, Tetrahedral Elements are utilized for meshingwith 5 mm for the Casing, 4 mm for the Wheel, 3 mm for the Shaft, and 2 mm for the Key. The Nozzle is represented as a point mass positioned at its center of gravity, connected to its designated location within the casing. The gravitational acceleration of 9.806 m/s² is applied to simulate the effects of the component masses. For interface modeling, a Frictional Contact approach with a Frictional Coefficient of 0.15 is employed for the Shaft-Casing interface, while Bonded Contact is used for all other interfaces, including Shaft-Key, Shaft-Wheel, and Key-Wheel..

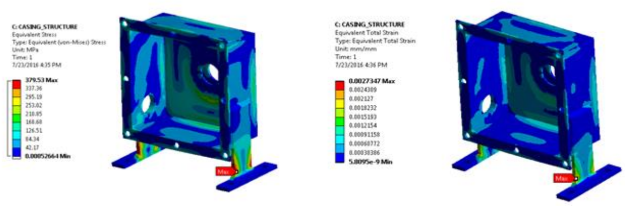

Fig. 10.Stresses and Strains in Casing

The Von-Mises stress in the Casing is found to be 379.5 MPa, slightly exceeding the material's yield limit of 370 MPa. However, the strain induced in the Casing is only 0.2%, well below the industry standard limit of 0.8% for steel components. It's important to note that the strain is concentrated near the legs of the Casing. Based on these Finite Element Analysis (FEA) results, it can be confidently concluded that the Casing Assembly is structurally sound and can safely withstand the component masses and pressure loads applied to it. The stress is only marginally above the yield limit, and the strain is well within acceptable limits, indicating a robust and reliable design.

## CONCLUSION:

The work presents a optimum design of a Pelton wheel to operate with organic gases. The turbine achieved hydraulic efficiency of 90.14%, resulting in a power output of 178.43 W under the specified input conditions. Meticulous 3D CAD models were developed for all components, relying on dimensions derived from the design phase. Additionally, FEA was conducted to evaluate critical aspects such as bucket deflection, nozzle and casing stresses and bucket modal frequencies.

Crucially, the FEA results indicated that all components experienced stresses well below their material yield limits. Although the casing stress slightly exceeded its yield limit, the recorded strain of 0.2% remained within acceptable bounds. Consequently, the detailed FEA analysis confirms the structural integrity and safety of the designed Pelton wheel under the specified operating loads.

Looking forward, future research avenues include experimental investigations, CFD analysis to study pressure distribution, and the estimation of losses due to friction. These endeavors will further enhance the understanding and optimization of the turbine's performance, facilitating its potential application in real-world scenarios.

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