INVESTIGATION OF STRESSES IN ALUMINIUM A356 ALLOY PELTON TURBINE BUCKETS

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Abstract

The need to predict performance of Pelton turbines in terms of efficiency and dynamic behavior under different operating conditions is paramount. It has been established that an increase in performance leads to a decrease of the ratio between the bucket width and the jet diameter. This leads to an increase in cyclic load and therefore an increase in stresses on the buckets and the turbine hub leading to failure. This paper seeks to demonstrate how to optimize the bucket profile in order to hence performance of Pelton turbine buckets for Pico hydro electric systems. Two models of 152mm pitch circle diameter Pelton turbine buckets were designed using Computer Aided Design (CAD) software and investigation on forces and stresses done using structural analysis on ANSYS software. The simulations results were compared with experimental results for the two models casted from recycled aluminium A356 alloy. The investigation indicated that there was a 14.2% reduction on stresses developed by introducing a ridge on the buckets backside. The result was a higher power output achieved by optimizing the buckets profile without altering the bucket width in relation to water jet diameter.

Key words: Pelton turbine buckets, Simulation using ANSYS, Bucket profile optimization

Nomenclature

P_{jet}	Kinetic power of the water jet	[W]
m	Mass flow rate	[kg/s]
C	Linear velocity of water jet	[m/s]
P_H	Hydraulic power	[W]
Q	Flow rate	[m^3/s]
r	Density	[kg/m^3]
g	Gravitational acceleration	[m/s^2]
H	Head	[m]
σ	Stress	[N/m^3]
$[D]$	Flexibility matrix	
ϵ	Elastic strain	
E	Young's modulus	[N/m^2]
DT	Change in temperature	[^{0}]
n	Poisson's ratio	
u v	Velocity in x and y direction	
G	Shear modulus	[Pa]
xyz	Special coordinates	

1.0 Introduction

The reduction in use of fossil fuel as energy sources in rural areas which are not connected to power grid is a major concern in order to stop a further decline in the environment [1]. The design of Pelton runner, patented in 1880 by Lester Allen Pelton, has changed very little but notable technological changes have taken place aiming at increasing the power output [4].

Based on affordability small turbines are forced to operate above optimum conditions in order to maximize power predisposing them to fail. Overworking the turbine subjects the runner buckets to a combination of stresses caused by centrifugal forces and cyclic loads. Centrifugal forces are induced by the mass of fast rotating turbine runner. Cyclic loads are induced as the water jet impinges on the buckets at high speed. Individual buckets undergo high repetitive forces as the turbine operates making the runner vulnerable to excessive loading failure. A study was done on castings of Pelton turbine buckets from recycled aluminium A356 alloy.

In order to increase power output of Pelton turbines different researchers have used different approaches. Mayse *etal*., developed a new design of Pelton wheel called hooped Pelton turbine. The design was based on redistribution of function on hoops to which the buckets are mounted. These allowed stresses to be more efficiently distributed around the runner. The percentage reduction of VON MISES stresses was of the order of 67.19%. The challenge with this method is that more material is used during fabrication and requires a more advanced fabrication technique.

(Vesely, *et.al*) conducted the upgrading of a 62.5 MW Pelton turbine by refurbishing the runner and nozzles. The rated capacity increased by 9% and efficiency increased by 1.4%. Heinz et.al [10] carried experiment to estimate the influence of splashing water distribution in turbine casing. They showed that the casing has a great influence to the operation of Pelton turbines and so it is very important to include casings as important factor in all investigations. Zhang *et al.,* concluded that the quality of a jet of a Pelton turbine has major impact on the overall efficiency of the turbine. They modified the jet needle tip angle and nozzle seat ring to achieve an increase in efficiency by 2%.

Few researches have been done on Pelton turbine buckets profiles and this paper seeks to address this issue. By modifying the bucket profile of a Pelton bucket one can significantly improve on power output without much effect on the turbine operation. This research investigates on how modification of the bucket profile reduces stresses induced by the water jet on buckets making it possible to use smaller sized turbine to produce more power.

2.0 Mathematical Description

In order to increase power output of a Pelton turbine of a given size two parameters were varied. These are head and nozzle diameter. Numerical calculations of jet force as a result of head and nozzle diameter variation were done in order to get the equivalent force impacted on the inner surface of the buckets. The results of forces from the numerical calculations were used as inputs conditions to do the structural analysis using ANSYS software. Stresses developed on the buckets were obtained. To validate the results, bending experiments were carried out on bending testing machine.

2.1 Numerical Calculations

Calculations based on the empirical formulas of Pelton turbines were done in order to determine the optimum operating conditions for a 0.1524m pitch circle diameter turbine.

$$P_{jet} = \frac{1}{2}\dot{m}c_1^2 \quad (1)$$

$$P_H = Q.\rho.g.H \quad (2)$$

Then the parameters are varied in order to increase the turbine power output and the jet loading on the buckets monitored. Investigations were done for power output between 1kW and 10kW.

2.2 Modeling

Solid modeling was used to develop the models for finite element analysis used in ANSYS structural analysis. This was done using Autodesk Inventor software. The development of parameterized design data in the form of CAD solid models for the bucket was directly imported into the ANSYS software and a block-structured grid analyzed. The simulations on stresses were performed on ANSYS v12 software based on the following formulas.

$$\{\sigma\} = [D]\{\epsilon^{el}\} \tag{3}$$

$$\sigma_y = \frac{E_y}{h}\left(\upsilon_{xy} + \upsilon_{xz}\upsilon_{yz}\frac{E_z}{E_y}\right)(\epsilon x - \alpha_x \Delta T) + \frac{E_y}{h}\left(1 - (\upsilon_{xz})^2\frac{E_z}{E_x}\right)(\epsilon_y - \alpha_y \Delta T) + \frac{E_z}{h}\left(\upsilon_{yz} + \upsilon_{xz}\upsilon_{xy}\frac{E_y}{E_x}\right)(\epsilon_z - \alpha_z \Delta T) \tag{4}$$

$$\sigma_x = \frac{E_x}{h}\left(1 - (\upsilon_{yz})^2\frac{E_z}{E_y}\right)(\epsilon x - \alpha_x \Delta T) + \frac{E_y}{h}\left(\upsilon_{xy} + \upsilon_{xz}\upsilon_{yz}\frac{E_z}{E_y}\right)(\epsilon_y - \alpha_y \Delta T) + \frac{E_z}{h}(\upsilon_{xz} + \upsilon_{yz}\upsilon_{xy})(\epsilon_z - \alpha_z \Delta T) \tag{5}$$

$$\sigma_z = \frac{E_z}{h}(\upsilon_{xz} + \upsilon_{yz}\upsilon_{xy})(\epsilon x - \alpha_x \Delta T) + \frac{E_z}{h}\left(\upsilon_{yz} + \upsilon_{xz}\upsilon_{xy}\frac{E_y}{E_x}\right)(\epsilon_y - \alpha_y \Delta T) + \frac{E_z}{h}\left(1 - (\upsilon_{xy})^2\frac{E_y}{E_x}\right)(\epsilon_z - \alpha_z \Delta T) \tag{6}$$

$$\sigma_{xy} = G_{xy}\epsilon_{xy}$$

$$\sigma_{yz} = G_{yz}\epsilon_{yz}$$

$$\sigma_{xz} = G_{xz}\epsilon_{xz} \tag{7}$$

The results obtained are compared with the experimental results in terms of:

i. Deflection and
ii. Stress on the bucket

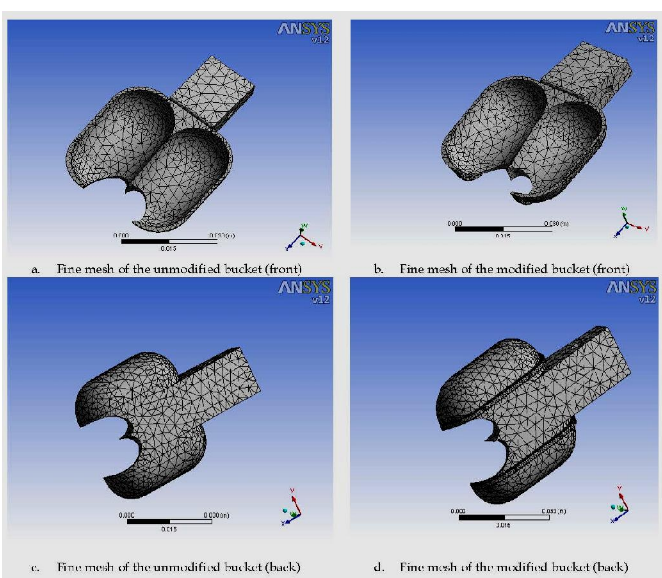


Figure 1: Meshed models used in the simulation

2.3 Experimental Validation

The traditional method to determine the stress in the Pelton bucket is a stress analysis based on classical beam theory [3]. This is known to deliver conservative results and so leads to very reliable mechanical designs for the normal range of Pelton turbines. The results are then compared with finite element analysis results in order to predict the stresses on the bucket. Three tests were carried out on a bending testing machine shown in Figure 2. The tests were:

i. Bucket bending deflection.
ii. Bucket bending strength.
iii. Bucket pull-out force.

The set up of the experiment is as shown in Figure 3. Selected ranges of results for deflection against applied load were used to counter-check the simulation results. A suitable load was used that would not take the bucket out of the elastic region, but would give a deflection that was large enough to be measured reliably.

Figure 2: Bending testing machine

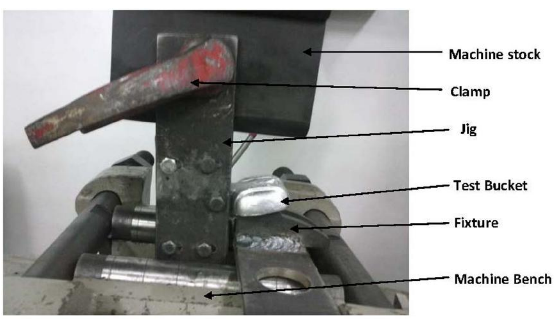


Figure 3: Set up of the bending experiment

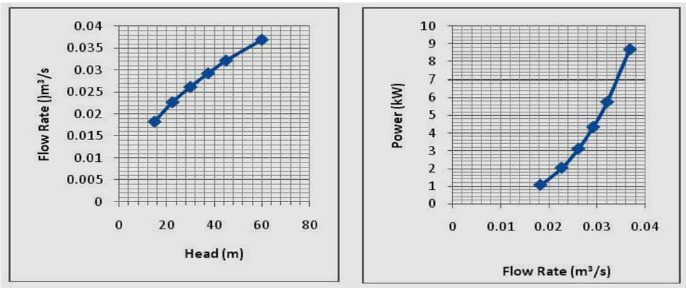
3.0 Results and Discussion

Modeling, experimental results of the two bucket models are explained below. Optimum operating conditions for 152.4mm pitch circle diameter Pelton turbine that would not fail under cyclic loading were obtained. ANSYS was used to simulate the behavior of a mechanical part under structural loading conditions and generated the results presented in this paper. Evaluation of the bucket designs was done by considering this information in conjunction with experimental tests results. A quality approach to engineering design usually mandates physical testing as the final means of validating structural integrity to a measured precision.

3.1 Modeling

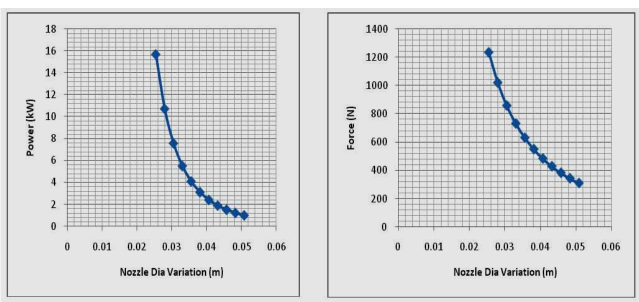
Stress analyses of unmodified and modified bucket were done with the help of ANSYS Workbench v12. Effect of modification on the bucket profile on stress developed was depicted. Jet force for head variations between 15m and 60m was obtained to be between 275.3N and 1105.2N and that of on nozzle diameter variations between 0.028m and 0.048m was obtained to be between 309.2N and 1236.8N using empirical formulas. These were based on power output from 1kW and 10kW. Figures 4 show variation of head between 15 meters and 55 meters with corresponding power output, flow rate and jet force. Figures 5 show variation of nozzle diameter between 0.025m and 0.055m with corresponding power output and jet force.

The different jet loads on the bucket deduced from the empirical formulas were used as the input conditions for the structural analysis in ANSYS simulation.



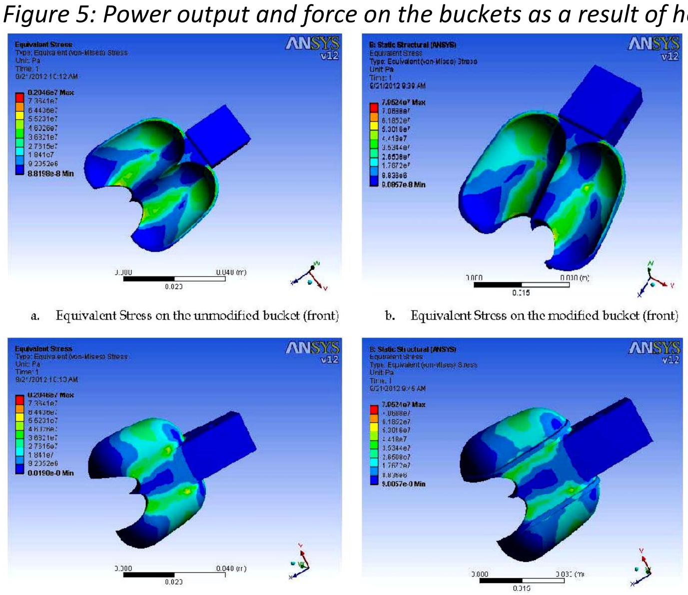
(a) Flow Rate Vs Head variation   (b) Power Vs Flow Rate

Figure 4: Flow rate and power output as a result of head varation



(a) Power Vs Nozzle Diameter Variation   (b) Force Vs Nozzle Diameter Variation

Figure 5: Power output and force on the buckets as a result of head varation



a. Equivalent Stress on the unmodified bucket (front)   b. Equivalent Stress on the modified bucket (front)

c. Equivalent Stress on the unmodified bucket (back)   d. Equivalent Stress on the modified bucket (back)

Figure 6: Simulation results on stresses

3.2 Laboratory Experiments

Three set of experimental analysis were done on the two types of buckets. The tests are explained below.

Bucket Bending Deflection

Full bending test was done on the buckets using bending testing machine and loading vs deflection graphs obtained.

Bucket Bending Strength

Using the calculations for equivalent Jet load that indicates the ultimate jet force on the bucket and applying equivalent amount of force to the clamped bucket until it fails in bending was done.

Bucket Pull-out Force

The force required to axially displace the clamp was measured and this indicated the level of security of the buckets upon the hub for vertical shaft arrangement.

Figure 7 shows how reduction in nozzle diameter i.e. in order to increase the velocity of the water jet (kinetic power) courses an increase in stresses on the buckets. Increase in the operation head means an increase in potential power and as a result more force will be induced on the buckets and thus there is an increase in stress as head increases as shown in Figure 8.

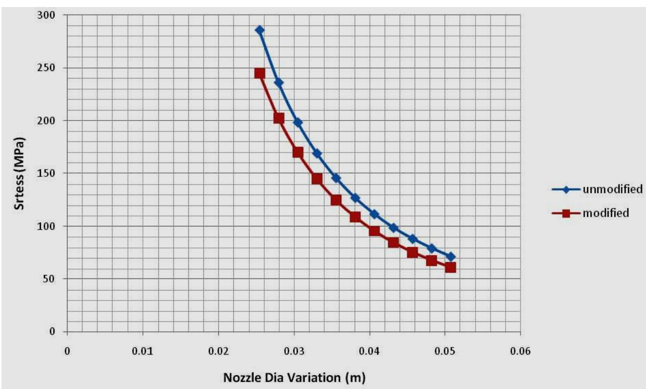


Figure 7: Variation on nozzle diameter with stress

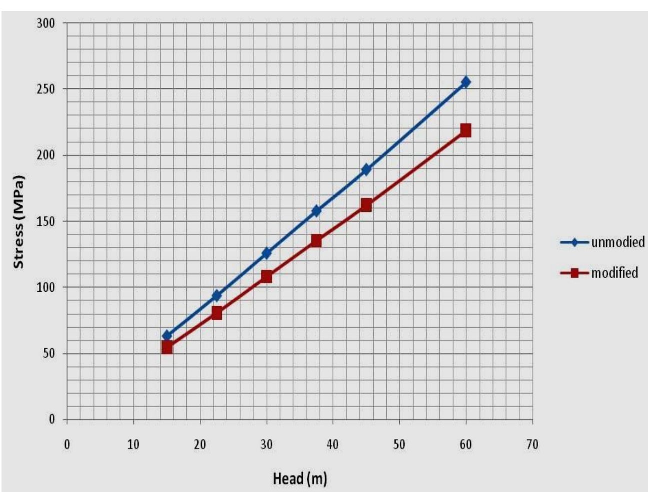


Figure 8: Variation on head with stress

4.0 Conclusion

The research involved modeling and validation of the results experimentally using material that are readily available in Kenyan market. The results showed that by modifying the Pelton bucket profile there was a 14.2%

reduction in stress. The modeling results showed that 152mm pitch circle diameter turbine can be operated to produce up to 5kW within a good safety factor without fear of failure in cyclic loading.

Experiments conducted verified the modeling results and at equivalent jet loading to produce 5kW power output, the buckets will expiries stress of 150MPa which is below the yielding strength of the recycled aluminium A356 alloy. The results will enhance existing knowledge on the performance of recycled aluminium A356 in production of Pelton turbine by castings. This will allow the next generation of Pelton turbines to be designed making use of a combination of empirical know-how from previous experience and an improved physical understanding of the complex Pelton bucket profile.

Acknowledgement
The authors would like to thank J.K.U.A.T for the support.

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