Open Access Journal

Journal of Sustainable Research in Engineering Vol. 2 (2) 2015, 70-75

Journal homepage: www.jkuat-sri.com/ojs/index.php/sri/index

Performance of Aluminium A356 Alloy based Buckets towards Bending Forces on Pelton Turbines.

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Abstract Based on affordability, small turbines are forced to operate above optimum conditions in order to produce maximum power making them to fail under excessive water jet loading. Performance of Aluminium A356 alloy based Pelton turbine was evaluated under bending forces condition. The bucket experiences vibration due to repeated loading since it is mounted as a cantilever onto a disc. If not properly designed or manufactured, a resonance may occur and severely increase the dynamic stress causing failure. It is important to note that the design of the bucket ensures conversion of jet force into tangential force that eventually causes rotation. However, during this conversion shear and bending forces are induced at the regions where buckets are mounted to the turbine ring. Bending strength was analyzed with different properties of the buckets. It was observed that bending resistance is an important parameter that controls the turbine failure due to cyclic loading on the buckets. Alumina content was found to have significant influence on the bending performance of the buckets. Bending forces were predicted using structural analysis on ANSYS modeling software. The results were observed to lie in close proximity to the experimental results.

Keywords Pelton buckets, Aluminium alloy, Structural analysis.

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_{jet}	Flow rate	[m^3/s]
r_w	Density	[kg/m^3]
g	Gravitational acceleration	[m/s^2]
H_n	Head	[m]
F_{jet}	Jet force	[N]
x	Ratio of bucket velocity to jet velocity	
ζ	Efficiency factor for flow in bucket	
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. Introduction

The socio-economic development and increased living standards with the fast growing industry has led to a major increase in electricity demand and generation. Being the basic input of all kinds of economic activity, electrical energy has become an indispensable component of social life. Less than 20% of the total population and 5% of the rural population in Kenya has access to electricity.

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].

Based on affordability small turbines are forced to operate above optimum conditions in order to maximize power predisposing them to fail [2]. 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 [3]. 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. Previous research consists mainly of numerical, experimental and analytical studies [4].This ranges from Pelton bucket profile optimization, flow analysis, studies on force redistribution on the bucket, analysis on stress distribution on the bucket, nozzle and casing modification and computer simulations to maximize operation conditions.

Models have been used to predict the power output of hydraulic machines using computational simulation aiming at reducing the time required at the design phase [5]. Varies developments on computational simulation for water turbines have already led to substantial improvement in the design and performance of hydro turbines. One of them is Computational Fluid Dynamics (CFD) which is a computer-based tool mainly used for simulating the behavior of systems involving fluid flow and heat transfer processes [6]. The challenge is that the software used is costly.

Mayse F. et al. [7] developed a design of Pelton wheel called hooped Pelton turbine. It is important to note that the design of the bucket ensures conversion of jet force into tangential force that eventually causes rotation. However, during this conversion, shear and bending forces are induced at the regions where buckets are mounted to the turbine ring. These effects were minimized by adding two hoops on either set of buckets. This calls for extra cost on the material used to fabricate the hoops.

Simplified charts have been developed to aid in selection of turbines especially for small scale plants. George A. et al. [8] presented a technology that accelerated the development of hydro turbines by fully automating the initial testing process of prototype turbine models and automatically converting the acquired data into efficiency hill charts. Unlike reaction turbines such as Propeller or Francis, the Pelton runner does not have to be designed for specific working conditions. A given turbine can be used for a range of heads and flows; this leaves a big gap for experimenting of performance of Pelton turbines based on material from which the runner is fabricated.

2. Theory and Modelling

In order to increase power output of a Pelton turbine of a given size two parameters were varied. These are:
i. Head
ii. Nozzle diameter

2.1. Numerical Calculations

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. 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 (pcd) turbine. Stresses developed on the buckets were obtained. To validate the results, bending experiments were carried out on bending testing machine. The jet force was calculated from [3]:

$$F_{jet} = \rho_w . Q_{jet} . C_v . \sqrt{2g.H_n} . (1-x)^2 . (1+\varsigma . \cos\gamma) \qquad (1)$$

The first consideration is the flow. The speed of the water in the jet is dependent only on the head, and the flow is determined by the speed, the area and the number of jets. Velocity of the jet is given by [3]:

$$V_{jet} = C_v . \sqrt{2g.H_n} \qquad (2)$$

$$P_{jet} = \frac{1}{2} \dot{m} c_1^2 \qquad (3)$$

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

2.2. Modelling

The results of forces from the numerical calculations were used as input conditions to do the structural analysis using ANSYS software. Solid modeling was used to develop the models for finite element analysis used in ANSYS structural analysis as shown in Figure 1. 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.

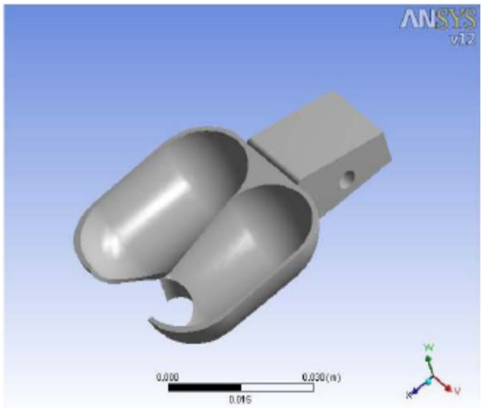


Fig. 1a. Bucket solid CAD model

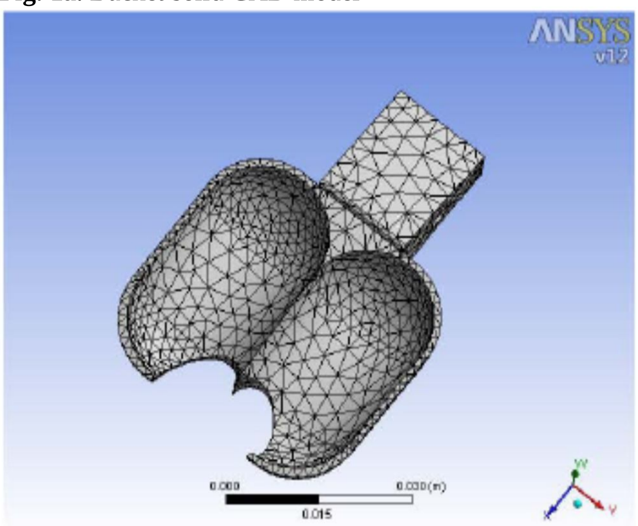


Fig. 1b. Pelton Bucket CAD meshed models

The simulations on stresses were performed on ANSYS software based on the following formulas [9].

$$\{\sigma\} = [D]\{\epsilon^{el}\} \tag{4}$$

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

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

$$\sigma_z = \frac{E_z}{h}(\upsilon_{xz} + \upsilon_{yz}\upsilon_{xy})(\epsilon x - \alpha_x \Delta T) + \frac{E_z}{h}\left(\upsilon_{yz} + \upsilon_{xz}\upsilon_{xy}\frac{E_y}{E_x}\right)(\epsilon_y - \alpha_y \Delta T) + \frac{E_z}{h}\left(1 - (\upsilon_{xy})^2\frac{E_y}{E_x}\right)(\epsilon_z - \alpha_z \Delta T) \tag{7}$$

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

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

$$\sigma_{xz} = G_{xz}\epsilon_{xz} \tag{8}$$

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

i. Deflection

ii. Stress on the bucket

3. Methodology

CAD drawing of the bucket was made with the dimensions of the bucket based on empirical relations for a 152.4 mm pitch circle diameter Pelton turbine. The dimensions were as follows; length = 2.3 times the diameter of the jet, width = 2.8 times the diameter of the jet, depth = 0.6 times the diameter of the jet, angular deflection of the jet = 165^o and bucket angle of setting = 15^0.

Computer aided manufacturing (CAM) was then implemented by running the computer software (the G-codes) to control related machinery in the manufacturing of master pattern. Computer Numerical Control (CNC) programming and simulation was developed using OneCNC software. An electronic database in form of G-codes used for machining the master pattern was created. This was then exported to a CNC machine as shown in Figure 2.

Production of the master pattern from a high density plastic as shown in Figure 3 was done and the pattern used to carry out casting.

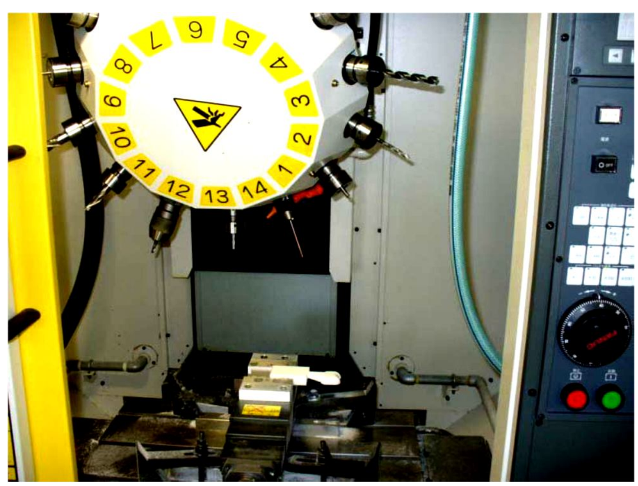


Fig. 2. Machining process of the master pattern

Fig. 3. Master pattern made from plastic

Casting was adapted as one of the viable fabrication techniques due to the complex shape of the bucket and the ease of reproducing the buckets. [9].

Aluminium A356 alloy of chemical composition as shown in Table 1 was used to cast the test buckets. Two methods were used to produce the test buckets i.e. investment casting and sand casting.

Table 1. Aluminium A356 alloy composition

Component	Si	Cu	Mg	Ti	Fe	Mn	Zn	Al
Wt %	6.5-7.5	Max. 0.2	0.25-0.45	Max 0.2	Max. 0.2	Max. 0.1	Max. 0.1	93

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 were then compared with finite element analysis results in order to predict the stresses on the bucket. Two tests were carried out on a bending testing machine as shown in Figure 4. The tests were:

i. Bucket bending deflection.
ii. Bucket bending strength.

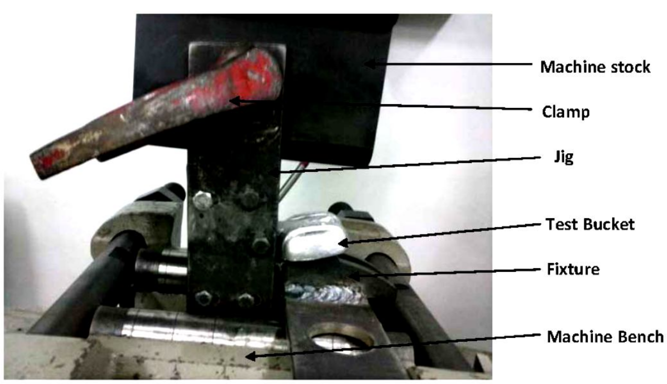


Fig. 4. Set up of the bending experiment

4. Results and Discussion

Optimum operating conditions for 152.4 mm pitch circle diameter Pelton turbine that would not fail under cyclic loading were obtained. Evaluation of the bucket designs was done by considering this information in conjunction with experimental tests results. Effect of modification of the bucket profile on stress developed was depicted. Jet force for head variations between 15m and 60 m was computed and found to be between 275.3 N and 1105.2 N. In the case of nozzle diameter variations between 0.028 m and 0.048 m jet force was computed and found to be between 309.2 N and 1236.8 N using empirical formulas [3]. The ranges of variations were based on power output between 1 kW and 10 kW for a 152.4 mm pitch circle diameter Pelton turbine.

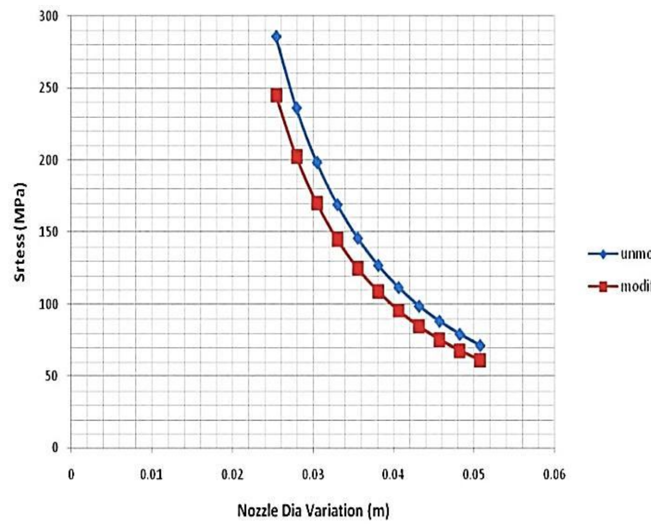


Fig. 5. Variation on nozzle diameter with stress

Figure 5 shows how reduction in nozzle diameter i.e. in order to increase the velocity of the water jet (kinetic power) causes an increase in stresses on the buckets. Based on the yield strength of the aluminium alloy used to fabricate the buckets optimum nozzle diameter was found to be 0.016 m. As a percentage of the turbine pitch circle diameter of 152.4 m this was 10.93% which is in line with the empirical value that states that the nozzle diameter should be about 11% of the turbine pitch circle diameter. At this nozzle diameter the equivalent power output was calculated to be 5.36 kW.

Figure 6 shows how stresses vary with increase in head. 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. It indicates that an increase in head creates more potential energy to run the turbine at accelerated speeds causing more cyclic loading and thus the increase in stress. Between the range of 15m and 60m the variation of stress against head was found to be linearly proportional. Based on the yield strength of the aluminium alloy A356 that was used which is around

206.84MPa, limits were reached at an average head of 52.93m with an equivalent power output of 6.80kW.

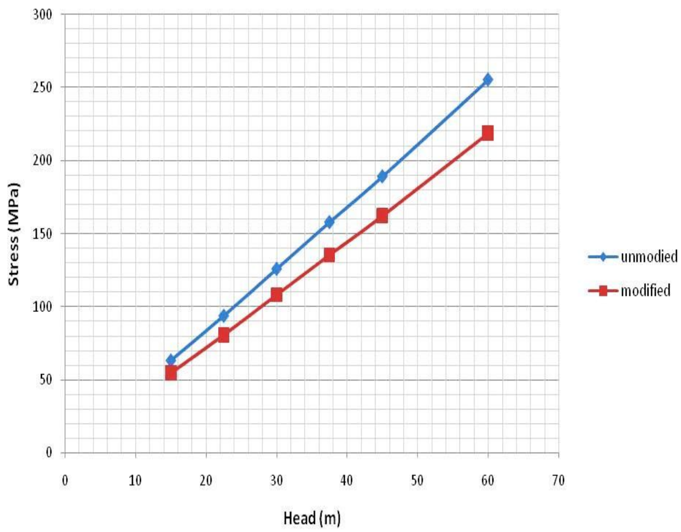
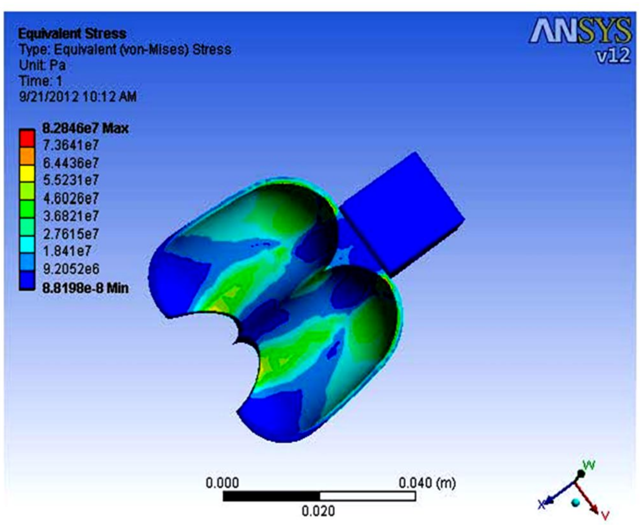
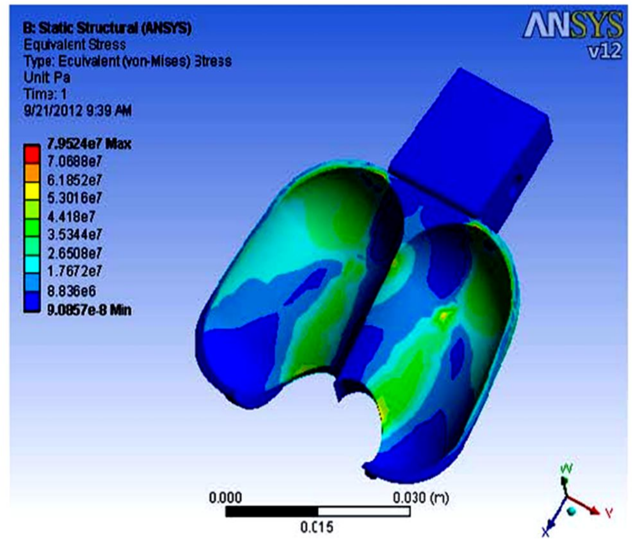


Fig. 6. Variation on head with stress

Figure 7 show ANSYS results on stress distribution patterns on the two buckets models. High stress values were noted at the point where the bucket joins the root and at the center line of curving within the bucket. Also noted was that for the modified buckets the stress values were less as compared to the unmodified bucket. This was due to even distribution of stress along the back ridge introduced on the bucket.



a. Equivalent Stress on the unmodified bucket (front)



b. Equivalent Stress on the modified bucket (front)

Fig. 7. Simulation results on stresses

The main fatigue load on a bucket comes from the jet force. This creates a bending stress in the stem every time a bucket passes a jet. Most turbines will exceed one million bending cycles on the bucket within the first few weeks of operation. The point at which the worst fatigue stress occurs depends on how the buckets are fixed. For a single piece casting the maximum bending moment occurs at the section where each bucket joins the disk, x-x as shown in Figure 8. This gives a maximum tensile bending stress at the point.

Stresses on the stem of the bucket are caused by two load cases that can cause the bucket to break off.

i. Runaway- this occurs when the external load is removed from the turbine and the runner accelerates to a high speed. This produces a large centrifugal force in the buckets, which can snap the stem of the bucket.
ii. Fatigue load- caused by bending stress on the stem due to the water hitting the bucket every time it passes a nozzle.

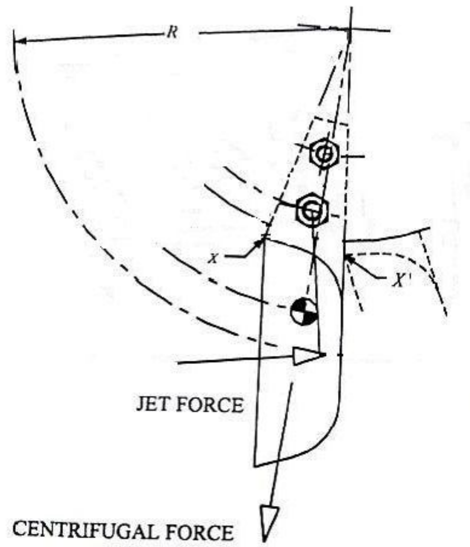


Fig. 8. Main forces on the bucket

The main cause of failure is the fatigue load. The runaway load is easier to address by introducing a clamping mechanism on the stem of the bucket.

The jet force was drawn for the maximum bending moment position, and has an effective moment arm of 0.163 x D about the neutral axis (based on the empirical data). 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.

5. Conclusions

The research involved modeling and validation of the results experimentally using material that are readily available in Kenyan market.

1. The results showed that by modifying the Pelton bucket profile there was a 14.2% reduction in stress. The modification was done by introducing ridge at the buck of the bucket. This was to ensure even distribution of the stress and thus the modified buckets could handle more stress compared to the unmodified.
2. The modeling results showed that 152 mm pitch circle diameter turbine can be operated to produce 5 kW within a good safety factor without fear of failure in cyclic loading at an optimum head of 35 m and a nozzle diameter of 0.016 m.
3. Experiments conducted verified the modeling results and at equivalent jet loading to produce 5 kW power output, the buckets will expiries stress of 150 MPa which is about 70% of 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 thank Prof. Nemoto and Prof. Ando in Ashikaga Institute of Technology, for technical and experimental set support. Mr. Mwai, Mr. Muigai and Mr. Nyanjui for casting technology transfer. This work was also supported by JICA through the Bright Project.

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