Investigate Curvature Angle of the Blade of Banki’s Water Turbine Model for Improving Efficiency by Means Particle Swarm Optimization

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Abstract—Turbines are used to convert potential energy into kinetic energy. Turbine blades are designed expertly with specific curvature angles. The output power, speed, and efficiency of a water turbine are affected by the curvature angle of the blade because water energy is absorbed by the blade in contact with the water flow. The particle swarm optimization (PSO) algorithm can be used to design and optimize micro hydro turbines. In this study, the blade curvature angle in a Banki’s water turbine model is investigated using the particle swarm optimization algorithm to obtain the highest output power, speed, and efficiency in the water turbine. Mathematical and experimental models are employed to investigate the blade curvature angle. The result shows that a curvature angle of 15° provides higher output power, speed, and efficiency than angles of 16° and 17°, despite the fact that 16° is commonly used in commercial production.

Keywords blades, turbine, PSO, hydropower.

1. Introduction

Today, energy resources play an important role in economics, politics, social life, and the sciences. This is especially so in Indonesia where conventional energy resources are decreasing and prices are tending to rise[1]. Renewable energy [2] sources can potentially substitute for conventional energy resources and to smoothly solve these problems [3]. Indonesia has many renewable energy resources, including river water, sea water flow, tides, wind, and geothermal and solar energies, however hydropower plants are predominantly used for renewable energy supply as compared with the other alternatives [4], [5], [6]. One type of hydropower is micro-hydro power, which has become popular since it is simpler in its design, easier and cheaper to operate, faster to install, and has lower environmental impacts than the others [7].

The water turbine is a simple machine usually made of wood or steel with a fixed blade attached to a surrounding wheel [8], [9]. This blade is driven by a stream of water flowing around the wheel. Water pressure on the blade produces torque on the shaft to make the wheel spin [10]. The potential energy in the accumulated water continuously applied to the blade attached to the turbine wheel generates kinetic energy on the turbine shaft [11]. Sometimes the water flows vertically at the top, the middle, or the bottom of the turbine wheel. The curvature angle of the blade is a key issue in the energy conversion process.

One type of water turbine model is called the Banki’s model. The theory behind the Banki’s model, written by Mockmore and Merryfield in 1949, can be found in reference In the Banki’s water turbine, the nozzle[12], whose cross-sectional area is rectangular, discharges a jet into the full width of the wheel, and is oriented at an angle of 16 degrees to the tangent of the wheel’s periphery. More simply stated,
the curvature angle of the blade is 16 degrees. Based on reference[12], the authors argued that the curvature angle is the key for obtaining the highest possible output power, speed, and efficiency of the water turbine.

This study explores mathematical and experimental design models of the Banki’s water turbine to investigate the performances resulting from 15°, 16°, and 17° blade curvature angles. Our goal is to determine the optimal turbine parameters for achieving the highest possible power output, efficiency, and speed. The mathematical models explored in our research are governed by principles of conservation of mass, momentum, and energy, and are described in detail in section 2. From the mathematical model we use, optimum parameter characteristics are simulated by the particle swarm optimization (PSO) [13], method to obtain the optimal blade curvature angle. Then, experimental design models are used to clarify the blade’s curvature angle obtained by PSO to assess whether or not it yields maximum values for all of the parameters. In order to validate the performance of each experimental turbine model, it was necessary to utilize the same turbine model parameters, including wheel diameter, thickness, material, number of blades, water discharge, and load generators.

2. The Banki Formula

2.1 Theoretical Background

The Banki’s turbine, as described in[12], consists of a nozzle and a turbine runner. The runner is built of two parallel circular disks joined at their rims with a series of curved blades. The nozzle, whose cross-sectional area is rectangular, discharges a jet of water the full width of the wheel, which enters the wheel at an angle of 16 degrees to the tangent of the wheel’s periphery. The shape of the jet is rectangular, wide, and fairly shallow. The water strikes the blades on the rim of the wheel, as shown in Figure 1, flows over the blade, then leaves it, passing through the empty space between the inner rims, enters a blade on the inner side of the rim, and is then discharged at the outer rim. The wheel is an inward jet wheel. Since the flow is essentially radial, the diameter of the wheel does not practically depend on the amount of water impact, and neither does the desired wheel breadth depend on the quantity of water.

The path of the jet through the turbine is assumed to be that at the centre of point A, as shown in Figure 1, at an angle of α1 tangent to the periphery. So the velocity of the jet path water through the turbine can be written as follows:

\[ V_1 = C \sqrt{2gH} \]  

(1)

The relative velocity of the water at the entrance, \( v_1 \), can be found at \( u_1 \) if the peripheral velocity of the wheel at that point is known. \( \beta_1 \) is the angle between the forward directions of the two latter velocities.

For maximum efficiency, the angle of the blade should equal \( \beta_1 \). Line AB represents the relative velocity of blade \( v_2 \) and \( \beta_2 \) represents the peripheral velocity of the wheel at that point. An absolute velocity of the water at the exit of blade \( v_2' \) can be determined by \( v_2', \beta_2', \) and \( v_2 \). The angle between the absolute velocity and the velocity of the wheel at this point is \( \alpha_2' \). The absolute path of the water when it is flowing over blade AB can be determined, as can the actual point when it leaves the blade. An absolute velocity \( V_2' \) assumes a constant value, so that \( V_2' \) can be obtained at point C. This means the absolute velocity \( V_1' \) is equal to \( V_2' \). Based on this statement, we can state that blade CD is equal to blade AB and \( \alpha_1' = \alpha_2'; \beta_1' = \beta_2' \).

The efficiency of the output power is given as

\[ HP_{out} = \left( \frac{qU_2}{g} \right) (V_1 \cos \alpha_1 - u_1) (1 + \psi \cos \beta_2 / \cos \beta_1) \]  

(2)

And the input power may be stated as \[ HP_{in} = \frac{qQ}{g} \] 

![Fig. 1. Path of water through turbine][11]

![Fig. 2. Velocity diagram][11]

Based on Equation (1), we obtain \( H = \frac{V_2^2}{2g} \). And substituting into Equation (3), we obtain \[ HP_{in} = qQV_1^2 / C^2 g \]. Efficiency is defined by the ratio of the output and input power, \( \eta = \frac{H_{out}}{H_{in}} \). \[ \eta = (2C^2u_1/V_1) \left( \cos \alpha_1 - u_1/V_1 \right) (1 + \psi \cos \beta_2 / \cos \beta_1) \] . When \( \beta_1 = \beta_2 \),

\[ \eta = (2C^2u_1/V_1) \left( 1 + \psi \right) \left( \cos \alpha_1 - u_1/V_1 \right) \]  

(3)

Assuming that Equation (3) is a function and \( u_1/V_1 \) is an observing variable. The first differentiation of Equation (3) is
equal to zero, and is given by, $u_1/V_1 = \cos \alpha_i/2$. Maximum efficiency is given by $\eta_{\text{max}} = (\beta C^2) \left( 1 + \psi \right) \left( \cos^2 \alpha_i \right)$ Based on Figure 2, the direction of $V_2$ is not radial when $u_1 = \frac{1}{2} V_1 \alpha_i$. The water flow of the outer of rim should be radial. Therefore, the value of $u_1$ is given by

$$u_1 = \left[ \frac{c}{(1+\psi)} \right] (V_1 \cos^2 \alpha_i)$$

(4)

When there are no losses on the head due to friction in the nozzle or the blades, $\psi$ and $C$ have unity values.

Variations in the nozzle roughness coefficient $C$ is a quadratic function, as stated at Equation (4). There are two types of hydraulic losses from this nozzle aspect that occur when water is striking the outer and inner peripheries. These losses are relatively small when the blade thickness is sufficiently thin and should not exceed the jet so, as shown in Figure 3. Therefore, the water energy will strike the outer and inner peripheries of the blade. The blade system can provide the optimal blade roughness coefficient $\psi$, 0.98, when the number and thickness of the blades is accurately obtained.

2.2 Design Construction Formula

2.2.1 Blade Angle

As stated in Figures 1 and 3, the blade angle $\beta_1$ can be calculated by $u_1 = \frac{1}{2} V_1 \cos \alpha_i$ and $\tan \beta_1 = \tan \alpha_i$. Based on Figures 4 and 5, the velocity of $V_1$ can be obtained as $V_1 = [2gh_2 + (V_2)^2]^\frac{1}{2}$

Fig. 5. Velocity Diagram [11]

2.2.2 Width of the Radial Rim

The thickness of the entrance jet $s_1$ for a distance of blade $t$ can be calculated by

$$s_1 = t \sin \beta_1$$

(5)

The inner exit blade spacing $s_2$ for $a = r_1 - r_2$ can be obtained by $s_2 = t (r_1/r_2)$ where $s_2 = v_1 s_1 / v_2$ to find the value of $\beta_1$ requires a velocity $v_1$ influenced by the centrifugal force, and is given by

$$(v_1)^2 - (v_2)^2 = (u_1)^2 - (u_2)^2$$

$$(v_2) = v_1 \left( \frac{u_2}{v_2} \right) = v_1 \left( \frac{u_2}{r_2} \right) \sin \beta_1$$

$$x^2 - \left[ 1 - \left( \frac{v_2}{u_2} \right)^2 \right] x - \left( \frac{v_2}{u_2} \right)^2 \sin^2 \beta_1 = 0$$

$$\frac{v_1}{u_1} = \frac{1}{\cos \beta_1}$$

where $x = (r_2 / r_1)^2$

Based on Figure 6, the central angle BOC can be calculated by

$$(v_1)^2 - (v_2)^2 = (u_1)^2 - (u_2)^2$$

$$(v_2)^2 = v_1 \left( \frac{u_2}{v_2} \right) = v_1 \left( \frac{u_2}{r_2} \right) \sin \beta_1$$

$$\tan \alpha' = \frac{v_2}{u_2}$$

$$\text{boC} = 2 \alpha'$$

(6)

2.2.3 Wheel diameter and axial wheel breadth

The wheel diameter can be determined from Equations. $u_1 = \pi D_1 N/(12) (60)$ and $D_1 = 360 C (2gh)^\frac{1}{2} \cos \alpha_i / \pi N$ To find the breadth of the wheel (L), where so = kD, Equations (26), (27) and (28) may be used.

$$Q = \left( \frac{C_0 L}{144} \right) (2gh)^\frac{1}{2}$$

(7)

$$L = 144 Q N/(862H^2 \pi k(2gh)^\frac{1}{2})$$

(8)

where $k = 0.075$ and 0.10, respectively.
In accordance with the construction design equations given above, our calculation results are shown in Table 1, using alpha angles of 15°, 16°, and 17°.

Table 1. Calculation results using the construction design

<table>
<thead>
<tr>
<th>Blade Angle</th>
<th>Radial Rim Width</th>
<th>Central Angle</th>
</tr>
</thead>
<tbody>
<tr>
<td>α₁</td>
<td>β₁</td>
<td>a</td>
</tr>
<tr>
<td>15°</td>
<td>28°</td>
<td>0.64 r₁</td>
</tr>
<tr>
<td>16°</td>
<td>30°</td>
<td>0.66 r₁</td>
</tr>
<tr>
<td>17°</td>
<td>31°</td>
<td>0.668 r₁</td>
</tr>
</tbody>
</table>

2.3 Particle Swarm Optimization (PSO) Algorithm

American psychologist Kennedy and electrical engineer Eberhart developed the PSO algorithm based on the behavior of individuals in a swarm of birds [14]. The PSO algorithm is an optimization technique and also a kind of evolutionary computation technique. PSO is initialized to a random solution, using an iterative search for the optimal value [15]. Each individual in the group is called a particle in a D-dimension solution space. The position vector of the i<sub>th</sub> particle is represented as X<sub>in</sub>=(X<sub>i1</sub>, X<sub>i2</sub>,…,X<sub>in</sub>). The best position found by the i<sub>th</sub> particle so far is denoted as P<sub>i</sub>=(P<sub>1</sub>, P<sub>2</sub>,…,P<sub>n</sub>), and is known as P<sub>best</sub>. Accordingly, the best position found by the whole swarm is denoted as P<sub>s</sub>=(P<sub>1</sub>, P<sub>2</sub>,…,P<sub>n</sub>), and is known as G<sub>best</sub>. The velocity vector of the i<sub>th</sub> particle is represented as V<sub>i</sub>=(V<sub>i1</sub>, V<sub>i2</sub>,…,V<sub>in</sub>). The velocity and position of the i<sub>th</sub> particle can be defined according to the following equations:

V<sub>in</sub> = wV<sub>in</sub> + c₁r₁(P<sub>best</sub>-X<sub>in</sub>) + c₂r₂(P<sub>best</sub>-X<sub>in</sub>)
X<sub>in</sub> = X<sub>in</sub> + V<sub>in</sub>

where w is a constriction factor, c₁ and c₂ are learning factors, and r₁ and r₂ are random numbers generated uniformly in the range [0-1].

A linearly decreasing inertia weight was first introduced by Shi and Eberhart in [14]. The linearly decreasing inertia weight is modified as

w(ITER) = w<sub>max</sub> - (w<sub>max</sub>-w<sub>min</sub>)/(ITER<sub>max</sub>) ITER

where w<sub>max</sub> is the initial inertia weight, w<sub>min</sub> is the final inertia weight, ITER<sub>max</sub> is the iteration maximum in the evolution process, and ITER is the current value of the iteration. Normally, w<sub>max</sub> is set to 0.9 and win is set to 0.4.
The following gives the design step for the improved version of the PSO algorithm [15],[16]

### 3. Problem Formulation

#### 3.1 Optimization Formulation

The input power and output power equations of the Banki turbine are affected by the value of the parameters $H$, $g$, $c$, $v$, $\alpha_1$, $\beta_1$, and $\beta_2$. Constant parameters, such as $H$, $C$, and $g$ do not change their values when optimization is performed. The values we can optimize are the values of $\alpha_1$, $\beta_1$, and $\beta_2$ with the goal of changing the angle of the corner to affect the efficiency of the turbine. The equation to optimize these parameters is as follows, by maximizing the overall value of equations (3), (1), (4), (2) The values of influence are the equality constraints:

\[
\frac{H_{p_{out}}}{H_{pin}} = \frac{WQ}{(V_1 \cos \alpha_1 - \nu_1) (1 + \psi) \cos \beta_2 / \cos \beta_1}
\]

The inequality constraints are

\[
\begin{align*}
\cos \alpha_{1_{\min}} & \leq \cos \alpha_1 \leq \cos \alpha_{1_{\max}} \\
\cos \beta_{1_{\min}} & \leq \cos \beta_1 \leq \cos \beta_{1_{\max}} \\
\cos \beta_{2_{\min}} & \leq \cos \beta_2 \leq \cos \beta_{2_{\max}}
\end{align*}
\]

#### 3.2 Proposed Method Optimization

According to the block diagram of Figure 8 above, the value of $V_1$ will be influenced by the parameters $H$, $g$, and $C$. Furthermore, the value of $V_1$ is the input for $U_1$, $H_{pin}$, and $H_{pout}$. The value of $U_1$ will be determined by the values of the parameters $C$, $\psi$, $\alpha_1$, and $V_1$. The $H_{pin}$ value is determined by the values of $Q$, $C$, $g$, and $V_1$. The $H_{pout}$ value is determined by the values of $Q$, $V_1$, $g$, $\omega$, $u_1$, $\psi$, $\alpha_1$, $\beta_1$, and $\beta_2$. The results of a comparison of the $H_{pin}$ and $H_{pout}$ values determine the value of the efficiency. We used MATLAB simulation in the optimization process to obtain the maximum value of the efficiency ($\eta$) by the value optimized angles $\alpha_1$, $\beta_1$, and $\beta_2$.

### 4. Simulation Result

By the method proposed above in section 3.2, Equation (8) is optimized by MATLAB simulation. The initialization of the parameter values in the PSO algorithm includes the number of birds $n = 50$, dimensions = 3, steps = 50, $c_2 = 0.3$, $c_1 = 0.1$, $w = 0.75$ momentum, the angle of $\alpha_1 (15 \leq \alpha_1 \leq 17)$, the angle of $\beta_1 (28 \leq \beta_1 \leq 31)$, and the angle of $\beta_2 (29 \leq \beta_2 \leq 32)$. The optimization results of Equation (8) are provided in Table 2.

First, PSO simulations were performed with the input values of $H$ and $Q$, then the PSO calculates the values of $\alpha_1$, $\beta_1$, $\beta_2$, $V_1$, $H_{pin}$, $U_1$, $H_{pout}$, and $\eta$. The range of $H$ values is $1 \leq H \leq 60$ and the $Q$ value is either 30, 35, or 40. Every pair of values of $H$ and $Q$ is given 180 combinations, and for each combination of $H$ and $Q$, the PSO will produce 30 lines of data. The next step is to identify combinations of $H$ and $Q$ with the highest efficiency ($\eta$). Of the 180 combinations, Table 2 shows that only 5 of these were chosen as having the highest $\eta$ in the PSO simulation results.

Please note that the simulated data generated by the PSO values for $17^\circ$ and $16^\circ$ angles never appeared. On the basis of these findings, we had to conduct experiments (section 5) to validate the PSO results and prove that the $15^\circ$ $\alpha_1$ angle is better than $16^\circ$ and $17^\circ$ angles.

#### Table 2. Simulation results of the PSO algorithm

<table>
<thead>
<tr>
<th>Data</th>
<th>1.0</th>
<th>2.0</th>
<th>3.0</th>
<th>4.0</th>
<th>5.0</th>
</tr>
</thead>
<tbody>
<tr>
<td>H</td>
<td>56.0</td>
<td>47.0</td>
<td>26.0</td>
<td>27.0</td>
<td>58.0</td>
</tr>
<tr>
<td>Q</td>
<td>35.0</td>
<td>30.0</td>
<td>35.0</td>
<td>40.0</td>
<td>30.0</td>
</tr>
<tr>
<td>$\alpha_1$</td>
<td>15.0004</td>
<td>15.0004</td>
<td>15.0004</td>
<td>15.0002</td>
<td>15.0008</td>
</tr>
<tr>
<td>$\beta_1$</td>
<td>28.1936</td>
<td>29.8677</td>
<td>29.8677</td>
<td>28.8056</td>
<td>29.2837</td>
</tr>
<tr>
<td>$\beta_2$</td>
<td>31.6861</td>
<td>31.8705</td>
<td>31.8705</td>
<td>29.2072</td>
<td>30.0975</td>
</tr>
<tr>
<td>$V_1$</td>
<td>10.3713</td>
<td>9.5015</td>
<td>7.0669</td>
<td>12.4734</td>
<td>14.9269</td>
</tr>
<tr>
<td>$H_{pin}$</td>
<td>1960000.0</td>
<td>1410000.0</td>
<td>910000.0</td>
<td>1080000.0</td>
<td>1740000.0</td>
</tr>
<tr>
<td>$u_1$</td>
<td>5.0</td>
<td>4.5</td>
<td>3.4</td>
<td>6.0</td>
<td>7.1</td>
</tr>
<tr>
<td>$H_{pout}$</td>
<td>1738531.6</td>
<td>1250676.3</td>
<td>807173.2</td>
<td>957961.8</td>
<td>1543379.2</td>
</tr>
<tr>
<td>$\eta$</td>
<td>0.887006</td>
<td>0.887004</td>
<td>0.887004</td>
<td>0.887002</td>
<td>0.887000</td>
</tr>
</tbody>
</table>

#### 5. Experimental Results

We constructed prototype turbine models with axles placed on two pillows so the turbine could spin with little friction, as shown in the photographs in Figure 11.
The prototype models were each made of a waterwheel supported by two pillows placed on the left and right sides of a holder perpendicular to the shafts, with axles made of iron, a pulley as the fan belt connected to the generator, and a water pump to circulate the air from the tank to the nozzle. In this study we used as many as 16 blades with 50 cm wheel diameters and wheels 12 cm thick. The curvature of the waterwheel’s blade angle is the factor responsible for generating a maximum efficiency value (\( \eta \)).

The angle value of \( \alpha_1 = 15^\circ \), obtained from the simulation results in section 4, was tested, as were angles \( \alpha_1 = 16^\circ \) and \( 17^\circ \).

To generate the water flow, we installed 2 pumps (Shimizu, Model PS-116 BIT, U: 1 x 220V 50Hz with Q = 10–24 l/min). Power input was calculated as \( P_{in} = Q \times \rho \times g \times h \), where \( P_{in} = \) power input (watts), \( Q = \) volumetric water flow rate (m\(^3\)/s) = 0.00064, \( \rho = \) density of water (kg/m\(^3\)) = 1000, \( g = \) acceleration due to gravity (= 9.81 m/s\(^2\)), and \( h = \) total energy difference between the upstream and downstream line of wheel (m) = 50 cm = 0.5 m. so, \( P_{in} = 1000 \text{kg/m}^3 \times 9.81 \text{m}^2/\text{s}^2 \times 0.00064 \text{m}^3/\text{s} \times 0.5 \text{ m} = 3.14 \text{ W} \), as the power input value for the efficiency value results.

Each turbine model was tested with curvature angles of \( \alpha = 15^\circ , 16^\circ , \text{and } 17^\circ \). The values of the other parameters were kept exactly the same in all models. Using a water pump to generate water flow to the nozzle, we changed the angle of the nozzle and measured the current I (amperes), voltage V (volts), and rpm. The power generated was calculated from \( P = I \times V \) (watts).

An output efficiency graph can be drawn for the angle (\( \theta \)) within the range 0\(^\circ \)–50\(^\circ \). Figure 12 shows a comparison of the efficiency output for blade angle curvatures (\( \alpha \)) of 15\(^\circ \), 16\(^\circ \), and 17\(^\circ \), respectively. The parameters of a water flow velocity of 0.00064 m/s, a 50 cm diameter water wheel, 16 blades, and a 12 cm-wide water wheel were the same in all models. The goal was to carefully investigate the effect of the curvature blade angle (\( \alpha \)) on water wheel efficiency. Experiments were performed sequentially for each wheel at a certain curvature angle \( \alpha \) to determine their influence. The value of the curvature of the blade angles (\( \alpha \)) selected for testing was based on the results of the PSO (section 3). The best angle (\( \alpha \)) according to our PSO is 15\(^\circ \). According to the results of the plans [12], 16\(^\circ \) was the optimum, and 17\(^\circ \) was the authors optimal experimental result by their own research. Our experimental results verified our results that a water wheel with a blade curvature angle (\( \alpha \)) of 15\(^\circ \) produces the highest nozzle angle (\( \theta \)) efficiency in the range 25\(^\circ \) < \( \theta \) < 45\(^\circ \), compared with water wheel angle curvatures of 16\(^\circ \) and 17\(^\circ \).
The RPM output can be ascribed to the nozzle angle $\theta$. In Figure 14 we can see that the RPM output value increases with an increasing nozzle angle from 10$^\circ$ to 30$^\circ$. These results show that a 15$^\circ$ blade curvature angle ($\alpha$) yields a higher RPM than the water wheels with 16$^\circ$ and 17$^\circ$ blade curvature angles ($\alpha$). Other parameters were held constant in the three experiments.

The improved results obtained with a 15$^\circ$ blade curvature angle $\alpha$ compared with 16$^\circ$ and 17$^\circ$ blade curvature angles $\alpha$ include (1) RPM value, (2) output power, (3) output current, and (4) voltage output. We can conclude that water wheels with a 15$^\circ$ blade curvature angle have higher efficiency compared with those with 16$^\circ$ and 17$^\circ$ blade curvature angles. Thus this study contradicts the research results of [12], which concluded that a 16$^\circ$ blade curvature angle was optimal.

6. Conclusion

The angle of the blade greatly affects water wheel efficiency where the blade is in direct contact, such that the thrust of water causes the wheel to spin. A 15$^\circ$ blade curvature angle is shown to produce higher efficiency values when compared with curvature angles of 16$^\circ$ [12] and 17$^\circ$. By utilizing the PSO algorithm to find the optimal blade curvature angle, the water wheel is proven to generate higher output power and higher RPM with its nozzle set at this angle ($\theta$) value. The energy generated by the water wheel with a 15$^\circ$ blade curvature angle proved to be higher than those with curvature angles of 16$^\circ$ [12] and 17$^\circ$. These results were obtained from a mini-generator operating with a water wheel-mounted fixed load, for which the output current and voltage were measured. The output power of each model were calculated, and the calculation results provide the evidence that the 15$^\circ$ blade curvature angle ($\alpha$) produces the highest energy, when compared with angles of 16$^\circ$ and 17$^\circ$.

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References


