

Torque and Power Consumption of Paddlewheel Aerator With Movable Blade

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Abstrak – *The development of movable blade is based on fact that power is required only when blade of paddle wheel aerator entering water and in contrary action of aeration effect only when the blade is about leaving the water. This study was carried out to design and simulate paddle wheel aerator with movable blade which will open when entering water and close when leaving water. Wheel closed at quadrant I to IV (entering water surface) and was about to open at quadrant III to II (leaving water surface). The blade was designed referring to commonly used Taiwan wheel type. The component of movable blade mechanism consisted of cam and shaft, velg, velg cap, blade holder, follower, spring and bearing. Follower was able rotate with angle of rotation was 125°, rotational displacement was 50 mm, maximum velocity was 0.55 m/s and acceleration was 6.09 m/s². Testing without a load at 115 rpm shows the torque that occurred 43.05 N and the electric power used 511.72 Watt. The gain is smaller than the increase of torque and power needed for movable blade paddlewheel aerator mechanism. Copyright © 2015 Department of Mechanical Engineering. All rights reserved.*

Keywords: *Paddle wheel aerator, movable blade, cam-follower mechanism, torque and power of aerator*

1 Introduction

Aeration is a mechanism of adding some amount of oxygen into water to provide sufficient amount of oxygen. Aeration is carried out by increasing water and air contact using aerator device. One type of aerator device which widely used in pond farming is paddle wheel aerator (Laksitanonta 2003). Paddle wheel aerator is considered as the most appropriate aerator device due to aeration mechanism and wide usable driven power (Romaine & Merry 2007).

Some of parameters including water and air surface contact, differential oxygen concentration, film surface coefficient and turbulence influence aeration rate (Boyd 1998). Aeration performance was influenced by geometry, size and wheel velocity (Moulicket al. 2002). Higher size tends to have higher aeration which simultaneously followed by higher driven power needs due to higher drag force. This condition creates certain problem in utilizing paddle wheel aerator as it may increase operational cost including electrical and fuel consumption.

Various models of paddle wheel aerator are offered in

market. Aerator made by Taiwan is widely used by consumers due to affordable price, light in weight and corrosion-resistant but has low efficiency (Wyban 1989). Aerator that was designed and fabricated by Taiwan has SAE (standard aeration efficiency) value of 1.063 kg O₂ kW h⁻¹ (Peterson & Walker 2002). Bhuyar et al (2009) designed aerator with SAE value 2.269 kg O₂ kWh⁻¹. The most appropriate paddle wheel aerator was designed by Moore and Boyd with SAE value 2.54 kg O₂ kWh⁻¹. Some of fabrications use aerator design with specification 2.25-7.5 kW and SOTR 17.4- 23.2 kg O₂ h⁻¹ and average value of SAE was 2.2 kg O₂ kW h⁻¹ (Moore & Boyd 1992).

Up to now, the development of paddle wheel aerator still uses non-movable blade which result in less optimum power consumption because power is linear with the increasing of aeration rate. Therefore, development of movable blade is needed due to aeration power is only required when blade entering water and in contrary the aeration effect only occurs when blade is about to leaving the water. Therefore movable blade was designed to open when leaving water and close when entering water. This study was aimed to design and simulate paddle wheel aerator with

movable blade to reduce drag force acting on blade as well as power consumption.

2 Functional Design and Testing Method

2.1. Functional Design

The wheel was designed to rotate clockwise with movable blade that enabled to open and close. The blade was about to close at quadrant I to IV (entering water surface) and open at quadrant III to II (leaving water surface). Blade opened to 45° from its close position which parallel to rim. Wheel dimension was designed similarly with commonly used wheel size i.e. 20 cm width, 30 cm rim diameter and 60 cm total dimension.

2.2. Testing

Torque measurement is done using a strain gauge mounted on the wheel shaft. The sensor is connected to the strain amplifier (DAS-406B Strain Amp DC) through the slip ring and the bridge box recorded with the data logger (minilab 1008) and stored on the computer just as shown in Figure 1. Data measurement results in the form of voltage (mVolt) is converted into value strain (μst) and torque measurement values (nm) to the value of the calibration has been done previously.

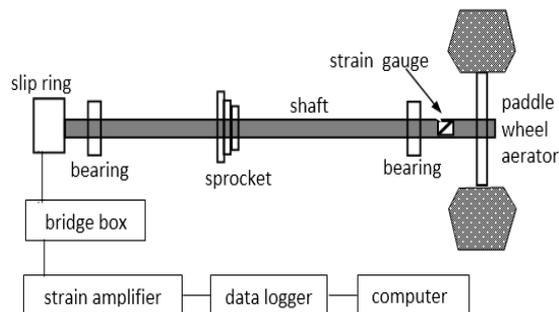


Figure 1. Torsion instrument scheme

The measurement of paddlewheel power is done by measuring the power consumption of electric motors using Ammeter (DO2A) which is connected to an electrical output. To read the power measurement values (Watt) is done using a digital camera video recording on the screen Ammeter. Rated power is taken on an average value which often arises from the reading showing on video recordings.

3 Result and Discussion

3.1. Structural Design

The wheel structure consisted of two main components i.e. stationary and rotary component as shown in Figure 2.

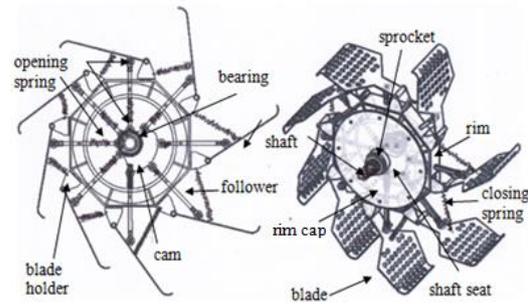


Figure 2. Wheel structure with movable blade

and 17.5 mm, respectively. Cam was mounted to shaft with diameter of 25 mm and attached to machine frame. Rotary component consisted of the rim, rim cap, blade holder, follower, bearing and spring. The rim was octagonal-shape encircling tube with diameter of 218 mm and height of 30 mm. One side of the tube was enclosed with metal sheet, shaft seat and bearing with diameter of 25 mm. Outside the shaft seat, sprocket that engage onto chain was attached for transmission purpose. The rim cap was a shaft seat made from metal sheet and similar bearing with rim tube which mounted to rim tube using bolt. Blade was used to directly bursting up water. Blades formed 30° of angle towards rim with radius of curvature was 40 cm. The size of the blade was 15 cm of width, 20 cm of length, trapezoid-shape with 15° of bottom side and 30° of top side, had 40 holes with diameter of 1.6 cm. Blade holder was used to place blade with shaft of 8 mm and height of 25 mm and bolted at the end side of rim. The follower stem was used to push blade to open and close adjusting to cam profile. The follower stem was 150 mm of height and bearing with 19 mm of external diameter was attached on the two end-sides. Spring consisted of opening blade and closing blade. The opening spring was inserted to follower stem with diameter of the spring was 10.5 mm, length was 60 mm, wire diameter was 1 mm and spring constanta was 0.35 Nm. The closing spring of blade was attached on the front blade holder with diameter of 10 mm, length of 45 mm, wire diameter of 1 mm and spring constanta of 0.5 Nm.

3.2. Movable Components Mechanism

Movable blade were driven using cam mechanism. The cam is a simply mechanism that can provide almost all types of follower movement. The movement analysis of cam mechanism is shown in Figure 3.

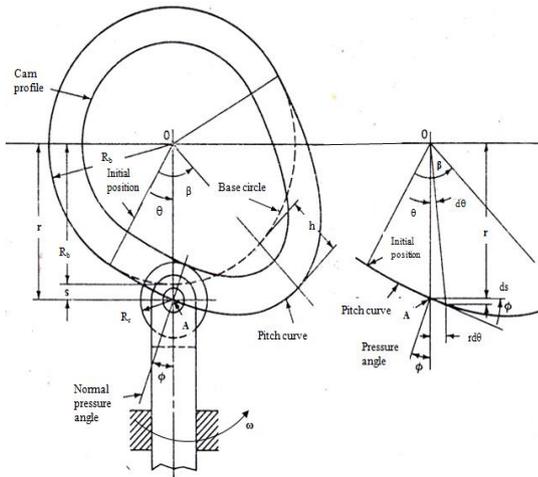


Figure 3. Profile analysis of cam-follower

Diagram of displacement, velocity and acceleration of cam is important factors in determining cam design (Martin 1982). Equation of cam displacement is written as follows:

$$\begin{aligned} \text{for } \frac{\theta}{\beta} \leq 0.5 \quad s &= 2h \frac{\theta^2}{\beta^2} \\ \text{for } \frac{\theta}{\beta} \geq 0.5 \quad s &= h \left[1 - 2 \left(1 - \frac{\theta}{\beta} \right)^2 \right] \end{aligned} \quad (1)$$

Equation of cam velocity is written as follows:

$$\begin{aligned} \text{for } \frac{\theta}{\beta} \leq 0.5 \quad \frac{ds}{dt} &= \frac{4h\omega\theta}{\beta^2} \\ \text{for } \frac{\theta}{\beta} \geq 0.5 \quad \frac{ds}{dt} &= \frac{4h\omega}{\beta} \left(1 - \frac{\theta}{\beta} \right) \end{aligned} \quad (2)$$

Equation of cam acceleration is written as follows:

$$\begin{aligned} \text{for } \frac{\theta}{\beta} \leq 0.5 \quad \frac{d^2s}{dt^2} &= \frac{4h\omega^2}{\beta^2} \\ \text{for } \frac{\theta}{\beta} \geq 0.5 \quad \frac{d^2s}{dt^2} &= -\frac{4h\omega^2}{\beta^2} \end{aligned} \quad (3)$$

The result of follower displacement, velocity and acceleration is shown in Table 1.

Table 1. Normal force of cam-follower

θ (deg)	$2\pi\theta/\beta$ (deg)	t (s)	s (mm)	ds/dt (m/s)	d^2s/dt^2 (m/s ²)
0	0	0.00	0	0.00	6.09
12.5	36	0.05	1	0.11	6.09
25	72	0.10	4	0.22	6.09
37.5	108	0.16	9	0.33	6.09
50	144	0.21	16	0.44	6.09
62.5	180	0.26	25	0.55	6.09
62.5	180	0.26	25	0.55	-6.09
75	216	0.31	34	0.44	-6.09
87.5	252	0.37	41	0.33	-6.09
100	288	0.42	46	0.22	-6.09
112.5	324	0.47	49	0.11	-6.09
125	360	0.52	50	0.00	-6.09

The maximum displacement of follower for one rotation 50 mm with angle of rotation 125°. The maximum velocity of follower was 0.55 m/s. The constant acceleration was 6.09 m/s².

Angle of pressure determines the smoothness of cam movement. The analysis of angle of pressure was illustrated in Figure 3. Angle of pressure (ϕ) for every angular position was equated as follows:

$$\begin{aligned} r &= R_b + s \\ \tan \phi &= \frac{ds}{r d\theta} \end{aligned} \quad (4)$$

The magnitude of pressure angle for every angle of rotation is shown in Table 2.

Table 2. Normal force of cam follower

θ (deg)	$2\pi\theta/\beta$ (deg)	ϕ (deg)	N (N)	N_x (N)	N_y (N)
0	0	0	239.64	239.64	0
12.5	36	18.13	-380.96	-362.05	-118.54
25	72	30.60	-146.87	-126.41	-74.77
37.5	108	37.38	-120.09	-95.43	-72.91
50	144	40.46	-122.97	-93.57	-79.79
62.5	180	41.40	-144.87	-108.68	-95.80
62.5	180	41.40	-144.87	-108.68	-95.80
75	216	31.01	-324.70	-278.29	-167.29
87.5	252	22.02	-3141.54	-2912.37	-1177.88
100	288	14.10	612.74	594.28	149.26
112.5	324	6.88	328.35	325.99	39.32
125	360	0	239.64	239.64	0

The largest angle of pressure between cam and follower was 41.40°. This magnitude was too large and not necessary for cam-follower mechanism as it required high force and caused mechanism failure that led to machine damage.

Each blade had two types of spring i.e. blade-closing spring and blade-opening spring. Blade-closing spring (s_1) worked against drag force (F_d) and gravity of the blade (w), while blade-opening spring (s_2) worked against force of blade-closing spring from cam pressure due to wheel rotation. Analysis of spring force is shown in Figure 4.

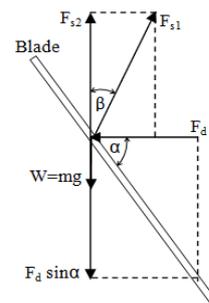


Figure 4. Spring force analysis

Based on the calculation, some of spring data were collected, including installed length, operating length, operating force, springs material, wire diameter, average diameter, inside diameter, outside diameter, free length, number of coils and allowable shear stress for blade-opening springs. The magnitude was 75 mm, 25 mm, 2.45 N, 49.05 N, chromium-vanadium A231, 2 mm, 2 mm, 10.5 mm, 14.5 mm, 80 mm, 12 coils and 922.74 MPa, respectively. The magnitude for blade-closing spring was 124 mm, 38 mm, 2.45 N, 264.50 N, chromium-vanadium A231, 2 mm, 2 mm, 8 mm, 125 mm, 130 mm, 20 coils and 815.75 MPa, respectively.

Inertia and torque analysis are shown in Figure 5. The influencing parameters consisted of force acting on follower (P), inertia force of follower (f), force of gravity on follower (W), shear stress acting on follower (F), normal force on rim towards follower (F_1, F_2), normal force of cam toward follower (N), follower overhang (a), distance between bearing surface (b), diameter of follower stem (d), pressure angle (ϕ) and friction coefficient between follower (μ).

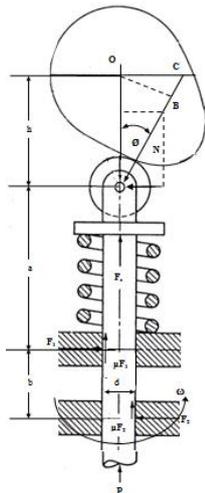


Figure 5. Analysis of angle of pressure

Total vertical force was:

$$N \cos \phi = F + \mu (F_1 + F_2) \quad (6)$$

Total horizontal force was:

$$F_1 = F_2 + N \sin \phi \quad (7)$$

Summing moments to a point where F_1 works, gave:

$$F_2(b - \mu d) = N_a \sin \phi + \frac{d}{2}(F - N \cos \phi) \quad (8)$$

By neglecting F_1 and F_2 at the last 3 equations, normal force of cam acting on follower was:

$$N = \frac{F_b}{b \cos \phi - (2\mu a + \mu b - \mu^2 d) \sin \phi} \quad (9)$$

Torque required to rotate paddle wheel was calculated as follow:

$$T = N (OB) \quad (10)$$

The maximum torque required to activate blade mechanism was 80.09 N-m.

Based on the required torque, using equation 6 and neglecting other mechanical loss, as much as 0.96 kW was required to rotate movable blade on paddle wheel aerator.

3.3. Torque and Power Consumption

Testing without a load at 115 rpm showed the torque that occurred 43.05 N and the electric power used 511.72 Watt. Torque occurring reduction compared with calculation results due to modification follower. Follower rod are bent to minimize the contact angle that happened. However, these test results showed the friction force still high occurred between the cam and the follower that are caused by a cam profile which must follow the functional design of the paddlewheel aerator. According to Bahri (2014) blade drag and torque reduction occurred because the movable blade simulation results showed average 34.20 % and 31.51 %. The gain is smaller than the increase of torque and power needed for movable blade paddlewheel aerator mechanism.

4 Conclusion

Structure of the wheel consisted of two main components i.e. stationary and rotary component. Stationary component consisted of cam and shaft. Rotary component consisted of a rim, a rim cap, blade holders, followers, bearings and springs. The follower was able to rotate with angle of rotation was 1250, rotational displacement was 50 mm, maximum velocity was 0.55 m/s and acceleration was 6.09 m/s². The follower had constant acceleration. Testing without a load at 115 rpm shows the torque that occurred 43.05 N•m and the electric power used 511.72 Watt. These test results showed the friction force still high occurred between the cam and the follower that are caused by a cam profile which must follow the functional design of the paddlewheel aerator.

References

- [1] S. Laksitanonta, S. Singh and G.A. Singh, *A review of aerators and aeration practices in Thai Aquaculture*, Agricultural Mechanization in Asia, Africa and Latin America, 34 (4):64-71. 2003.
- [2] J.M. Moore and C.E. Boyd, *Design of small paddle wheel aerators*, Aquaculture Engineering, 11:55-69. 1992.
- [3] S. Moulick, B.C. Mal and Bandyopadhyay, *Prediction of aeration performance of paddlewheel aerators*, Aquaculture Engineering, 25:217-237.2002.

- [4] E.L. Peterson and M.B. Walker, *Effect of speed on Taiwanese paddlewheel aeration*, Aquaculture Engineering, 26:129-147. 2002.
- [5] J.A. Wyban, G.D. Pruder, and K.M. Leber, *Paddle wheel effect on shrimp growth, production and crop value in commercial earthen ponds*, Journal of the World Aquaculture Society, 20:18-23. 1989.
- [6] L.B. Bhuyar, S.B. Thakre, and N.W. Ingole, *Design characteristics of curved blade aerator w.r.t. aeration efficiency and overall oxygen transfer coefficient and comparison with CFD modeling*, International Journal of Engineering, Science and Technology 1: 1-15. 2009.
- [7] C.E. Boyd, *Pond water aeration systems*, Aquaculture Engineering, 18:9-40. 1998.
- [8] G.H. Martin, *Kinematics and dynamics of machines*, USA: McGraw-Hill, Ltd. 194-382, 1982.
- [9] Munson, Young, and Okiishi, *Fundamentals of fluid mechanics*, USA, John Wiley & Sons, Inc. 518-538, 2006.
- [10] S. Bahri, R.P.A. Setiawan, W. Hermawan and M.Z. Junior, *Design and simulation of paddle wheel aerator with movable blades*, International Journal of Engineering Research and Technology. 4 (2):994-999, 2015.