

Design Optimization of Savonius Wind Turbine Using CFD-Particle Swarm Optimization with Power Flow Validation Experimentally


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ABSTRACT

Air wheel generator or wind turbine generator has capabilities in generating electricity even at the lowest speed. Recently, the Savonius wind turbine with combined wind deflector and housing that applied at the top of car roof. In this paper, the unidirectional vertical axis wind turbine with two blades of Savonius type and belting system have been numerically optimizes using Computational-Particle Swarm Optimization (CFD-PSO). The 5-step optimization was done systematically to determine the related formulation based on the geometrical dimension of wind turbine blade and design constraints. The simplified model of the classical cantilever beam was used in formulating the bending stress and deflection as a constraint. Then the simulation analysis in Matlab was performed using Particle Swarm Optimization (PSO) method in minimization the mass of blades. The current generation was compared to the initial design OPTION 1 for the blades with a mass of 1.3414 kg. From simulation analysis results, the researcher found OPTION 3 has $m_3 = 0.8387$ kg with 37.5% mass reduction of blades. The result of OPTION 3 = [0.1 0.001] with belting unit were proposed to be used in prototype development that achieving the objective in minimization the mass of blades but not fulfil the minimum power and current generated caused by the existing generator type not enough rotational speed with minimum of 1400 for cut in speed for standard automotive application.

Keywords:

 Savonius; Wind Turbine; Optimization;
 Simulation Analysis; Power Flow

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1. Introduction

The term of air wheel generator has been used by researchers since the rotor and blades have appeared as a wheel physically and well-known called as wind turbine generators for electric generation. There are two types of wind turbine as Horizontal Axis Wind Turbine (HAWT), and Vertical Axis Wind Turbine (VAWT) [1, 2]. The benefit of wind turbine that using wind energy that available in 24 hours, not depend on the sun and weather. VAWT became widely used, and much

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attention in the 1970s as renewable energy during petroleum oil crisis [1, 2]. The Savonius wind turbine has the advantages than other wind turbine includes; ground installation, reducing maintenance cost, simple structure, low cost, high starting torque, and low noise emissions [3, 4]. In recent years, low speed of the wind will rotate the Savonius blades in generating power for many applications, enable the intermittent generation, and small power storage for small electronics, charging of cellular phone, and lighting [5]. The Savonius wind turbine power production depends on interaction between the rotor and the wind. The design factors are wind turbine rotor, including aerodynamics, generator characteristics, blade strength and rigidity, noise level etc. [6]. Blade is the key component of a wind turbine that consist of two stages; aerodynamic design and structural design. In reducing the raw materials of blade in manufacturing, it requires the optimization steps to overcome the aerodynamic loads [2]. For Savonius wind turbine, the Coefficient of Power, $C_p=0.2$ [6, 7], 0.125 [8], 0.1 to 0.25 [9], 0.24 [10] and 0.25 [11] is lower when compared to aero blade type wind turbine about 0.3 to 0.45 causing it only applicable for low power application [12].

The optimization on blades merged the significance studies; the interested method was performed using Computational Fluid Dynamic (CFD) simulation and wind tunnel test with the analytical method for structural analysis on the material used are ABS 3D print [13] and Acrylic [14, 15]. The Matlab simulation were studied by Shah *et al.*, [12] and the mathematical formulation were carried out in improving the new blade shape with techno-economic performance. The conversion of electrical voltage against wind speed were carried out by Parti *et al.*, [16]. This paper will perform the simulation analysis on AWG to enhance the mass of blade selection to ensure the structural design has sufficient strength under wind loads when vehicle speed moving at 80km/h. The researcher performing the 2 blades Savonius VAWT with combined one-directional flow housing of a modern passenger car for electrical energy generation. The optimization using Matlab (PSO) combined with Computational Fluid Dynamic (CFD) simulation became the cheapest and faster by reducing the cost of manufacturing where for further comparison with experimental results.

2. Methodology

2.1 Mathematical Formulation

A mathematical model was developed in Matlab in optimization the blade thickness, thus reducing the overall blade mass. The output power of a wind turbine depends on the wind speed, the swept area, aerodynamics, mechanical, and electrical. The initial design of Savonius wind turbine blade has a length $L=0.2$ m, height, $h=0.1$ m and thickness, $t=0.002$ m. Therefore, the aim of the research to minimize the mass of blades to highly rotate in aerodynamic load minimization the blade thickness. In preparing the mathematical formulation for both simulation and analytical analysis, the five step optimization [17] will be used as follows.

2.1.1 Optimization description

The optimization requiring the researcher to maximize energy harvesting in term of the current generated at the given speed of vehicle, $V=80$ km/h. The wind turbine has two design variable for dimension as shown in Figure 1 where the height of the blade is in the range ($0.1 \leq x(1) \leq 1$)m and minimum thickness of the blade varies from 0.001 m, 0.0015m and 0.002 m. However, in order to manufacturing consideration, the need to minimize the mass of blades and overcome the aerodynamic loads during rotational when passenger car is moving at the given speed = 80kmh⁻¹. The line pressure on blade $w=60$ Nm⁻¹, where the length of blade is fixed =0.2 m and material were made from the stainless steel 304. The mass of blades, m_b , includes a shaft (0.34 kg) and n is the number of

the blade was defined in Eq. (12) as; where the parameters include the variable of thickness in the design of the blade was summarized in Table 1.

Table 1
Parameter design of Savonius blades (curved shape)

Parameters	Values	Notes/References
Single blade diameter, D_b	0.2 m	Fixed
Blade length, L (as straight beam)	0.2 m	Fixed
Height, h	(0.1 to 1) m	Variable x(1)
Thickness, t	(0.002 ; 0.0015; 0.001) m	Variable x(2)
Material	Stainless steel 304	Fixed
Modulus of elasticity, E	193 GPa	Fixed
Density of stainless steel 304, ρ	8000 kgm^{-3}	Fixed
Maximum deflection	0.02 m	Fixed
Gear ratio (Belting system)	1:4 (WT: Alternator)	Fixed
Maximum rotor speed, N	212.2 RPM	Experiment data [18]
Coefficient of Power, C_p	0.2	Experiment data [6, 7]
Tip speed ratio (TSR), λ	0.2	Experiment data [18]
Maximum pressure on blades	600 Pa	CFD data [18]
Vehicle speed, V	80 kmh^{-1} (22.22 ms^{-1})	Fixed
Number of blades. n	2	Fixed
Mass of shaft	0.34 kg	Fixed

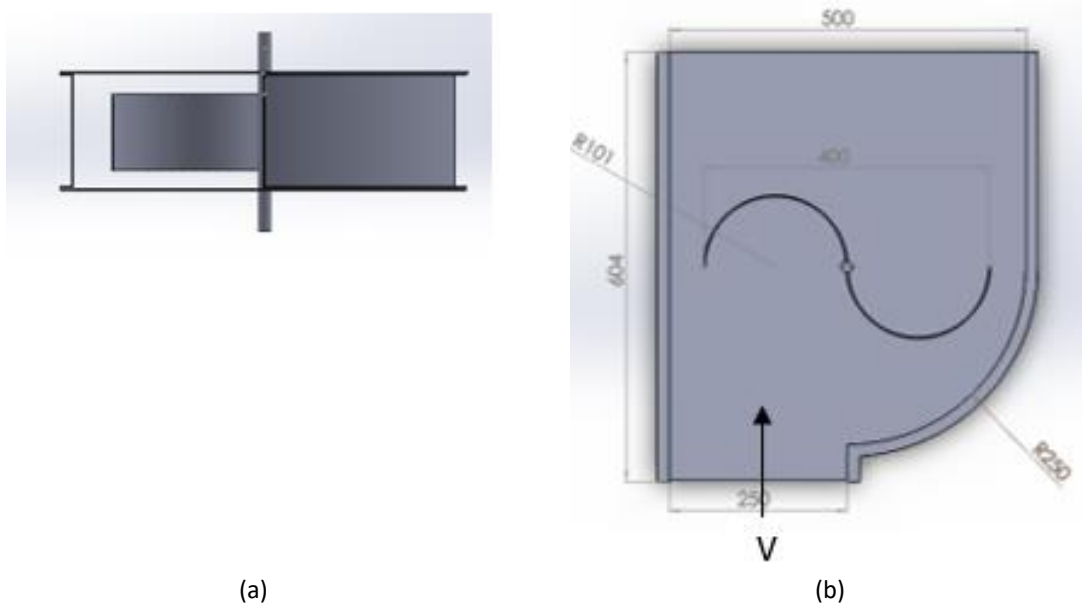


Fig. 1. Air wheel generator with curved shape blades (a) front view (b) top view

2.1.2 Data and information collection

The performance of Savonius wind turbine is measured using the mechanical power or wind turbine output and conversion into electrical power. The power flow analysis in section 3.4.1 will be used in validating the simulation results using analytical analysis that based on the as below.

$$P_m = \frac{1}{2} C_p \rho A V^3 \quad (1)$$

$$A = Dh$$

where C_p is coefficient of wind power, ρ is density, A is the drag area on blade, V is wind speed, D is rotor diameter and h is rotor height.

Tip speed ratio (TSR) as,

$$TSR, \lambda = \frac{V_{Rotor}}{V} = \frac{\omega r}{V} \quad (2)$$

Rotational speed of rotor as,

$$N = \frac{\omega \times 60}{2\pi} \quad (3)$$

where N is revolution per minute (RPM) and ω is angular velocity in rad/s.

From the simulation using ANSYS Fluent as done previously at the velocity of 80 km/h = 22.22 m/s, the maximum pressure on blade = 600 Pa according to Ansys Fluent results, where the existing blade as initial design has fixed rotor diameter, $D_r=0.4$ m where the single blade has diameter, $D_b=0.2$ m, height, $h = 0.1$ m and three optional blade thickness (0.002; 0.0015; 0.001) m as depicted in Table 2. By calculating the pressure of 600 Pa as line pressure at fixed blade length, we get; the line pressure acting on blade $w = 60$ N/m. The parameters in Table 2 will be used in simulation analysis in Matlab.

The structural design constraints:

The maximum tip deflection (blade) $\delta_m=10\%L$, where L is blade length or radius [19] while SF=2 will be used for maximum bending stress [20] as;

Maximum Tip deflection:

$$\delta_0 - \delta_m \leq 0 \quad (4)$$

Maximum Bending stress:

$$\sigma_{max} - SF\sigma_y \leq 0 \quad (5)$$

where δ_0 is refer to tip deflection for beam, δ_m is the maximum tip deflection, σ_{max} is maximum stress, SF is safety of factor and σ_y is yield stress for the material used.

Assuming the single blade is simplified as beam as below with acting pressure in line $w=60$ N/m, where L is the diameter or length of the blade as shown in Figure 2. The value of pressure line was based on the average pressure acting on the blade area from CFD results that converted into line pressure.

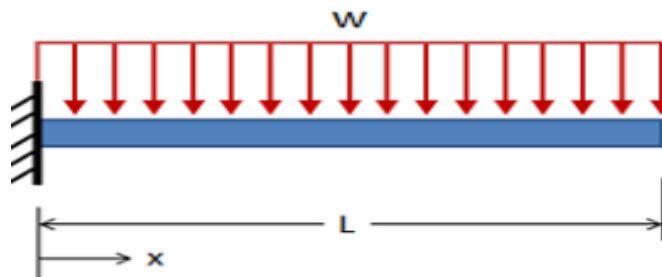


Fig. 2. Beam cantilever with line loads

The Eq. (6) for deflection of beam as,

$$\delta_0 = wL^4[8EI]^{-1} \quad (6)$$

Thus,

$$wL^4[8EI]^{-1} - \delta_m \leq 0$$

The Eq. (7) for bending stress of beam as,

$$\sigma_{max} = \frac{My}{I} \quad (7)$$

$$M = \frac{wL^2}{2} \quad (8)$$

$$I = \frac{1}{12}(x(1) \times x(2)^3) \quad (9)$$

$$y = 0.5 \times x(2) \quad (10)$$

For Eq. (3) to Eq. (9) where L is blade length, w is line pressure acting on blade (as a beam), E is modulus of elasticity, I is moment of inertia for beam, y is the vertical value in y axis as refer to blade thickness.

By substituting Eq. (8), Eq. (9) and Eq. (10) into Eq. (7) then Eq. (5) at the below became as,

$$\frac{My}{I} - SF\sigma_y \leq 0$$

$$\frac{wL^2}{2} \times 0.5 \times x(2) \left[\frac{x(1) \times x(2)^3}{12} \right]^{-1} - SF\sigma_y \leq 0$$

$$3wL^2[x(1) \times x(2)^2]^{-1} - SF\sigma_y \leq 0 \quad (11)$$

2.1.3 Definition of design variables

Both variables may vary in the fixed boundary range in order to obtain the optimum value for minimum mass of the blade, suit with housing sizes and not became the obstacle that increasing the aerodynamic friction where $x(1)$ = blade height (m) and $x(2)$ = blade thickness (m).

2.1.4 Optimization criterion

The mass of blades (curve shaped) includes shaft (for the same material) in Eq. (12) became the objective function (OF) in the design optimization.

$$m_b = \rho(n \times \pi \times x(1) \times x(2) \times 0.5 \times L) + 0.34 \quad (12)$$

2.1.5 Structure design constraint

There are four design constraints used in the simulation; where C1 and C2 for structural strength while C3 and C4 for boundary limitation for mounting on the roof of the car where all the relevant design constraint is depicted in Table 2.

Table 2
Design constraint for structural strength and its limitation

Design Constraint	Details
C1	$3wL^2[x(1) \times x(2)^2]^{-1} - SF \times \sigma_y \leq 0$
C2	$wL^4[8EI]^{-1} - \delta_m \leq 0$
C3	$0.1 \text{ m} \leq x(1) \leq 1 \text{ m}$
C4	OPTION 1: $0.002 \text{ m} \leq x(2) \leq 0.005 \text{ m}$ OPTION 2: $0.0015 \text{ m} \leq x(2) \leq 0.005 \text{ m}$ OPTION 3: $0.001 \text{ m} \leq x(2) \leq 0.005 \text{ m}$

2.2 Computational Fluid Dynamic (CFD)

The number of 2 blades with curved shape were decided early as the initial design in optimization the Savonius wind turbine. The simulation was done using ANSYS R18.1 in determining the average maximum pressure on blades where 600 Pa as depicted in Table 1.

2.2.1 Pressure acting on blades

Assuming the air flowed around the rotor was assumed turbulent. The momentum, turbulent kinetic energy, and dissipation rate were simulated using the Second Order Upwind method. Simulation using CFD provided the pressure and velocity values at all nodal points of the flow domain around the rotating blades. The same convergence criteria were used for all the models in the case of continuity, X and Y velocity, Kinetic energy (k) and dissipation rate (ϵ). A systematic iterative simulation study on various cross-sections of the blades was performed using the commercial software package CFD (ANSYS Fluent). Pressure based solver and transient solution were used to simulate airflow around the rotor.

Mesh motion technique was used to rotate the rotor. The pressure-velocity coupling was achieved using the well-known SIMPLE (Semi-Implicit Method for Pressure-Linked Equations) method. The standard k - ϵ turbulence model in FLUENT was used for the analysis of turbulent flow around the rotor models. Turbulent kinetic energy (k) and turbulent dissipation rate (ϵ) second order upwind scheme was chosen for the momentum equation solution. The standard k- ϵ turbulence model is a semi-empirical model based on model transport equations for turbulence kinetic energy (k) and its dissipation rate (ϵ).

2.2.2 Material and boundary condition

There is one type of material; air as set fluid boundary. The “mesh motion” was used to provide the contact with air and rotor so that the rotation of the rotor is simulated when air flowing onto the rotor and collision happen onto the rotor blade area. The value of rotation speed was set based on Table 3 to determine the pressure profile of the blade. The estimation of rotor speed is based on the experimental test stand whereby using Eq. (1), Eq. (2) and Eq. (3).

Table 3
Rotor rotational speed set up in mesh motion

No.	Rotor Speed (rad/s)	Model of blade
1	22.22	D=400mm

2.2.3 Meshing and grid independence study

The number of nodes and element were depicted in Table 4 for meshing as shown in Figure 3. The tetrahedron fine mesh was used for both inner blade and inner housing. The total of nodes and elements for inner blade and inner housing are 72665 and 129713 respectively.

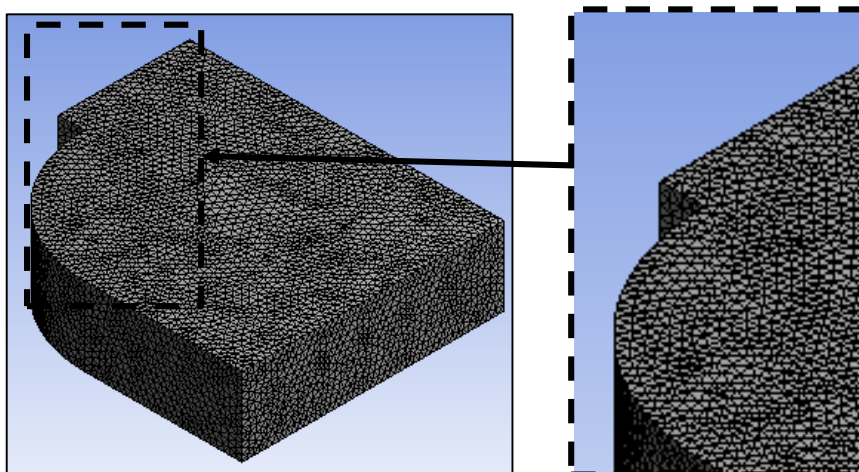


Fig. 3. Mesh Model using tetrahedron in the large view

Table 4
Number of nodes and elements

	Inner Blade	Inner housing
Nodes	12323	22926
Elements	59942	106787
Total	72665	129713

2.2.4 Run the simulation

The number of iterations is 25 for each for overall time time step. Then the calculation will repeat for next time step until finish number of time steps of 10 to complete the overall calculation. Time step size is 0.018s where the researcher refer to the movement of rotor every 45 degree rotational as detail in Table 4. For 22.22 rad/s require 7.07 rotation per second. Then in 1 rotation require 0.14 s and each 45 degree rotation increment time has 0.018s.

2.3 Writing Particle Swarm Optimization using Matlab

The particle swarm optimization algorithm was written using Matlab 18.1 that includes three main inputs; objective function, constraints and boundary of variables as detailed in this section.

2.3.1 Objective function

The objective function (of) was written as a mass of blades and the shaft (kg) as mentioned previously in Eq. (12);

$$\text{of} = \rho(n \times \pi \times x(1) \times x(2) \times L \times 0.5) + 0.34;$$

where length of the blade, $L = 0.2$ m; density, $\rho = 8000$ kg/m³ and number of blades, $n=2$.

2.3.2 Constraint conditions

The structural design constraints were written as;

Bending stress: $\sigma(1) = 3wL^2[x(1) \times x(2)^2]^{-1} - S_y$

Maximum tip deflection: $\delta(2) = (wL^4) [8E(x(1) \times x(2)^3/12)]^{-1} - \Delta$

A classical beam is proposed in the plane surface as simplified where the three dimensional distortion of the blade shell structure is not accounted [2]. Table 5 depicted the input parameter for constraint condition.

Table 5

Input parameter for constraint conditions

Input Parameter	Value
Loading, w (N/m)	60
Young's Modulus, E (Pa)	193e9
Length of the Blade, L (m)	0.2
Yield Stress, σ_y (Pa)	215e6
Allowable Deflection, δ_m (m)	0.02

2.3.3 Boundary of variables

Boundary 1, Boundary 2 and Boundary 3 in term of lower and upper bounds of the variables as depicted in Table 6. This boundary will be used separately in Matlab code for the PSO main file to obtain the optimum blade design structure. The PSO code can be changed easily, it reduces the time and cost computational.

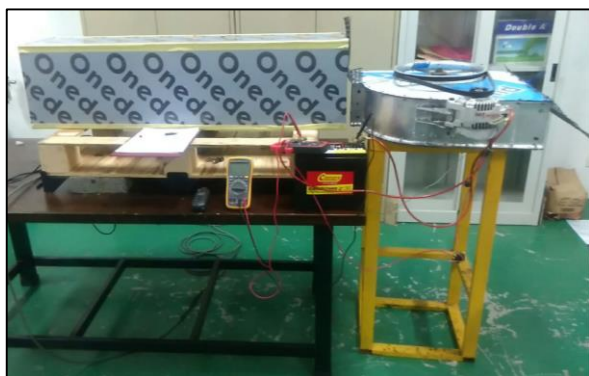
Table 6

Input Parameter for Boundary of Variables as refer to three blade thickness

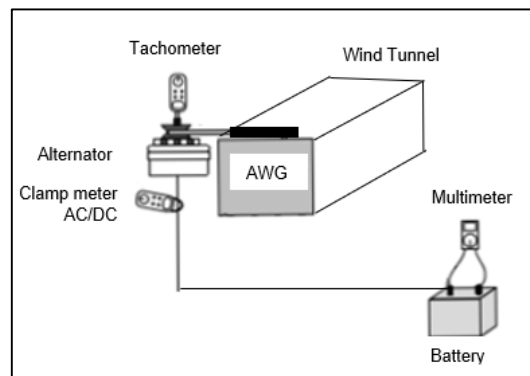
Optimization type	Input parameter (blade thickness)	Lower bounds, LB (m)	Upper bounds, UB (m)
OPTION1	Boundary 1 for $t=0.002$ m	[0.1 0.002]	[1 0.005]
OPTION2	Boundary 2 for $t=0.0015$ m	[0.1 0.0015]	[1 0.005]
OPTION3	Boundary 3 for $t=0.001$ m	[0.1 0.001]	[1 0.005]

2.4 Experimental Studies

The experimental studies were conducted using two type of belting system; V belt and V ribbed as performed by Abdul Qoiyum *et al.*, [21] based on ISO 8854 and SAE J (as shown in Figure 4).



(a)



(b)

Fig. 4. Experimental set up (a) Wind tunnel test (b) Schematic diagram [21]

3. Results

3.1 Pressure Distribution on Blade

In this study, the average pressure was used as input for simulation analysis using Matlab. Only several pressure contour were shown for $T=0$ s and $T=0.018$ s at time step =1 where it refers to angle of 0° and 45° respectively (as shown in Figure 5).

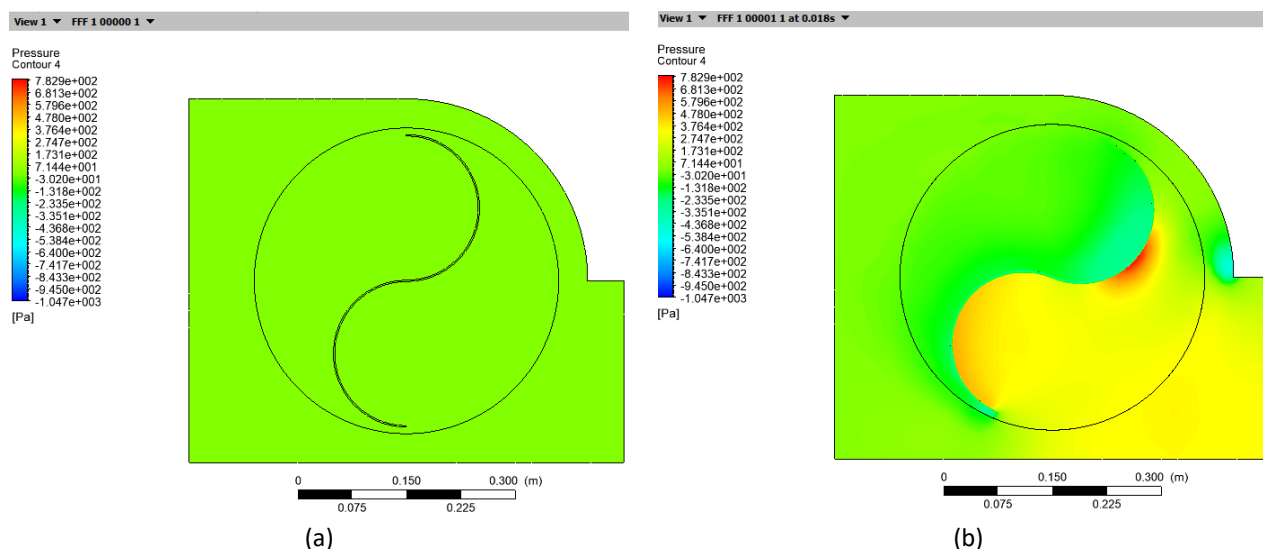


Fig. 5. Pressure distribution on the blade for different simulation time a) $T=0$ s b) $T=0.018$ s time step =1

3.2 Verification Simulation using CFD

The simulation results using ANSYS Fluent were visualized in the form of contour pressure. Thus, allow us to verify the rotations of the blade were simulated correctly where the pressure profile was different for each rotation angle. Hence, the movement of the rotor can be displayed with different colour of graph where the maximum of pressure and drag force can be traced using report file. The input data of rotational speed of rotor used in section 2.2.3 were referred to experimental results using blade thickness, $t=0.0009$ m [18, 21]. From the test stand, the low thickness performing higher rotational of the rotor, thus proven it has both sufficient strengths due to aerodynamic loads but requiring blade guider as holder to limit the wider opening collision area.

3.3 Simulation Results using Matlab

The simulation analysis results show the best value for the mass of 2 blades have a minimum value of $m_3 = 0.8387$ kg with the radius and thickness of blade are $x(1) = 0.1$ m and $x(2) = 0.001$ m respectively. Table 7 shows the mass of blade based on optimization type, OPTION 1 has $m_1=1.3414$ kg as the initial design while OPTION 2 has $m_2=1.0901$ kg and OPTION 3 has $m_3=0.8387$ kg which have the reduction of mass about 18.7% and 37.5% respectively as compared to the initial design OPTION 1. OPTION 3 [0.1 0.001] became the best design optimization for the minimization mass of blades.

Table 7
Mass of blade results based on optimization types

Optimization type	X (1) m	X(2) m	m_b (kg)	% Reduction
OPTION 1	0.1	0.002	$m_1=1.345$	0.0
OPTION 2	0.1	0.0015	$m_2= 1.0901$	18.7
OPTION 3	0.1	0.001	$m_3= 0.8387$	37.5

*Note: % reduction as compared to OPTION 1

3.4 Validation of Simulation Results

3.4.1 Power flow study

The analytical calculation for the power flow study was done to compare both simulation results and experimental results. The power output, aerodynamic, mechanical, and electrical performance are summarized in Table 8. The data of simulation analysis using Matlab were dependent on CFD results, thus all the grid independent study is may refer as the simulation results. The analytical calculations were compared also for V belt type and V ribbed type. The simulation results below show the tip speed ratio, TSR is 0.2 as the input data that referred to the test stand for maximum rotational speed of rotor, $N= 212.2$ rpm for wind speed, $V= 22.22$ m/s. The simulation data simplified the wind turbine rotor has the loads from alternator using belting system with Gear Ratio, $G=4$ with the blade thickness, $t = 0.0009$ m~0.001 m as refer to the Matlab best results even it lower than the OPTION 3 as the best blade thickness.

Table 8

The simulation and experiment results on AWG performance in harvesting current on passenger car [18]

No.	Parameters	Simulation Results	Experimental Result	
		TSR=0.2 on turbine	V-Belt	4PK V Ribbed-Belt
1	Mechanical power or wind turbine output	53.76 W	53.76 W	53.76 W
2	Angular velocity, ω_m of rotor	22.22 rads ⁻¹	26 rads ⁻¹	20.25 rads ⁻¹
3	Speed of rotor RPM	212.2 RPM	248.6 RPM	193.8 RPM
	Velocity of Rotor, $V=\omega_m r$	4.44 m/s	5.2 m/s	5.06 m/s
4	The torque of wind turbine	2.42 Nm	2.068 Nm	2.655 Nm
5	Gear ratio, G	4	4.3	6.4
6	Alternator speed	848.8 RPM	1073.8 RPM	1051.6 RPM
7	Angular velocity of alternator	88.88 rads ⁻¹	112.45 rads ⁻¹	110.12 rads ⁻¹
8	Tip speed ratio TSR	0.2	0.234	0.182
9	Power input to alternator	53.76 W	86.6 W	73 W
10	Current generated	0.158A	0.4 A	0.3A
11	Voltage output	0.093 V	0.1 V	0.1 V
12	Generator output power ($P=VI$)	0.0147 W	0.04 W	0.03 W

According to the results from Table 8, the corresponding output voltage versus alternator speed of experimental data were used in determining the current output for simulation performance. The interpolation calculation can be used for the similar alternator that has functioned in rotational speed as depicted in Table 9. Both simulation and experimental results proved that no sufficient charging current from the alternator to the battery with current values varies in the range of 0.158 to 0.4 A as compared to standard alternator that has, standard charging current, $I= 2A$ as referring to ISO 8854 and SAE J 56 [22, 23].

Table 9
Data of power with respective current generated for interpolation method

Data	Power (W)	Current (A)
Experiment on V belt	86.6	0.4
Experiment on V Ribbed Belt	73	0.3
Simulation on AWG	53.76	x

3.4.2 Recommendations

From the power flow study performed in section 3.4.1 shows that the achievement of Particle Swarm Optimization in minimization the blade thickness. However, it depends on experimental data to get the estimation of the input data that can be referred as listed in the Table 1 that using two variables only. The optimization was successfully validated in term of minimum blade thickness as refer to the given OPTION1, OPTION 2 and OPTION 3. The power flow study is necessary for validation in finding the best blade thickness. The voltage and current generation as measured using a clamp meter in the experimental setup depend on the power output and function in rotational speed of minimum rotational requirement. Based on the low power output less than 1 Watt as depicted in Table 8 has proven that the car alternator is not a sufficient application for the existing wind speed that resulting the less than 1400 rpm for minimum alternator rotational speed for generating voltage above than 12.6 V as standard cut off voltage for the alternator.

In the future, the PSO analysis need to consider the effect of blade guider that acting as blade holder, thus make the deflection =0 since it prevents over widening the blade collision area. The consideration the number of blades is 4 as refer to the actual number blade used in the final prototype should be done using the same step in design optimization of Savonius wind turbine using CFD-PSO with power flow study validation. Otherwise, the speed of wind may vary up to 120 km/h since it may apply for high speed application for energy harvesting on vehicle for reducing the dependency on petroleum based fuel that still contribute to the global warming and acid rain from combustion gas exhausted to the atmosphere.

4. Conclusions

The optimization study was summarized several conclusions, where from simulation analysis using MATLAB, the researcher found that:

- i. OPTION3 has the 37.5 % of mass reduction (for the blades with $m_3=0.8387$ kg). The researcher decided to use the result for OPTION3 with the height of blade, h as $x(1)=0.1$ m and thickness, t as $x(2)=0.001$ m with belting system with gear ratio, $G= 4$ for the proposed prototype development.
- ii. The simulation analysis using Particle Swarm Optimization (PSO) based on pressure profile on the blade from simulation model using ANSYS Fluent that successfully compromised the load of the alternator and rotational speed of the wind turbine that found the similar to experimental results. Thus, by using pressure profile according to mesh motion in CFD (ANSYS Fluent) may predict the value of torque on turbine and the alternator shaft with the common value of the tip speed ratio of blade (TSR), $\lambda =0.2$.
- iii. For economic reason, the blade thickness, $t=0.0009$ m is suggested to be used for prototype material as the wide application, thus make it was selected instead of use $t=0.001$ m.

- iv. In comparing the simulation results and experimental results, Table 7 proved that for air speed, $v=22.22$ m/s (80 km/h), not sufficient charging current for battery with alternator current values varies in range of 0.158 to 0.4 A as compared to standard alternator current, $I=2$ A as refer to ISO 8854 and SAE J. Thus, the AWG design must vary the air speed up to 120 km/h as done from stand test for air speed, $v=22.22$ m/s to 32.8 m/s (80 km/h to 118 km/h) that were successfully done based on the recommendations.

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