Gear Design of A 100 Watt Horizontal Axis Wind Turbine

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Abstract— The aim of this paper is the gear design of a 100 Watt horizontal axis wind turbine (HAWT). This is for integration into a prototype HAWT as a starting point to facilitate improved electricity supply. The procedure involved the use of gear design principles, bending theory, applicable codes and standards to achieve the design and material selection, and the gears were modelled with AutoCAD. Based on the determined module of 2mm, the design results indicate a gear diameter of 26.6mm with 13.3 teeth on the output shaft coupled to the generator to deliver a rotational speed of 800 revolutions per minute (rpm) would mesh with a 28 teeth, 56mm diameter rotor shaft gear speed running at a speed of 381rpm. The gears face width and tooth thickness were also being found to be 20mm and 3.14mm respectively. Due to bearing boundary dimension standardization constraint, 17mm and 7mm were selected for the main and output gears internal diameters respectively which should correspond to shafts nominal diameters. Reduction in bending stress was found as pressure angle increased. The induced bending stress at pressure angle of 20^0 was 23.58N/mm2, a lower value compared to the design bending stresses for AISI 1018 and A6061 respectively for the fillet stress concentration factors.

Keywords— Gear, Design, Horizontal Axis Wind Turbine, Pressure Angle, Bending Stress.

INTRODUCTION

The indispensability of energy in virtually every aspect of human affairs cannot be overemphasized. These days the way to go favours the patronage of renewable energy of which wind turbines are essential part. Renewable energy sources have gained much attention due to the recent energy crises, and the urge to get clean energy (Iliyasu, Iliyasu, Tanimu, and Obada, 2017); the duty to curb the undesirable climate change. Wind power is capable of providing a competitive solution to battle global climate change (Wang, Kolios, Luengo, and Liu, 2016). Horizontal axis wind turbines (HAWTs) have gained dominance over vertical axis wind turbines (VAWTs) as it is today.

Although VAWTs do not become short of wind, their tip speed ratio is not high and are associated with difficulty of self-starting. This work looks at the design of the spur gear drives for a 100 Watt (W) HAWT. The objectives basically include determining the speed and shaft sizes requirements for the gears and carrying out the design while considering appropriate material selection; evaluating the bending stress at the gears root and modelling the gears with AutoCAD.

Justifying this study include the step towards harnessing local expertise, providing improved electricity supply, as well as the promotion of clean and renewable energy.

The function of the gears is to increase the rotational speed of the main shaft to that required at the output shaft coupled to the generator. This is necessary since the wind speed is much lower.

Engineering design principles, codes and standards deployable to specific component design are critical to achieving the aim. Materials selection for the fabrication of each part will normally take into account parameters such as magnitudes and types of forces acting, mechanical properties, environmental condition, availability, and importantly cost among other factors. As a result, these set of factors or variables often presents tradeoffs that make the design process even more complex (Ogunkah and Yang, 2012).

Applicable design standards and codes should also be consulted, among others include the American Institute of Steel Construction (AISC), American Gear Manufacturers Association (AGMA), Japanese Industrial Standards (JIS). Gears are toothed wheels which are utilized for power transmission from one shaft to another.

They can be used to increase or decrease shaft speed depending on what speed or torque is required at the point of usage. Gears can also be applied to change the direction of motion during power transmission to achieve desired effect. There are various types of gears with their specific functions, pros and cons. Spur gears have their teeth straight, and two spur gears can transmit power between parallel shafts.

Helical gear teeth are cut along a helix, while bevel gears are used to transmit motion at right angles. Other types of gears include worm and wheel, and finds application in steering systems.

Due to the low speed of the wind and rotor of HAWT, gearing is applied to increase the speed using a required gear ratio based on the designed output shaft rotational speed that matches the speed of the generator.

THEORETICAL ANALYSIS

2.1 Design Calculation

2.1.1 Gear Design

Gear teeth, rotational speed, diameter and torque relation is

$$\frac{Z_2}{Z_1} = \frac{N_1}{N_2} = \frac{\omega_1}{\omega_2} = \frac{D_2}{D_1} = \frac{r_2}{r_1} = \frac{T_2}{T_1} = G_r \quad (1)$$

where, $Z_1 = 28$ number of input gear teeth; $Z_2 =$ number of output gear teeth; $N_1 = 381 =$ revolutions per minute (rpm) = main shaft rotational speed; $N_2 = 800\ rpm =$ output shaft rotational speed; $\omega_1 =$ main shaft rotational speed (rad/s) ; $\omega_2 =$ output shaft rotational speed (rad/s); $D_1 =$ input gear diameter (mm); $D_2 =$ output gear diameter (mm); $r_1 =$ input gear radius (mm); $r_2 =$ output gear radius (mm); $T_1 =$ main shaft torque (N/m) ; $T_2 =$ output shaft torque (N/m); $G_r = 0.476 = gear\ ratio.$

From equation (1)

$$Z_2 = \left(\frac{N_1}{N_2}\right) * Z_1$$

$$Z_2 = \left(\frac{381}{800}\right) * 28$$

$$Z_2 = 13.3$$

Shaft design results from previous research which this work follows are shown in Table 1.

Table I: Shaft Design Results for a 100W HAWT

Power, P (W)	100
Main Shaft Rotational Speed, N_1 (rpm)	381
Main Shaft Design Torque, T_d (Nm)	2.62
Main Shaft Brake Torque, T_{bi}, (Nm)	5.28
Main Shaft Total Torque, T_t, (Nm)	7.92
Output Shaft Rotational Speed, N_2 (rpm)	800
Output Shaft Torque, T_o (Nm)	1.26
Output Shaft Brake Torque, T_{bo}, (Nm)	2.52
Output Shaft Total Torque, T_{to} (Nm)	3.78
Main Shaft Diameter, D_1, (mm)	15.7
Output Shaft Diameter, D_2, (mm)	5.6

Source: Ayadju and Ujevwerume, 2020

Gear materials include steels, aluminium alloys and plastics with their advantages and disadvantages respectively. Mild steel strength properties, low cost and availability are desirable factors for its selection for the gears manufacture. The mechanical properties of mild steel (AISI 1018) include yield tensile strength of $370N/mm^2$ and ultimate tensile strength of $440N/mm^2$. For surface hardness $(BHN) \leq 350$ and life cycle $\geq 10^7$, the design bending stress σ_b is given in equations (2) and (3) respectively,

$$\sigma_b = (1.4k_{b1}\sigma_{-1})/nk_\sigma \quad (2)$$

(for single direction of rotation), and

$$\sigma_b = (k_{b1}\sigma_{-1})/nk_\sigma \quad (3)$$

(for rotation in both directions),

where, $k_{b1} = 1 =$ life factor for number of cycles $> 10^7$; $k_\sigma = 1.4 =$ fillet stress concentration factor; $n = 2$ (taken) $=$ safety factor; $\sigma_{-1} =$ endurance limit in reversed bending (Amaldhasan and PonPaul, 2013; Maitra, 1989) ,

$$\sigma_{-1} = (0.22(\sigma_u + \sigma_y) + 50)N/mm^2 \quad (4)$$

where, $\sigma_u = 440N/mm^2$, and $\sigma_y = 370N/mm^2$ are ultimate and yield stress accordingly.

The minimum module, m_{od} for cutting the gears is given as

$$m_{od} \geq 1.26(M_t/\sigma_b\varphi_m Zy)^{1/3} \quad (5)$$

where, $Z =$ number teeth; $M_t = (7.92 * 10^3\ Nmm = M_t$ (Table 1) taken for safety; $\varphi_m = b/m_{od} = 10$ (assumed); $b =$ face width; $y = 0.3$.

Applying equation (5) in respect of the input gear, then

$$m_{od} = 1.26\,(7.92 * 10^3/114.2 * 10 * 28 * 0.3)^{1/3}$$

$$m_{od} = 1.18mm$$

and in respect of the output gear then,

$$m_{od} = 1.26\,(7.92 * 10^3/114.2 * 10 * 13.3 * 0.3)^{1/3}$$

$$m_{od} = 1.51mm.$$

Again, applying this in equation (5) then,

$$m_{od} = 1.1mm \text{ and } m_{od} = 1.4mm$$

take $2mm$ as module.

Tooth circular pitch, thickness, centre distance, reference, tip (outside) and root diameters are given in equations (6), (7), (8), (9), (10) and (11) accordingly,

$$p = \pi m_{od} \quad (6)$$

$$t = \pi m_{od}/2 \quad (7)$$

$$a = m_{od}(Z_1 + Z_2)/2 \quad (8)$$

$$D = Zm_{od} \quad (9)$$

$$D_t = D + 2m_{od} \quad (10)$$

$$D_r = D - 5/2 * m_{od} \quad (11)$$

where, $m_{od} = 2mm =$ module used; $p =$ pitch (mm); $t =$ thickness; $a =$ centre distance; $D =$ reference diameter; D_1 and D_2 are reference diameters of the meshing gears (fig. 1); $D_t =$ diameter at tip; $D_r =$ diameter at root;

$$p = \pi * 2$$

$\boldsymbol{p = 6.28mm}$

But, $\boldsymbol{m_{od} = D/Z}$

so that,

$\boldsymbol{D = pZ/\pi}$ **(12)**

Putting values in to equation (12), then

$\boldsymbol{D_1 = 6.28 * 28/\pi = 56mm, and D_2 = 6.28 * 13.3/\pi}$

$\boldsymbol{= 26.6mm,}$

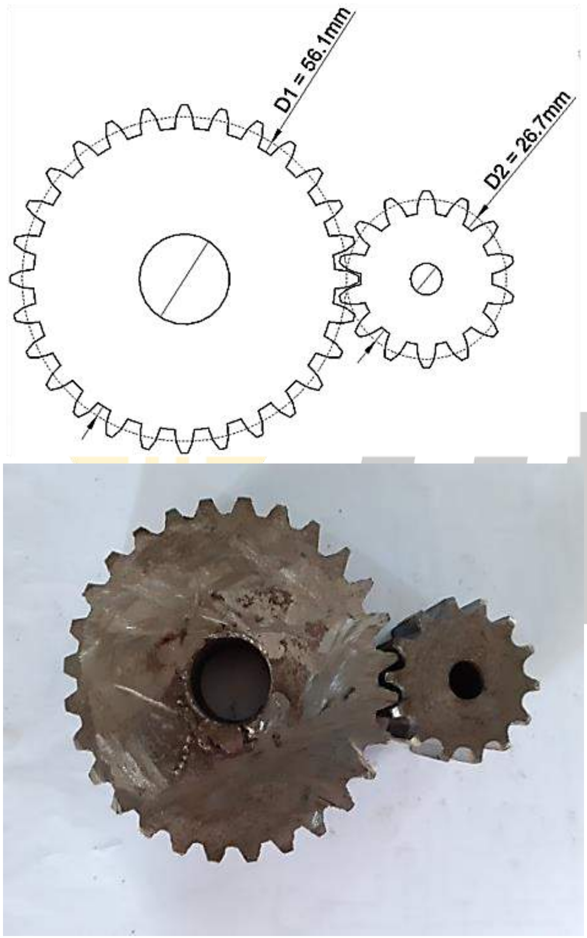

Fig. 1: AutoCAD 2-D Model and Manufactured Designed Spur Gears

equation (8) gives

$\boldsymbol{a = (56 + 26.6)/2 = 41.3mm}$

and using equation (7),

$\boldsymbol{t = 3.14mm}$**, and**

from equations (10) and (11) the tip and root diameters of the input gear, $D_{ti} = 60mm$ D_{ti} = 60mm and $D_{ri} = 52mm$ respectively, while those for the output gear are $\boldsymbol{D_{to} = 30.6mm}$ **and** $\boldsymbol{D_{ro} = 22.6mm}$.

The gear face width, b is given as

$\boldsymbol{b = \varphi_m m_{od}}$ **(13)**

or

$\boldsymbol{b = \varphi a}$

where, $\varphi = 0.3$; putting value in equations, then

$\boldsymbol{b = 10 * 2 = 20mm,}$ **or** $\boldsymbol{b = 0.3 * 41.3 = 12.39mm.}$ **Take** $\boldsymbol{b = 20mm.}$

2.1.2. Tangential Force

The tangential force F_t acting on teeth of input and output gear in meshed operation is given as

$\boldsymbol{F_t = T/r}$ **(14)**

or

$\boldsymbol{F_t = T * 2/D}$

with $T = T_d = 2.62 * 10^3$ (Table 1), and $r = D/2 = 56/2 = 28mm$, hence,

$\boldsymbol{F_t = 2.62 * 10^3/28}$

$\boldsymbol{F_t = 93.6N,}$

using the relation in equation (15),

$\boldsymbol{F_t = 102 * P/v}$ **(15)**

where, $v = d_b n/19100$ = tangential speed of working pitch (m/s); $P = 0.1kW$ = power; $d_b = 56$ = working pitch diameter (mm); $n = 381$ = rotational speed (rpm), so that

$\boldsymbol{F_t = (102 * 0.1)/(56 * 381/19100)}$

$\boldsymbol{F_t = 9.13kf/mm^2 = 89.6N}$, this value is close to value got from equation (14).

applying $\boldsymbol{T = T_t = 7.92 * 10^3}$ (Table 1), and $r = D/2 = 56/2 = 28mm$, hence,

$\boldsymbol{F_t = 7.92 * 10^3/28}$

$\boldsymbol{F_t = 282.9N,}$

considering the total torque and choosing $F_t = 282.9$, the resultant force, F_R on the gear tooth is

$\boldsymbol{F_R = F_t sec\emptyset}$ **(16)**

$\boldsymbol{F_R = F_t * 1/cos\emptyset}$

where, $\emptyset$ is pressure angle = 20^o, although, other pressure angles exist including standard 14.5^o as well as

25^o. In the work, effect of pressure angle and tip relief on the life of speed increasing gearbox: a case study, the analysis was carried out for three different pressure angles of 15º, 20º and 22.5º (Shanmugasundaram, Kumaresan and Muthusamy, 2014).

$\boldsymbol{F_R = 297.46N}$.

2.1.3 Bending Stresses

For the AISI 1018, the design bending stress can be obtained from putting parameter values in equations (3) and (4). The endurance limit in reverse bending is 228.2N/mm^2. Again, using equations (3), the designed bending stress for aluminium alloy (A6061-T6) can be evaluated, with values of tensile strength, ultimate tensile strength and fatigue limit (fatigue strength) also called endurance limit being $276N/mm^2$, $310N/mm^2$ and $96.5N/mm^2$ respectively.

From the Lewis equation the induced bending stress is given in equation (17),

$$\boldsymbol{\sigma_{bi} = F_t/b * m_n * y} \quad \textbf{(17)}$$

(Osakue and Anetor, 2020),

where, $\sigma_{bi} =$ induced bending stress; $b = 20mm =$ face width; $m_n = 2 =$ normal module; $y = 0.3 =$ Lewis factor, the induced bending stress from inputting parameters in equation when pressure angle is set to 20^o is

$$\boldsymbol{\sigma_{bi} = 282.9/20 * 2 * 0.3}$$

$$\boldsymbol{\sigma_{bi} = 23.58N/mm^2},$$

MATERIALS AND METHODS

Rotor and output shaft speeds were applied to gear design principles to determine the number of teeth on gears. Pitch and minimum module for cutting the gears were calculated to obtain input and output gear diameters.

The gears internal diameters were also sized based on the shafts diameters which were previously got from taking into consideration torsion and bending loads, as well as standardisation. Modelling of the gears was carried out using AutoCAD.

The design bending stresses for AISI 1018 and A6061-T6 were evaluated, as well as to understand how design stress is influenced by fillet stress concentration factor. The induced bending stress at the root of the gears teeth was also determined following applicable theory (Osakue and Anetor, 2020).

RESULTS AND DISCUSSION

Table II: Variation of Bending Stress with Pressure Angle

Pressure angle, Ø (º)	Tangential Force, F_t (N/mm^2)	Bending Stress, σ_{bi} (N/mm^2)
14	290.30	24.19
15	289.24	24.10
16	288.11	24.01
17	286.92	23.91
18	285.65	23.80
19	284.31	23.69
20	282.90	23.58
21	281.42	23.45
22	279.87	23.32
23	278.26	23.19
24	276.57	23.05
25	274.82	22.90

Table III: Effect of Stress Concentration Factor on Design Bending Stress

Fillet Stress Concentration Factor, k_σ	AISI 1018 Design Bending Stress, σ_b (N/mm^2)	A6061-T6 Design Bending Stress, σ_b (N/mm^2)
0.7	228.22	96.50
0.8	199.69	84.44
0.9	177.50	75.06
1.0	159.75	67.55
1.1	145.23	61.41
1.2	133.13	56.29
1.3	122.89	51.96
1.4	114.11	48.25
1.5	106.50	45.03
1.6	99.85	42.22

Table IV: Designed Spur Gears Specification

Component	Input Gear, D_1	Output Gear, D_2
Inner Diameter, D_i (mm)	17	7
Outer Diameter, D_o (mm)	56	26.6
Face Width, b (mm)	20	20
Tooth Thickness, t (mm)	3.14	3.14
Number of Teeth, Z	28	13.3
Module, m_{od} (mm)	2	2
Pressure Angle, Ø (º)	20	20
Rotational Speed, N (rpm)	381	800
Induced Bending Stress (N/mm^2)	23.58	23.58
Pressure Angle (º)	20	20

The gear mechanism has been designed to obtain an increased rotational speed at the output shaft coupled to the generator. The gears specification for a 100W HAWT from results (Table IV) indicate a smaller sized output gear (pinion) of 26.6mm in diameter with 13.3 teeth, rotating at 800rpm, and the meshing driver gear with 28 teeth, 56mm diameter, about twice the size of the pinion. The gears face width and tooth thicknesses have also been found to be 20mm and 3.14mm respectively, promoting ease of mass production since these parameters are the same for the gear and pinion. Due to bearing boundary dimension standardization constraint, 17mm and 7mm have been selected for the main (input) and output gears internal diameters respectively which will also correspond to shafts diameters. The induced bending stress at the mating gears root decreased with increasing pressure angle (Table II). An induced bending stress of 23.58N/mm^2 on account of a tangential force of 282.9N was predicted to exert using a standard pressure angle of 20^o (Table II), a lower value compared to the design bending stresses for the mild steel material and A6061-T6 (Table III), thus keeping the design safe. This selected pressure angle would offer a good trade off on contact ratio and tooth strength of the meshing gears. The AISI 1018 material was considered for the gears manufacture above the A6061-T6 because of its higher strength, stress withstanding capacity (Table III) and it is relatively cheaper to procure.

CONCLUSION

A 100W HAWT rotor with rotational speed of 381rpm carrying a gear of 56mm diameter would use a gear of 26.6mm diameter and 13.3 teeth to deliver a rotational speed of 800 rpm at the generator end. The sizes of the main and output gears were also being found to be 17mm and 7mm inner diameter respectively. Due to a tangential force of 282.9N, the induced bending stress was obtained as 23.58N/mm^2 at a pressure angle of 20^owhich is much smaller than the design bending stress for both materials for the fillet stress concentration factors. Further A6061-T6 was found to have a much lower design strength compared to AISI 1018 which was selected for the gear manufacture because it is cheaper to buy and has higher strength.

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