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Design of Windmill Power Generation Using Multi-Generator and Single Rotor (Horizontal Blade)

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Abstract

Wind energy is the environmental free and one of the best renewable energy for generation of electric power. The main aim of the paper is "to produce current using multi generator and single rotor". This paper proposes multi-generator to address potential challenges: dimension, cost and reliability. The two permanent magnet D.C. generators are desired to share the single shaft through straight bevel gears. These poles of the two generators will be changed as alternate to parallel. This paper discussed about the design procedure of gears, gear life and wind turbine rotors. The output current is stored in series of battery to appliances through converter and step up transformer. The performances and practicalities of the proposed architecture are verified in simulation using prototype wind turbine.

Keywords: permanent magnet D.C. generator, wind turbine, straight bevel gear, poles of generator.

1. Introduction

The wind energy is an environment-friendly and efficient source of renewable energy. The kinetic energy of the wind can be used to do work. This energy is harnessed by windmill in the past to do mechanical work. This is used for water lifting pump and generating electricity. To generate the electricity, the rotary motion of the windmill is used to turn the turbine of the electric generator. The output of single windmill is quite small and cannot be used for commercial purposes. Therefore, a number of windmills are erected over a large area, which is known as wind energy farm. The each and every windmill is coupled together to get a electricity for commercial purposes. The wind speed should be higher than 15 Km/hr.

2. Literature review

The three-bladed rotor proliferates and typically has a separate front bearing, with low speed shaft connected to a gearbox that provides an output speed suitable for the most popular four-pole (or two -pole) generators. This general architecture commonly, with the largest wind turbines, the blade pitch will be varied continuously under active control to regulate power in higher operational wind speeds. Support structures are most commonly tubular steel towers tapering in some way, both in metal wall thickness and in diameter from tower base to tower top. Concrete towers, concrete bases with steel upper sections and lattice towers, are also used but are much less prevalent. Tower height is rather site specific and turbines are commonly available with three or more tower height options.

3. Design of windmill

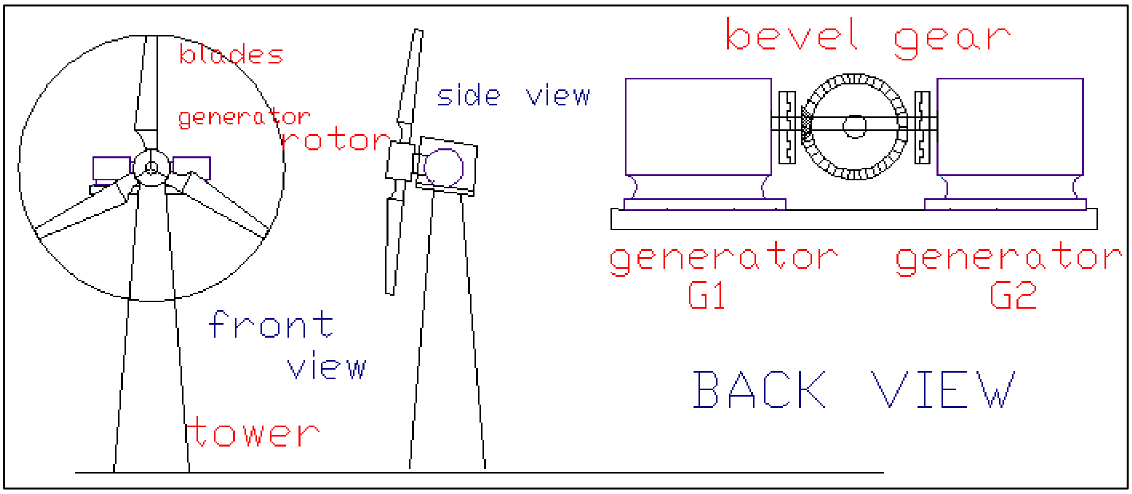


Figure 1. Experimental setup

The design is based on five steps as follows:

1. Design of wind turbine rotor; 2. Design of tower; 3. Yawn control; 4. Selection or Design of generator; 5. Design of gear.

3.1 Design of wind turbine rotor

3.1.1 Selection of location

Place: Mailam Engineering College

Latitude: 12.117^0; Longitude: 79.615^0

Hub height: 20m; Wind speed: 4.8 ± 0.9 m/s

Min. & Max. Wind speed: 3.9 m/s; 5.7 m/s

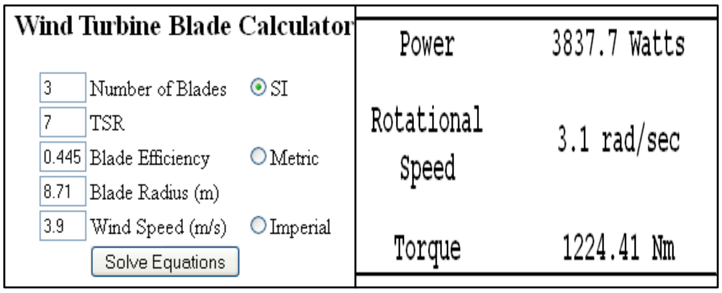


Figure 2. Software calculations

Nomenclature for Windmill: P_O = power contained in wind; η_E = efficiency of electrical generation; η_M = efficiency of mechanical transmission; C_P = power co-efficient; P = Required power output = 3600 W; V_∞ = wind speed velocity = 3.9 m/S; D = diameter of rotor; R = radius of rotor; a= axial interference factor; υ = Wind turbine velocity; V_2 = exit velocity of wind; ω = angular velocity of blade; v = n – bladed velocity; n = number of blades = 3 (assume); N = speed of bladed rotor; θ = Angle separated between two blades; t_a = Time taken by one blade move into the position of preceding blade; TSR = tip speed rate; A = Area of blade; L = Length of blade; f = Width of blade; t = Thickness of blade; H.R = Hub radius; V = Volume of blade; ρ = density of air = 1.225Kg/m^3; A = area of rotor; P_{MAX} = Maximum exactable power; C_F = Force coefficient; β = 1/3 (feasible) or 1 (constant); u = Aerofoil velocity; I = Angle of inclination; (L/D) = Ratio of lift to drag; C_L = Lift coefficient; C_D = Drag co-efficient; $C_{mc/a}$ =Moment co-efficient; C_{DO} = Profile drags co- efficient; i = Angle attack for infinite aspect ratio; w = relative velocity; A_b = Blade area; F_L = Lift force; F_D = Drag force; ϵ = Eiffel polar; i = Angle attack for infinite aspect ratio; α = Pitch angle; P_M = power speed characteristic; C_P = power co-efficient; C_T = Torque co-efficient = C_P / λ =0.1666; T_M = Torque speed characteristic; b_P = width of profile = 0.106m; υ' = kinematic viscosity; μ = absolute dynamic viscosity = 10^{-5} Pa = 10^{-5}N/m^2 (for air); μ = 8.16 x 10^{-6}; η_a = aero dynamic efficiency

3.1.2 Calculation of rotor diameter

Power, $P = P_O \eta_E \eta_M C_P$ (1)

In the absence of above data we use fast or slow rotor formula as shown below

For slow rotor, $P = 0.15\, D^2\, V_\infty^{\,3}$ (2)

For fast rotor, $P = 0.2\, D^2\, V_\infty^{\,3}$ (3)

We use the fast rotor formula to calculate diameter

D = 17.42 m; R = 8.71m

3.1.3 Circumference, swept area of rotor

Circumference of rotor = $\pi \times D$ = 54.72m (4)

$A_S = \pi \times R^2 = 238.33 m^2$ (5)

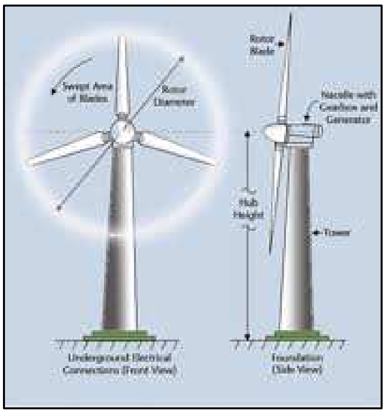
3.1.4 wind variation calculation

Axial interference factor, $\upsilon = V_\infty (1 - a)$ (6)

$= \frac{2}{3} V_\infty = 2.6$ m/s (7)

a = 0.3333

$V_2 = 2\upsilon - V_\infty = 1.3$ m/s (8)



3.1.5 Calculation of number of blades

Figure 2. windmill specification

$$\frac{\omega}{v} \approx \frac{2\pi}{nD} \quad (9)$$

ω = 0.3125 rad/sec

$$\omega = \frac{2\pi N}{60} \quad (10)$$

N = 2.985 rpm

3.1.6 velocity of wind turbine rotor

$$V = \frac{\pi D N}{60} = 2.723 \text{ m/s} \quad (11)$$

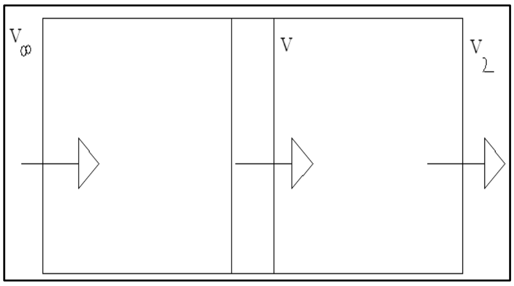


Figure 3. wind transfer specification

3.1.7 Angle separated between two blades

$$\theta = \frac{360}{n} = 120^0 \quad (12)$$

3.1.8 Time taken by one blade move into the position of preceding blade

$$t_a = \frac{2\pi}{n\omega} = 6.7 \text{ sec} \quad (13)$$

.

3.1.9 Time taken for turbine disturbed by wind

$$t_b = \frac{D}{v} = 6.7 \text{ sec} \quad (14)$$

3.1.10 Calculation of hub radius, Length of blade, thickness of blade, width of blade

H.R = 0.14 x R = 1.22m; L = 0.86 x R = 7.49m; t = 0.2 x L = 1.50m; f = 0.1 x t = 0.15m

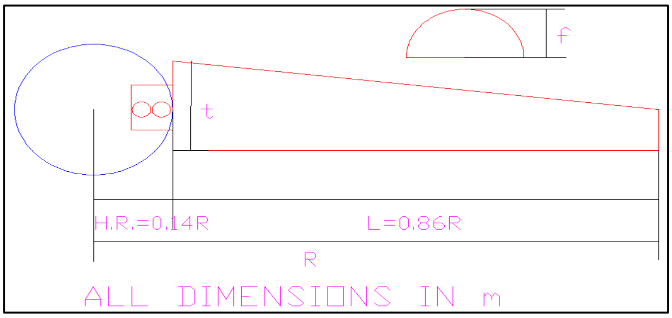


Figure 4. blade specification

3.1.11 Area and Volume of blade

A = Lf = 1.1235 m^2 (15)

V = Lft = 1.69 m^3 (16)

3.1.12 Power contaminated in wind

$$P_O = \frac{\rho A V_\infty^3}{2} \quad (17)$$

A = π x R^2 = 238.33 m^2

P_0 = 8659.37 W

3.1.13 Maximum exactable power

$$P_{MAX} = \frac{16}{27} P_O = 5131.47 \text{ W} \quad (18)$$

3.1.14 Calculation of tip speed rate

$$revolutions(rpm) = \frac{V_\infty \times TSR \times 60}{6.28 \times R} \quad (19)$$

TSR = 7

$$\text{Tip speed rate} = \frac{\text{blade tip speed}}{\text{wind velocity}} \quad (20)$$

Blade tip speed = 27.23 m/s

3.1.15 Force coefficient

$$C_F = \frac{27}{8} \frac{P_{MAX}}{P_o} = 2 \quad (21)$$

3.1.16 Aerofoil velocity

$$\frac{u}{V_\infty} = \beta \qquad (22)$$

u = 1.3

3.1.17 Angle of inclination

$$I = \cot^{-1}\frac{u}{v} = 63.4^0 \qquad (23)$$

3.1.18 Power co-efficient

$$C_P = \frac{\text{Power output from wind turbine}}{\text{power contaminated in wind}} \times 100 \qquad (24)$$

$C_P = 0.4142 \times 100 = 41.42 \approx 40\%$

From aerofoil data sheet National Advisory Committee of Aeronautics NACA 4412,

(L/D) = 20; $C_L = 1$; $C_D = 0.20$; $C_{mc/a} = -0.08$; $C_{DO} = 0.01$; i = -2

3.1.19 Lift force

$$F_L = \frac{1}{2}\rho A_b w^2 C_L \qquad (25)$$

$$w = \sqrt{u^2 + v^2} = 2.91 \text{ m/s} \qquad (26)$$

$F_L = 5.824$ N

3.1.20 Drag force

$$F_D = \frac{1}{2}\rho A_b w^2 C_D = 1.16 \text{ N} \qquad (27)$$

3.1.21 Aerodynamic forces in aerofoil moving in direction of the wind

where, F_D = Drag force; F_L = Lift force; u = Aerofoil velocity; V = wind velocity

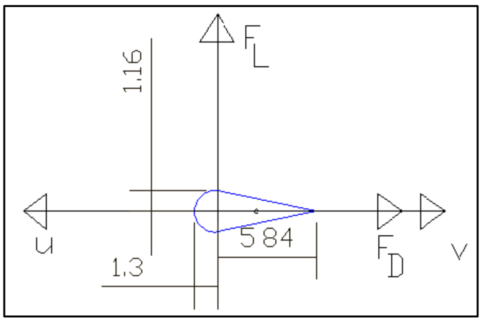


Figure 5. Aerodynamic force in aerofoil

3.1.22 Tangent to the Eiffel polar

$$\text{Tan} \in = \frac{C_D}{C_L} \qquad (28)$$

$\epsilon = 21.8^0$

3.1.23 Calculation of pitch angle

$\alpha = I - i \quad = 65.4$ (29)

3.1.24 Calculation of specific rated capacity

$$SRC = \frac{\text{power rating of the generator}}{\text{rotor swept area}} = 7.553 \quad (30)$$

3.1.25 Calculation of power speed characteristic

$$P_M = \frac{\rho C_p \pi R^2 V_\infty^{\ 3}}{2} = 3593.57 \text{ W} \quad (31)$$

3.1.26 Torque speed characteristic

$$T_M = \frac{P_M}{\omega} \text{ or } T_M = \frac{1}{2}\rho C_T \pi R^3 V_\infty^{\ 3} = 12570.51\text{N-m} \quad (32)$$

3.1.27 Calculation of Reynolds number

$$R_E = \frac{V_\infty b_p}{\nu'} \quad (33)$$

$$\upsilon' = \frac{\mu}{\rho} \quad (34)$$

$R_E = 2.1 \times 10^5$

3.1.28 Calculation of aero dynamic efficiency

$$\eta_A = \frac{1 - (\text{Tan} \in \text{Cot I})}{1 + (\text{Tan} \in \text{Tan I})} \times 100 = 44.45\% \quad (35)$$

3.2. Design of Tower

From the table 1, required power is 3.6KW. So, we taken as 20m

Table 1. Tower height selection

Power	**Tower height**
Upto 100 KW	Upto 30m
100 - 300 KW	30 - 35m
300 - 500 KW	35 - 40m
Above 500 KW	Above 40m

3.3. Yaw control

The yaw control mechanism is used to control the speed of rotor. when fan tail is placed perpendicular to the main turbine. so that the thrust force automatically pushes the turbine in the direction of wind.

3.4. Design or Selection of Generator

If the required power of different generators are present in market at low speed, we used that generator to make a gear design. Until, we design the gear to design the generator. The selected generator is Permanent Magnet D.C. Generator and required power is 3600W present in market at low speed of 480rpm and model name is GL – PMG – 1800. Specification is given below:

Electrical Specification

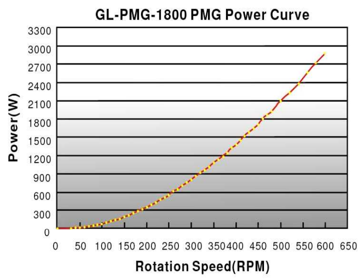
Rated Output Power(W):	1800
Rated Rotatoin Speed (RPM):	480
Recified DC Current at Rated Output (A):	6
Requied Torque at Rated Power:	44.5
Phase Resistance (Ohms):	5.0
Output Wire Square Section (mm^2):	4
Output Wire Length (mm):	600
Insultation:	H Class
Generator configuration:	3 Phase star connected AC output
Design Lifetime:	>20 years

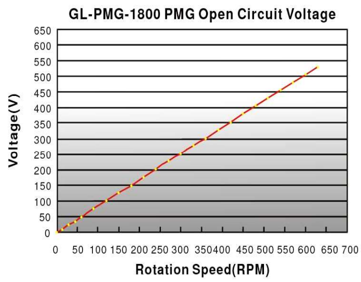
Mechanical Specification

Weight (Kgs):	19.5
Starting Torque (NM):	<0.9
Rotor Inertia (Kg.m^2):	0.013
Bearing Type:	High standard NSK 6207DDUC3 (Front) NSK 6207VVC3 (Rear)

Material Specification

Shaft Material:	High standard Stainless Steel
Shaft Bearing:	High standard SKF or NSK bearing
Outer Frame Material:	High standard Aluminium alloy with TF/T6 heat treatment
(TF/T6 full heat treatment for increasing the performance of aluminium alloy as follows. Heat 4-12 hours at 525-545 degrees Celsius, quench with hot water, and precipitation heat treatment for 8-12 hours at 155-175 degrees Celsius.)	
Fasteners (nuts and bolts):	High standard Stainless Steel
Windings Temperature Rating:	180 degrees Celsius
Magnet Material:	NdFeB (Neodymium Iron Boron)
Magnets Temperature Rating:	150 degrees Celsius
Lamination Stack:	High specification cold-rolled Steel





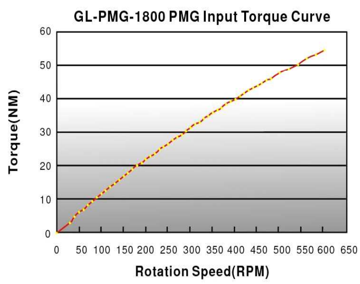


Figure 6. Specification of D.C. generator

3.5. Design of gear

3.5.1 Gear name: Straight Bevel Gear.

(NOTE: From PSG data book pg. no: 8.38, 8.39, 1.40, 8.53, 8.18, 8.51, 8.15, 8.17, 8.14, 8.16, 8.13, 8.13A)

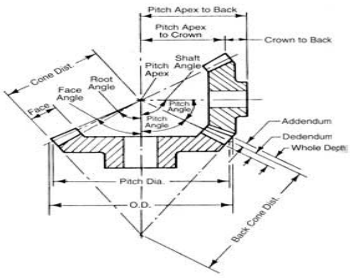


Figure 7. Specification of straight bevel gear

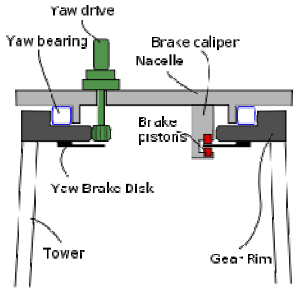


Figure 8. Yawn control

Nomenclature for Gears: N_2 = driver gear speed (Turbine speed = 2.985 rpm); N_1 = driven gear speed = 480rpm; i = Transmission ratio; Z_1 = No. of teeth in driven gear $\geq 7 = 7$; Z_2 = No. of teeth in driver gear; δ_2 = Pitch angle of driver gear; δ_1 = Pitch angle of driven gear; Z_{V1} = Virtual number of teeth of driven gear; Z_{V2} = Virtual number of teeth of driver gear; P = Rated power = 3600 W; K_O = Correction factor, 2 (assume high shock); V = Velocity of driven gear; F_T = tangential load on tooth; d_1 = diameter of driven gear, (M_T x Z_1) millimeter; M_T = transverse module; F_D = dynamic load; C_V = velocity factor; b = face width = 10 M_T; σ_B = bending stress for alloy steel 126 N/mm^2; R = cone distance; y' = lewis form factor; C = errors in action; e = Errors of tooth profile; V = mean velocity; Q' = Ratio factor; K_w = Wear factor; n = speed of driven gear; T = life time in hrs; N = Life time in cycles; , M_τ = nominal twisting moment transmitted by driven gear; K_d = dynamic load factor; K = load concentration factor; K_{BL} = life factor for bending; σ_{-1} = endurance limit stress in bending ($\sigma_U + \sigma_Y$) + 120; σ_U = ultimate tensile stress = 700 N/mm^2; σ_Y = yield stress = 360 N/mm^2; n = factor of safety = 2; k_σ = fillet stress concentration of factor = 1.2; C_R = coefficient depending on the surface hardness, 22; HRC = brinell or rockwel hardness number, 55 – 63 ≈ 59; K_{CL} = life factor, 1; ψ_Y = 1; E_{eq} = Equivalent young's modulus; [σ_C] = Compressive stress; M_T = nominal twisting moment transmitted by driven gear; K_d = dynamic load factor = 1.2; K = load concentration factor = 1.02.

3.5.2 Calculation of transmission ratio

$$i = \frac{N_1}{N_2} = 160.80 \approx 161 \qquad (36)$$

3.5.3 Calculation of number of teeth

$$Z_2 = i\, Z_1 = 1127 \qquad (37)$$

3.5.4 Material

For pinion: C 45 (Forged steel, Case hardened)

Allowable static stress σ_B = 126 N/mm^2; Compersible static stress σ_C = 1150 N/mm^2; Tensile strength σ_U = 700 N/mm^2; Yield point stress σ_y = 360 N/mm^2; BHN = 229

For wheel: Case steel graded

3.5.5 Calculation of pitch angles and virtual number of teeth

Pitch angles, $\delta_2 = \mathrm{Tan}^{-1}\, i = 89.64^0$ (38)

$\delta_1 = 90^0 - \delta_2 = 0.356^0$ (39)

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Virtual number of teeth, $Z_{V1} = \dfrac{Z_1}{Cos\delta_1} = 7$ (40)

$$Z_{V2} = \frac{Z_2}{Cos\delta_2} = 179368.8 \approx 179369 \quad (41)$$

3.5.6 Calculation of tangential load on tooth (F_T)

$$F_T = \frac{P}{V} \text{諉}\ _O \quad (42)$$

$$V = \frac{\pi \text{諂}_I \text{諆}_1}{60} \quad (43)$$

V = 0.1759 M_T m/s.

$$F_T = \frac{40925.557}{M_T} \text{ N}$$

3.5.7 Calculation of dynamic load (F_D)

$$F_D = \frac{F_T}{C_V} \quad (44)$$

$$C_V = \frac{5.6}{5.6 + \sqrt{V}} \quad (45)$$

Where, V = velocity, 5m/s (assume)

$$C_V = 0.715; \quad F_T = \frac{57238.541}{M_T} \text{ N} \rightarrow (1)$$

3.5.8 Calculation of beam strength (F_S)

$$F_S = \pi M_T b \sigma_B y' \frac{(R - b)}{R} \quad (46)$$

$$y' = 0.154 - \frac{0.912}{Z_{v1}} \quad (47)$$

y' = 0.1084 [for 20^0 involute]

$$R = 0.5 \text{譓}\ _T \text{讀}\sqrt{\ _1^2 + Z_2^2} = 563.511\ M_T \quad (48)$$

$F_S = 421.476\ M_T^2 \rightarrow (2)$

3.5.9 Calculation of transverse module

From (1) and (2), $F_S \geq F_T$

$M_T \geq 4.59$ mm ≈ 5 mm

3.5.10 Calculation of b, V, d

Face width, $b = 10\ M_T = 50$ mm = 0.05 m (49)

Reference diameter, $d_1 = M_T \times Z_1 = 35$ mm = 0.035 m (50)

$d_2 = M_T \times Z_2 = 5635$ mm = 5.635 m (51)

Velocity, $$V = \frac{\pi 譫_I 諜_1}{60} = 0.879 \text{ m/s} \quad (52)$$

3.5.11 Revision of beam strength

$$F_S = \pi M_T b \sigma_B y' \frac{(R - b)}{R} \quad (53)$$

$R = 563.511\ M_T$ = 2817.555 mm = 2.8176 m

$F_S = 10536.9$ N

3.5.12 Calculation of accurate dynamic load

$$F_D = F_T + \frac{21V(bC + F_T)}{21V + \sqrt{(bC + F_T)}} \quad (54)$$

C = 8150 e; e = 0.022; C = 179.3 mm (55)

$$F_T = \frac{P}{V} = 4.09556 \text{ N (in mm)} \quad (56)$$

$F_D = 8927.4$ N → (4)

3.5.13 Check the beam strength

From (3) and (4), $F_D \leq F_S$; So, design is safe and satisfactory.

3.5.14 Calculation of maximum wear load

$$F_W = \frac{0.75 d_1 b Q'(K_W)}{\cos\delta_1} \quad (57)$$

$$Q' = \frac{2Z_{V2}}{Z_{V1} + Z_{V2}} = 1.980 \quad (58)$$

$K_w = 0.919$ N/mm^2; $F_W = 2388.297$ N → (5)

3.5.15 Check for beam strength

From (5) and (4), $F_D \leq F_W$

So, design is safe and satisfactory.

3.5.16 Gear life

Life time: 10, 00, 000 hrs (assume)

$N = 60n\,T = 2.88 \times 10^{10}$ cycles (59)

3.5.17 Calculation of twisting moment

$[M_\tau] = M_\tau \times K \times K_d$ (60)

WhereK. $K_d = 1.3$

$$M_T = \frac{P?\ \ 0}{2\pi N_1} = 71.61972 \text{ N-m} \quad (61)$$

$[M_\tau] = 93.105$ N-m

3.5.18 Calculation of E_{eq}, $[\sigma_B]$, $[\sigma_C]$

Equivalent young's modulus $E_{eq} = 2.15 \times 10^5$ N/mm^2

Bending stress $[\sigma_B] = \dfrac{K_{BL}\sigma_{-1}}{nK_\sigma}$ (62)

$\sigma_{-1} = 0.22\,(\sigma_U + \sigma_Y) + 120 = 353.2$ N/mm^2 (63)

bending stress $[\sigma_B] = 103.01$ N/mm^2

Compressive stress, $[\sigma_C] = C_R \times HRC \times K_{CL} = 1122$ N/mm^2 (64)

3.5.19 Calculation of cone distance

$$R \geq \psi_Y \sqrt{(i^2+1)}\sqrt[3]{\frac{E_{eq}M_T}{i} \times \left(\frac{0.72}{(\psi_Y - 0.53)[\sigma_C]}\right)^2} \quad (65)$$

$R \geq 0.5688$

3.5.20 Revision of center distance

R = cone distance = $0.5\text{譓}\;{}_T\,\text{讀}\sqrt{{}^2_1 + Z_2^2} = 563.51$ mm (66)

Design is satisfactory

3.5.21 Calculation of ψ_Y

$\psi_Y = b/d_1 = 1.428$ (67)

3.5.22 Select the suitable quality

The preferred quality is 10 or 12.

IS quality is coarse.

3.5.23 Revision of design torque

$[M_\tau] = M_\tau \times K \times K_d = 87.6$ N-mm (68)

3.5.24.Calculation of revision of bending stress

$$\sigma_B = \frac{R\sqrt{i^2+1}[M_T]}{(R-0.5b)^2 bM_T Y_{V1}} \quad (69)$$

Where, $Y_{V1} = 0.389$

$\sigma_B = 0.0455$ N/mm^2

3.5.25 Check for bending

$\sigma_B \leq [\sigma_B]$; Therefore, design is safe and satisfactory

3.5.26 Calculation of revision of wear strength

$$\sigma_C = \frac{0.72}{R-0.5b}\left[\frac{\sqrt{(i^2+1)^3}}{ib}E_{eq}[M_T]\right]^{\frac{1}{2}} = 132.1 \text{ N/mm}^2 \quad (70)$$

3.5.27 Check for wear strength

$\sigma_C \leq [\sigma_C]$; Therefore, design is safe and satisfactory

3.5.28 Basic calculations

Transverse module, $M_T = 5$ mm

No. of teeth, $Z_1 = 7$; $Z_2 = 1127$

Cone distance, R = 563.51 mm

Face width, b = 50 mm

Pitch angle, $\delta_1 = 0.356^0$; $\delta_2 = 89.64^0$

Reference diameter, $d_1 = M_T \times Z_1 = 35$ mm

$d_2 = M_T \times Z_2 = 5635$ mm

Tip diameter, $d_{a1} = M_T (Z_1 + \text{Cos}\ \delta_1) = 20.995$ mm

$d_{a2} = M_T (Z_2 + \text{Cos}\ \delta_2) = 200.099$ mm

Height factor, $f_0 = 1$

Addendum angle, $\text{Tan}\theta_{a1} = \text{Tan}\theta_{a2} = \frac{M_T f_0}{R}$

$\theta_{a1} = \theta_{a2} = 0.57^0$

Dedendum angle, $\text{Tan}\theta_{f1} = \text{Tan}\theta_{f2} = \frac{M_T (f_0 + C)}{R}$

$\theta_{f1} = \theta_{f2} = 0.0119^0$

Shift angle: $\sum = \delta_1 + \delta_2 = 90^0$

Whole depth: $h = 1.2\ M_T = 1.2$ mm

Middle circle diameter:

$d_{m1} = d_1 - b \sin \delta_1 = 10.05$ mm

$d_{m2} = d_2 - b \text{ Cos}\ \delta_2 = 100.05$ mm

Addendum: $h_{a1} = h_{a2} = M_T = 1$ mm

Dedendum: $h_{f1} = h_{f2} = 1.2\ M_T = 1.2$ mm

Face angle:

$\delta_{a1} = \delta_1 + \theta_{a1} = 6.281^0$

$\delta_{a2} = \delta_2 + \theta_{a2} = 71.63^0$

Back cone distance:

$R_{a1} = R \tan \delta_1 = 10.05$ mm

$R_{a2} = R \tan \delta_2 = 1005.1$ mm

Middle module: $M_m = d_{m1} / Z_1 = 0.5025$ mm

Crown height:

$C_{h1} = (d_2 / 2) - (M_T \sin \delta_1) = 99.9$ mm

$C_{h1} = (d_1 / 2) - (M_T \sin \delta_2) = 9.0$ mm

4. Conclusions

This paper presents a new methodology for power generation using two same generators of single rotor, further advantage of the method is cost efficient and generating high power with a same torque. Theoretical analysis and experimental work is carried out confirm validity of the analytical work.

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