DESIGN AND ANALYSIS OF WIND INATOR

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Abstract - This project is based on Green Energy where energy is obtained from the wind source. Hear wind energy is converted into mechanical energy where further converted into electrical energy. During this process there is no emission of carbon or greenhouse gases. By adopting this method they do not produce any harmfulness to environmental. This is one of the best source because, the consumed energy which has not created or destroyed during this energy conversion. Now days the use of electrical energy is getting more and more in day today life. So the use of fossil fuel are consumed in a greater extent. So we need to fine some alternate method.

Many research works are taking place in conventional energy method so I have focused on the wind source. Wind energy is available, abundantly in this planet. So use of that wind is appropriate method which leads to energy production. By installing wind turbines the wind are captured by the wind blades. Those taped wind energy rotates the blade. This rotation is made constant so that this wind energy which help to convert into mechanical form of energy. Due to this rotation there is a torque conversion. Torque is converted into speed that speed is use to run the generator. The generator produces electricity. That electricity is used to produce useful work.

So I have made a study on wind speed and wind direction. Based on the work I have planned to design the wind turbine which works efficiently based on the wind speed. By installing this wind turbine at the top of the house, tall buildings, street lap and near the coastal area. We can get optimal energy which will be very useful rather than using other forms of energy.

I. INTRODUCTION

Windmills were used in Persia (present-day Iran) as early as 200 B.C. The wind wheel of Hero of Alexandria marks one of the first known instances of wind powering a machine in history. However, the first known practical windmills were built in Sistan, an Eastern province of Iran, from the 7th century. These "Panemone" were vertical axle windmills, which had long vertical drive shafts with rectangular blades. Made of six to twelve sails covered in reed matting or cloth material, these windmills were used to grind grain or draw up water, and were used in the gristmilling and sugarcane industries.

Windmills first appeared in Europe during the middle Ages. The first historical records of their use in England date to the 11th or 12th centuries and there are reports of German crusaders taking their windmill-making skills to Syria around 1190. By the 14th century, Dutch windmills were in use to drain areas of the Rhine delta.

The first utility grid-connected wind turbine to operate in the UK was built by John Brown & Company in 1951 in the Orkney Islands.Despite these diverse developments, developments in fossil fuel systems almost entirely eliminated any wind turbine systems larger than supermicro size. In the early 1970s, however, anti-nuclear protests in Denmark spurred artisan mechanics to develop microturbines of 22 kW. Organizing owners into associations and co-operatives lead to the lobbying of the government and utilities and provided incentives for larger turbines throughout the 1980s and later. Local activists in Germany, nascent turbine manufacturers in Spain, and large investors in the United States in the early 1990s then lobbied for policies that stimulated the industry in those countries. Later companies formed in India and China. As of 2012, Danish company Vestas is the world's biggest wind-turbine manufacturer.

II. DISADVANTAGE OF LARGE WIND TURBINE

a. Requires large open areas for setting up wind farms.
b. Noise pollution problem is usually associated with wind mills.
c. Wind energy can be harnessed only in those areas where wind is strong enough and weather is windy for most parts of the year.
d. It can be a threat to wildlife. Birds do get killed or injured when they fly into turbines.
e. Maintenance cost of wind turbines is high as they have mechanical parts which undergo wear and tear over the time.

III. ADVANTAGE OF SMALL WIND TURBINE

a. Low Cost.
b. No need of high installation setup like crane or fork lift.
c. Low manufacturing techniques.
d. Low inventory.
e. Reliable power output compared to large wind turbine.
f. Easy installation setup.
g. Lead to large production and give business opportunity to young Entrepreneur.

IV. ADVANTAGE OF VAWT OVER HAWT

a. The maximum aerodynamic efficiency of any VAWT will be lower than available HAWT designs. This difference is likely to be between 15 to 25%.
b. Due to the lower efficiency, the VAWT will capture less energy for the same swept area.
c. For a given swept area, the mass of the rotor and support structure of a VAWT will be greater than that of an equivalent HAWT. This mass difference is likely to translate into a cost difference

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d. The savings that a VAWT may enjoy due to lower drive train and maintenance costs are unlikely to balance the lower energy capture and higher initial rotor costs.

e. The same diseconomies of scale apply, in theory, to both HAWT and VAWT configurations.

f. Both HAWTs and VAWTs suffer from fatigue loads. The basic aerodynamic principles of the VAWT lead to fatigue loads at the harmonic frequencies, but the VAWT is not as sensitive as is the HAWT to the effects of turbulence. However, the VAWT is more likely to suffer resonant conditions especially if operated at variable speeds.

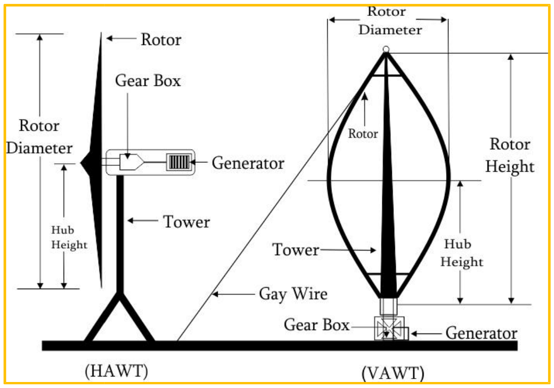

Fig 1. HAWT & VAWT

V. AVERAGE WEATHER FOR CHENNAI, INDIA

This report describes the typical weather at the Chennai International Airport (Chennai/Madras, India) weather station over the course of an average year. It is based on the historical records from 1996 to 2012.

Chennai/Madras has a tropical savanna climate with dry winters. The area within 40 km of this station is covered by croplands (60%), oceans and seas (33%), built-up areas (3%), and lakes and rivers (3%).

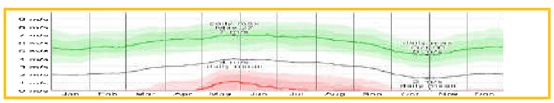

Fig 2. Wind Speed

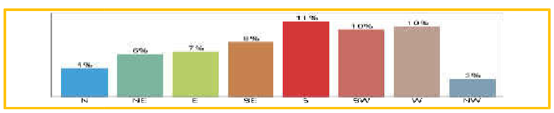

Fig 3. Wind Directions over the Entire Year

VI. .THEORY OF BETZ'S LAW

Betz's law calculates the maximum power that can be extracted from the wind, independent of the design of a wind turbine in open flow. It was published in 1919, by the German physicist Albert Betz. The law is derived from the principles of conservation of mass and momentum of the air stream flowing through an idealized "actuator disk" that extracts energy from the wind stream. According to Betz's law, no turbine can capture more than 16/27 (59.3%) of the kinetic energy in wind. The factor 16/27 (0.593) is known as Betz's coefficient. Practical utility-scale wind turbines achieve at peak 75% to 80% of the Betz limit.

Betz formula P = _ $^{3}C_p$

VII. CALCULATIONS OF POWER

By given the following data:

Blade length, $l = 2$ m

Wind speed, $v = 4$ m/sec

Air density, $\rho = 1.23$ kg/m3

Power Coefficient, $Cp = 0.4$

Inserting the value for blade length as the radiusof the swept area into equation (6) we have:

$l = r = 2$m

$A = \pi r^2$

$A = \pi \times 2^2$

$A = 12.566\ m^2$

We can then calculate the power converted from the wind into rotational energy in the turbine using equation (5);

P = _ $^{3}C_p$

P = _ $\times 1.23 \times 12.566 \times 4 \times 0.4$

P = 197.839 W

Table 1.Based on the wind speed and blade length, watts are calculated

Wind speed	l = 0.5m	l = 1m	l = 1.5m	l = 2m
2 m/s	1.545	6.182	13.910	24.730
4 m/s	12.365	49.461	111.287	197.844
6 m/s	41.732	166.931	375.596	667.726
8 m/s	98.922	395.689	890.302	1582.759
10 m/s	193.207	772.831	1738.871	3091.327
12 m/s	333.863	1335.453	3004.770	5341.813
14 m/s	530.162	2120.650	4771.463	8482.601

VIII. CALCULATION OF SPEED AND TORQUE

By given the following data:
Blade length, l = r = 2 m
Wind speed, v = 4 m/sec
Power, P = 197.839 W

$v = r\,\omega$ (1)

$= \omega$

$P = T\,\omega$ (2)

Substitute equation (2) in (1)

$P = \frac{}{}$ (3)

$P = 2\ \ n\,T$ (4)

Where
n = Speed in rps (rps - Revolution per second)
P = Power in w (w - watts)
T = Torque in Nm (Nm - Newton meter)

Equate equation (3) and (4)

$\frac{}{} = 2\ \ n\,T$ (5)

$n = \frac{}{\pi}$

Substitute the given data in equation (5)

$n = \frac{}{\pi}$

n = 0.318 rps
N = 19.08 rpm

Now take the value of n = 0.318 rps and substitute in equation (4)

$P = 2\ \ n\,T$

$T = \frac{}{\pi}$

$T = $

T = 99.015 Nm

Table 2. Based on the wind speed and blade length, theoretical value of speed in (rpm) are calculated

Wind speed	l = 0.5 m	l = 1 m	l = 1.5 m	l = 2 m
2 m/s	38.19719	19.09859	12.7324	9.549297
4 m/s	76.39437	38.19719	25.46479	19.09859
6 m/s	114.5916	57.29578	38.19719	28.64789
8 m/s	152.7887	76.39437	50.92958	38.19719
10 m/s	190.9859	95.49297	63.66198	47.74648
12 m/s	229.1831	114.5916	76.39437	57.29578
14 m/s	267.3803	133.6902	89.12677	66.84508

IX. CALCULATION OF TWO STAGE GEARS BOX

Calculation of stage one gear speed
Notations:
T_a, T_b & T_c represent the tooth available in the gears.
N_a, N_b & N_c represent the speed available in the gears.
By given the following data:
$T_a = 70$
$T_b = 30$
$T_c = 10$
Blade length, l = r = 2 m
Wind speed, v = 4 m/sec
Input speed N_a = 19.09859 rpm
Formula

$$\frac{}{} = \frac{}{}$$

70 =

30 19.09

N_b = 44.54 rpm

=

30 =

10 44.54

N_c = 133.63 rpm

Calculation of stage two gear speed

Now, the value of N_c is taken as input of N_a

=

70 =

30 133.63

N_b = 311.80 rpm

=

30 =

10 311.80

N_c = 935.41 rpm

The output speed is calculated as 935.41 rpm

Table 3. Based on the wind speed and blade length, two stage gear box (rpm) are calculated
ta = 70, tb = 30, tc = 10

Wind speed	l = 0.5 m	l = 1 m	l = 1.5 m	l = 2 m
2 m/s	1871.66231	935.83091	623.8876	467.915553
4 m/s	3743.32413	1871.66231	1247.77471	935.83091
6 m/s	5614.9884	2807.49322	1871.66231	1403.74661
8 m/s	7486.6463	3743.32413	2495.54942	1871.66231
10 m/s	9358.3091	4679.15553	3119.43702	2339.57752
12 m/s	11229.9719	5614.9884	3743.32413	2807.49322
14 m/s	13101.6347	6550.8198	4367.21173	3275.40892

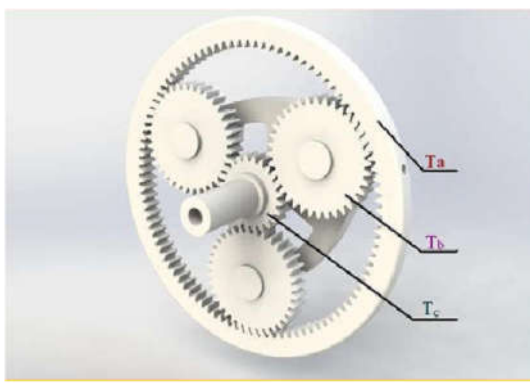

Fig 4. Diagrammatic representation of planetary gears

X. DESIGN OF PLANETARY GEAR

By given the following data:

Blade length, l = 2 m

Wind speed, v = 4 m/sec

Power, P = 197.839 W

Input speed in rpm, N_a = 19.09

Number of teeth in Z_a = 70

Number of teeth in Z_p = 30

Z_p = i x z_a (from DD book Page No: 8.22)

i = 0.42

=

From this we have calculated the value of

Output speed in rpm, N_p = 44.54 Table 40)

Calculation of Form factor

When Z_p = 30

Y_p = 0.358 20^0 FD (from DD book Page No: 8.53

y_p = Y_p /

Take $\sigma_p = 196$ N/mm^2 (from DDT's book Page No: 5.31 Table 5.4)
$y_p = (196 \times 0.358) / \pi$
$\sigma_p \times y_p = 22.33$
Calculation of Form factor
when $z_a = 70$
$Y_a = 0.429$ 20^0 FD (from DD book Page No: 8.53 Table 40)
$y_a = Y_a /$
Take $\sigma_a = 224$ N/mm^2(from DDT's book Page No: 5.31 Table 5.4)
$y_a = (224 \times 0.429) / \pi$
$\sigma_a \times y_a = 330.58$
Since the value of ($\sigma_a \times y_a$)is greater than ($\sigma_p \times y_p$)
So the planetary gear is weaker than the annular gear
Selection of Material
C55 Mn75 Carbon steel (from DD book Page No: 1.12)
Calculation of Module (M)
Since the center distance (a) is give as a = 250 mm.
we need not to equate F_s and F_d.
$a = M (Z_a + Z_p) / 2$
$250 = M$

$M = 5$
To check the recommended series of module in
(DD book Page No: 8.2 Table 1)
Calculation of b,d and V
Face width (b) = 10 x M
= 10 x 5
= 50 mm
Pitch circle diameter $d_a = M \times Z_a$
= 5 x 70
= 350 mm
$d_p = M \times Z_p$
$d_p = 5 \times 30$
= 150 mm
Pitch line velocity V =

V =

V = 0.3498 m/s
Calculation of beam strength (F_s)
$F_s =$ x M x b x σ_p x y_p
$F_s =$ x 5 x 50 x 196 x
$F_s = 17542$N
Calculation of dynamic load (F_d)
$F_d = F_t +$

$F_t =$
$F_t =$
$F_t = 565.57$N
C – Deformation factor for 20^0 Full depths (from DDT's book Page No: 8.53 Table 41)
C = 8150e
e – Expected errors
e = 0.0125
C = 8150 x 0.0125
C = 101.875 N/mm
$F_d = 5601.95$N
Check for beam strength or tooth breakage
We din F_s is greater than F_d it means the gear tooth has adequate beam strength and it will not fail by breakage thus the design is satisfactory.
Calculation of wear load (F_w)
Maximum wear load $F_w = d_a \times b \times q \times k_w$
q =

q = 0.59
k_w = Load stress factor (from DDT's book Page No: 5.36 Table 5.9)
$k_w = 20^0$ FD Cast iron = 1.42 N/mm^2
$F_w = 350 \times 50 \times 0.59 \times 1.42$
$F_w = 14661.5$N
Check for wear
We find F_w is greater than F_d it means gear tooth has adequate wear capacity and it will not wear out. Therefore the design is satisfactory.
Even the same approach is applied for stage two gear box.
Basic dimension of planetary gear (from DDT's book Page No: 8.22 Table 26)

Module $M = 5$ mm
Number of teeth: $Z_a = 70$
$Z_p = 30$
Pitch circle diameter: $d_a = 350$ mm
$d_p = 150$ mm
Centre distance: $a = 250$ mm
Face width: $b = 50$ mm
Height factor $f_o = 1$ for 20^0 FD
Bottom clearance: $C = 0.25$ M
$C = 0.25 \times 5$
$C = 1.25$ mm
Tip diameter: $d_{a1} = (Z_a + 2f_o)$ M
$d_{a1} = (70 + 2)\ 5$
$d_{a1} = 360$ mm
$d_{p1} = (Z_p + 2f_o)$ M
$d_{p1} = (30 + 2)\ 5$
$d_{p1} = 160$ mm
Root diameter:
$d_{af} = (Z_a - 2f_o)\ M - 2C$
$d_{af} = (70 - 2)\ 5 - 2 \times 1.25$
$d_{af} = 337.5$ mm
$d_{pf} = (Z_p + 2f_o)\ M - 2C$
$d_{pf} = (30 + 2)\ 5 - 2 \times 1.25$
$d_{pf} = 137.5$ mm

XI. DESIGN OF SUN GEAR

By given the following data:
Blade length, $l = 2$ m
Wind speed, $v = 4$ m/sec
Power, P = 197.839 W
Input speed in rpm, $N_p = 44.54$ rpm
Number of teeth in $Z_p = 30$
Number of teeth in $Z_s = 10$
$_{Zp} = i \times _{Zp}$ (from DD book Page No: 8.22)
i = 0.33

— = —

From this we have calculated the value of
Output speed in rpm, $N_s = 133.63$ rpm
Calculation of Form factor
when $Z_p = 30$
$Y_p = 0.358$ 20^0 FD
(from DD book Page No: 8.53 Table 40)
$y_p = Y_p /$
Take $\sigma_p = 196$ N/mm^2
(from DDT's book Page No: 5.31 Table 5.4)
$y_p = (196 \times 0.358) / \pi$
$\sigma_p \times y_p = 22.33$
Calculation of Form factor
when $Z_s = 10$
$Y_a = 0.201$ 20^0 FD
(from DD book Page No: 8.53 Table 40)
$y_a = Y_a /$
Take $\sigma_a = 140$ N/mm^2
(from DDT's book Page No: 5.31 Table 5.4)
$y_a = (140 \times 0.201) / \pi$
$\sigma_a \times y_a = 8.95$
Since the value of ($\sigma_p \times y_p$)is greater than ($\sigma_a \times y_a$)
So the sun gear is weaker than the planetary gear
Selection of Material
C55 Cr75Carbon steel (wear resisting, hardened and tempered)
(from DD book Page No: 1.12)
Calculation of Module (M)
Since the center distance (a) is give as a = 250 mm.
we need not to equate F_s and F_d.
$a = M (Z_p + Z_s) / 2$
250 = M ——
M = 12
To check the recommended series of module in (DD book Page No: 8.2 Table 1)

Calculation of b,d and V
Face width (b) = 10 x M
= 10 x 12
= 120 mm
Pitch circle diameter $d_p = M \times Z_p$
= 12 x 30
= 360 mm
$d_s = M \times Z_s$
$d_p = 12 \times 10$
= 120 mm
Pitch line velocity V =

V =

V = 0.8396 m/s
Calculation of beam strength (F_s)
$F_s = \ \times M \times b \times \sigma_p \times y_p$
$F_s = \ \times 12 \times 120 \times 140 \times$
$F_s = 40521.6$ N
Calculation of dynamic load (F_d)
$F_d = F_t + \frac{(\ \ )}{\sqrt{}}$
$F_t =$
$F_t =$
$F_t = 235.63$ N
C – Deformation factor for 20^0 Full depths (from DDT's book Page No: 8.53 Table 41)
C = 8150e
e – expected errors
e = 0.022
C = 8150 x 0.022
C = 179.3 N/mm
$F_d = 21806.82$ N
Check for beam strength or tooth breakage
We din F_s is greater than F_d it means the gear tooth has adequate beam strength and it will not fail by breakage thus the design is satisfactory.
Calculation of wear load (F_w)
Maximum wear load $F_w = d_a \times b \times q \times k_w$
q =
q = 0.49
k_w = Load stress factor (from DDT's book Page No: 5.36 Table 5.9)
$k_w = 20^0$ FD Cast iron = 1.31 N/mm^2
$F_w = 360 \times 120 \times 0.49 \times 1.31$
$F_w = 27730.08$ N
Check for wear
We find F_w is greater than F_d it means gear tooth has adequate wear capacity and it will not wear out. Therefore the design is satisfactory.
Even the same approach is applied for stage two gear boxes.
Basic dimension of planetary gear
(from DDT's book Page No: 8.22 Table 26)

Module	M = 12 mm
Number of teeth:	$Z_p = 30$
	Zs = 10
Pitch circle diameter:	$d_p = 360$ mm
	$d_s = 120$ mm
Centre distance:	a = 250 mm
Face width:	b = 120 mm
Height factor f_o = 1for 20^0 FD	
Bottom clearance:	C = 0.25 M
	C = 0.25 x 12
	C = 3 mm
Tip diameter:	$d_{p1} = (Z_p + 2f_o)$ M
	$d_{p1} = (30 + 2)$ 12
	$d_{p1} = 384$ mm

$d_{s1} = (Z_s + 2f_o)$ M
$d_{s1} = (10 + 2)$ 12
$d_{s1} = 144$ mm
Root diameter:
$d_{pf} = (Z_p - 2f_o)$ M – 2C
$d_{pf} = (30 - 2)$ 12 – 2 x 3
$d_{pf} = 330$ mm
$d_{sf} = (Z_s - 2f_o)$ M – 2C

$d_{sf} = (10 - 2)\ 12 - 2 \times 3$
$d_{sf} = 90$ mm

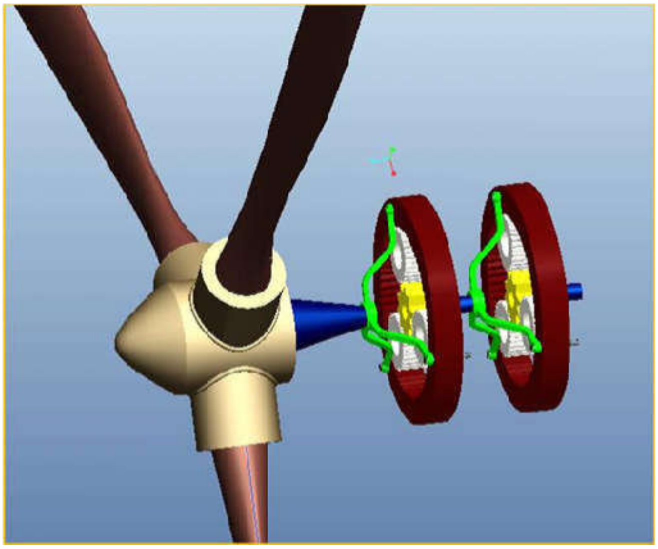

Fig 5. Total Assembly of Wind Inator

CONCLUSION

I have made design and calculation of wind turbine, which is used to produce optimal power in small wind blade. With this help of wind turbine we can able to produce sufficient energy which is helpful for day to day life.
This is a green energy, because there is no emission or any harmful gases are produced. If we employ this type of wind turbine in costal are we get a constant wind speed. People and government have to focus on this type of renewable energy to produce power so that we can eradicate the power cut, which leads to less utilization of fossil fuel.
Due to this size of the wind turbine is small we no need any special equipment's to install such as crane and fork lift. Due to this we can save time and money. When we bring this wind turbine to the real time we can improve the entrepreneurship.

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