

Gearbox Designing System of Dual Rotor Wind Turbine (DRWT) - a Technology of Future

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1. INTRODUCTION

A wind resource is one of the clean energy resources and has vast potential that covers more than two hundred times of annual world energy consumption [4]. It is very important to use this wind resource as an energy source for a reduction of fossil fuel dependency and sustainable development. To use the wind resource, a wind turbine that converts wind energy to mechanical energy through rotation of a rotor is used. A worldwide installed capacity of wind turbines has shown a high growth rate because power generation of a wind turbine has lower cost of generation and higher technology maturation than that of other renewable energy resources [3]. In order to reduce cost of wind energy further and maintain continued growth of wind power, it is required to improve the energy conversion efficiency of a wind turbine. The energy conversion efficiency of a wind turbine is usually characterized by its power coefficient that is the ratio of the power extraction from a wind to the power available in the wind. Based on the classical momentum theory, the maximum power coefficient of a wind turbine having an ideal single rotor without any losses is about 59% that is known as the Betz limit [5]. In practice, it is found that the maximum power coefficient of conventional horizontal axis wind turbines having a single rotor is about 40 to 50% due to some losses such as viscous loss, three dimensional loss, and transmission loss. Over the past few decades, many different concepts and blade designs of a wind turbine have been proposed to improve the maximum power coefficient. A Dual-rotating wind turbine having two rotors rotating in same or in opposite direction at the same axis has been proposed as a new concept to enhance the maximum power coefficient of a wind turbine. Using the classical momentum theory, Newman found that the maximum power coefficient of a wind turbine having two rotors without any losses was increased to about 64% [3]. Recently, based on this result, many researches for a counter-rotating wind turbine have been carried out to obtain more power from a wind than a conventional wind turbine having a single rotor.

The majority of wind turbines currently in operation have the conventional concept design. That is a single-rotor wind turbine system which is connected through transmission system (gearbox) to a generator. Recently, the research on dual-rotor wind turbine is undergoing in several companies and individual researchers have been introduced to the market. It has been proven that the steady state performance of the dual-rotor wind turbine system for extracting energy is better than the single-rotor wind turbine system [1]. The counter-rotating wind turbine has two rotors rotating in opposite or in same directions on the same axis. It has been proposed on the basis of the theory which states that a configuration of two rotors having the same swept area on the same axis has higher maximum power coefficient than a conventional configuration of a wind turbine having a single rotor. In this paper author has designed a gearbox of dual rotor wind turbine system. This gearbox composed of both compound and planetary gear train. The wind speed rotates the rotor of wind turbine 60-70 RPM in case of large wind turbines and 100-300 in small wind turbines. The large wind turbine requires efficient gearboxes to convert small rotational speed of 70 RPM to the high rotational speed of 1600-1800 RPM. The Author had designed a gearbox on CATIA which has two input shafts and one output shaft. The dual rotors in front and rear side capture wind energy. The captured wind energy is transformed into high speed rotational energy by transmission system. In this Wind turbine the radius of the wind blades for the dual rotor is taken as 1.5m. The rated speed for the wind turbine is taken 10m/s. Cut in speed is about 2.5m/s and Cut out speed is taken 25m/s. Lot of research has been done in application of gearless drive in dual rotor wind turbine but few researches has been done in application of mechanical gearbox in dual rotor wind turbine.

2. LITERATURE REVIEW

In order to increase the power efficiency of wind turbines on Counter Rotating Wind Turbine research have been carried out by many investigations and also comparison of power output in Counter Rotating Wind Turbine with that of Single rotor wind turbines was reported.

Jung[7] has obtained power curve experimentally and numerically for a 30 kW Counter Rotating Wind Turbine system and also the effects of distance and diameter ratio between two rotors by using Blade Element Momentum (BEM) theory.

Appa Energy Systems[1] has measured the rotor performance and numerical predictions using BEM theory for a prototype of 6 kW counter Rotating Wind Turbine. The field by Appa Technology tests conducted in this study demonstrated that power conversion efficiency could be increased by 25 to 40 % by means of a contra rotating rotor system. At low rotor speeds the net power coefficient is seen exceed Betz limit of 59%. This might be possible since the two rotors are in different planes having velocity compounding

characteristics. Possibly the interference effect between two slowly moving rotors seems to be minimal. Therefore there is a need to revisit this test using grid-connected models. Moreover, the buffeting phenomenon that is believed to be resulting from the interaction of the dual rotors was not observed in these tests. The second observation suggests that the contra rotation of two rotors appears to benefit large-scale wind turbines that operate at 15 to 20 rpm.

J.D. Booker[4] has designed A compact, high efficiency contra-rotating generator suitable for wind turbines in the urban environment design, development and performance testing of a permanent magnet generator for wind turbine applications in urban areas. The radially interacting armature windings and magnet array are carried on direct drive, contra-rotating rotors, resulting in a high torque density and efficiency. This topology also provides improved physical and mechanical characteristics such as compactness, low starting torque, elimination of gearboxes, low maintenance, low noise and vibration, and the potential for modular construction. The design brief required a 50 kW continuous rated prototype generator, with a relative speed at the air-gap of 500 rpm. A test rig has been instrumented to give measurements of the mechanical input (torque and speed) and electrical output (voltage, current and power) of the generator, as well as temperature readings from inside the generator using a wireless telemetry device. Peak power output was found to be 48 kW at a contra-rotating speed of 500 rpm, close to the design target, with an efficiency of 94%. It is anticipated that the generator will find application in a wide range of wind turbine designs suited to the urban environment, e.g. types sited on the top of buildings, as there is growing interest in providing quiet, low cost, clean electricity at point of use[6]. Seungmin Lee has calculated the Effects of design parameters on aerodynamic performance of a counter-rotating wind turbine In this study, a modified blade element momentum theory for the counter-rotating wind turbine is developed to investigate the effects of these design parameters such as the combinations of the pitch angles, rotating speeds and rotors' radii on the aerodynamic performance of the counter-rotating wind turbine[7]. Azhumakan Zhamalovich Zhamalov has designed Simulation Model of Two-Rotor Wind Turbine with Counter-Rotation. [2]

3. MATERIAL SELECTION

The gear material should have the following properties:

- Good manufacturability
- High tensile strength to prevent failure against static loads
- High endurance strength to withstand dynamic loads
- Low coefficient of friction

The material used for the manufacture of gears depends upon the strength and service conditions like wear, noise etc. The gears may be manufactured from metallic or non-metallic materials. The metallic gears with cut teeth are commercially obtainable in cast iron, steel and bronze. The non-metallic materials like wood, rawhide, compressed paper and synthetic resins like nylon are used for gears, especially for reducing noise. The cast iron is widely used for the manufacture of gears due to its good wearing properties, excellent machinability and ease of producing complicated shapes by casting method. The cast iron gears with cut teeth may be employed, where smooth action is not important. The steel is used for high strength gears and steel may be plain carbon steel or alloy steel. The steel gears are usually heat treated in order to combine properly the toughness and tooth hardness. We are using C60 plain carbon steel having tensile strength 600-700N/mm² [8]

4. METHODOLOGY

The Designing of gearbox is done on the powerful CAD software CATIA V-5R17.

For designing a gearbox first we have to determine its design considerations which are as follows:-

In the design of a gear drive, the following data is usually given [8]:

1. The power to be transmitted.
2. The speed of the driving gear,
3. The speed of the driven gear or the velocity ratio

Total Power of the wind on our front blades of Dual Rotor wind turbine system:-

$$P = \frac{1}{2} \rho A V^3 \dots \dots \dots (1)$$

Where ρ is density of air taken 1.25kg/m³

V is Rated wind velocity taken 10m/s

And A is Swept area of wind turbine rotors.

In this wind turbine we had taken the blade length of 1.5m

Hence the swept area of the wind turbine rotor is $A = \pi R^2$

On putting all these values in equation 1 we will get Total Wind power is

$$P = \frac{1}{2} * 1.2 * \pi * 1.5^2 * 10^3$$

$$P=4309.65W$$

Similarly we will calculate wind power in rear rotor also; however the wind speed on the rear rotor decreases due to obstruction of front wind blades. It has been found that maximum wind speed at the front rotor is $2/3^{rd}$ of the free stream hence only $1/3^{rd}$ of the wind speed will reach at the rear rotor. Hence the size of the rotor at the rear side is taken of larger size than of front one. In our rotor we had taken the blade size of the rear side equal to 3.5m.

Torque available at the gear,

$$T = \frac{P \cdot 60}{2 \pi N}$$

For our wind turbine system

Assuming tip speed ratio (λ) = 3

λ = speed of rotor/speed of wind

$$V = \omega \cdot R$$

$$V = (2 \times \pi \times N \times R) / 60$$

$$V = (2 \times 3.14 \times N \times 1.5) / 60$$

$$10 \cdot 60 = 8.164N$$

$$N = 63.69 \approx 65 \text{rpm}$$

This is rotational wind speed, Rotor speed will be

Tip speed ratio (λ) = speed of rotor/speed of wind

$$3 = \text{speed of rotor} / 65$$

$$N = 195 \text{ RPM}$$

Hence torque will be,

$$T = (4309.65 \cdot 60) / (2 \cdot \pi \cdot 195)$$

$$T = 211.15 \text{ N-m}$$

By the help of torque, power and gear ratio and by using the following formula's we can determine the design values of different gears, shafts and keys of our gearbox.

4.1 DESIGNING OF GEARS

In this gearbox design the assembly of sun and planet gears are bit different from other epicyclic train. The power from wind energy which is converted to rotational energy by rotor blades is transferred to the transmission system or its called gearbox system. The input wind energy from both front and rear side rotor in the form of torque transferred into the sun gear, the sun gears are in the direct mesh with planet gears. The torque from both sun gears is then transferred to crown wheel. The crown wheel is in direct mesh with output pinion gear. We had taken the gear ratio of crown wheel and output pinion gear as 1:4. From output pinion gear it goes to input spur gear.

4.2 DESIGNING OF THE SUN GEARS, PLANET GEARS AND BEVEL GEARS.

For designing any gear drive, first we have to assume the gear ratio. Let the gear ratio be G.

Minimum number of teeth to avoid interference

$$T_p = \frac{2A_w}{G \left[\sqrt{1 + \frac{1}{G} \left(2 + \frac{1}{G} \right) \sin^2 \delta} - 1 \right]}$$

T_p = Number of teeth on pinion

A_w = Fraction by which the standard addendum for the wheel should be multiplied,

G = Gear ratio of pinion and gear,

δ = Pressure angle of the gear,

we will take pressure angle 20° full depth involute system.

This will give us minimum number of teeth on pinion, and by using gear ratio problem we can calculate number of teeth on gear drive also as follows:-

$$\frac{T_g}{T_p} = G$$

Now we determine the pitch angle (Θ_{p1}) for the pinion and gear (Θ_{p2}).

$$\Theta_{p1} = \tan^{-1} \left(\frac{1}{v.r} \right) \dots \dots \dots (5)$$

$$\text{And } \Theta_{p2} = (90 - \Theta_{p1})$$

The Tangential load on the pinion is to be calculated, so that we can apply Lewis equation to determine the module.

Hence Tangential load on pinion is given by

$$F_T = 2T/D_p$$

Where T= Torque and D_p= pitch circle diameter of the pinion

We Knows that,

$$D_p = m.T_p$$

Here m is module of the gears,

The Length of the pitch cone element (L)of a bevel gear is calculated as follows,

$$L = \frac{D_p}{2 \sin \theta_{p1}}$$

The face width for the gear can be assumed between L/3 to L/4

In this gear we has assumed the face width (b)

$$b = L/3$$

Putting all these values in Lewis equation as follows,

$$F_T = (\sigma_{op} \times C_v) b \times \pi \times m \times y_p \times \frac{(L-b)}{L}$$

We will get the cubic equation in the form of m and by solving this equation we will get value of 'm'.The value of 'm' is used to calculate other dimensions of a gear. We had used above given formulae's to design the bevel gears, sun gear, planet gear, crown wheel and output pinion gear.

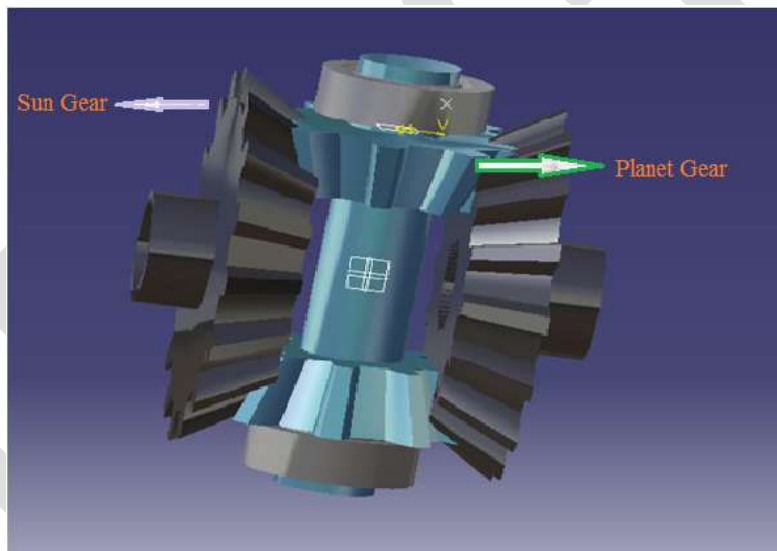


Figure 7 Sun & Planet Gears Designed on CATIA

4.2.1 DESIGNING OF SPUR GEARS

The Torque from output pinion gear is transmitted to input spur gear. The value of torque is calculated as follows.

$$\frac{\text{Torque on output pinion gear}(T)_1}{\text{Torque on input spur gear}(T)_2} = G$$

We know the values of torque on Output pinion gear; we can calculate torque on input spur gear by the help of gear ratio (G). After finding the torque we will apply the same procedure on spur gear calculation as in the case of bevel gear and find the dimensions of spur gears.

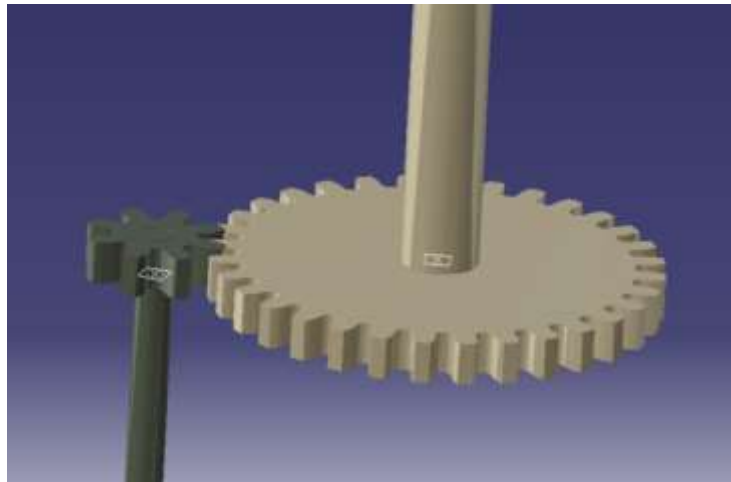


Figure 2 Spur Gear Drive on CATIA software

4.2.2 DESIGNING OF GEAR SHAFTS AND KEY SPLINES

A shaft is a rotating machine element which is used to transmit power from one place to another. The power is delivered to the shaft by some tangential force and the resultant torque (or twisting moment) set up within the shaft permits the power to be transferred to various machines linked up to the shaft. In order to transfer the power from one shaft to another, the various members such as pulleys, gears etc., are mounted on it. These members along with the forces exerted upon them causes the shaft to bending. In other words, we may say that a shaft is used for the transmission of torque and bending moment.

The shafts may be designed on the basis of

1. Strength, and
2. Stiffness & Rigidity

In designing shafts on the basis of strength, the following cases may be considered:

- (a) Shafts subjected to twisting moment or torque only,
- (b) Shafts subjected to bending moment only.
- (c) Shafts subjected to combined twisting and bending moments, and
- (d) Shafts subjected to axial loads in addition to combined torsional and bending loads.

In order to find the diameter of shaft for gears, the following procedure may be followed:-

1. First of all, find the normal load (F_N), acting between the tooth surfaces. It is given by

$$F_N = F_T / \cos \delta$$

2. The weight of the gear is given is calculated by

$$F_G = 0.00118 * T_G * b * m^2 \text{ (in N)}$$

3. Now the resultant load acting on the gear is calculated

$$F_R = \sqrt{F_N^2 + F_G^2 + 2F_N F_G \cos \delta}$$

4. If the gear is overhung on the shaft, then bending moment on the shaft due to the resultant load,

$$M = F_R \times X$$

Where X = Overhang i.e. the distance between the centre of gear and the centre of bearing.

5. Since the shaft is under the combined effect of torsion and bending, therefore we shall determine the equivalent torque. We know that equivalent torque,

$$T_e = \sqrt{(M)^2 + (T)^2}$$

6. Now the diameter of the gear shaft (d) is determined by using the following relation,

We also know that equivalent twisting moment (T_E)

$$T_E = (\pi/16) \times \tau \times d^3$$

τ = Shear stress of the shaft

We have used above Equations to calculate the diameter of Gear Shafts.

4.2.3 SPLINES

Sometimes, keys are made integral with the shaft which fits in the keyways broached in the hub. Such shafts are known as splined shafts as shown in Fig. These shafts usually have four, six, ten or sixteen splines. The splined shafts are relatively stronger than shafts having a single keyway.

$$d = 1.25D \text{ and } b = .25d$$

Where D is diameter of the shaft

d = depth of the spline slots

b = width of the spline slots

The splined shafts are used when the force to be transmitted is large in proportion to the size of the shaft as in automobile transmission and sliding gear transmissions.

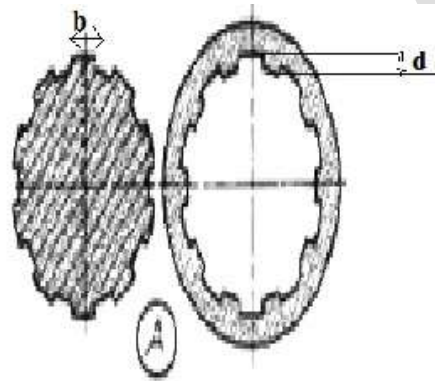


Figure 8 Shaft Spline

By using splined shafts, we obtain axial movement as well as positive drive is obtained.

4.2.4 BEARING DESIGN FOR GEARBOX

A bearing is a machine element which supports another moving machine element (known as journal). It permits a relative motion between the contact surfaces of the members, while carrying the load. Bearings in wind turbines operate at the extremes of operational environments in terms of temperature, load fluctuation, maintenance access and lubricant optimization. As rotor diameters increase, confidence in your bearings becomes even more critical.

Generally For wind turbine Gearbox spherical roller bearing for large wind turbine are used but for small gearboxes Ball bearing is used.

We are using radial ball bearings for our Gearbox.

In this Gearbox we have designed a radial ball bearing.

For Designing the Bearing we need basic dynamic load rating(C) & Diameter of Shaft.

Basic dynamic load rating(C)

$$C = W \times \left(\frac{L}{10^6} \right)^{1/6}$$

L = Life of bearing, in hours, Assuming of bearing life of wind turbine gearbox 100000 Hours.

Where W is the dynamic equivalent radial load, which is determined by radial load (F_R) and constant axial or thrust load (F_A)

$$W = X \cdot V \cdot F_R + Y \cdot F_A$$

Where V = A rotation factor,

F_R = Radial load,

F_A = Axial or thrust load,

X = radial load factor

Radial Load (F_A) and Axial Load (F_R) is Determined by following Formulae

$$F_A = F_T \tan \alpha \sin \delta$$

$$F_R = F_T \tan \alpha \cos \delta$$

α = Nominal angle of contact

δ = Pressure angle of gear

By putting all values in equation (14) we get dynamic load rating.

Hence we get the value of C and for the diameter of shaft of diameter d we can use standard value of Ball Bearing no. 6407 from PSG Design Data book.[12]

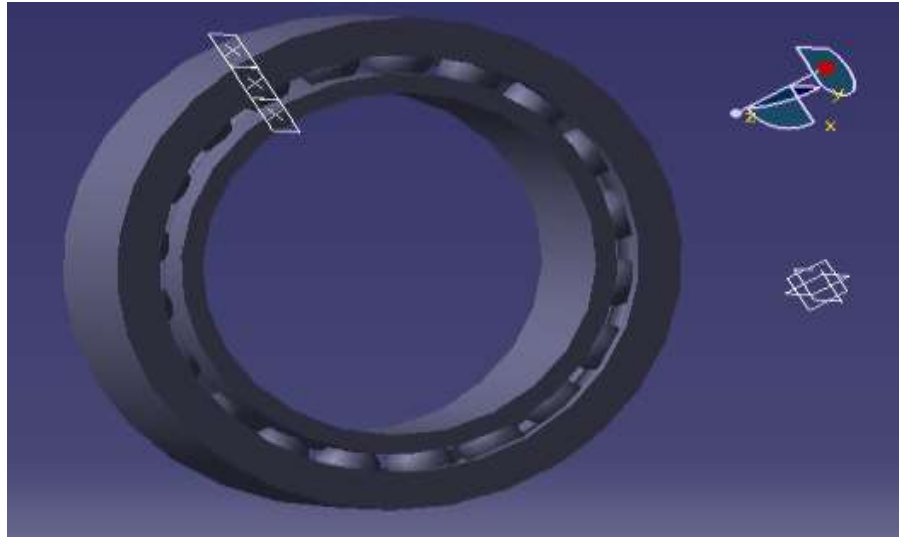


Figure 3 CATIA Design of Radial Ball Bearing

5. RESULT & DISCUSSION

The Design of the Dual Rotating Wind Turbine for is designed by using the CATIA V5 R17 Software. To Design the Gearbox of 1.5kW we have first calculated the Torque acting on the Gears. The author has assumed the value of blade length 1.5 m length, and Rated wind speed of RGPV Bhopal is 10m/s. These values are the base for designing of the Gearbox. The tables showing the design value of Gears such as crown wheel gear, output pinion gear, sun gear, planet gear, spur gears shafts and keys are shown in the Following Tables.

Table 2 Design perimeters for Spur Gears

s.no	Particulars	Standard proportion	Gear dimension (in mm)
1	Diameter of output spur gear	$m \cdot T_P$	49
2	Diameter input Spur gear	$m \cdot T_G$	245
3	Addendum	1m	$1 \times 3.5 = 3.5$
4	Dedendum	1.25 m	$1.25 \times 3.5 = 4.375$
5	Working depth	2 m	$2 \times 3.5 = 7$
6	Minimum total depth	2.25 m	$2.25 \times 3.5 = 7.875$
7	Tooth thickness	1.5708 m	$1.5708 \times 3.5 = 5.5\text{mm}$
8	Minimum clearance	0.25 m	$.25 \times 3.5 = .875\text{mm}$

Table 4 Design perimetres of Crown wheel Gear and output Pinion Gear.

S.N	Particulars	Standard proportions	Crown wheel Gear	Output Pinion Gear
1	Module	M	2mm	2mm
2	Outside diameter	$D_{OG} = D_G + 2 a \cos \Theta_p$	130	35mm
3	Pitch circle diameter	$D_G = m.T_G$	128	30
4	Number of teeth	T_G & T_P	64	16
5	Length of the cone (L)	$L = \frac{D_P}{2 \sin \theta_{P1}}$	66	66
6	Face width(b)	$b = L/3$	22	22
7	Addendum (a)	$a = 1m$	2	2
8	Dedendum(d)	$d = 1.2m$	2.4mm	2.4mm

Table 4 Design perimetres of sun and planet gear

S.no	Particulars	Standard proportions	Sun Gear	Planet gear
1	Module	m	2.5mm	2.5mm
2	Outside diameter	$D_{OG} = D_G + 2 a \cos \Theta_p$	52mm	35mm
3	Pitch circle diameter	$D_G = m.T_G$	45mm	30
4	Number of teeth	T_G & T_P	18	12
5	Length of the cone(L)	$L = \frac{D_P}{2 \sin \theta_{P1}}$	30mm	27mm
6	Face width(b)	$b = L/3$	10mm	10mm
7	Addendum (a)	$a = 1m$	2.5mm	2.5
8	Dedendum(d)	$d = 1.2m$	3.75mm	3.75mm

The Efficiency of Dual rotor wind turbine could be increased up to 40% than the Efficiency of a wind turbine with only one rotor. This increase in result will increased the power generation from a given tower installation. Energy conversion efficiency was high at low rotor speeds, suggesting application of Dual rotor at places where wind speed is low 4-5m/s.

6. AUTHORS' INFORMATION

The Author Mukesh Pandey is the Head of the Department of School of Energy and environmental management at Rajiv Gandhi Technical University (also known as RGPV). he has about 13 years of experience 14 years in academic and 4 years in industrial area. he had published 27 international journals and 58 national journals. he is also the writer of 6 books. author anurag gour is assistant professor in department of energy and environmental management in school of energy technology at Rajiv Gandhi Technical University. He has published 19 research paper in international journals and attended 2 international and national conferences. author Tipu sultan is research scholar in department school of energy technology at Rajiv Gandhi Technical University. He has completed his Graduation in Mechanical Engineering.

7. CONCLUSION

In this Research paper we have studied the designing of the Gearbox of dual rotor wind turbine system. We have used CATIA software for the designing of the gearbox, more accurate calculation for the dimensions of gear is done by basic formulae's. We assumed that the speed of wind in front rotors is reduced by 2/3rd times the original wind speed. Basic formulae's for calculating gear dimensions are discussed and tables of gear dimensions are drawn in this paper. This gearbox has two input shafts and one output

shaft. The dual rotors in front and rear side capture wind energy. The captured wind energy is transformed into high speed rotational energy by transmission system. In case of single rotor only single front rotor can convert wind energy into useful energy but in this research paper had designed a dual rotor wind turbine gearbox on CATIA which can generate about 5% more power than conventional wind turbine system.

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