Analysis of Eclipse Gearbox for Wind Turbine

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Abstract- *A wind energy translation system consists of a number of components to transform the wind energy to electrical energy. The rotor is one of the component of wind turbine that excerpts energy from the wind. One of the major component, Gearbox is used for transmission high torque generated by rotor to low torque required for generator. Gearbox Un-reliability and high repair costs pool to result in critical negative effects on the cost of wind energy production. The Eclipse Gearbox is suggested in this paper that can significantly decrease reliability problems occurred in traditional gearbox. The features of Eclipse gearbox is a shortened load path through a single pair of gears combined with linkages and a crankshaft. Multi-stage planetary system of traditional gearbox is reduced to single stage eclipse gearbox, helps to increase speed ratio, long endurance life, small size and light weight. Its size is like to a traditional gearbox weight reduced to half. Contact stress of gear tooth is considerably lower due the increase in the number of gear teeth that are simultaneously engaged. The minimum tooth contact stress finally increases the endurance life and torque capacity of the gears.*

Keywords- Eclipse gearbox, Reliability, Speed ratio, Traditional gearbox, Wind turbine

I. INTRODUCTION

A wind energy translation system consists of a number of components to transform the wind energy to electrical energy. A wind turbine working regime is divided into three areas.

Area 1(wind speed up to 4m/s) is the low wind speed area for which the turbine does not yield any power, the turbine is disconnected from the grid. If the turbine will be connected to the grid at low wind speeds, the generator will start employed as a motor.

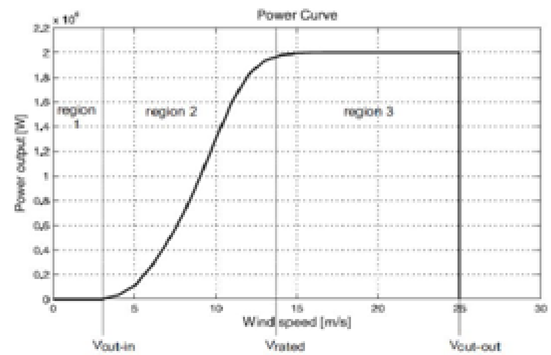

Fig. 1 Power o/p Vs Wind speed

The area 2(wind speed 4 to 14m/s), is the area in which turbine starts to operate (Vw;cut:in) and the wind speed at which extreme power is produced (Vw;rated). This is the area for which growing energy capture, but restriction of dynamic loads also becomes more important. In area 2 operation explanations for added than 50% of the annual energy capture. This designates the importance of efficient operation in this system. In area 3(wind speed 14 to 25m/s), which is the area from the rated wind speed to the wind speed at which the turbine is stopped to avoid damage (Vw;cut:out). In area 3, energy capture is limited such that the turbine and generator are not overloaded and dynamic loads do not result in mechanical failure. The limitation in energy capture is generally controlled by pitching the rotor blades, by appropriate methods. Because of blade pitching, less energy is extracted from the wind results in declining the efficiency.

II. PRESENT THEORIES AND PRACTICES

Power Electronics

The generator results in the creation of current with a mutable frequency. The frequency of the produced current is noted by the electrical angular speed of the generator. The frequency and phase of all power producing units must persist synchronous within constricted limits. If the frequency of the generator varies too much (2 Hz), circuit breakers cause the generator to separate from the system, avoid damage to the grid. Small deviances in the generator frequency can indicate instability in the grid.

Power electronics is a technology that is develop fast. High current and voltage ratings are available, efficiency maximizes and costs reduces. Therefore, power converters are largely used in the wind turbine industry to increase the concert of wind turbines. However, there are lots of disadvantages of using power electronics.

Disadvantages of Power Electronics

The largest drawback of power electronics is reliability. Mechanical components expose wear & tear and therefore any disasters in these components can be predict, maintenance can be scheduled before failure occurs. Power electronics do not show symbols of degrading, hence failures cannot be predict and these sudden failures are very costly to repair. Combine with high failure costs, power electronics tend to fail quite rapidly because they are very subtle to voltage spikes . In the wind energy industry about 25% of all failures is due to the power electronics. Traditional gearbox failures present major issues in the wind energy industry. Gearbox Un-reliability and high repair costscombine to result in critical negative effects on the cost of wind energy production

Lost revenues result from

- High down-times when energy cannot be formed,
- The considerable expense of the large crane needed to lift a replacement gearbox into place

Eclipse Gearbox Introduction

The gearbox is the dangerous component disposed to failure in the load path between the turbine and the generator. Traditional wind turbine gearboxes alter an indirect path through a multi-stage planetary system. Introduced here is a gearbox that structures a shortened load path through a single pair of gears collective with linkages and a crankshaft.

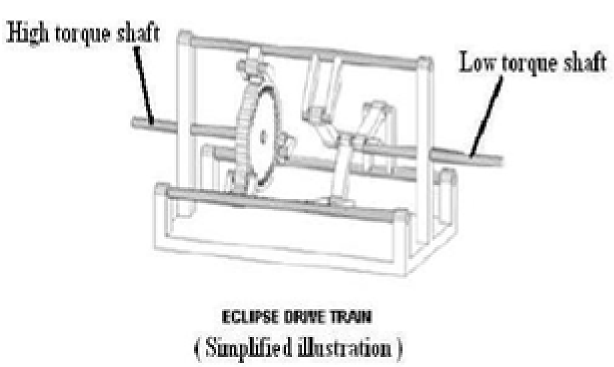

Fig 2 Eclipse Gearbox

The Eclipse Gearbox overcomes the restrictions of the planetary gear set and offers a practical, high-reliability gearbox. A basic version of the Eclipse Gearbox is illustrated in figure above. One gear revolves and delivers a circular path for another gear. The second gear oscillates on a circular path, gear is linked with linkages to the output crankshaft. The load path gets with the high torque shaft and ends with the low torque shaft[2].

Functionality and operation of the eclipse drive train

The crankshaft and a least of three linkages are required to control the translational motion of the translational gear. Additional linkages are used to distribute the translational gear reaction loads. The Eclipse accommodates speed ratios up to 150 to 1 in a single stage. This speed ratio is based on the practical limit to the gear tooth size.

Speed ratio = $-N_s / N_T - N_s$ to 1

Where N_S is the number of teeth on the spur gear and
N_T is the number of teeth of the translating gear.

The endurance life and power rating of the Eclipse Drive Train are dependent on the number of linkages and the sizing of the bearings and gears. In relating, for traditional gearboxes to be sized for effective operation in great power wind turbines, their cost, weight and size would be defensive. The link load cycle for a 1.6 MW gearbox is illustrated to show the distributed load through dissimilar linkagesrepresenting an input torque of 600,000 lb-ft. The addition of the linkage loads are equal to 75 percent of the bearing forces in the planetary gears of a traditional planetary gear set.

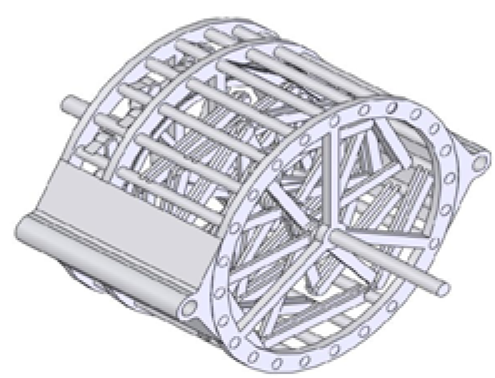

Fig. 3 1.6 MW Eclipse Gearbox

The linkages are designed with respect to production tolerances, joint free play and stiffness to maintain equally distributed linkage loads throughout the Eclipse system, unrelatedly of the loads applied to the windmill blades. The linkages act in parallel to allocate the translational gear loads. The gear loads are distributed over multiple bearings. The bearings in the linkages rotate back and forth about 15 degrees. The high and low torque shafts rotate a complete 360 degrees. The gear tooth stresses are significantly reduced due to the loads being distributed over a greater number of teeth. The lower gear tooth stresses significantly increase the fatigue life of the gears. The mechanical design efficiency of the Eclipse Drive train results in significantly greater efficiency than traditional planetary gearboxes, due to the reduced number of energy dissipating components and to the fact that energy travels though only one set of gears and bearings[1].

III. METHODOLOGY

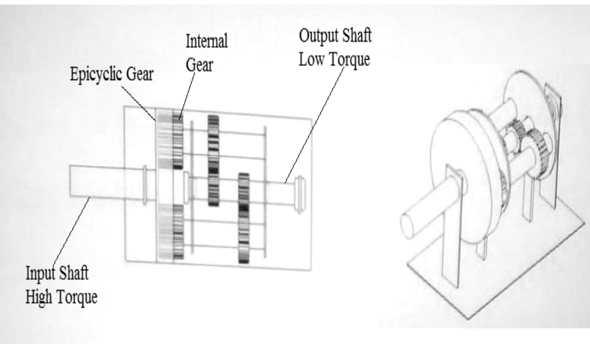

FIG. 4 LAYOUT OF ECLIPSE GEAR BOX

Epicyclic gear is linked to the input shaft (high torque). Two internal are connected to the epicyclic gear through two linkages and linkages are connected to output shaft (low torque) through gears. Motion carried by epicyclic to internal gear in 360 degree rotation of input shaft (by one pinion) is only during forward state due to one way clutch. During 0 degree -180 degree one pinion in onward transmission is continuous.

Output is mainly depends on : Number of linkages, Linkages dimensions, Gear ratio of epicyclic gear and internal gear, at the same time other pinion will be in reverse state, during next phase of 180 degree -360 degree condition reverse so motion gear.

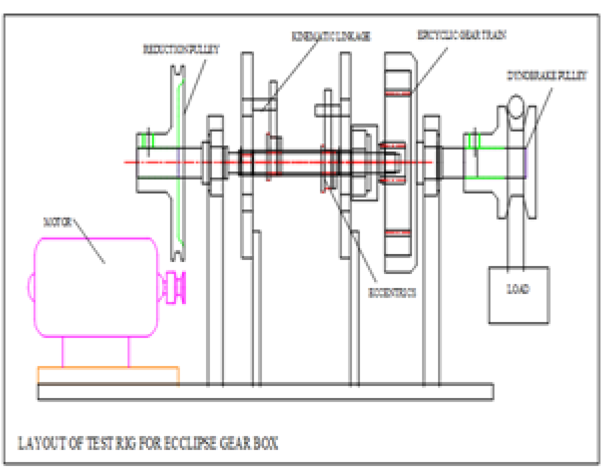

Fig. 5 Layout of Test Rig for Eclipse Gear Box

A standard internal gear and pinion are engaged without tooth interference . On the driving shaft A is attached an eccentric, the axis of the driving gear follows the motion of eccentric, but is kept from rotate about its own axis by pin, which works in the slot. Linkage is actuated by the eccentric, which continuously maintains slot in an perpendicular position through the action of parallel links, swiveled on studs. Since the axis of gear follows the motion of Eccentric and the gear does not rotate about its own axis , the motion imparted to the driven gear will be uniform. For testing purpose we take low torque shaft as i/p shaft by using motor and belt input motion is given. Two linkages are in motion through gear and epicyclic gear rotates high torque means output shaft at high torque, various loads are applied and change in rpm is noted.
Design and analysis of critical components of assembly namely : Internal Gear ring and External gear

Designand analysis of internal gear and external wobble gear

To Calculate Input Torque

Input data - Motor is an Single phase AC motor , Power 50 watt , Speed is continuously variable from 0 to 6000 rpm. Assuming operation speed = 800 rpm.

$$\text{Power} = \frac{2\Pi N T}{60}$$

$$T = \frac{60 \times P}{2 \times \Pi \times N}$$

$$T = \frac{60 \times 50}{2 \times \Pi \times 6000}$$

T = 0.5968 N.m.

Assuming 100% overload.

T design $= 2 \times T$
$= 2 \times 0.5968 \times 10^3$
$= 1.19 \times 10^3$ N.mm.
T-design = 1.19 N-m

Internal Gear Data :

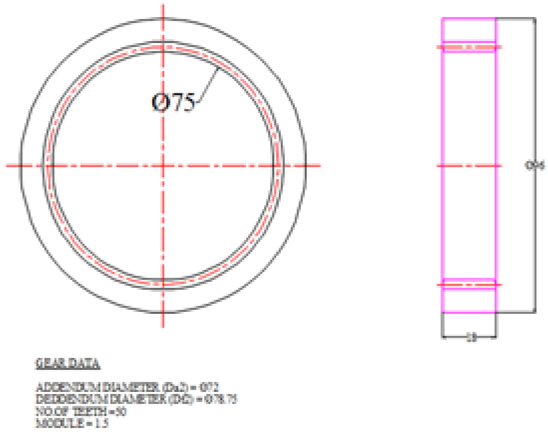

Fig. 6 Internal Gear

Addendum Diameter(Da2) = 96 mm
DeddendumDiameter(Df2) =78.75 mm
No. of Teeth = 50
Module = 1.5

Design of Internal Gear - Theoretical method

TABLE I:MATERIAL SELECTION FOR INTERNAL GEAR

Designation	Ultimate Tensile strength N/mm^2	Yield strength N/mm^2
EN 24	800	680

As Per ASME Code;
$fs_{max} = 108\ N/mm^2$

Check for torsional shear failure:-

$$T = \frac{\Pi fs_{act}}{16} \times \left(\frac{Do^4 - Di^4}{Do} \right)$$

$$1.19 \times 10^3 = \frac{\Pi \times fs_{act}}{16} \times \left(\frac{96^4 - 75^4}{96} \right)$$

$fs_{act} = 0.01\ N/mm^2$

As; $fs_{act} < fs_{all}$

Gear is safe under torsional load

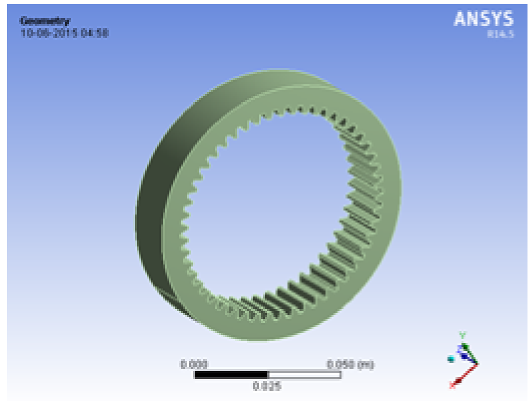

Fig. 7 Geometry of Internal gear

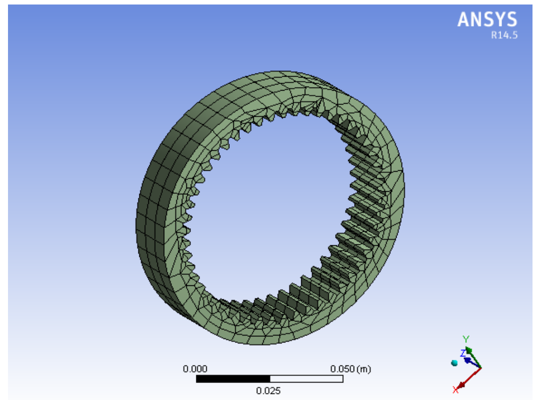

Fig. 8 Meshing of Internal gear

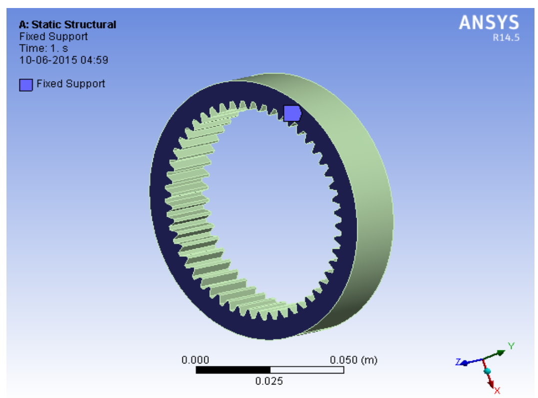

Fig. 9 Boundary Condition of Internal gear

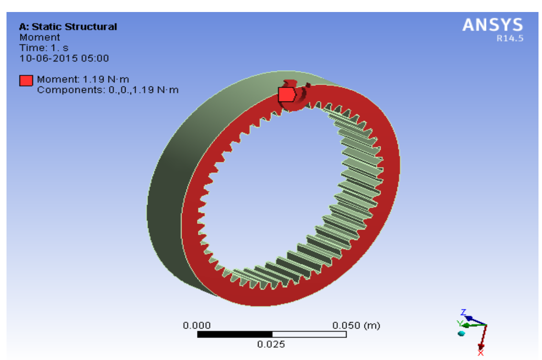

Fig. 10 Moment of Internal gear

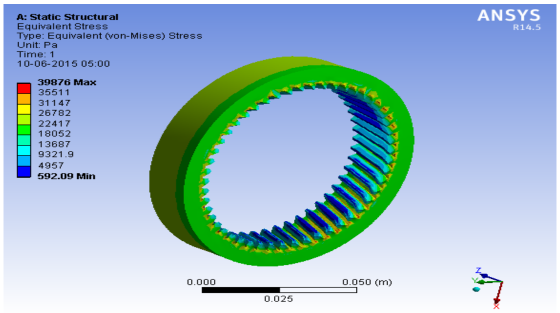

Fig. 11 Equivalent stresses in Internal gear

Maximum stress by analytical methods is well below the allowable limit of 108 N/mm2 hence the internal gear is safe.

External Gear Data -

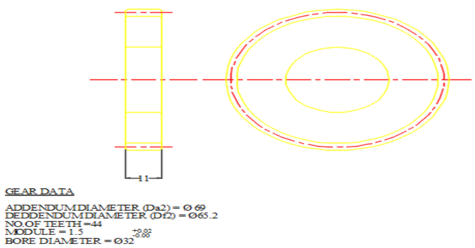

Fig. 12 External Gear

Addendum Diameter(Da2) = 69 mm
DeddendumDiameter(Df2) = 65.2 mm
No. of Teeth = 44
Module = 1.5
Bore Diameter = 32 mm

Design of External gear - Theoretical method

TABLE II MATERIAL SELECTION FOR EXTERNAL GEAR

Designation	Ultimate Tensile strength N/mm²	Yield strength N/mm²
EN 24	800	680

As Per ASME Code;

fs_{max} = 108 N/mm^2

Check for torsional shear failure:-

$$T = \frac{\Pi fs_{act}}{16} \times \left(\frac{Do^4 - Di^4}{Do}\right)$$

$$1.19 \times 10^3 = \frac{\Pi \times fs_{act}}{16} \times \left(\frac{69^4 - 32^4}{69}\right)$$

fs_{act} = 0.02 N/mm^2

As; $fs_{act} < fs_{all}$

Gear is safe under torsional load

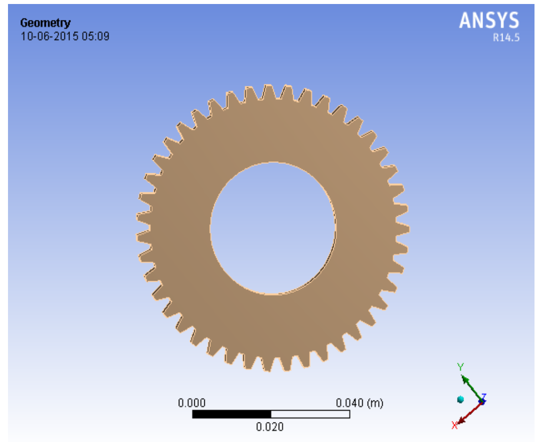

Fig. 13 Geometry of External gear

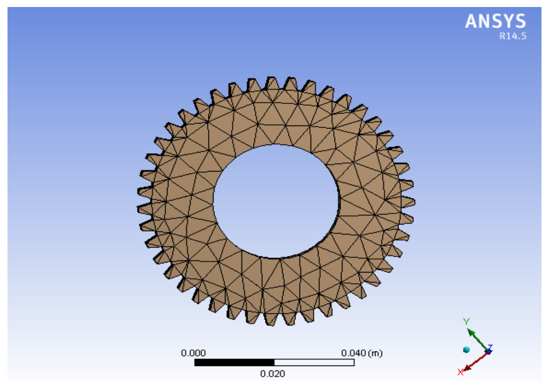

Fig. 14 Meshing of External gear

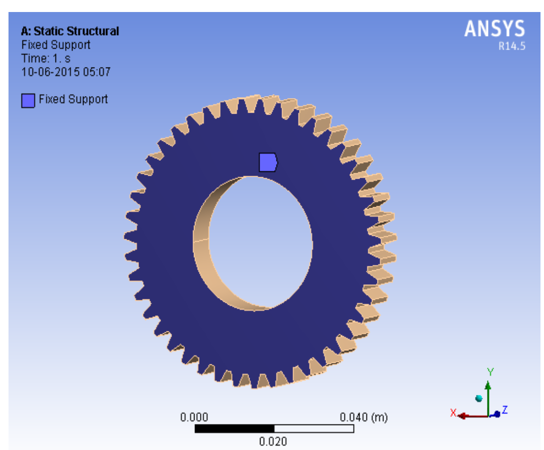

Fig. 15 Boundary Condition of External gear

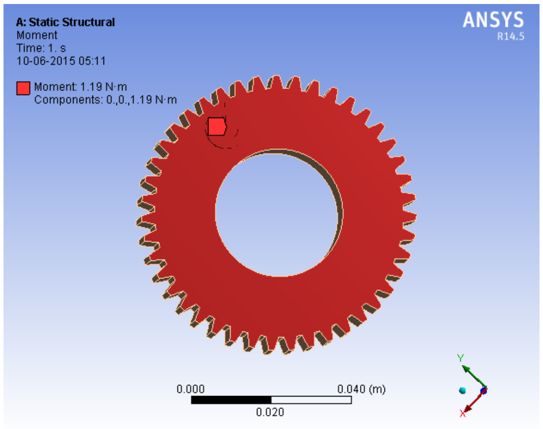

Fig. 16 Moment of External gear

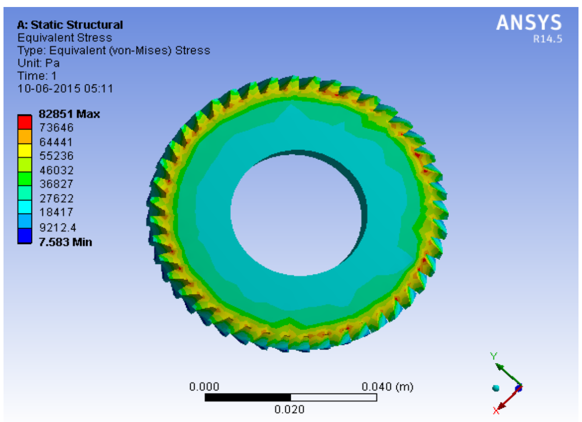

Fig. 17 Equivalent stresses in External gear

Maximum stress by analytical methods is well below the allowable limit of 108 N/mm2 hence the external gear is safe.

Result and Discussion -

TABLE III RESULTS OF INTERNAL GEAR AND EXTERNAL GEAR

Gear type	Maximum stress N/mm^2	Theoretical N/mm^2	Result
Internal gear	0.3896	0.01	Safe
External gear	0.82851	0.02	Safe

IV. CONCLUSIONS

- Determined stress by theoretical and analytical methods are well below the allowable limit of 108 N/mm^2 hence the internal gear is safe
- Determined stress by theoretical and analytical methods are well below the allowable limit of 108 N/mm^2 hence the external gear is safe
- Eclipse gearbox can effecting change the existing one, due to its higher speed ratio, strength and lesser weight.

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