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Analysis of Multistage Linkage Based Eclipse Gearbox for Wind Mill Applications

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Abstract— A wind energy conversion system consists of number of components to transfer the wind energy to electrical energy. Presently wind mills employ epicyclic gearbox for transfer of high torque generated by rotor to low torque required for generator. Gearbox is expensive and critical component of wind turbine. Moreover the high speed ratio and fluctuating loads put limitation on module of gear teeth that lead to failure of conventional gearbox due to breaking of teeth. The three stage overlay kinematic linkage system and epicyclic gearbox combination provides a reliable solution to above problem where in high strength, robustness and friction less performance of kinematic linkage ensure a fail safe operation and high efficiency and maximum life.

Index Terms—**Eclipse gearbox, Reliability of gearbox, Crankshaft, Traditional gearbox, Wind turbine.**

I. INTRODUCTION

A wind energy conversion system consists of a number of components to transform the energy in the wind to electrical energy. One of these components is the rotor, which is the component that extracts energy from the wind. The operating regime of a wind turbine is divided into three regions.

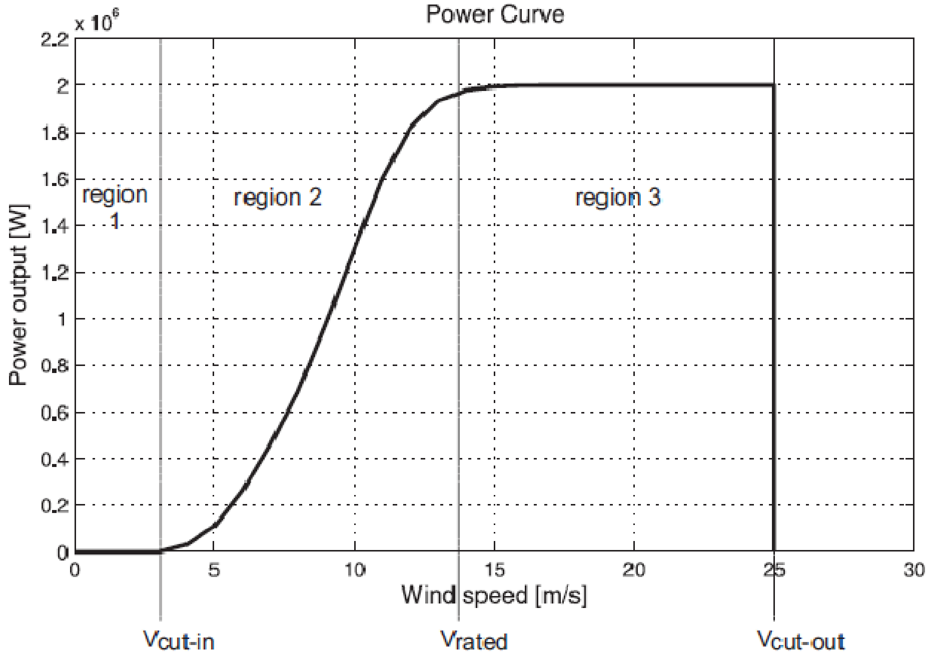

Fig. 1: Power O/p Vs Wind speed

Region 1(wind speed up to (4m/s) is the low wind speed region for which the turbine does not produce any power, the rotor is standing still and the turbine is disconnected from the grid. When the turbine would be connected to the grid at these low wind speeds, the generator would start working as a motor, driving the turbine. The turbine would then actually be working as a huge fan, consuming energy instead of producing. The second region, region 2(wind speed 4 to 14m/s), is the region between the wind speed at which the turbine starts to operate (Vw;cut¡in) and the wind speed at which maximum power is produced (Vw;rated). This is the region for which maximizing energy capture is very important, but limitation of dynamic loads also becomes more important. In a typical wind turbine, region 2 operation accounts for more than 50% of the annual energy capture. This indicates the importance of efficient operation in this regime. Finally there is region 3, which is the region from the rated wind speed to the wind speed at which the turbine is stopped to prevent damage (vw;cut¡out). In this region, 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. [1]

II. Blade Pitch Control

Blade pitch control is used to control the aerodynamic power captured from the wind. By pitching the rotor blades along their longitudinal axis, the aerodynamic efficiency of the rotor is changed. This change is caused by a modification in the aerodynamic angle of attack. The aerodynamic angle of attack is defined as the angle between the chord line of the rotor blade and the direction of the approaching wind, as seen by the rotor blade. When decreasing the angle of attack, called feathering, the lift capacity of the blades is reduced and therefore the power captured by the rotor decreases. Conversely, an increase in the angle of attack, with respect to the operational position, will lead to a higher power capture due to a reduction in drag. When the critical aerodynamic angle of attack is reached, the airflow separates at the surface of the rotor blades, limiting the power. This effect is called stall.

Below rated wind speed, the pitch angle can be controlled to change the tip speed ratio. However, application of blade pitch control to follow the desired tip speed ratio is limited by the rate at which the blades can be pitched and the reaction time of the rotor speed to change. A disadvantage of using blade pitching below rated speed is that less energy is extracted from the wind, decreasing the efficiency. For this reason, blade pitch control is generally not used below the rated wind speed.

Above rated wind speed, blade pitch control is used to limit the angular speed of the rotor by capturing less power from the wind then available and to protect the system from excessive forces. By pitching the blades, the operating range of the turbine in terms of wind speed is increased. Without blade pitch control, the maximum operating speed would be rated wind speed. Above rated, the dynamic loads on the mechanical components would become too high and the angular speed of the rotor would exceed its maximum. With blade pitch control, the operating range is typically increased up to 25 m/s. [2]

III. Methodology

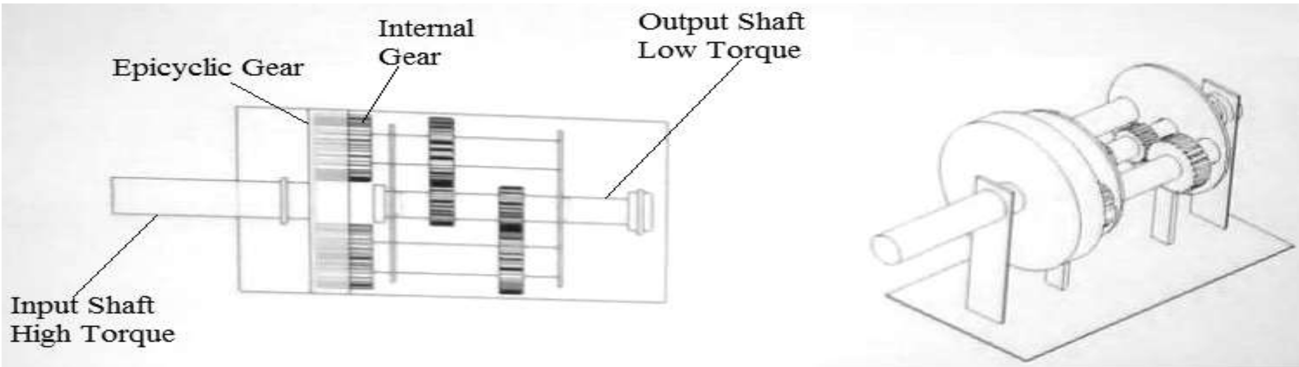

Fig. 2: Layout of Eclipse Gear Box

Epicyclic gear is connected to the input shaft (high torque) and linkages are connected to output shaft (low torque). Motion delivered 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. Output is mainly depends on: Number of linkages, Linkages dimensions, Gear ratio of epicyclic gear and internal gear.

A standard internal gear and pinion are meshed without tooth interference. On the driving shaft A is mounted an eccentric, the axis of the driving gear follows the motion of eccentric, but is kept from revolve about its own axis by pin, which works in the slot. Linkage is actuated by the eccentric, which constantly maintains slot in a perpendicular position through the action of parallel links, pivoted 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.

IV. Design And Analysis Of Various Components Of Eclipse Gearbox

Input Shaft

To Calculate Input Torque,

Input data - Motor is a Single phase AC motor, Power = 60watt, Speed is continuously variable from 0 to 6000 rpm. Considering the operating speed of motor = 600 rpm at the input shaft and assuming 75 % efficiency of belt drive.

Input power = (60 /6000) x (600 x 5) x 0.75 = 22.5 watts

Where, 5= reduction ratio of belt drive

& 0.75 is efficiency.

Motor Torque,

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

$$T = \frac{22.5 \times 60}{2 \Pi \times 600}$$

$$T = 0.35 \text{ N-m}$$

Considering 50 percent overload on system

$T_{design} = 1.5 \times 0.35 = 0.54$ N-m

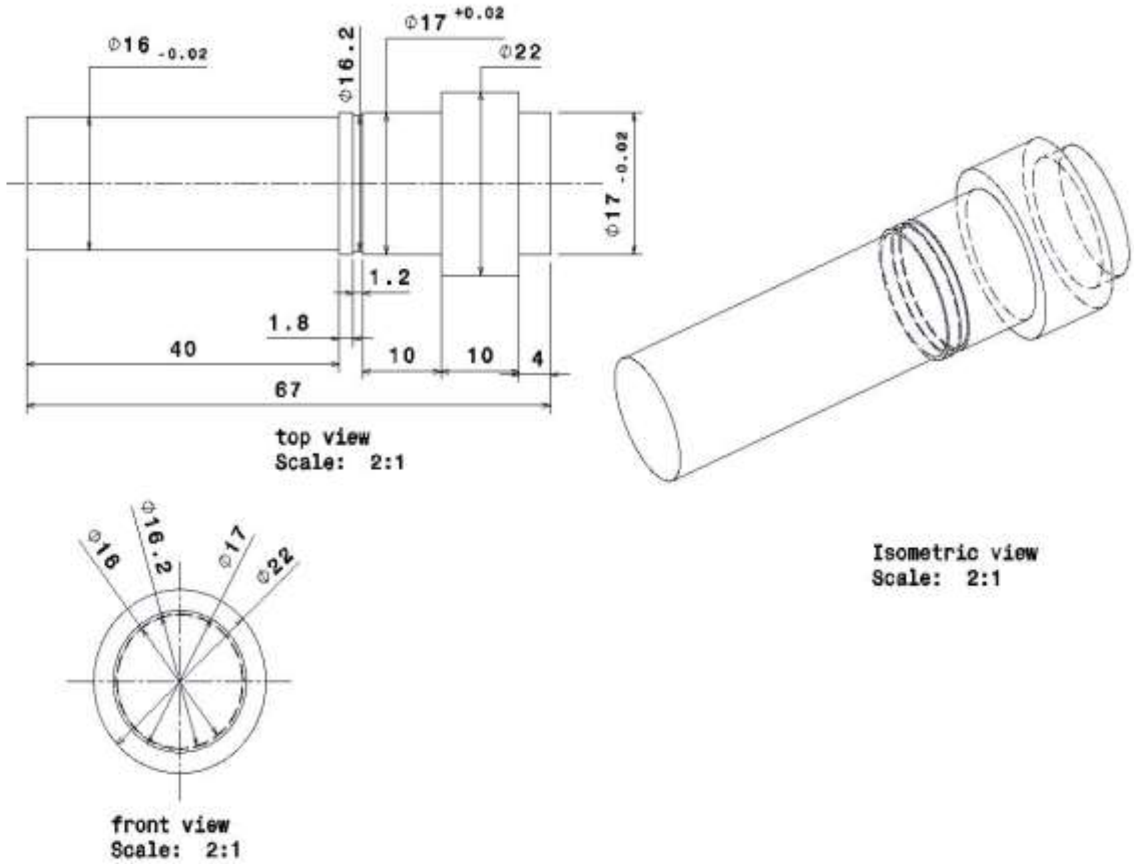

Fig. 3: Design of Input Shaft

Design of Input Shaft - Theoretical method

Table 1: Material Selection for Input Shaft

Designation	**Ultimate Tensile Strength (N/mm^2)**	**Yield Strength (N/mm^2)**
EN 24	800	680

As per ASME code,
$f_{s\,max} = 108\ N/mm^2$
Check for torsional shear failure:
$f_{s\,act} = 0.67\ N/mm^2$
As; $f_{s\,act} < f_{s\,all}$
Input Shaft is safe under torsional load.

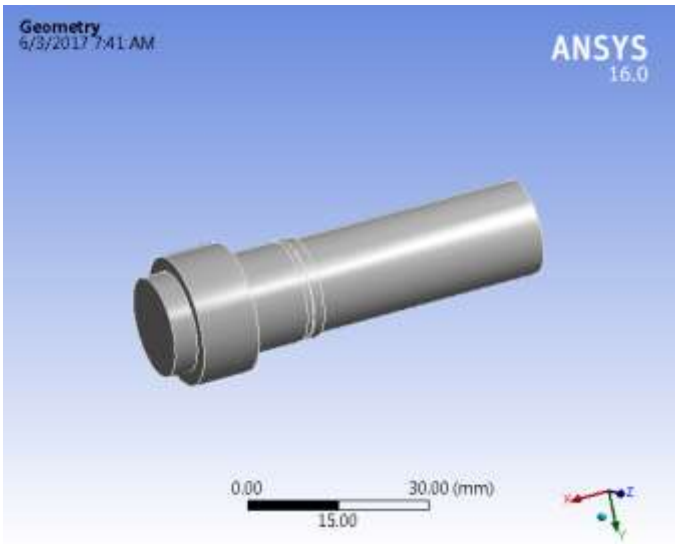

Fig. 4: Geometry of Input Shaft

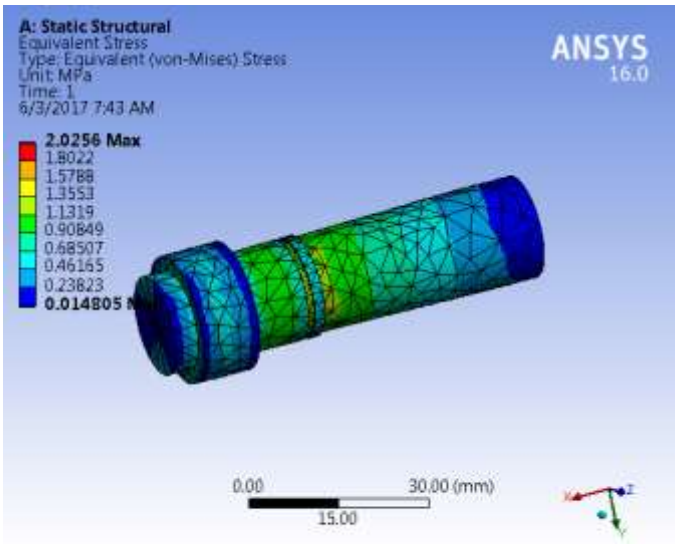

Fig. 5: Equivalent Stresses in Input Shaft

Maximum stress by analytical methods is well below the allowable limit of 108 N/mm^2, hence the input shaft is safe.

Yoke

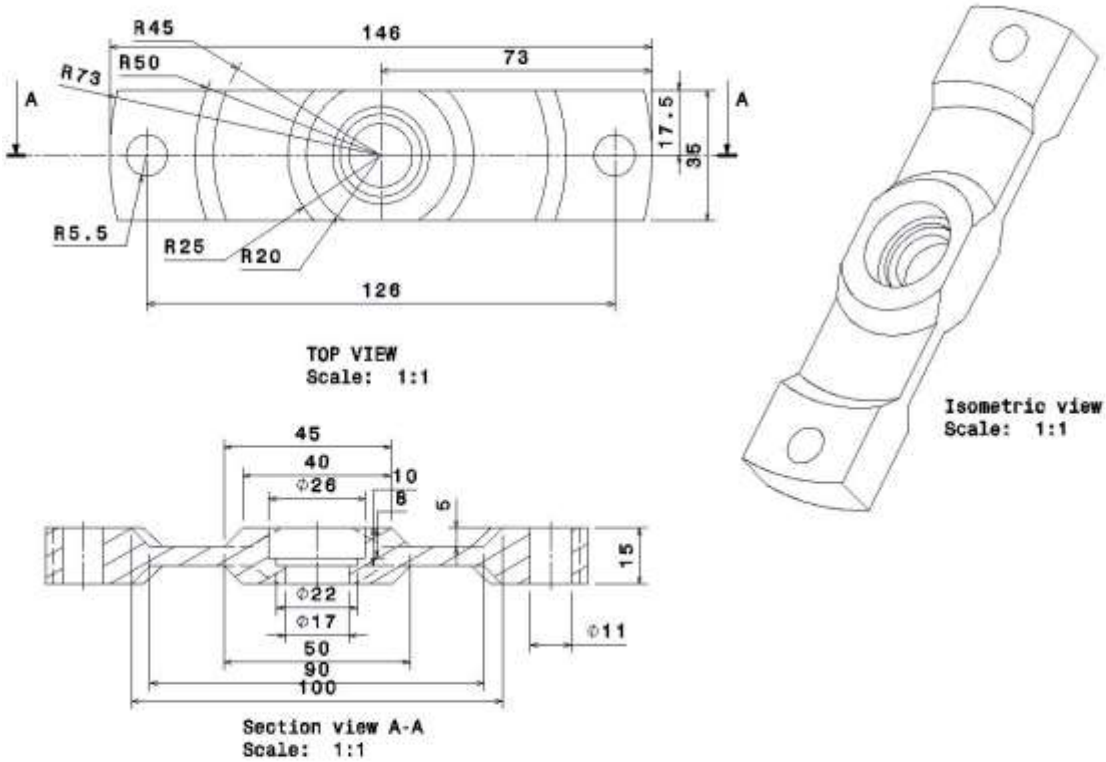

Fig. 6: Design of Yoke

Design of Yoke - Theoretical Method

Table 2: Material Selection for Yoke

Designation	**Ultimate Tensile Strength(N/mm^2)**	**Yield Strength(N/mm^2)**
EN9	600	380

As per ASME code,
$f_{s\ max} = 100$ N/mm^2
Check for torsional shear failure:
$f_{s\ act} = 0.073$N/mm^2
As, $f_{s\ act} < f_{s\ all}$
Yoke is safe under torsional load

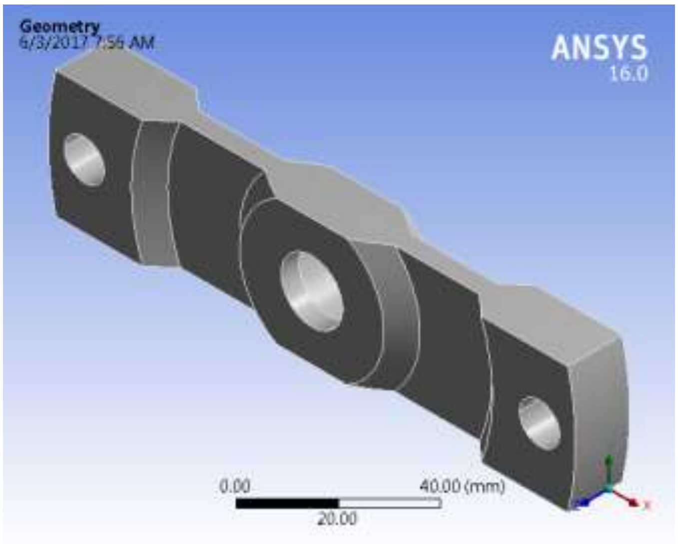

Fig. 7: Geometry of Yoke

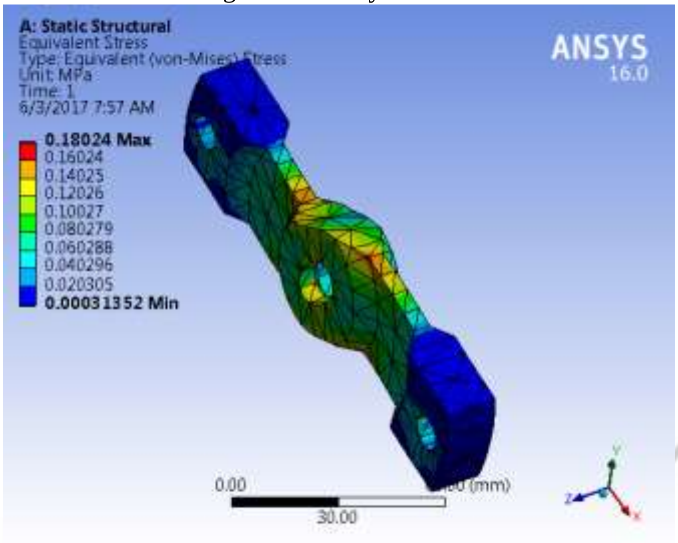

Fig. 8: Equivalent Stresses in Yoke

Maximum stress by analytical methods is well below the allowable limit of 100 N/mm^2, hence the yoke is safe.

Crank

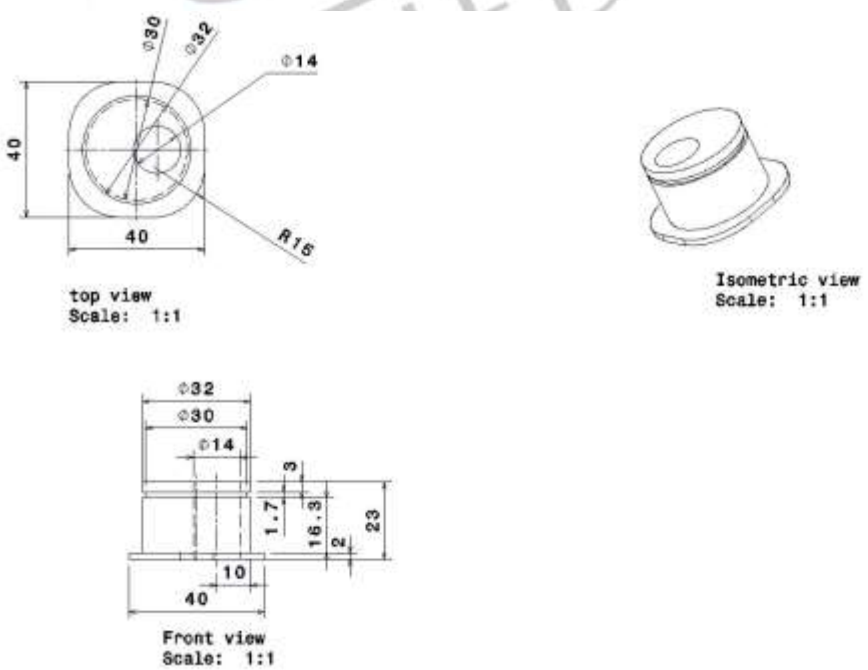

Fig. 9: Design of Crank

Design of Crank - Theoretical method

Table 3: Material Selection for Crank

Designation	Ultimate Tensile strength (N/mm^2)	Yield strength (N/mm^2)
EN9	600	380

As per ASME code,
$f_{s\,max}$ = 100 N/mm^2
Check for torsional shear failure:
$f_{s\,act}$ = 0.087 N/mm^2
As $f_{s\,act} < f_{s\,all}$
Crank is safe under torsion load.

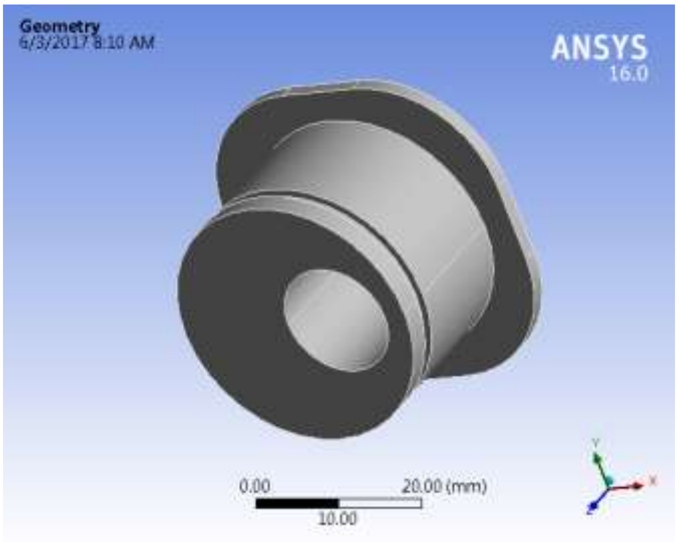

Fig. 10: Geometry of Crank

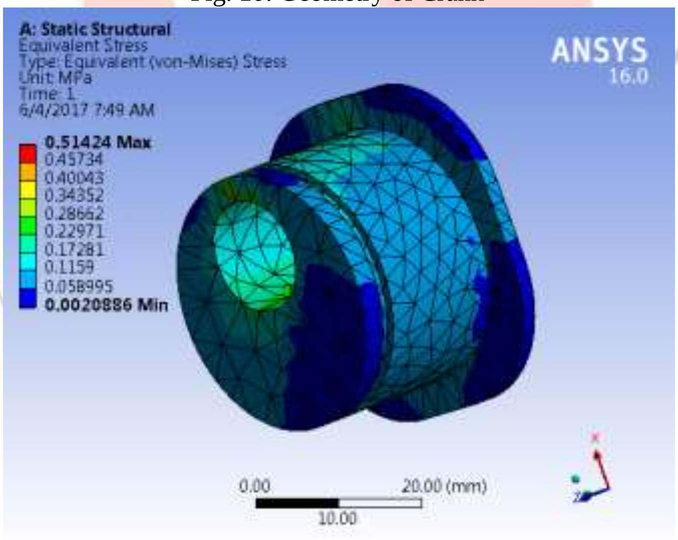

Fig. 11: Equivalent Stresses in Crank

Maximum stress by analytical methods is well below the allowable limit of 100 N/mm^2, hence the crank is safe.

External Gear
To calculate gear torque,
Torque at the external and internal gear pair = Torque (without overload) x reduction ratio due to kinematic linkage x Gear ratio of internal external
Where,
Linkage reduction ratio is such that each link converts 360 degree to 45 degree hence total three links are these which means the net motion of the system is (45+45+45 = 135) and reduction ratio of linkage = 360 /135 = 2.93--- approximately 3
T_{design} for gear system = 0.35 x 3 x (50/44)
= 1.19 N-m
Hence, the torque used to design gear pair is 1.19 N-m.

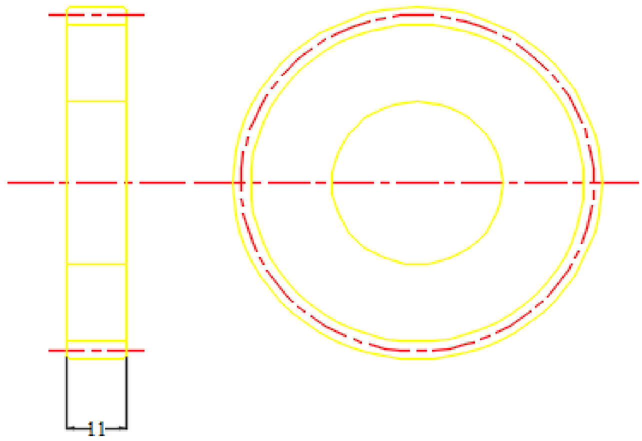

Fig. 12: Design of External Gear

Design of External Gear - Theoretical method

Table 4: Material Selection for External Gear

Designation	Ultimate Tensile Strength (N/mm^2)	Yield Strength (N/mm^2)
EN 24	800	680

As per ASME code,
$f_{s\ max}$ = 108 N/mm^2

Check for torsional shear failure:
$f_{s\ act}$ = 0.02 N/mm^2

As, $f_{s\ act} < f_{s\ all}$
External gear is safe under torsion load

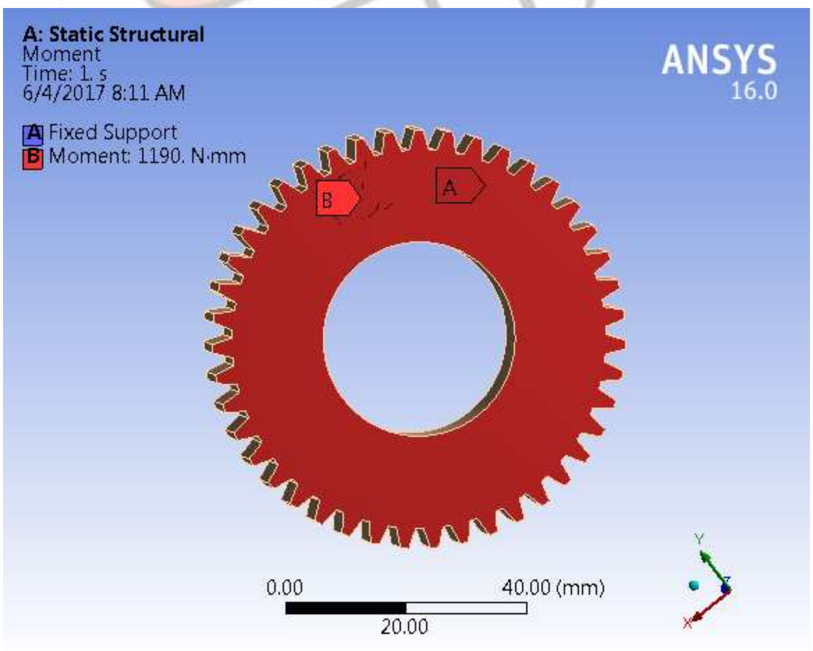

Fig. 13: Moment of External Gear

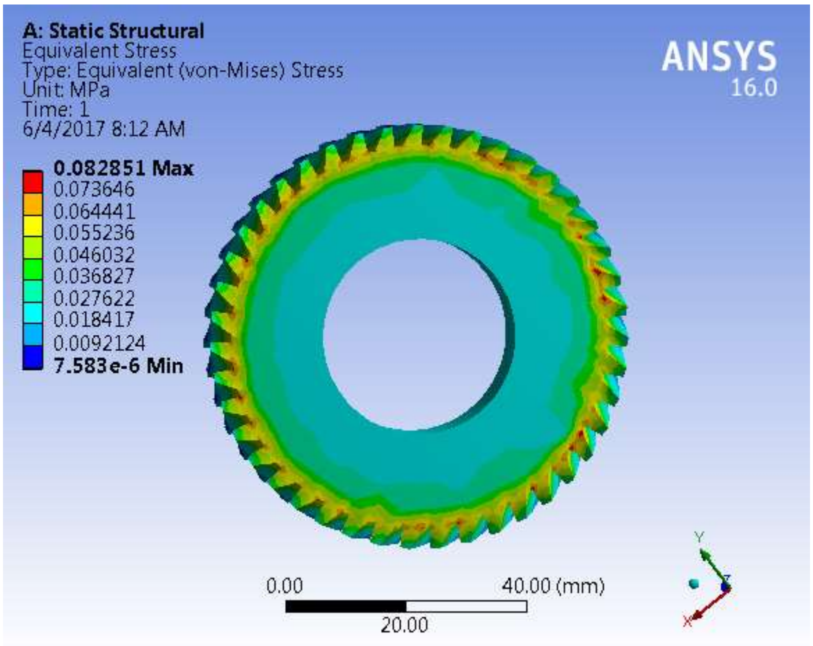

Fig. 14: Equivalent Stresses in External Gear.

Maximum stress by analytical methods is well below the allowable limit of 108 N/N/mm^2, hence the external gear is safe.

Internal Gear

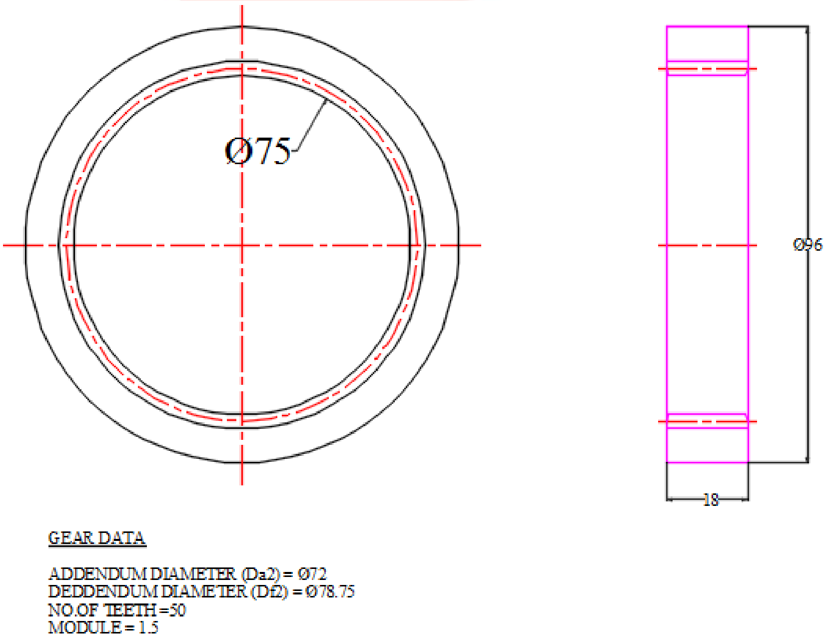

Fig. 15: Design of Internal Gear

Design of Internal Gear- Theoretical Method

Table 5: 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,

$f_{s\,max}$ = 108 N/mm^2

Check for torsional shear failure:

$f_{s\,act}$ = 0.01 N/mm^2

$f_{s\,act} < f_{s\,all}$

Internal gear is safe under torsion load.

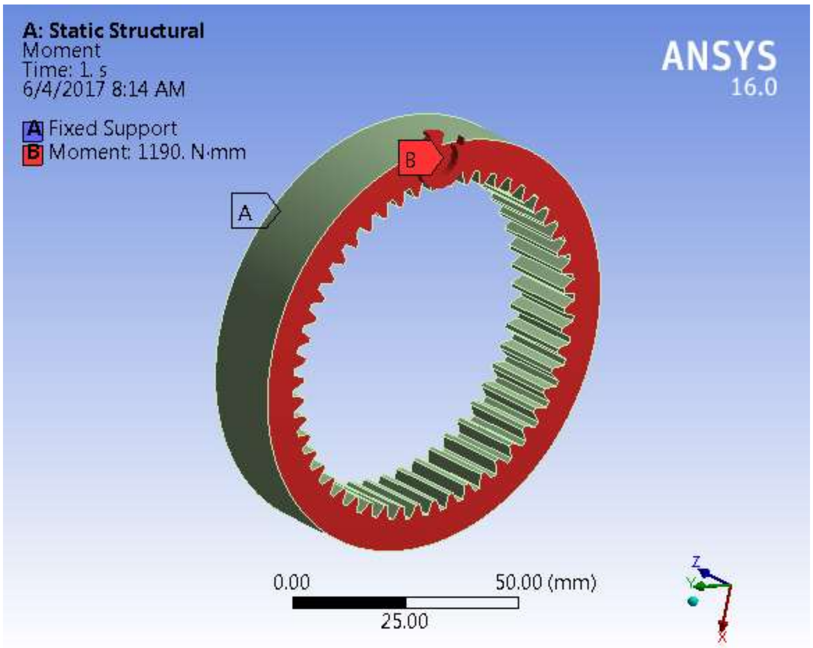

Fig. 16: Moment of Internal Gear

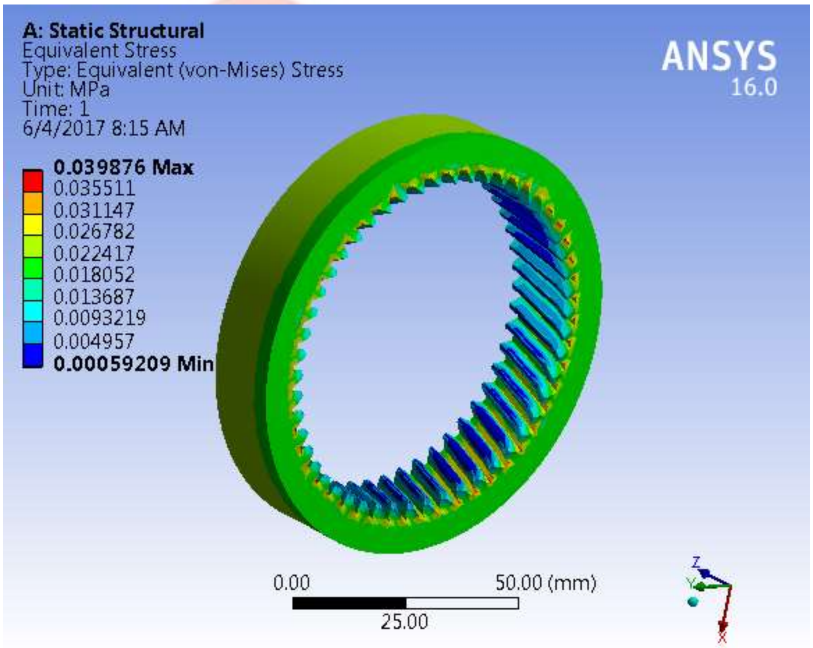

Fig. 17: Equivalent Stresses in Internal Gear

Maximum stress by analytical methods is well below the allowable limit of 108 N/mm^2, hence the internal gear is safe.

V. RESULTS

Table 6: Results of Components of Eclipse Gearbox

Name of the Components	Maximum Allowable Stress (N/mm^2)	Theoretical Stress (N/mm^2)	Maximum Analytical Stress (N/mm^2)	Results
Input shaft	108	0.671	2.0256	Safe
Yoke	100	0.073	0.18024	Safe
Crank	100	0.087	0.51424	Safe
External gear	108	0.02	0.08251	Safe
Internal gear	108	0.01	0.039876	Safe

VI. Conclusions

- Determined stress by theoretical and analytical methods is well below the allowable limit of 108 N/mm^2 hence the input shaft, external gear and internal is safe.
- Determined stress by theoretical and analytical methods is well below the allowable limit of 100 N/mm^2 hence the yoke and crank is safe.
- All the components shows stresses well below the permissible limit indicating safety of the component which solve the reliability problems.
- The service life for the proposed eclipse gear box will be marginally higher as compared to the existing gearbox.

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