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Finite Element Analysis of Telescopic Roller Screw Mechanism Actuator

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***Abstract*— Hydraulic Actuators in spite of all its disadvantages have been used for many years. Electromechanical actuators can efficiently replace the hydraulic actuators. Different electromechanical actuators like the ball screws, roller screws can be used in place of hydraulic actuators. Roller screws when compared to ball screws can carry higher loads, have a longer life, and can provide higher speeds and accelerations. Roller screw provides finer leads as compared to ball screws. Roller screw consists of different components like screw, shaft, roller, ring gears and spacer rings. If the screw shaft is made lengthy the screw becomes a long column and it fails due to buckling. Hence to avoid buckling and to get more stroke length telescopic roller screw actuators is designed. Initially the literature survey is carried out. Then the CAD modelling of the telescopic roller screw actuator is carried out in CATIA software. The deformations and stresses induced in the telescopic roller screw actuator is calculated from finite element analysis and the results are compared with the theoretical results. The stresses induced in helical gears and planetary gears is calculated by finite element analysis and the results are compared with the theoretical results.**

I. INTRODUCTION

The performance of hydraulic actuator used in missile applications depends on oil properties and operating pressure. In missile application rapid stoke is achieved by using high operating pressures, however due to high pressure, there is possibility of leakage of oil at junction of assembly, moreover the periodic replacement of oil is essential for satisfactory function of missile. Due to high maintenance cost and its vulnerability to high temperatures and pressures, hence, hydraulic actuator is proposed to be replaced by telescopic roller screw actuator mechanism. A roller screw is a mechanical device used for converting rotary motion into linear motion or vice versa and to transmit the power. A roller screw coupled with a motor and a gearbox can be used as an electromechanical actuator to replace the hydraulic system. By making a mechanism telescopic larger stroke lengths can be obtained and it also prevents in avoiding the buckling phenomenon. With the help of telescopic mechanism, specific angles can be obtained. Planetary roller screw consists of screw shaft, rollers, nut, ring gears and spacer rings. The efficiency of the planetary roller screw has been analyzed by Steven.A.Velinsky[1]. It was observed that the load carrying capacity of planetary roller screw was 69% greater than the ball screws. Moreover, maximum efficiency of 98.5 % was observed for planetary roller screw at certain conditions of contact angle and the helix angle. Guan Qiao et al.[2] have developed electromechanical actuator for aerospace applications. The roller screw actuators were compared with the hydraulic actuators and they found that the maintenance cost of the hydraulic actuator is high and it is vulnerable to high temperatures and pressures. Mattew.H.Jones et al.[3] have developed a kinematic model to predict the axial migration of the rollers relative to the nut in the planetary roller screw mechanism. They have proved that the axial migration is an undesirable phenomenon that causes binding and eventually leads to the destruction of the roller screw mechanism. They also proved that the axial migration of rollers is due to slip at the nut-roller interface, which is caused by pitch mismatch between the spur-ring gear and the effective nut-roller helical gears. Direct stiffness method to construct a stiffness model of the roller screw mechanism was used by Mattew.H.Jones et al. [4] . They modelled the entire roller screw mechanism as a large spring system composed of individual springs representing various compliances. In addition to predicting the overall stiffness of the mechanism, direct stiffness method was used to distribute load across the threads of the individual bodies. The nature of contact between the load transferring surfaces was studied by Mattew.H.Jones et al.[5]. They developed theory which was used to design a roller screw, which would result in higher stiffness and less contact stress. A three dimensional clearance vector for mating threads was developed by Xiaojun Fu et al,[6] .With the help of this clearance vector the contact positions of the mating surfaces have been calculated and the clearances were calculated in all the directions. The model of the planetary roller screw mechanism according to force equilibrium was developed by Wenjie Zhang et al. [7]. The main aim was to calculate the load distribution occurring on the threads and as well as to calculate the shaft sections deformation, threads deformation and contact deformation on the contact points of threads. The relationship between different parameters of planetary roller screw was given by Shangjun Ma et al. [8]. It was proved that by knowing the correct relationship between different components of the planetary roller screw, better design of the planetary roller screw was possible which resulted in increased stiffness and decreased stresses. Analytical method for calculating the stresses in spur gears was derived by N.D.Naryankar et al.[9].An axial stiffness model was prepared by Jianan Guo et al, [10] to improve the accuracy of the planetary roller screw. They have concluded that as the number of rollers increases, the axial stiffness increases.

The main objective of this work is to carry out the finite element analysis of the critical components of the telescopic roller screw actuator. Initially the various configuration of the

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roller screw actuator is studied and based on space availability and functional requirements, one of the configuration is selected for finite element analysis. Finite element analysis of the critical components is carried out to check the stress and deformation induced in the telescopic roller screw actuator.

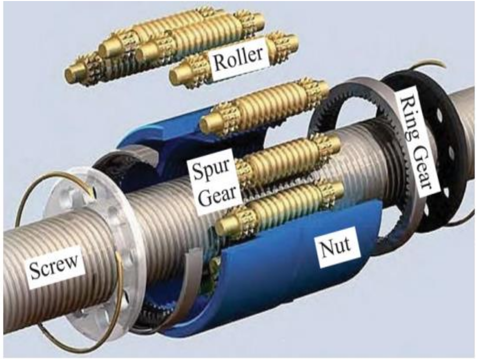

Fig 1 Standard Planetary Roller Screw

II.CONSTRUCTION AND WORKING OF ROLLER SCREW MECHANISM FOR ACTUATOR

Initially the configuration of the telescopic roller screw actuator is prepared. The schematic diagram of the telescopic roller screw actuator is shown in fig 2,

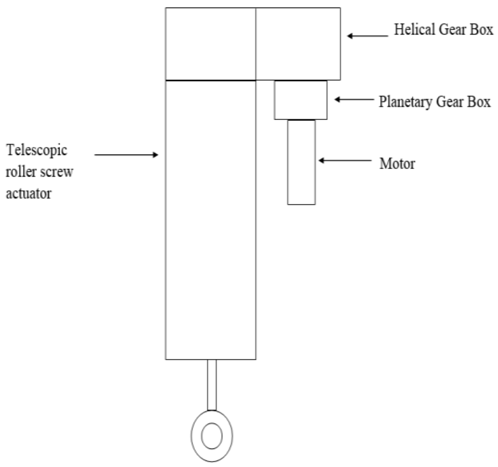

Fig 2 Schematic of the telescopic roller screw actuator

A servo motor is used to supply power to the telescopic roller screw actuator. The motor provides high speed and low torque. Hence for converting low torque of the motor to the required torque, gear boxes such as planetary and helical gear boxes are used. Torque from the servo motor is supplied to the planetary gear box which increases the torque by using the required reduction ratio. Planetary gear box is further coupled to the helical gear box to increase the torque. This torque at the output of the helical gears is used as the input to the outer solid screw shaft. Based on literature it is observed that, the pitch of screw, roller and nut is in the range of 1.2 to 2.5, hence pitch of 2 mm is selected. The ratio of number of starts on screw and number of starts on nut is equal to number of starts on roller. Hence number of starts on roller is taken as 1. As the pitch of screw and nut is 2mm and number of starts is 5, for 1 revolution of the screw, the nut advances by a distance of 10 mm.

$$d_r = d_n - d_s \tag{1}$$

Diameter of Screw=d_s=30mm

Let d_n=Diameter of nut and d_r=Diameter of roller

$$\frac{d_n}{d_r} = \frac{Z_n}{Z_r} = \text{Number of Starts of threads} \tag{2}$$

Where,

Z_n=Number of teeth on ring gear

Z_r=Number of teeth on the spur gear

By solving the above two equations, the diameter of roller is obtained as 10 mm and the diameter of nut is as obtained 50 mm. The number of threads on roller is taken as 20. The reduction ratio obtained by dividing nut diameter and roller diameter is 5. Hence, the number of teeth on spur gear is taken as 16 and number of teeth on ring gear is taken as 80.

Diameter of the inner hollow screw is taken as 72.5 mm with pitch 2mm and number of starts 6. Hence for one revolution of screw, nut has a lead of 12 mm. By solving equation (1) and (2) diameter of roller is obtained as 18 mm and diameter of nut is obtained as 110 mm.

The planetary gear consists of sun gear, planet gears and ring gears having 18, 45 and 108 teeth respectively with module of 1 mm. The helical gear consists of 18 and 126 teeth on pinion and gear with normal module 2.5 mm. A servomotor having a torque of 1.2 Nm is selected which runs at 3000 rpm.

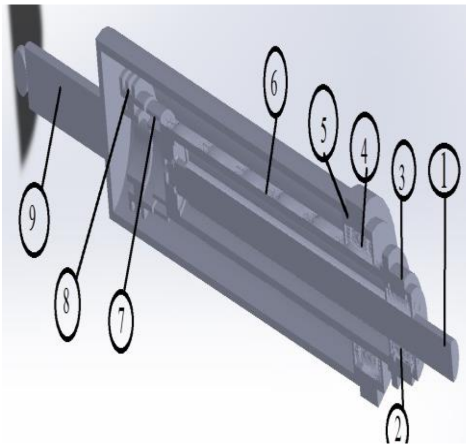

Fig 3 Sectional view of the complete telescopic roller screw actuator assembly

Figure 3. 1. Central screw; 2. Roller; 3. Nut; 4. Roller in Inner nut; 5. Inner Nut;6. Inner hollow screw; 7. Locking arrangement for central screw; 8. Locking arrangement for inner hollow screw; 9. Actuator

As the outer screw rotates, the nut moves in axial direction. After travelling a certain distance, at a certain position the nut gets locked. Hence axial motion of nut is converted to rotary motion. This rotary motion is further supplied as input to the inner hollow screw. The total stroke length of 1200 mm is

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achieved in two stages. In the first stage of engagement of the outer screw and the nut 600 mm and in further subsequent engagement of inner hollow screw with nut, 600 mm stroke length is achieved.

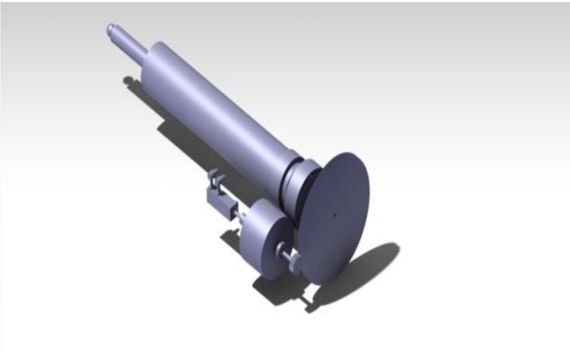

Fig 4 Model of the complete telescopic roller screw assembly

III. FINITE ELEMENT ANALYSIS OF CRITICAL COMPONENTS OF ACTUATOR

Finite Element Analysis Critical Components of Actuator is carried out using commercial software such as ANSYS. Finite element analysis is carried out between the screw roller and between the nut roller arrangement.

A) *Finite Element Analysis of Screw Roller*

The linear static structural analysis of screw roller is carried out in order to check stresses and deformation induced in screw and roller. The material selected is structural steel. For meshing, tetra mesh was selected as time required for obtaining the solution is less. Due to tetra meshing, 74760 nodes and 44375 elements are created. The mesh model of roller screw arrangement is shown in Fig.5. In actual assembly there are 9 rollers, however due to symmetry of the components and to reduce the computational time only one roller is considered in contact with the screw.

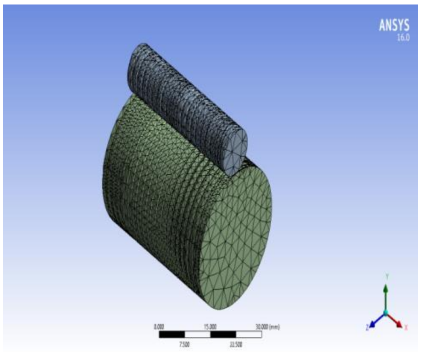

Fig 5 Meshing of screw roller

It is assumed that the total load acting on the telescopic roller screw actuator is equally distributed on each roller. For analysis purpose, only the part of the screw which is in contact with the roller is considered. For calculating the deformation one end of the screw is fixed while force is applied at other end of roller.

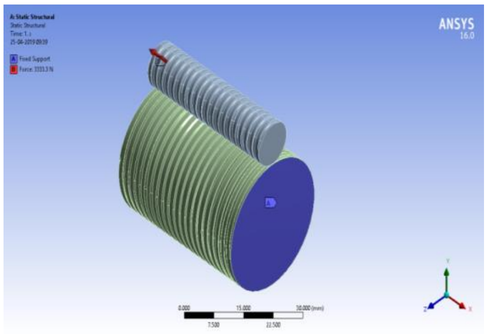

Fig 6 Boundary conditions applied for screw roller

The deformation in screw and roller is calculated and it is considered that the deformation on all rollers is same. Maximum deformation is obtained at the face where the force is applied. After calculating deformation, stress is calculated.

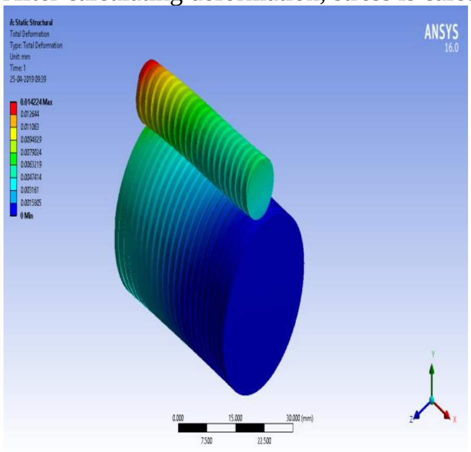

Fig 7 Deformation in screw and roller

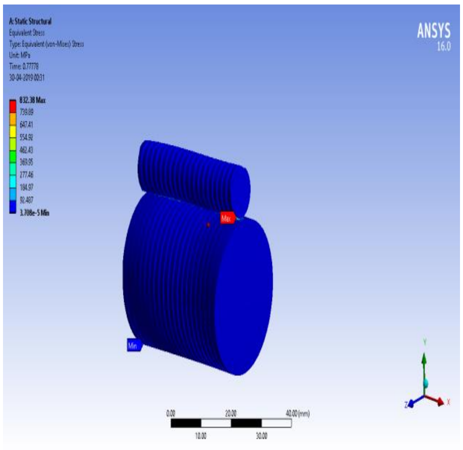

Fig 8 Maximum stress in screw roller arrangement

It is observed that the maximum stress occurs at the contact between the screw and the roller. Similar analysis is carried out between the inner hollow screw and the roller and the stress and the deformation are calculated.

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B) *Finite Element Analysis of Nut and Roller*

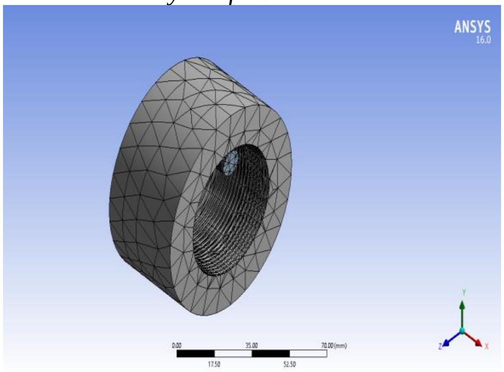

Fig 9 Meshing of nut roller

After carrying out finite element analysis of screw and roller, finite element analysis of nut and roller is carried out. For reducing the solution time, only one roller is considered in contact with the nut. The meshing of the nut and roller arrangement is carried out in ANSYS Workbench. Tetra meshing was carried out with element size 10 mm, which generates 48762 nodes and 29683 elements. Element size of 10 mm was taken because the nut contains multiple threads and it is difficult to properly mesh these threads. Hence to capture the nut surface properly element size of 10 mm was selected.

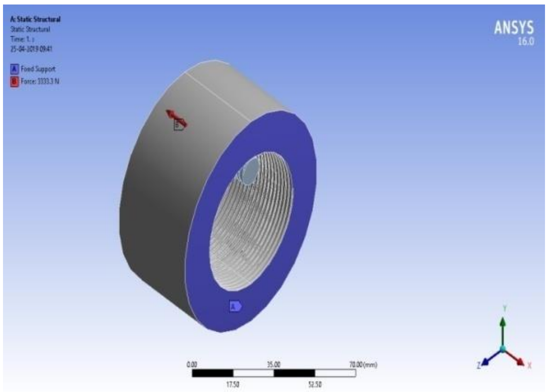

Fig 10 Boundary conditions applied at nut and roller interface

While applying the load it is considered that the load is equally distributed on all rollers. For calculating the deformation one end of the nut is fixed while force is applied on the roller.

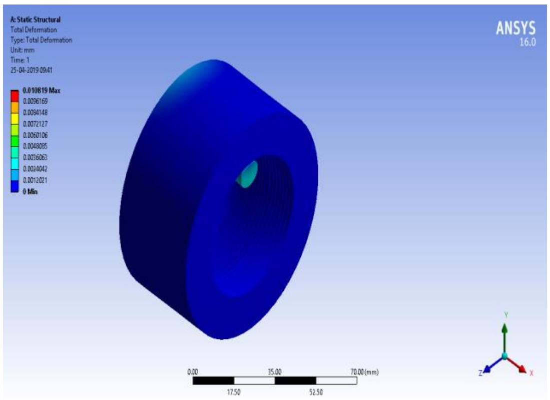

Fig.11.Deformation in nut and roller

The deformation on the nut and roller is calculated and it is considered that the deformation on each roller is same. The total deformation induced in the telescopic roller screw is addition of deformation between the screw and roller and deformation between nut and roller The maximum stress is generated at nut and roller contact. The total stress generated in the telescopic roller screw mechanism is the addition of stress generated between the screw and roller and between nut and roller. Similar analysis is carried out at the inner roller and nut assembly and deformation and stress is calculated.

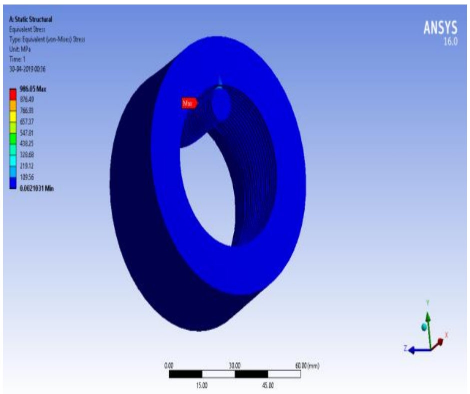

Fig 12 Stress generated at nut and roller

C) *Finite Element Analysis of Helical Gear and Planetary gear*

Initially bending stress in helical gear is calculated. Tetra Meshing of a single helical gear is carried out with 6253 nodes and 1074 elements.

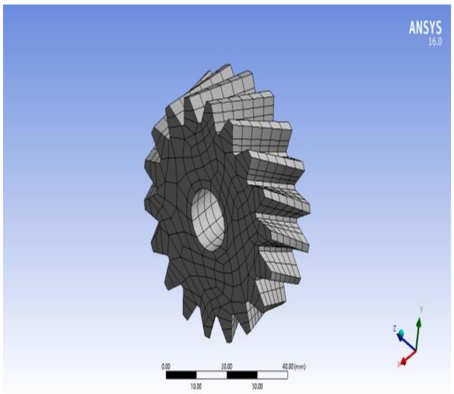

Fig 13. Meshing of a helical gear

After meshing tangential force of 343.656 N is applied on one teeth and the gear is allowed to move freely in X direction but motion of gear is restricted in Y and Z direction.

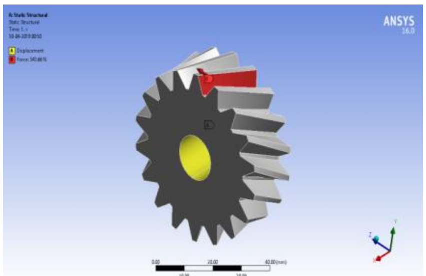

Fig.14.Boundary Conditions for helical gears

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It is observed that bending stress of 11.85 MPa is generated in helical gear which is within the elastic limit.

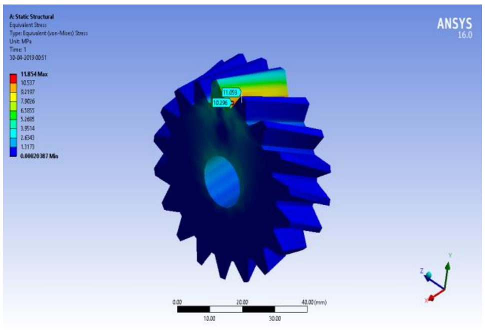

Fig.15.Bending stress in helical gear

For calculating the contact stress in helical gears, two gears in contact are selected. To reduce solution time only the portion in which helical gears are in contact is selected for analysis. Tetra meshing of the two helical gears in contact is carried out.

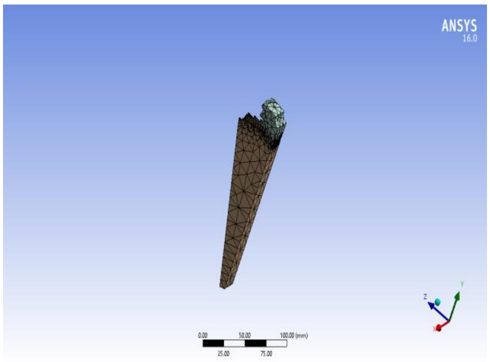

Fig.16.Meshing of the helical gear in contact

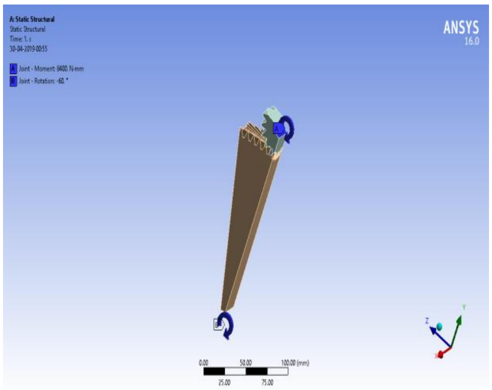

Fig.17.Boundary conditions for helical gears

A moment of 8400 Nmm is applied to the small helical gear and the larger helical gear is rotated by 60 degrees for generating the contact stress. Maximum contact stress in the helical gear is generated at the contact between the two gears.

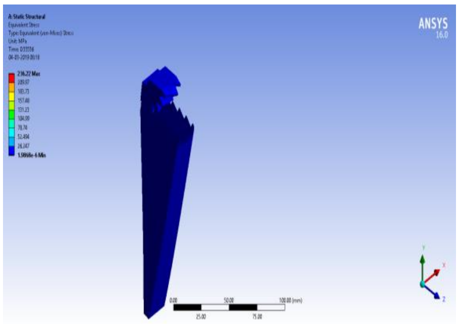

Fig.18. Contact Stresses in Helical gear

In the Planetary gear box initially sun and planet gear is considered. Hex mesh of sun and planet gears is carried. Fixed support is applied at the planet gears and moment of 1200 Nmm is applied at the sun gear. The maximum stress is generated at the contact between the sun gear and the planet gear. Similar analysis is carried out for planet gears and ring gears and the stress is calculated.

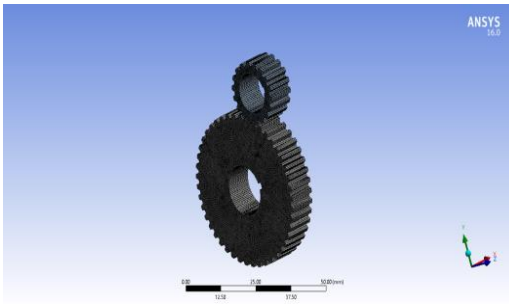

Fig.19.Meshing of the sun and planet gears

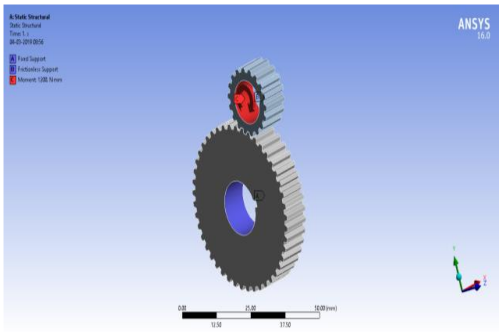

Fig.20.Boundary Conditions for sun and planet gear

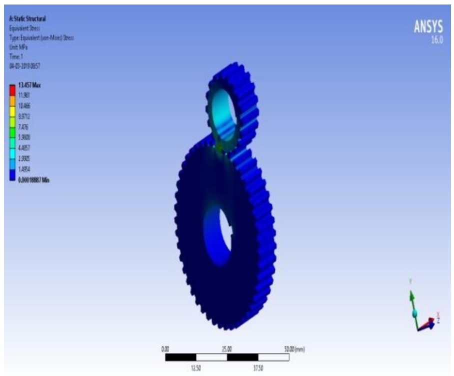

Fig.21Stress in sun and planet gears

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IV. RESULTS AND DISCUSSION

Stress and deformation induced in the telescopic roller screw is calculated from finite element analysis. The results obtained from finite element analysis is compared with theoretical results. It is observed that stress and deformation obtained from theoretical analysis and finite element analysis is nearly equal.

Table I Simulation Results

Parameters	Results
Deformation in outer roller screw	0.2253 mm
Stress generated in roller screw	1818.43 MPa
Bending stress in helical gear	11.854 MPa
Contact stress in helical gear	236.22 MPa
Stress between sun and planet gear	13.2 MPa
Stress between planet and ring gear	22.244 MPa
Deformation in inner roller screw	0.162 mm
Stress generated in inner hollow roller screw	1700 MPa

V. CONCLUSION

It is observed that the maximum stress in a roller screw mechanism occurs between the screw and roller interface and between the nut and roller interface, as roller is in contact with the screw and nut simultaneously. In case of helical gears, the maximum stress is generated at the contact between the two gears. Electromechanical actuators like telescopic planetary rollers screws can successfully replace the hydraulic system for various engineering applications.

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