International Journal for Ignited Minds (IJIMIINDS)

Design and Analysis of Motion Simulator Structure Used in Antenna Mounting System

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ABSTRACT

Antennas are the most used equipment for analyzing celestial objects as well as ground communications. In this paper, a motion simulator structure is theoretically designed for strength, modeled in three dimensional space and analyzed for fatigue loading. Initially the wind load calculations are carried out based on the standard formulas for 200Kmph speed. Every individual component of assembly is theoretically checked for the design safety. Further three dimensional modeling is carried out using CATIA software. Meshing is done through HYPERMESH and imported to ANSYS in 'inp' file format for further analysis to find the safety of the assembly for the given loading conditions. The analysis results are captured for both the loading conditions corresponding to the maximum and minimum loads. The result shows safety of the assembly for static conditions under both the cases. Similarly the fatigue analysis carried out on the structure also shows complete safety under the fluctuating loading conditions. Further modal analysis carried out to find the dynamic safety shows very high fundamental natural frequency compared to the operational frequency. All the results are represented with corresponding pictures to show structural safety.

Keywords *- Antenna, Analysis, Alt-azimuth, X-V mount, Fatigue.*

1. INTRODUCTION

Antennas are the most used equipment for analyzing celestial objects as well as ground communications. Antenna rotates for certain angles by rotating mechanism provided to trace the object and the whole assembly is supported on frames. Most of the present antenna technology is based on satellite tracking system. Satellite tracking systems are employed to track fast moving weather or earth resources satellites, space shuttles and unmanned deep space probes on interplanetary voyages. Over the years, various tracking system designs have been developed and employed to suit the application. The design of a satellite communication system is a complicated process. A great deal of research has been carried out on design techniques for improving the efficiency of large antenna dishes. To steer a large antenna dish with the required pointing accuracy a sophisticated antenna mount system must be employed. Present ground based satellite tracking stations use a two axis mounting of either Alt-Azimuth type or X-V type represented in fig 1 shows elevation and declination axis.

For a ship based tracking system the antenna is mounted on a stabilized platform which isolates the antenna from the dynamic motion of the ship. The ideal antenna mounting mechanism is kinematically capable of moving the antenna dish through the visible hemisphere and is strong enough to withstand the wind and other loads. Such a mechanism was found in the form of a Stewart platform modified for a large angular range. It is essentially a closed link mechanism consisting of six parallel variable length actuators constrained between a fixed base and a movable platform. This mechanism offers six degrees of freedom. The closed link structure results in a very strong mechanism capable of fast and accurate movements. When equipped with a closed loop control system and controlled through a computer, an antenna mount based on this parallel link mechanism offers a novel tracking system design. This antenna mount with proper joint designs enables large antenna dishes to track a moving target through the zenith without target loss, even during heavy weather.

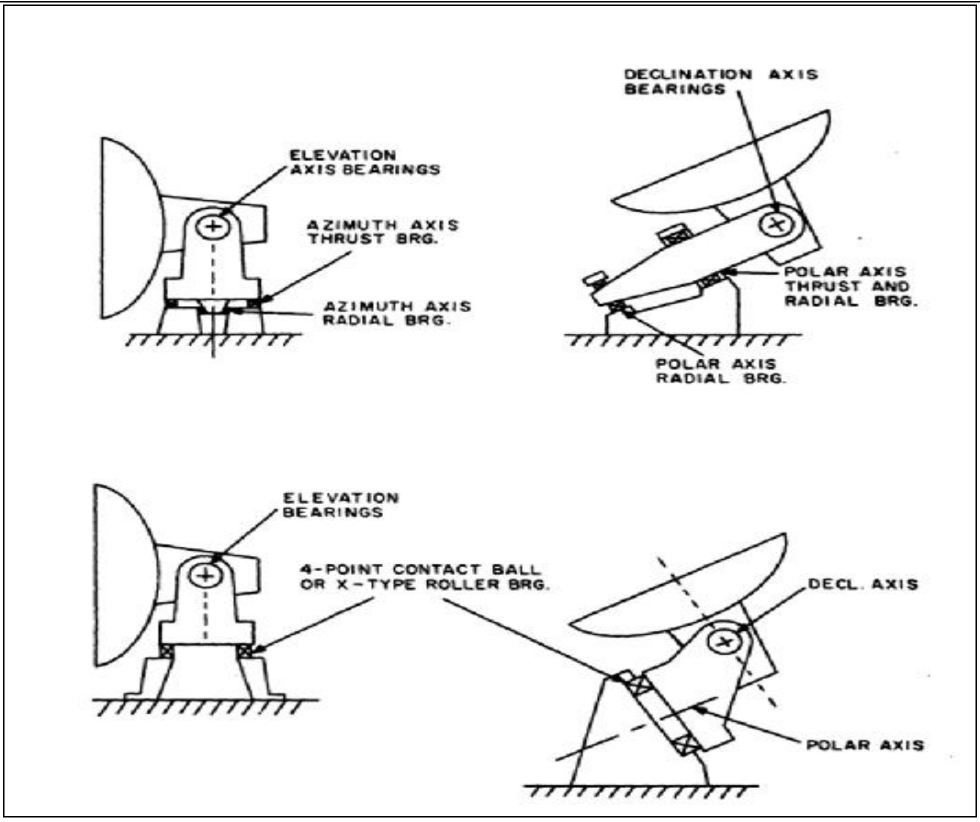


Fig 1: Types of Mounts for Antenna Tracking System.

1.1 Alt-Azimuth Mount and Associated "Keyhole"

This is the most common type of mounting system and has independently controlled azimuth and elevation axes. It consists of a vertical axis revolute joint which carries a horizontal axis revolute joint. The antenna dish is mounted on the horizontal axis. The antenna bore sight axis is positioned by rotating the vertical joint through the azimuth angle from the North and then rotating the horizontal joint through the elevation angle from the horizon as shown in fig 2. Once the satellite is acquired, there is a direct 1:1 mapping of the tracking errors and a control computer is not required. This mount has a singular position (keyhole) near the zenith. If a moving satellite is tracked through the zenith, or very close to it, then as the elevation angle reaches 900 the azimuth angle must rotate through 1800. The satellite can move on out of the antenna beam while this rotation takes place and the contact is lost. This region is called the "keyhole" of the Alt-Azimuth mount system.

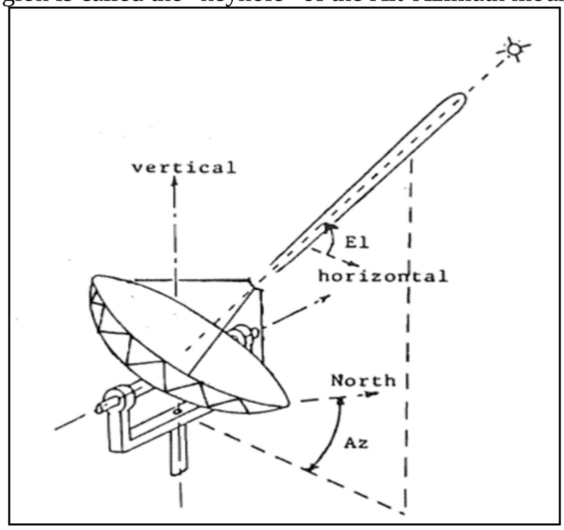


Fig 2: The Alt-Azimuth Mount.

When this mounting system is used on a ship and the antenna bore sight axis is pointing at the zenith, the rolling motion of the ship will produce an azimuth change of 1800 in a relatively short time. This results in an excessive speed requirement for the servo mechanism and effectively the ship loses contact with the satellite. For a high gain, narrow beam width antenna, the rolling and pitching action of the ship will cause the singularity of the Alt-Azimuth mount to trace out a flattened conical region around the zenith. Communication with a satellite within this region will be unreliable. This region is the effective "keyhole" and is greatly enlarged by the motion of the ship. For ground stations, prediction can be used to reduce the severity of the keyhole problem. As the elevation angle starts approaching 900, the azimuth axis begins to rotate so that the 1800 azimuth rotation can be completed within a larger time interval. This type of mount is suitable for high latitudes operations. This mounting system has the keyhole problem near the zenith.

1.2 X - Y Mount and Associated "Keyholes

The X-V mount consists of two controlled orthogonal axes represented in fig 3. A horizontal axis revolute joint which carries another revolute joint at right angles, which in turn supports the antenna. Each look angle is a function of both the controlled angles. Thus the two axes are not decoupled, as is the case with Alt-Azimuth mount and control is therefore more complex. The tracking errors do not map directly to each axis and computer control is necessary. This mounting system does not have a keyhole problem about the vertical axis. However it does have keyhole problems. The keyholes are located at each end of the horizontal axis, so ships near the polar region could have difficulty in communicating with geostationary satellites. Also, contact with a sub synchronous satellite making a low pass could be difficult. This keyhole problem can be overcome by installing two X-V mount antennas perpendicular to each other. Each antenna covers the singularities of the other. This doubles the cost of the tracking system, but the extra reliability plus full hemispherical coverage is sometimes worthwhile.

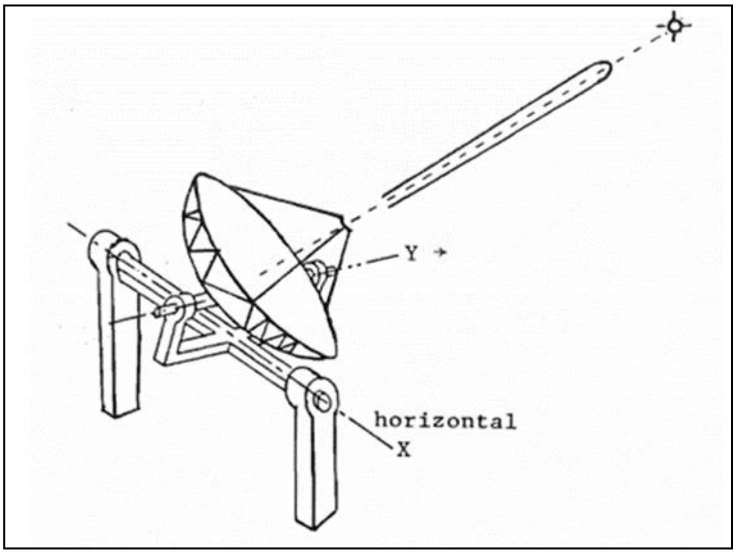


Fig 3: X-V Mount.

2. MODELING OF MOTION SIMULATOR

The fig 4 represents three dimensional representation of the motion simulator with its major components. Since main work is limited to moving frame, base frame and gear calculations remaining parts are modeled to the requirements to get a view of the problem. Catia with it’s extremely and user friendly Graphical interface, the geometries built using sketcher, part modeler and assembler.

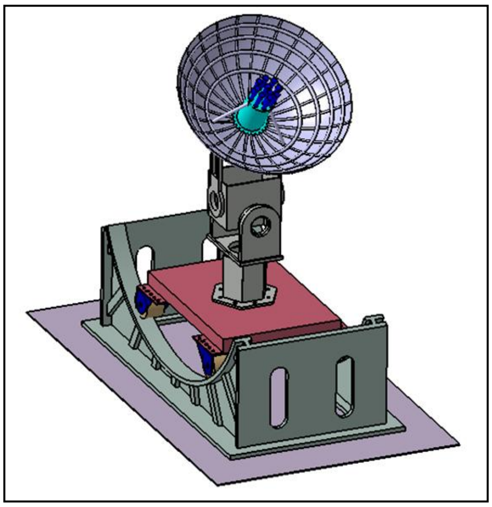


Fig 4: Geometry of Motion Simulator.

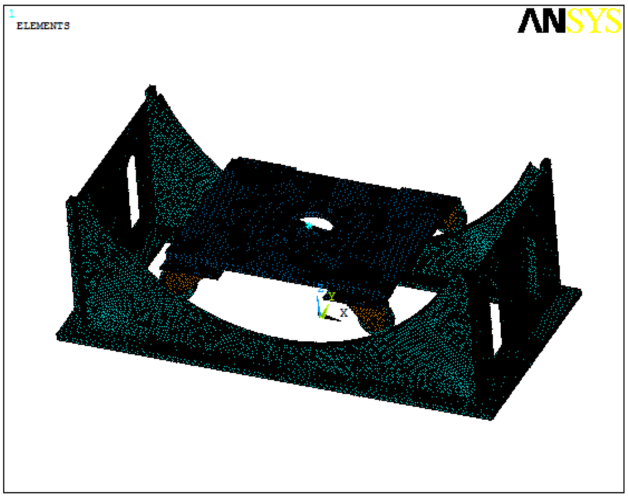


Fig 4: FE Model of Motion Simulator.

2.1 Meshed Model

The fig 5 shows meshed model of the problem. 4 noded tetrahedral elements are used for representation of the problem. The structures are free meshed to get 4noded tetrahedral (Solid45) and coupling equations are used to define the contact. A Mass element is built at the center of the moving frame to apply the loads through RBE3 connection. A total of 187805 elements and 58425 nodes are used for representation of the problem.

2.2 Boundary Conditions Plot:

The figure 6 shows applied boundary conditions on the problem. Through RBE3 element which connected to the antenna support structure, a load of 180000N vertical load along with 206958000N-mm moment is applied in the tilting direction. The base of the platform is constrained in all the directions.

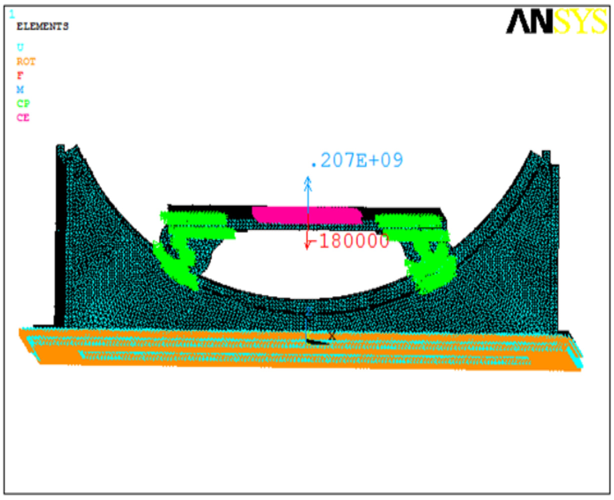


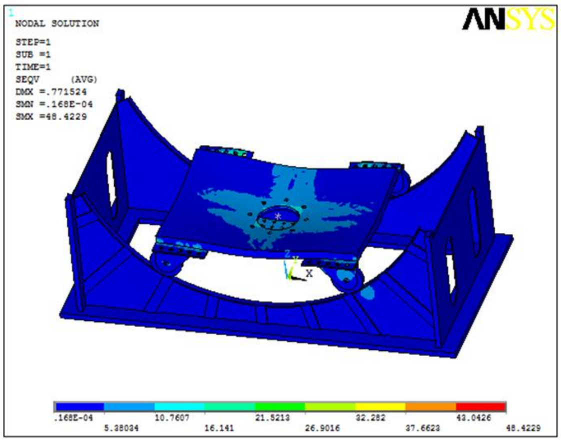
Fig 6: Boundary Conditions Plot.

3. RESULTS AND ANALYSIS

3.1 FE Analysis

Finite Element Analysis has been carried out on the motion simulator to find the fatigue strength along with the structural strength. Fatigue analysis requires at least two load cases for finding the life of the component. In the problem, both the cases are considered based on the maximum projected area condition for the wind load. One at 30^0 which has highest projected area and the other minimum projected i.e. 90^0 to the azimuth condition. Since the component weights will not vary by position, only torque load acting on the structure will vary by the position. So two load cases representing the maximum and minimum loading conditions are written to separate files. The results are as follows. Structural safety analysis is generally carried out by checking the stress and deformation conditions. These two parameters decide the structural safety by checking for allowable limits under the given loads. The results are as follows.

The fig 7 shows maximum von-Mises stress of around 48.4229 N/mm^2. This stress is less than the allowable stress of the material. The stresses are concentrated around the holes region. So structure is safe for the given maximum loading conditions.



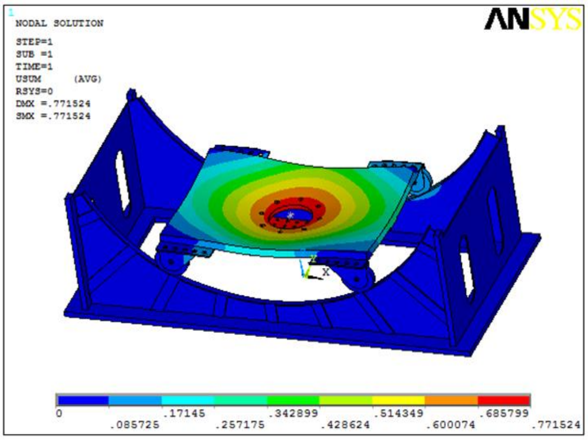


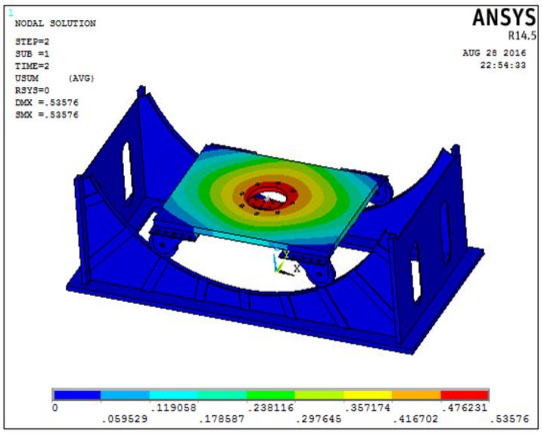
Fig 7: Max von-Mises Stress Plot for Assembly. Fig 8: Displacement Plot.

The fig 8 shows maximum displacement of 0.771524mm. Maximum allowable deflection as per the IS standard is minimum unsupported length/750 mm. Here minimum unsupported length is 2750mm. Therefore maximum allowable deflection for the problem is 2750/750=3.66mm. The developed deformation of 0.771524mm is less than the allowable deflection of the problem. So the structure is safe for the given loading conditions under maximum loading conditions.

3.2 Minimum Load Analysis

Further analysis is carried out for the minimum loading conditions. The results are again captured for the overall stress, deformation and stress in the individual components to find the structural safety for static condition. Again the load is applied through RBE3 element simulating the vertical column at the center.

The fig 9 shows deformation of the structure for the given minimum load. The deformation value is 0.53576mm at the center. This can be attributed to a simply supported configuration of the problem. The displacement is spreading from the center to the outer support regions. The deformation value is reduced near the support regions.



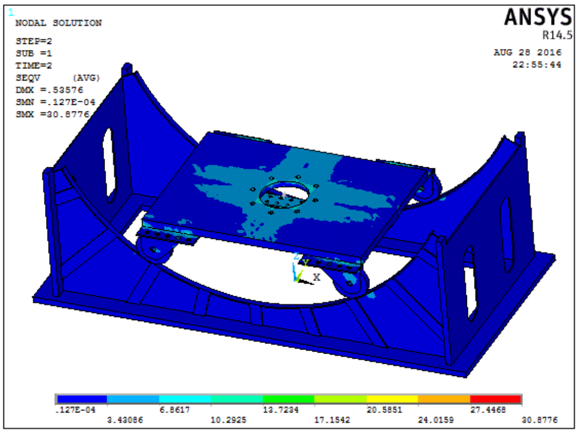


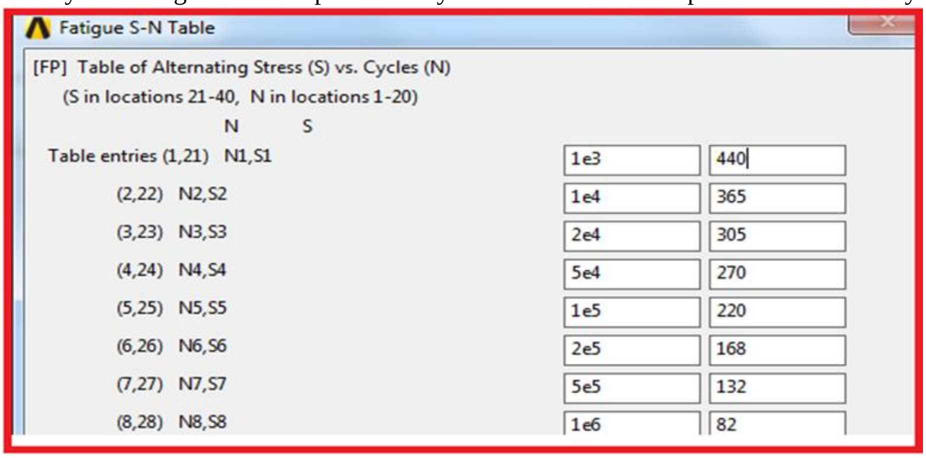
Fig 9: Deformation for the Minimum Load. Fig 10: Max von-Mises Stress for Min Loading Conditions.

The fig 10 shows von-Mises stress in the problem. Maximum stress value is 30.8MPa which is less than the allowable stress of the material. So structure is statically safe for the given loading condition.

3.3 Fatigue Life Estimation

Even though the structure is safe for the given static loading conditions, it can’t be concluded that the structure is safe for the fatigue load. Fatigue load is the failure of the structures under the cyclic or fluctuating loads. The same static loads applied repeatedly on the structures forms fatigue loading on the structure. Since the operation of antenna is continuous, the base structure is subjected to fatigue loads. Fatigue estimation is carried out by selecting the node corresponding to the maximum stress. The stress values for both load cases will be obtained at the same node to find the alternating stress and fatigue life of the assembly.

The fig 11 shows fatigue data of the given material. The fatigue stress or alternating stress value is reducing with the number of cycles. Fatigue data is represented by S-N curve which is represented in the Ansys menu.


Fatigue S-N Table

[FP] Table of Alternating Stress (S) vs. Cycles (N)

(S in locations 21-40, N in locations 1-20)

		N	S
Table entries (1,21)	N1,S1	1e3	440
(2,22)	N2,S2	1e4	365
(3,23)	N3,S3	2e4	305
(4,24)	N4,S4	5e4	270
(5,25)	N5,S5	1e5	220
(6,26)	N6,S6	2e5	168
(7,27)	N7,S7	5e5	132
(8,28)	N8,S8	1e6	82

Fig 11: Fatigue Data for the Material.

The fig 12 shows stress values listed by the Ansys software for the given loading conditions. The stress values are represented for 6 stress components of S_x, S_y, S_z, S_{xy}, S_{yz}, and S_{zx} for both the load cases.

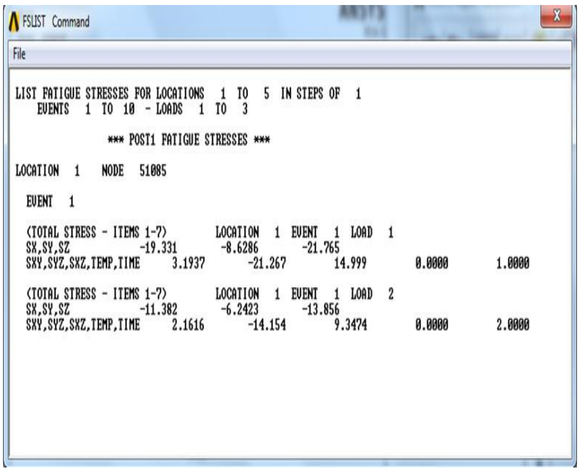


Fig 12: Stress Value of Node Number 51085.

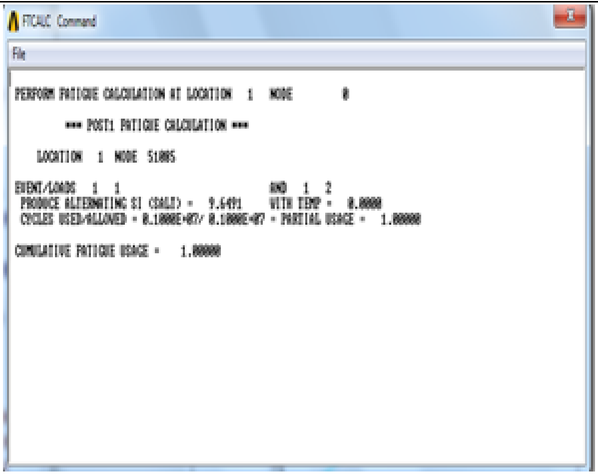


Fig 13: Fatigue Output from the Table.

The fatigue calculation from the software is shown above. The fig 13 shows alternating stress development of 9.6491Mpa which is less than the fatigue limit of 82Mpa for the material. So the component can with stand the loads without failure for the given 1 million cycles. So the system is safe for the fatigue loading conditions.

3.4 Modal Analysis

Modal Analysis is carried out to find the natural frequencies of the system subjected to dynamic conditions. Finding the natural frequencies allows the design changes to prevent possible resonance in the system. Natural frequency values directly depend on the mass distribution and stiffness in the structure. High dense material has less frequency and low mass has generally high frequency. It also influence by elastic modulus of the material. High elastic material has high stiffness. Also the frequencies are varied by support conditions. But it allows the designer to decide the speed or operational range of the problem.

Set No	Frequency
1	68.861
2	83.53
3	269.96

Table 1: Natural Frequencies.

The table 1 shows natural frequencies obtained in the process. The frequency is much higher than the operational frequency of the system. So system is safe for resonance conditions. The fig 14, 15 and 16 shows the mode shapes corresponding to Natural frequency. Maximum stress values vary with Natural frequencies.

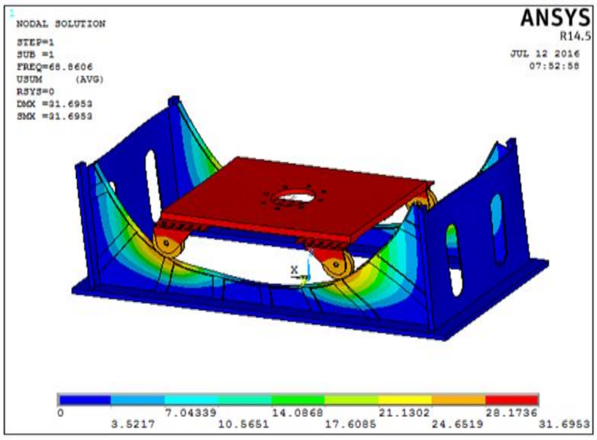


Fig 14: 1st Mode with Natural frequency of 68.86Hz.

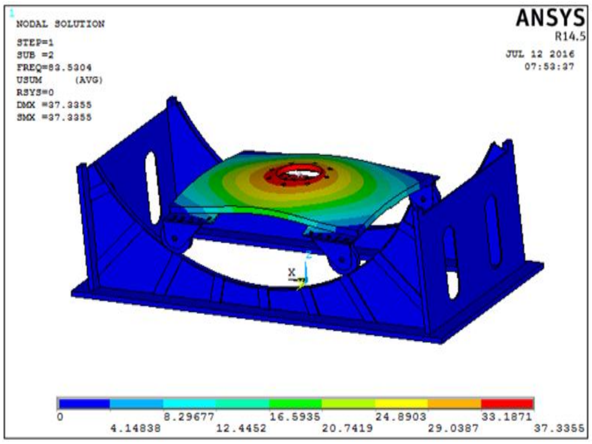


Fig 15: 2nd Mode with Natural frequency of 83.53Hz.

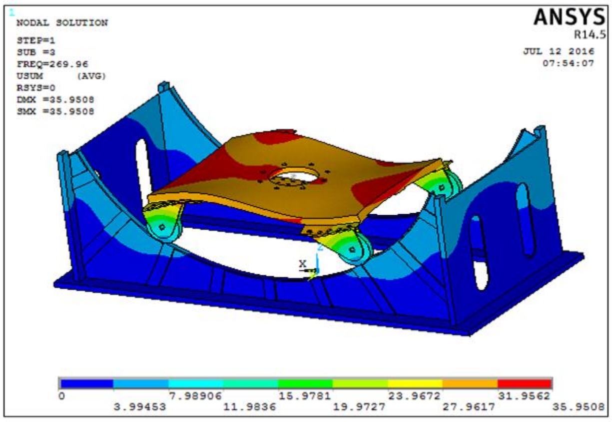


Fig 16: 3rd Mode with Natural frequency of 269.96Hz.

Although Analysis has been carried out on motion simulator structure is safe for given conditions, the individual results are represented below the table 2.

Components	Von-Mises Stress, MPa	Displacement, mm	Maximum Stress, MPa	Deformation, mm
Moving plate	39.1	0.7715	5.52	0.4154
Base plate	11.48	0.9194	25.65	0.5357
Wheels	8.55	0.9220	5.7	0.417
Wheel supporting bracket	48.42	0.1290	30.8	0.7026

Table 2: Results for Individual Components.

3.5 Modal Analysis

The motion simulator used for Antenna mounting is designed with basic calculations for structural requirements. Initially the total vertical loads are calculated based on the design specifications. Total load is calculated as 180KN. Based on the CG position and mass, mass moment of inertia calculations are carried out. Total Mass moment of Inertia found to be 30484kg-m^2. Total wind load is calculated for 200KMPH speed. BS standards are used for calculation of wind load with the coefficients specified. Inertia torque is calculated based on the design input speed of 100/sec. Frictional torque is also considered for power estimations. Due to initial level calculations, the motor is not specified. Based on the moment and loads, the moving bracket, wheels, base structure dimensions are calculated. Through RBE3 element the loads are applied for the structure after meshing it using Ansys software. Fatigue loads corresponding to two load cases are represented. Both the results are obtained for maximum and minimum loading conditions. The result shows complete safety of the structure as the developed stresses are well within the working range of the material. Even fatigue calculations shows safety of the structure as the developed alternating stress value is much less than the allowable stress of the member. Even the modal analysis results for structural safety and dynamic conditions also shows the fundamental frequency is much higher than the operational frequency of the system. So system is safe for the resonant conditions.

4. CONCLUSION

The motion simulator is designed, modeled and analyzed for fatigue. The summary of process is as follows.

- Initially for the design conditions, wind loads, inertia loads, torque calculations are carried out.
- Similarly gears, brackets and moving plate calculations are carried out with basic design principles considering stress concentration factors.
- The motion simulator is meshed using tetrahedral elements. Using an RBE3 element loads are applied to the platform. The analysis is carried out fatigue safety of the structure. Analysis is carried out for both maximum and minimum loads. The result shows complete safety of the equipment for the given loads. The stresses on individual components are also represented.
- The fatigue calculations from the Ansys fatigue module shows development of alternating stress is smaller than the allowable stress of 82Mpa for the given material. So the motion simulator structure is safe for the fatigue loads.
- Even modal analysis results shows, very high natural frequencies compared to the operational frequency of Motion simulator. So the structure is safe dynamically.
- All the results are presented with necessary pictures.

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