DESIGN AND ANALYSIS OF ROCKET HARDWARE PREPARATION MACHINE

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Abstract: *Rocket hardware comes in multiple sizes. It is pivotal to design an efficient machine that can perform cleaning operations. The purpose of this project is to design and analyze the Rocket Hardware Preparation Machine which comprises of Roller-stand assembly with the drive system, Reciprocating Boom with the support structure, Boom drive system with support stand, Lance actuator mechanism to perform cleaning operations. The Machine has been designed using theoretical formulations, later 3D models were developed using Solid Woks and ANSYS Workbench was used for FE analyses. This design also addresses the future augmentation of rocket hardware.*

***Keywords* - Linear Actuator, Drive System, Reciprocating Boom, Roller Stand and Support Structure.**

I. INTRODUCTION:

Rocket hardware generally comes in multiple sizes. The hardware needs to be prepared for further operations. For this, the hardware has to go through cleaning operations. As hardware is of big size, machine is necessary to carry out the cleaning operations. Those machines are known as rocket hardware preparation machines. Design of hardware preparation machine should be focused on handling two different hardware on one machine because two machines for two different hardware's will take up lot of space in industry and money for manufacturing.

The machine has to accommodate two hardware on it and both the hardware varies in diameter and length. The hardware has to be placed on rollers for a relative rotation to complete the cleaning on the whole inner surface of the hardware, so drive is provided to the supporting rollers by geared motor. The support structure has to bear the load of hardware. Inclination is provided to structure to collect the materials used for cleaning. Skew arrestor is to be provided to bear the lateral load of the hardware in the roller stand assembly. To carry out the operations, lance is provided with nozzle attached to linear actuator which reciprocates in the boom to move the lance. The hardware is cylindrical hollow in shape and some hardware's comes with one end as closed ones with sphere shape, so the cleaning has to be done on that surface also, that's why the lance end link has to travel from -135° to +45° with respect to boom central axis. The actuator is fixed in boom to move the lance links. The boom is supported by boom support structure and drive is provided to boom by boom drive system to move the boom from one end of hardware to other end of hardware to accommodate the length of hardware. Modeling of a Cabinet is based on space availability with ergonomic considerations to prevent human interaction with cleaning materials coming from the hardware. It also houses recirculating mechanism for cleaning materials.

II. PROBLEM DEFINITION:

The project aims to design and analyze a machine that can prepare hardware and perform various cleaning operations. The cleaning operations will be performed by a nozzle attached to a linear actuator mechanism. The linear actuator mechanism will navigate through the inner surface of the hardware with a reciprocating boom and a drive system. The rocket hardware will be placed on a roller stand which comprises of a drive system to rotate the hardware. The project also aims at designing a hand wheel operated screw mechanism to accommodate hardware of multiple sizes.

III. MATERIAL:

The following are the important design specifications and standards that have been collected and used in this project. These have been collected from Indian standard codes. The specifications are mentioned in the following table from Mild Steel IS:2062 Gr. B (Fe 410), IS 800 and maximum shear stress theory.

Table 1: Material properties and design parameters

Parameters	Values	Units
Modulus of Elasticity	208000	MPa
Density	7850	Kg/m^3
Poisson's ratio	0.3	-
Ultimate Tensile stress	410	MPa
Yield stress	240	MPa
Allowable yield stress	158.4	MPa
Allowable shear stress	79.2	MPa
Co-efficient of rolling friction	0.02	-

IV. METHODOLOGY:

The purpose of this project is to design and analyze a Rocket Hardware Preparation Machine that can prepare hardware and make them ready for sub sequent processes in industry. The basic step is to complete theoretical calculations and to create 3D models by using SOLIDWORKS and later analysis by using ANSYS. The machine is to be designed based on space availability and by considering ergonomics.

The following are the steps involved in Methodology:

- Design calculations of roller stand assembly, skew arrestor, selection of suitable drive system and power screw to move the roller assembly for multiple hardware placement followed by modeling and analysis.
- Design calculations of reciprocating boom, boom support structure and selection of suitable drive system followed by modeling and analysis.
- Design of Lance mechanism using inverse kinematics and selection of linear actuator followed by modeling and analysis.
- Modeling of a cabinet based on space availability and ergonomics and selection of door hinges and analysis.

4.1 Design of Roller Stand Assembly

The rocket hardware has to be placed on the rollers for a relative motion with them. An angle of 55° was taken as hardware position made by the center of rollers with the center of the hardware. Roller has been designed to withstand wear of the roller surface and also the whole load is taken on two rollers because the power to rotate the hardware is provided to only two rollers and other two are idle. The circular plate of the roller will be in contact with the hardware, so by evaluating contact stress, plate has to undergo hardening. Hardening of steel is done by using carburizing where the roller surface would be heated for 4 hours at 900°C as carburizing temperature and then by subsequently quenched.

The design of shaft is necessary as it is subjected to combined bending and torsion. The shaft design depends on the bending and turning moments. The value of diameter obtained is 79.2mm which has been rounded to 80 mm as per R20 series. The required speed of the roller (N) is 29.29 rpm. The torque (T) required to rotate the hardware by using roller is equivalent to 305.52 Nm by taking the co-efficient of rolling friction (μ_R) as 0.02. Based on the torque, the power (P) obtained is 0.937 kW. The rated power (P_{RATED}) is 1.474 kW by substituting efficiency of motor as 95%, service factor of 1.3 and number of starts and stops factor of 1.15. The geared motor is selected based on the above parameters obtained.

The hardware is placed at an inclination to recirculate the materials used for cleaning. So, an inclination of 4 degrees is provided throughout the assembly. Because of this inclination a skew arrestor is designed by taking a lateral load of 10% of safe working load. Hand wheel is provided to reduce the human effort to move the drive roller assembly inside and out to accommodate both the hardware.

The support structure for the roller stand assembly has been designed based on the space availability, column buckling and availability of cross-sections of structural steel. After various trial and error methods, the column cross section has been standardized to 240 mm in depth, 120 mm in breadth and 8 mm in thickness as per TATA steel structures. The safe working load for the structure has been given as an input which is 159.41 kN. The unanticipated loading like impact loads and mishandling of crane equipment resulting in hardware dropping on the support structure, the environment which might be corrosive and also for the future augmentation we have to keep in mind and designed the support structure. The roller stand assembly with drive rollers, idle rollers, skew arrestor and hand screw can be seen in fig 1.

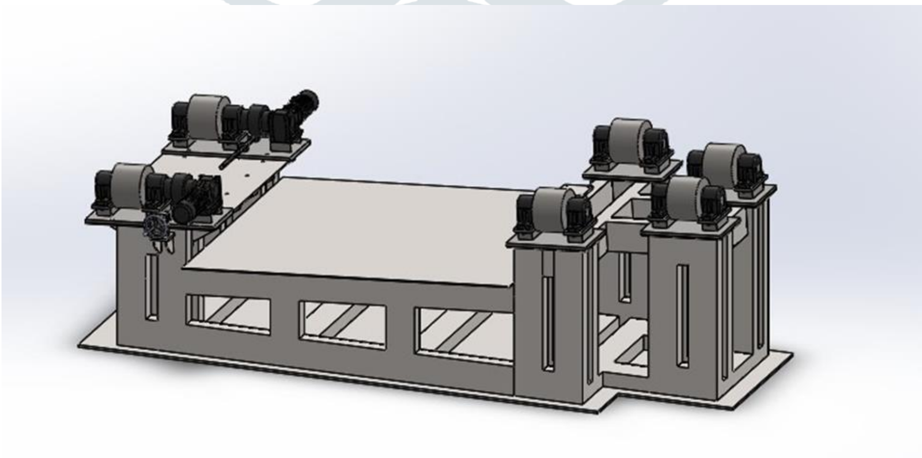

Fig 1: Roller Stand Assembly

4.2 Design of boom, support structure and drive system

The reciprocating boom is required to move along the hardware with a lance mechanism that navigates through the inner surface of the hardware. The boom has been standardized as a rectangular cross-section because it can take better vertical loads compared to a circular pipe. Also the other necessary structures can also be mounted easily in rectangular boom. The boom dimensions have been finalized by calculating various parameters like moment of inertia, sectional modulus, mass and volume for different cross-sections and the optimal dimension as per the requirement has been chosen as 450 mm X 250 mm X 10 mm Thick.

The boom has to be supported by rollers for smooth navigation of boom through the inner surface of the hardware. The unanticipated mass of 500 kg has been considered because of impact loads which can't be expected. The support rollers have been designed considering the fully extended and fully retarded position of the boom. The frictional force is 2198.91 N, by assuming it as a simply supported beam with partial uniformly distributed load, the diameter of the support roller is 49.15 mm. But considering for future augmentations and interchangeability, the diameter is rounded to 60 mm.

A chain drive powered by a geared motor has been selected to accommodate motion to the boom. The boom has to be moved forward and backward with a certain speed of 0.13 to 1.3 rpm. Factor of safety for breaking load is 8, breaking load is the frictional force which is 2198.91 N and number of sprocket teeth as 40 for silent chain. The chain needs to be driven by two sprockets with a center to center distance in this case with support rollers in between. The chain specification is 12B selected by using IS 2403. A geared motor of 0.25 kW is chosen for the drive system. The shaft for sprocket is designed based on both bending and torsion. The diameter of the sprocket shaft is 25.94 mm. Since, there is availability of four rollers of 60 mm diameter, and Plummer blocks for the same, they can be used for the sprocket shaft considering interchangeability as a factor. The chain needs to be provided with attachments which connect the reciprocating boom and chain drive. This attachment has been selected from IS 2403.

The boom drive system support structure acts as a continuity for the power transmission by enabling ease of functionality to withstand the load of drive system members namely geared motor, sprockets, shafts, coupling, base plates, support rollers for the chain drive, Plummer blocks and the weight of the boom. The alignment and movement of boom is taken care by the boom support structure. The column selected for the boom support structure and boom drive system support structure is 100 mm X 100 mm X 8 mm Thick and cross members on the boom drive system support structure is 40 mm X 40 mm X 6 mm Thick.

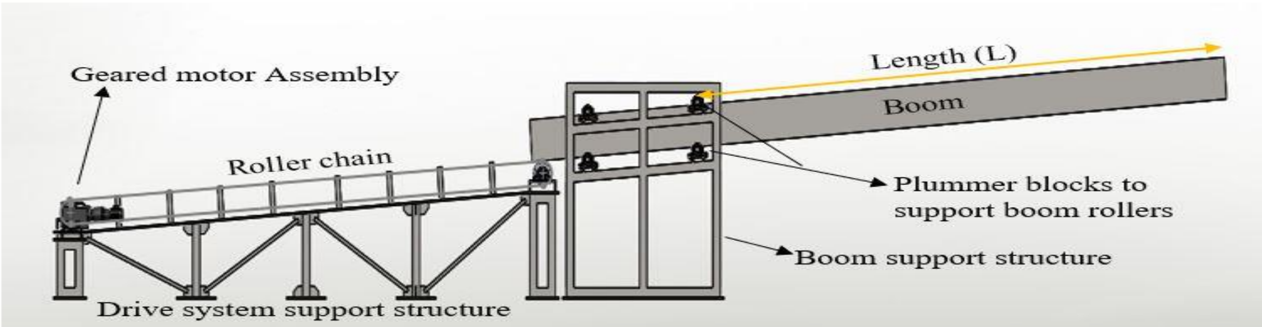

Fig 2: Boom with support structure and drive system support structure

4.3 Design of Lance and actuation mechanism

To perform the cleaning operations, a set of links and a linear actuator must be fixed to the boom. The set of kinematic links has been named as the Lance and to actuate the Lance to and fro, a linear actuator must be provided. This linear actuator provides a linear motion from its output link. The linear motion is then converted to rotary motion by the provided links. Inverse kinematic approach has been followed. Since the end effector position is given as input parameter, and the end link needs to navigate from -135° degree to + 45°, the link lengths decided by using this approach.

End effector's position needs to navigate from -135° degree to +45°. So, the links and joints must be decided in a manner such that the central axis of linear actuator is placed along the central axis of the boom. The inverse kinematic approach may have infinite solutions. One such solution has been identified and solved for link lengths according the requirements. A linear actuator from Duff Norton is selected for moving the links.

4.4 Cabinet

The cabinet is an enclosure which prevents human interaction with cleaning materials and helps in the collection of dust particles. This model doesn't require any design calculations except for the door hinges to withstand the self-weight of the cabinet doors without large deformation and stress. It is also required to recirculate the medium of cleaning materials. The door hinges selected are Steel Butt Hinges from IS 1341 which are heavy duty type.

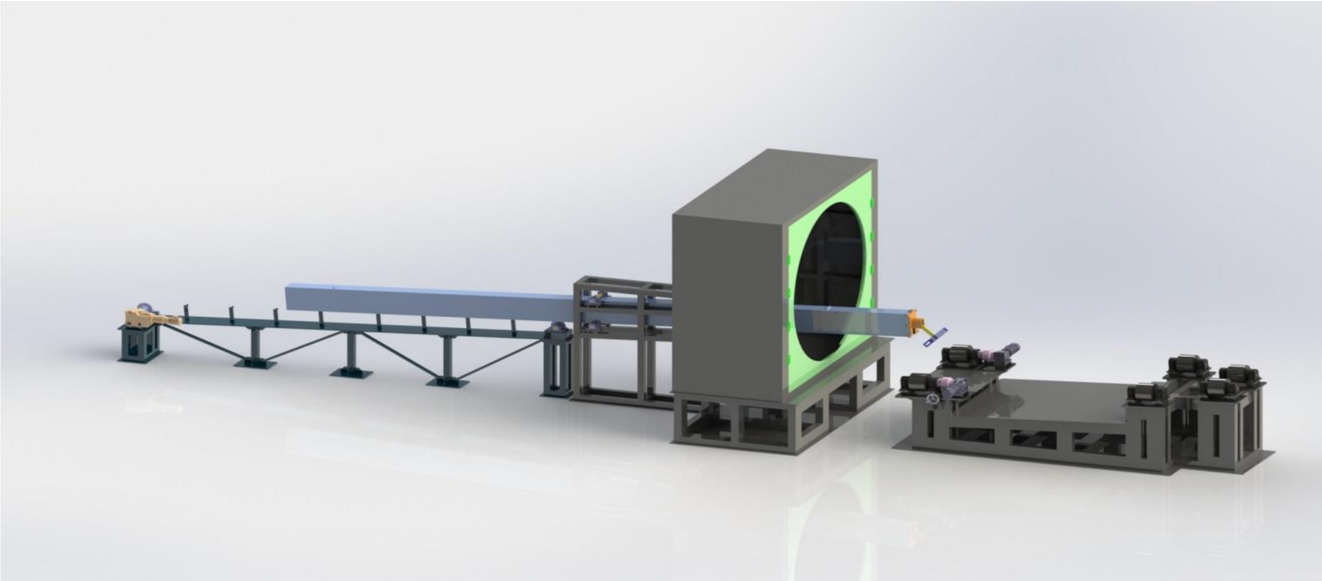

Fig 3: Rocket Hardware Preparation Machine

V. RESULTS FROM ANALYSIS:

Analysis for the structures and components in this machine has been carried out using ANSYS Workbench. The major structural members have been analyzed using the Structural analysis in ANSYS Workbench.

Material used for the whole project is structural steel, so the material properties for all structural analysis are same. Mesh element is tetrahedral and element type is 10 nodded tetrahedral element throughout the analysis of all structures. Supports and loads vary based on the structure.

5.1 Roller stand:

Roller stand has to bear the load of two hardware's, it needs to be analyzed for both of them. As the load of hardware acts on the roller stand at an angle, the loading condition is remote loading. Remote load is applied at the center of gravity of the roller stand by selecting the required faces where it should act. Remote load is 151904 N. The type of support is compression only support.

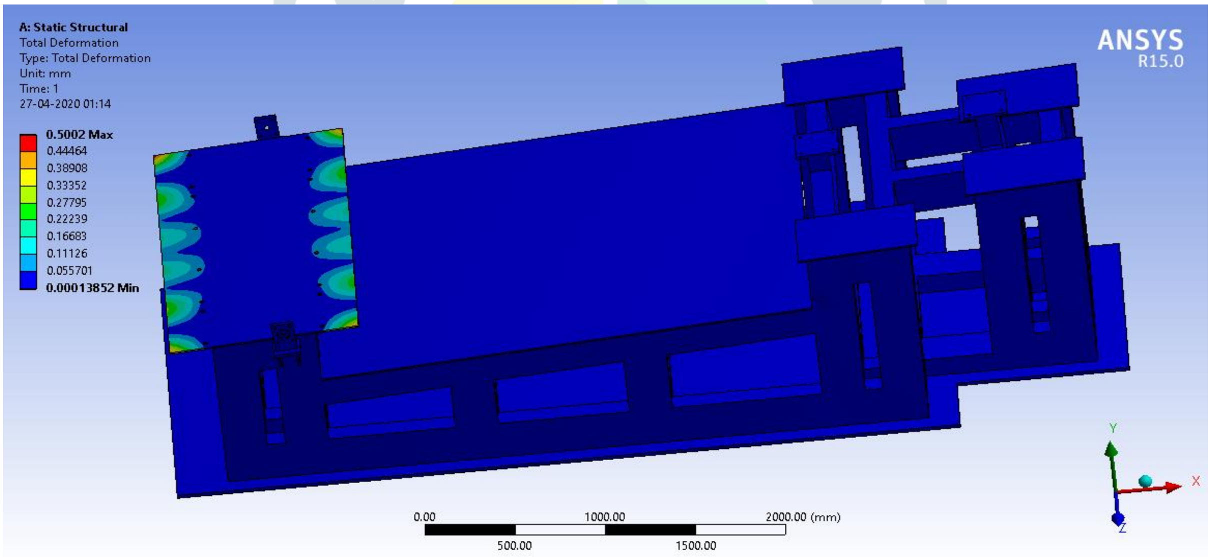

Fig 4: Total deformation when hardware is placed on Roller stand

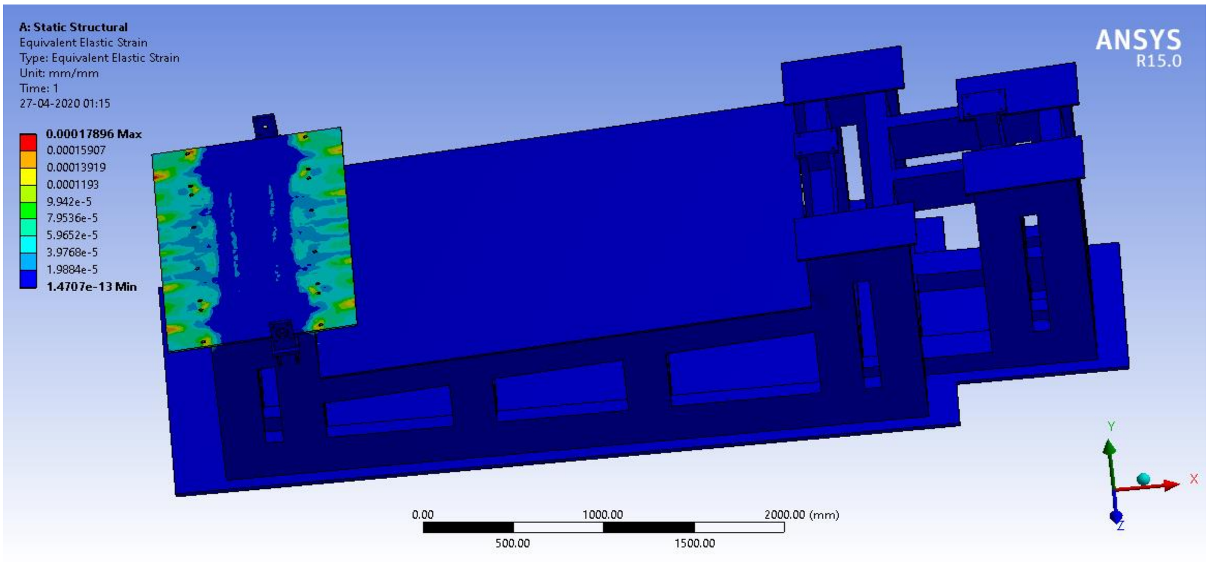

Fig 5: Equivalent elastic strain when hardware is placed on Roller stand

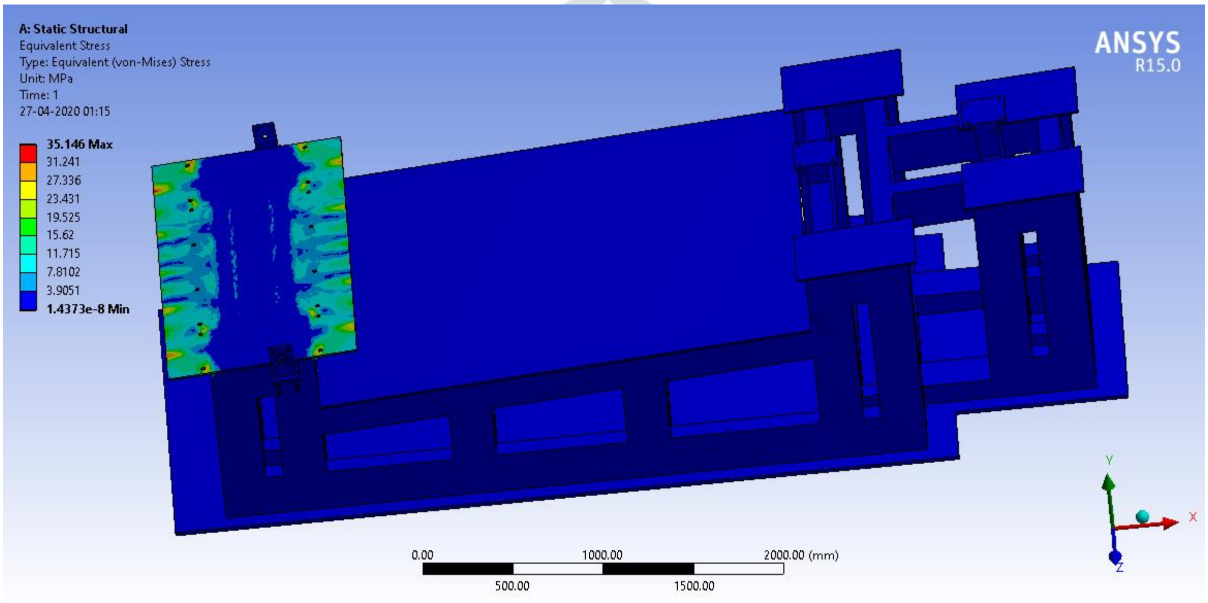

Fig 6: Equivalent stress when hardware is placed on Roller stand

Table 1: Results from Analysis of Roller stand

Result parameter	Maximum value	Minimum value
Total deformation	0.5002 mm	0.00011625 mm
Equivalent elastic strain (von-Mises)	0.00017896 mm	1.235 x 10^{-13} mm
Equivalent stress (von-Mises)	35.146 MPa	1.493 x 10^{-8} MPa

5.2 Skew Arrestor:

Skew arrestor is to bear the load produced due to an inclination of 4 degrees. The skew arrestor has been analyzed by applying the load of 15942 N and boundary condition as fixed support.

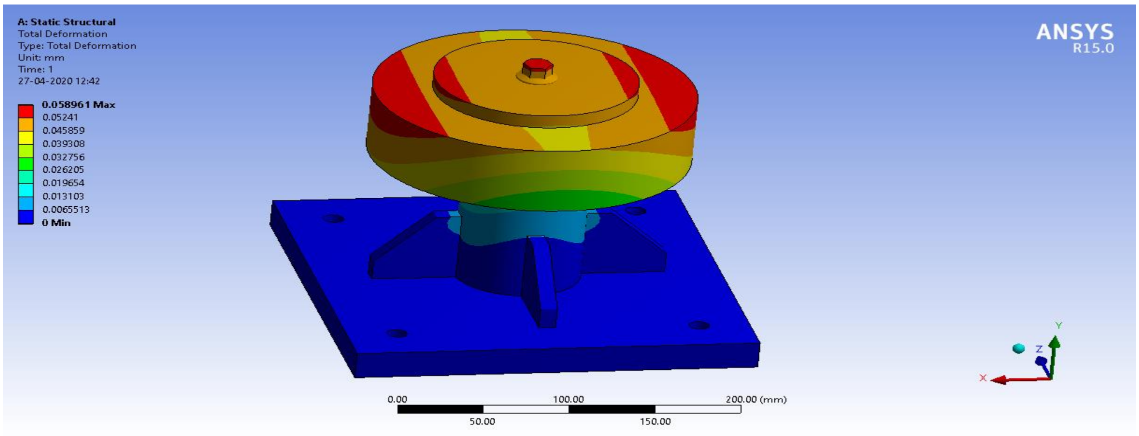

Fig 7: Total deformation of skew arrestor

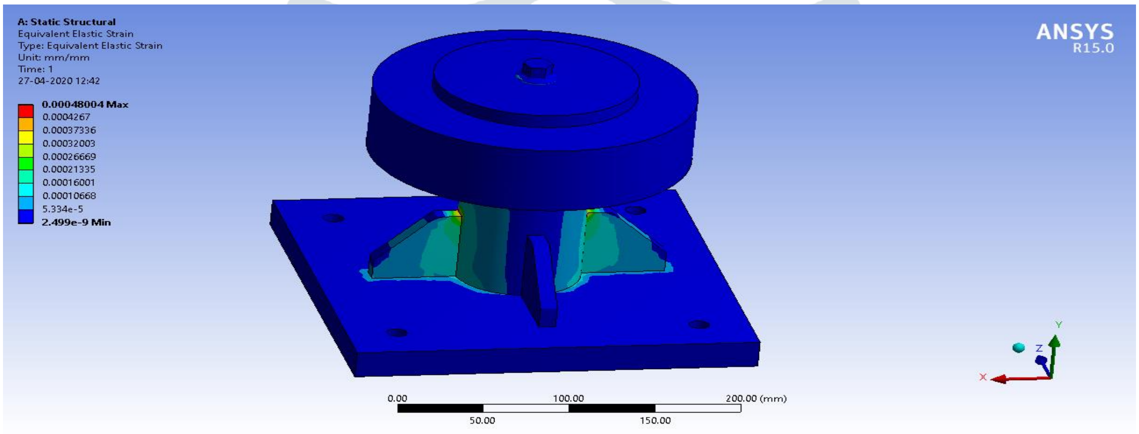

Fig 8: Equivalent elastic strain of skew arrestor

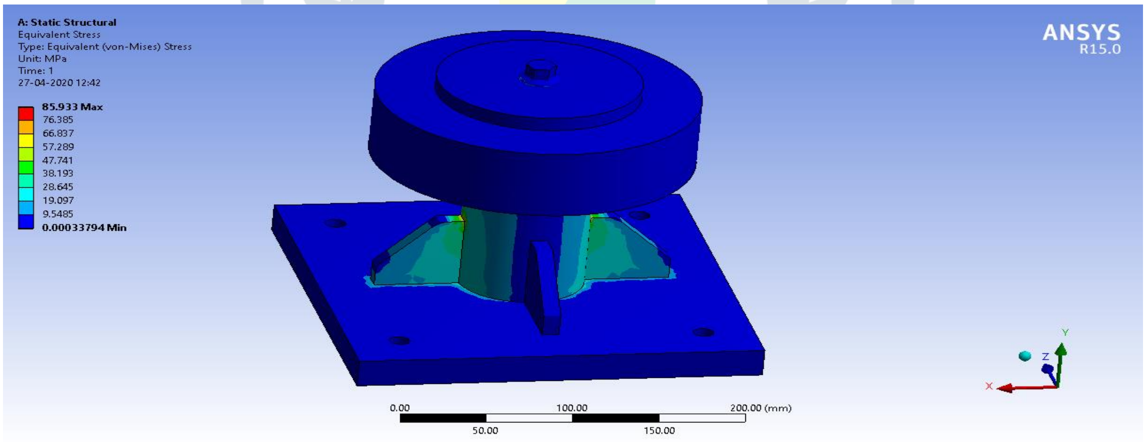

Fig 9: Equivalent stress of skew arrestor

Table 3: Results from analysis of skew arrestor

Result parameter	Maximum value	Minimum value
Total deformation	0.058961 mm	0 mm
Equivalent elastic strain (von-Mises)	0.00048 mm	2.499×10^{-9} mm
Equivalent stress (von-Mises)	85.933 MPa	0.000337 MPa

5.3 Boom:

The structural analysis for boom which reciprocates is important as it carries self-weight. The boundary condition is fixed support and load of standard earth gravity.

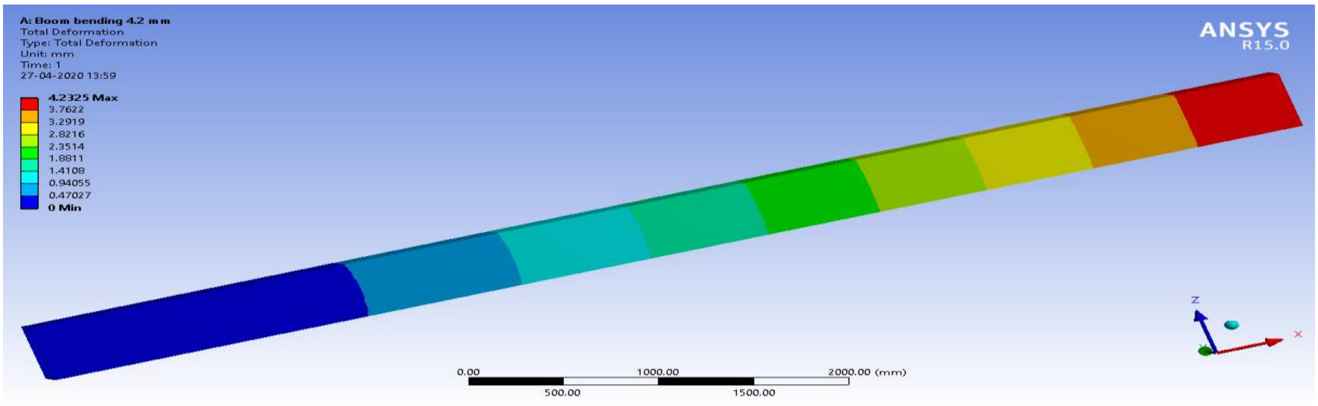

Fig 10: Total deformation of Boom

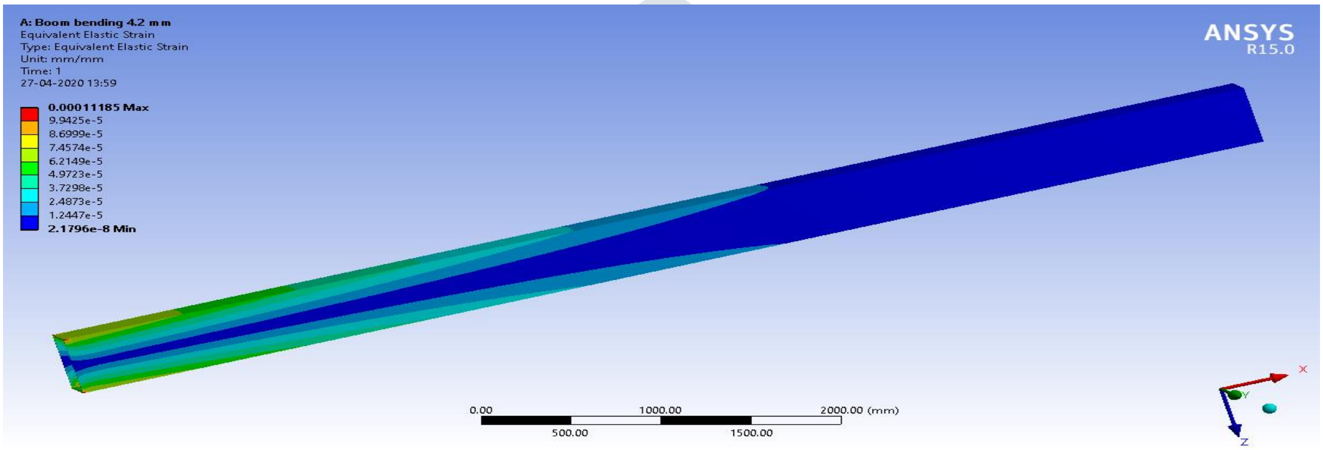

Fig 11: Equivalent elastic strain of Boom

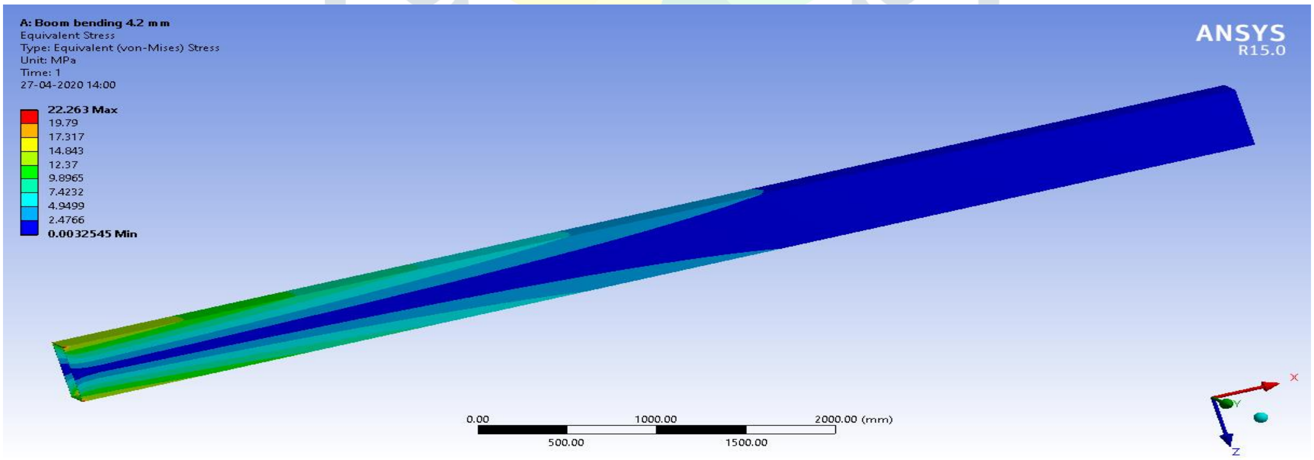

Fig 12: Equivalent stress of Boom

Table 4: Results from analysis of Boom

Result parameter	**Maximum value**	**Minimum value**
Total deformation	4.2325 mm	0 mm
Equivalent elastic strain (von-Mises)	0.000111 mm	2.1796×10^{-8} mm
Equivalent stress (von-Mises)	22.263 MPa	0.0032545 MPa

5.4 Boom Support Structure:

The boom with a certain self-weight is to be placed on a support structure and the support structure is responsible to bear the load during the entire process both during the extended and retracted positions. The actual calculated values of reaction forces have been considered with a safety factor of 1.25.

a. Extended position:

The boundary condition for the boom is compression only support, where the load is applied on the left side roller base on upper base and right side roller base on lower side because when the boom is in extended position, the load will be on those rollers. So the force on lower base is 29553 N and force on upper base is 21117 N.

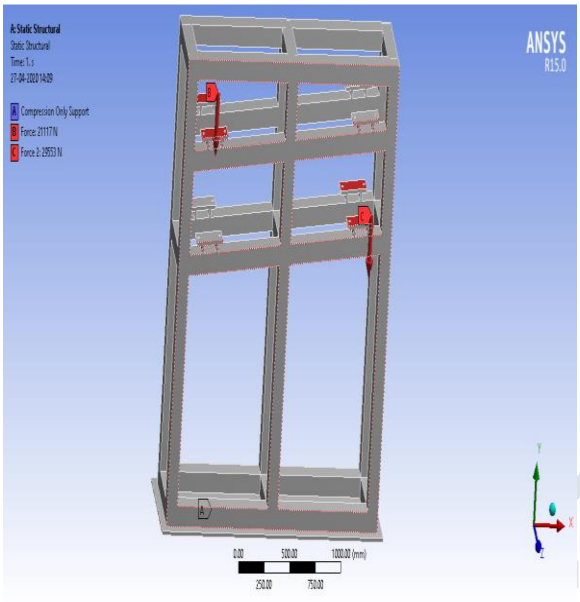

Fig 13: Boundary condition

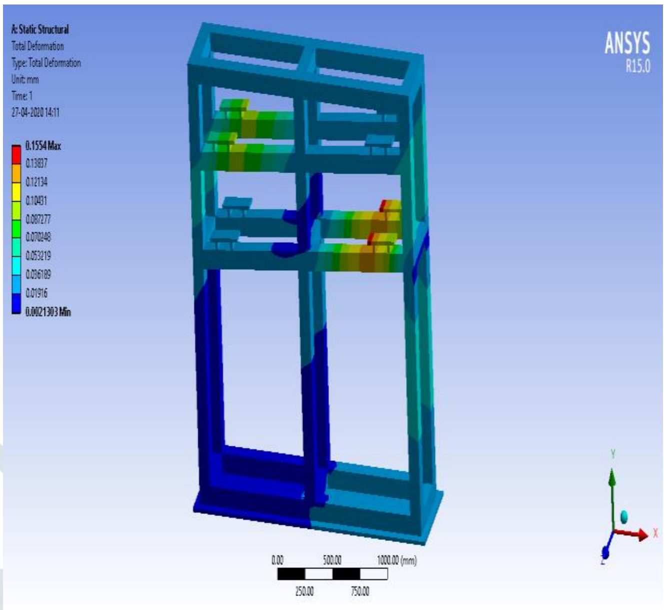

Fig 14: Total deformation

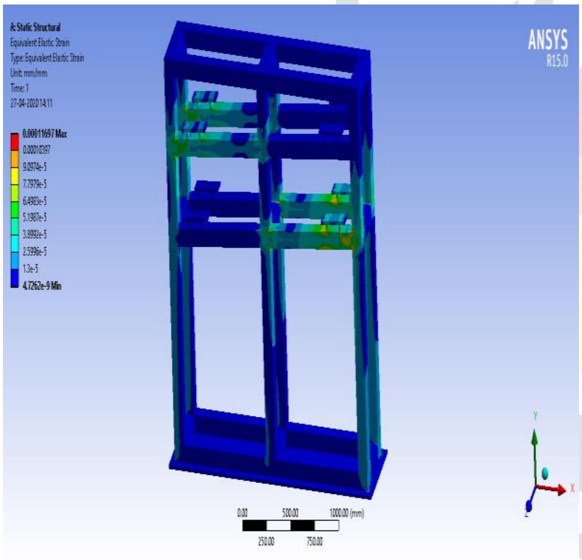

Fig 15: Equivalent elastic strain

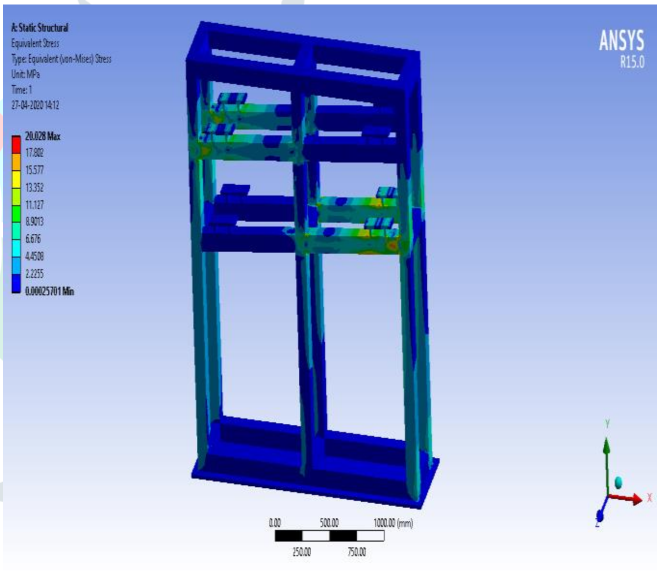

Fig 16: Equivalent stress

Table 5: Analysis results for Boom support structure (extended position)

Result parameter	Maximum value	Minimum value
Total deformation	0.1554 mm	0.0021 mm
Equivalent elastic strain (von-Mises)	0.000116 mm	4.726×10^{-9} mm
Equivalent stress (von-Mises)	20.028 MPa	0.000257 MPa

b. Retarded position:

The boundary condition for the boom is compression only support, where the load is applied on the right side roller base on upper base and left side roller base on lower side because when the boom is in retarded position, the load will be on those rollers. So the force on lower base is 26283 N and force on upper base is 19073 N.

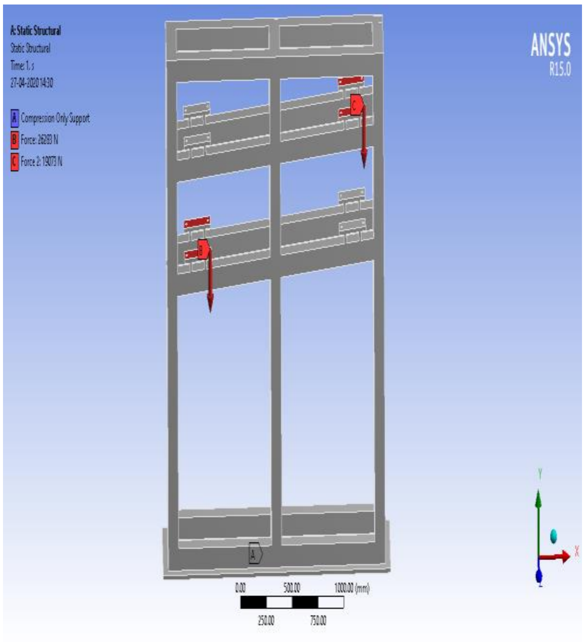

Fig 17: Boundary condition

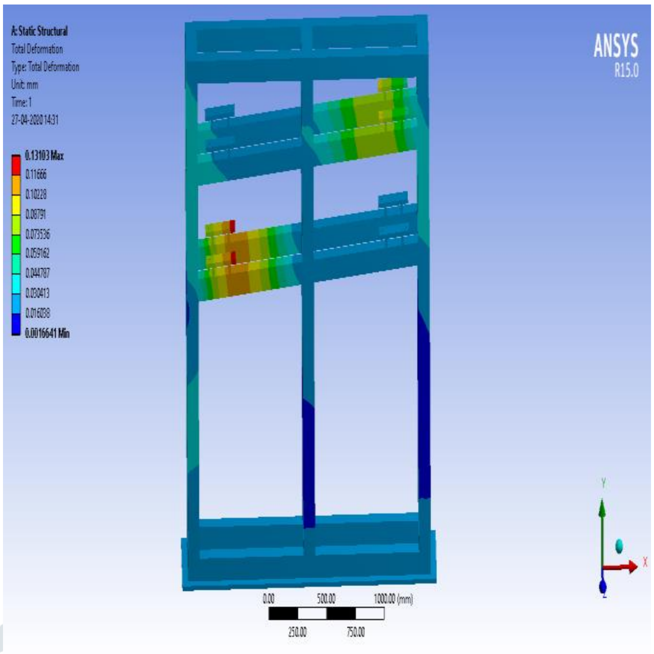

Fig 18: Total deformation

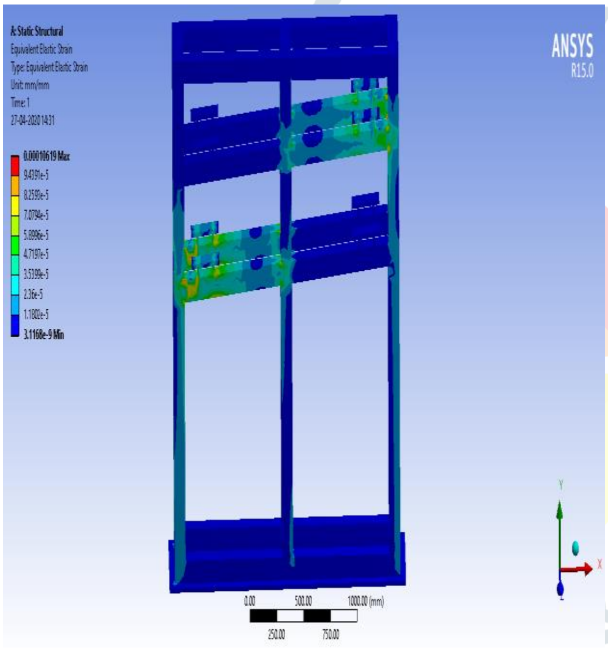

Fig 19: Equivalent elastic strain

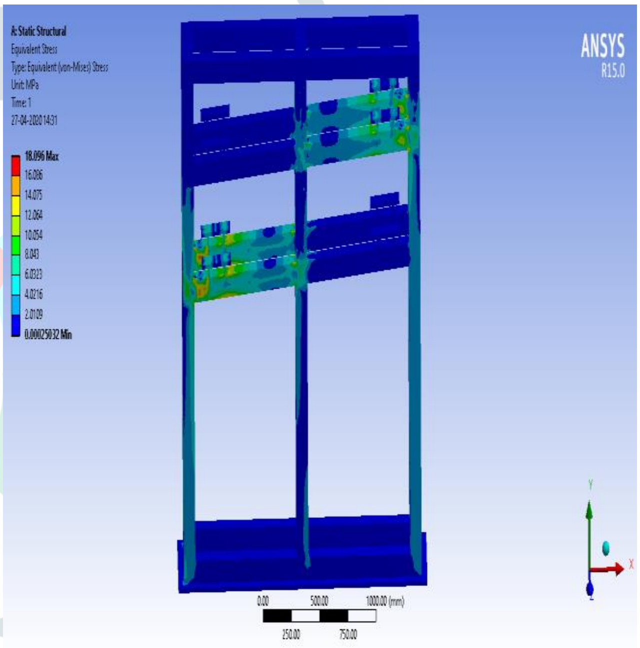

Fig 20: Equivalent stress

Table 6: Analysis Results for Boom support structure (retracted position)

Result parameter	Maximum value	Minimum value
Total deformation	0.13103 mm	0.00166 mm
Equivalent elastic strain (von-Mises)	0.000106 mm	3.1168 x 10^{-9} mm
Equivalent stress (von-Mises)	18.096 MPa	0.000250 MPa

VI. CONCLUSION:

This project has resulted in the Design and analysis of a rocket hardware preparation machine which can handle two hardware. The resulting machine has the following sub systems.

- Roller Stand assembly
- Boom Assembly with support structure
- Boom drive system with support structure
- Lance and actuation Mechanism

The Roller stand Assembly has been designed and analysed which is used for the placement and rotation of hardware. The design and analysis of boom assembly which navigates through the inner surface of hardware carrying the Lance and actuation mechanism has been completed. The boom support structure is responsible for supporting the reciprocating boom at any position. The boom drive system is responsible to move the boom to and fro using a chain drive and a geared motor. The support structure for drive system is responsible in withstanding the loads of the components of drive system. The lance with actuation mechanism is a combination of kinematic links attached to a linear actuator which can perform cleaning operations of hardware. The 3D modeling has been done by using SOLIDWORKS and analysis by ANSYS Workbench.

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