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DESIGN OF MOTOR CASING FOR SOLID PROPELLANT ROCKET AND EVALUATION BY ANALYTICAL AND NUMERICAL ANALYSIS COMPARISION.

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ABSTRACT
In aerospace,motor case design plays an important role as it shields the motor from external factors and also it should withstand high temperature and pressure generated by the motor. This project deals with the design of solid rocket motor casing mainly consists of determining the thickness of motor casing which includes the domes at head and cylindrical end welded joints. Modeling of solid rocket motor casing is done in CATIAV5R19. The structural analysis is done for design with different materials like maraging steel,D6AC Steel,austentite and martensitic steels by means of ANSYS 14.5.The design with the best deformation properties of considered materials is considered after comparison and the results are concluded accordingly. Also, comparison is done for Von-Mises stresses, Hoop and Longitudina/Axial stresses for evaluation of both numerical and analytical analysis of the model.

KEYWORDS: Maraging steel,D6ACSteel,austentite steel,martensitic steel,Von Mises stress,Hoop stress

INTRODUCTION
All the types of rocket use some or the other type of propellants until new types were introduced in the form of hybrid and liquid propellant rockets.Solid rockets are most commonly used compared to other forms mainly because it is easier to construct,maintain and is more reliable comparatively.Solid propellant rockets can be stored for large period of time and are more commonly used for military purposes.However, their performance characteristics are quite poor compared to liquid propellant rockets and hence are not the preferred choice for initial propulsion in any of the launch vehicle carrying large amount of payloads to the outer orbits.The main objective is to produce a motor casing by using basic sheet metal and assembly by means of welding.The main characteristics to be considered at the time of motor design are the characteristics of the type of motor used,type of materials used based on the yield strength of the design which is calculated and case design considerations for evaluations like case loads,deformation,stresses,structural analysis etc.Also, the case is designed to satisfy the performance requirements of the motor.A simple solid propellant rocket motor mainly consists of ignitor,insulator,grains,nozzle.The grain acts as the fuel in this case by igniting upon the sparks generated by the ignitor there by producing large mass of thrust through the nozzle upon combustion.The insulator acts as a sepration medium between the grain and the casing assembly.The gases passed through the nozzle is of the very high pressure range varying from 10-200 Bars.

LITERATURE REVIEW
Different researchers have discussed the design and analysis of solid propellant rocket motor casing in numerous ways.They are summarized below:

ASME PRESSURE VESSEL code section VIII division 1 gives the formula for determining the thickness for the hemispherical heads and also the formula for determining the thickness of the cylindrical sections of the casing.

Roy Hartfield In their A Review of Analytical Methods for Solid Rocket Motor Grain Analysis presents the Analytical methods for solid rocket motor grain design isproving to be advantageous to some recent studies to develop solid-rocket propelled missiles. The analytical approach is not favoured in recent years,however for some grain types the analytical methods are more favouring. Here a review for analytical methods of calculating burn area and port area for a variety of cylindrically perforated solid rocket motor grains. This set of geometries represent a variety of hopes for two-dimensional grain design.

NASA SP-8025 has given the details about material characteristics of various solid rocket motors. Based on these material properties certain materials are short listed for consideration in the case design.NASA has given the indepth understanding of the solid propellant rocket motor casing design review and structural analysis of the motor factory joint.Structural analysis is carried out to verify the structural stability of the solid rocket motor at certain working temperature. NASA has given the solid propellant performance prediction and analysis. Based upon this the performance the design is done by undertaking the loads that are acting on the solid rocket motor casing.

Siva Sankara Raju R In their Design and Analysis of Rocket Motor Casing by Using Fem Technique. studies the design of motor casing by determining the thickness of motor casing which contains the domes at head end, nozzle end and flange for bolted joints. Design of motor casing and its assembly is done in CATIAV5R19. Stress distributions is developed because of effect of working stress developed in the assembly. The max working stress is compared with allowable yield stress of the material. Final conclusion brings out a well modelled solid rocket motor for the effective holding of propellant for getting the required impulse. 2D Axi- Symmetric structural analysis for rocket motor Casing is carried out to determine the stress level of all components using ANSYS 12.0.

Mohamad Izwan Ghazali In their Design Fabricate and Testing Small Rocket Motor discussed the study on Solid Rocket Motor propellant. This project deals with the study of solid rocket motor characteristics including the types of the design consideration and manufacturing, analysis using static thrust testing. There are two main factors that need to be considered in the design selection and manufacturing which are performance and mechanical strength. The theoretical performance of the propellant was determined by using CHEM program.

MATERIALS AND METHODS

Materials

The materials considered for this paper are on the basis of the yield strength being more than 1500 Mpa and the materials considered are given in the below table along with some of their main properties:

Table 1. Material Properties

Material	Maraging Steel	D6AC Steel	Martensitic steel	Austenitic steel
Density (lb/in3)	0.29	0.28	0.28	0.29
Yield tensile strength (Mpa)	2300	1764	1947	2147
Young's modulus (Gpa)	210	208	195	190
Poisson's ratio	0.3	0.3	0.3	0.29

Case design assumptions and Design calculations

1) Design Pressure is assumed to be uniform throughout the inner casing.
2) Ultimate tensile strength of the material= 981 MPa (allowable stress)
3) Maximum expected operating pressure (MEOP)= 4.8 MPa
4) The design safety factor is specified = 1.25.
5) The motor case cylinder diameter D = 800mm.
6) Design Pressure(P) = MEOP X design safety factor=4.8 X 1.25 = 6 Mpa
7) According to ASMEfor hemispherical head thickness is calculated by,

$$t = \frac{PR}{2SE - 0.2P}$$

Where,L=Head Radius(mm)=400mm,S=Ultimate tensile strenght=983 Mpa.Therefore,thickness=2.44 mm

8) According to ASME for shell thickness calculation is given by

$$T_s = \frac{P \times R}{(2 \times S \times E + 0.4 \times P)}$$ Where, E=weld efficiency=1

9) The Max thrust/Force at which deformation takes place = 75000N
10) Propellant considered is ammonium perchlorate composite propellant.
11) Specific Impulse(Isp) = 303 s
12) Exhaust jet velocity(Cj) = specific impulse X acceleration due to gravity = 2972.43m/s
13) Propellant mass flow rate(Mp) = Thrust/Exhaust jet velocity = 25.23 N/m

Drawing and Model

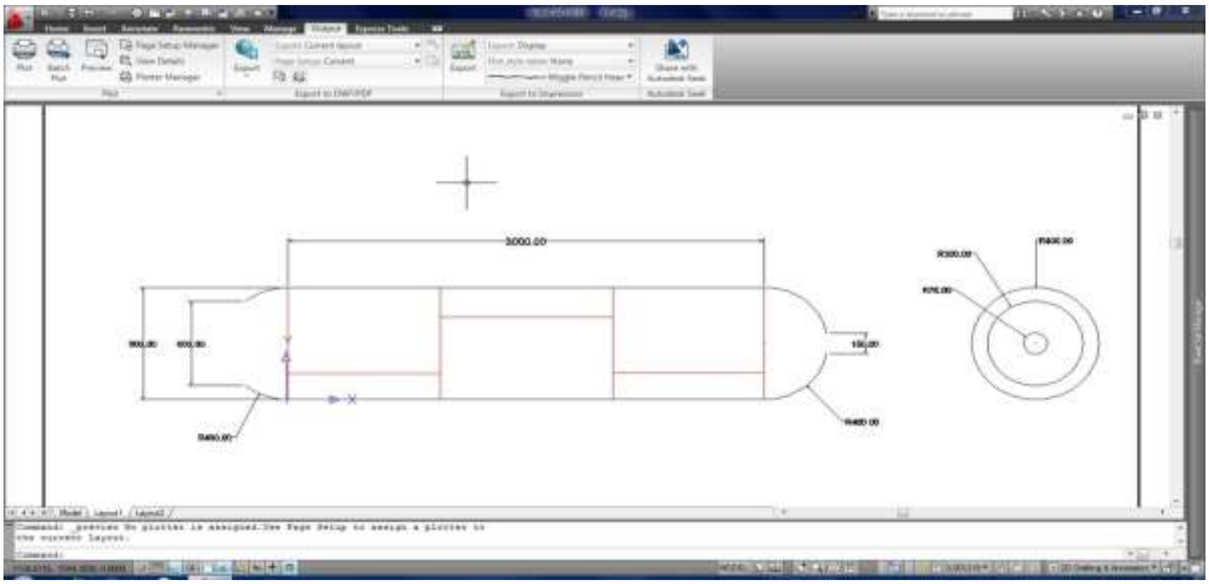

Figure 1: 2D drawing of the motor casing

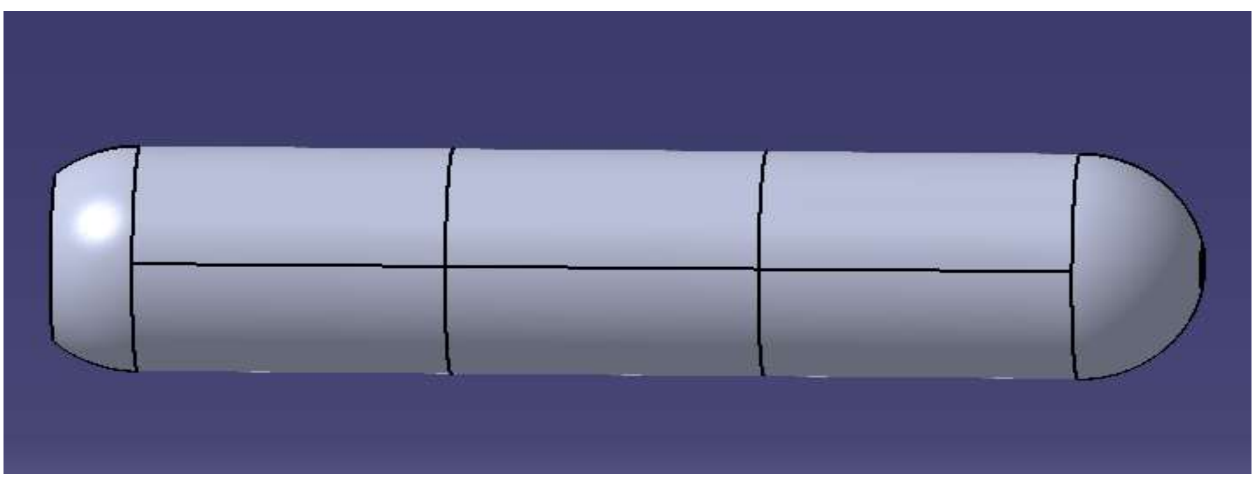

Figure 2: 3D Model of the motor casing in CATIA V5

Load distribution

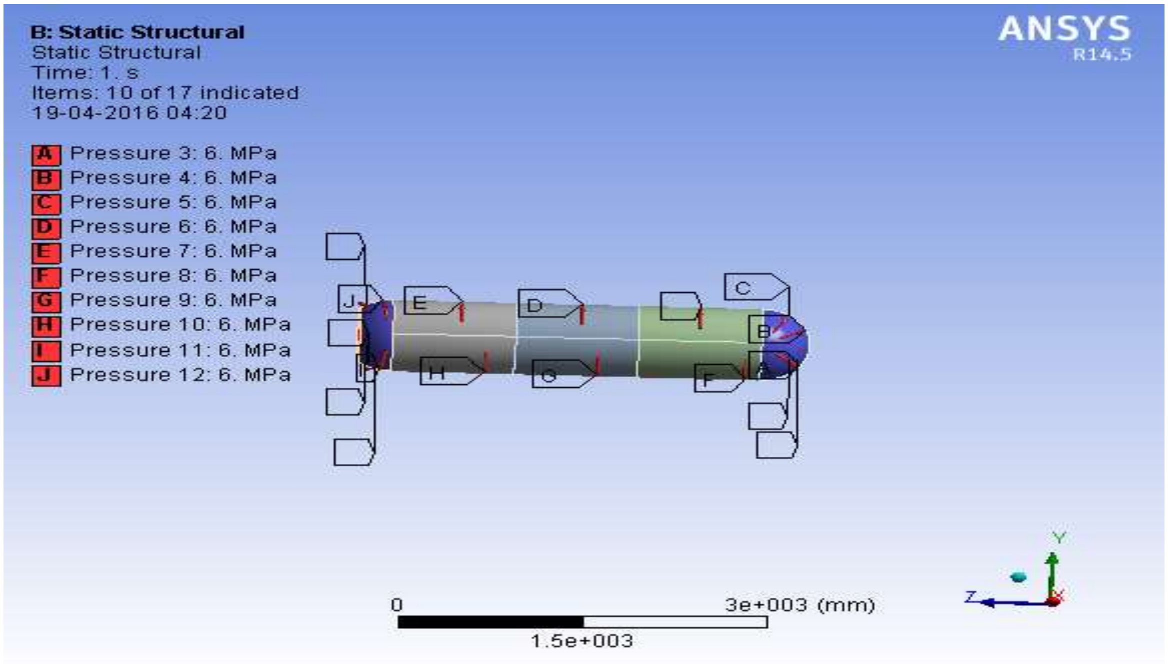

Figure 3: Pressure distribution inside the motor casing.

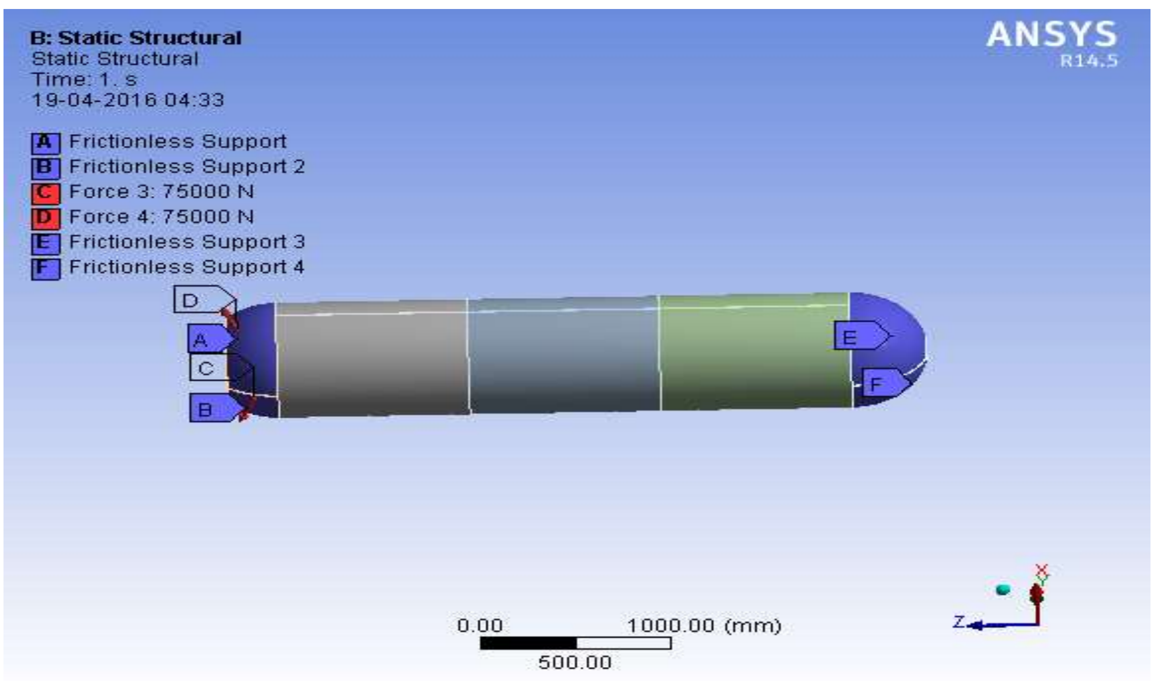

Figure 4: forces and supports.

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ANALYTICAL AND NUMERICAL ANALYSIS COMPARISION

Equivalent(Von Mises) stresses analytical and numerical analysis comparision

Von mises stress is mainly used to determine whether a design can withstand a given load condition.In simpler terms it can be used to determine that beyond a certain load a material tends to fail.This concept mainly arises from the distortion energy failure theory.This theory is based on the comparision mainly between two types of energies.i.e. distortion energy of the actual case and distortion energy of simple tension case in times of failure.The following figure gives the von mises stress for the motor casing.

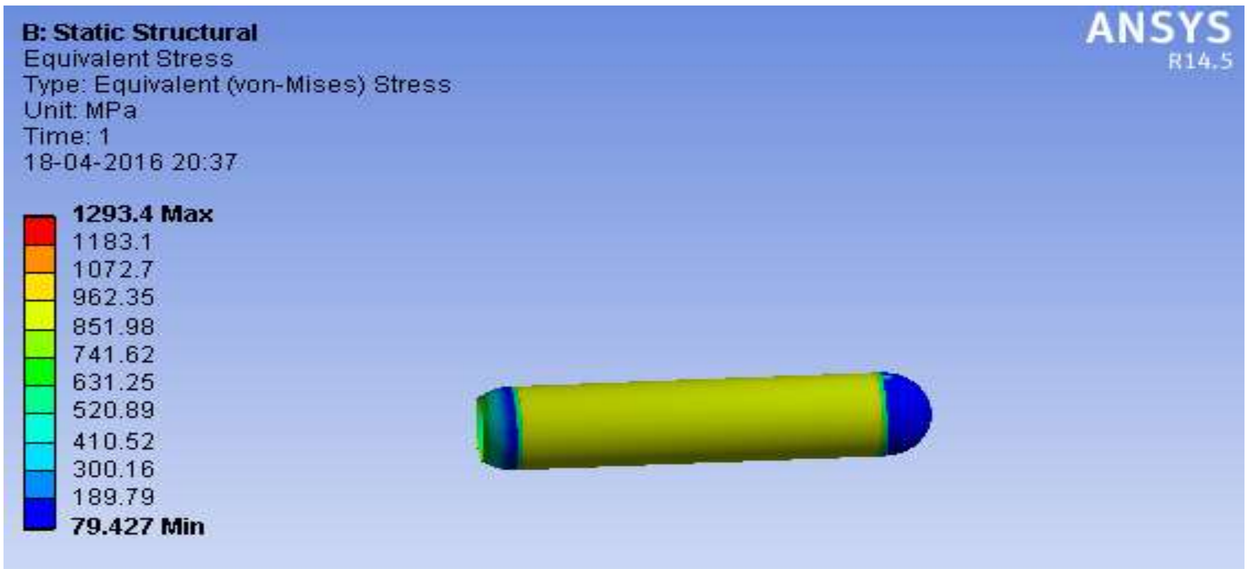

Figure 5: Equivalent(Von Mises) Stress

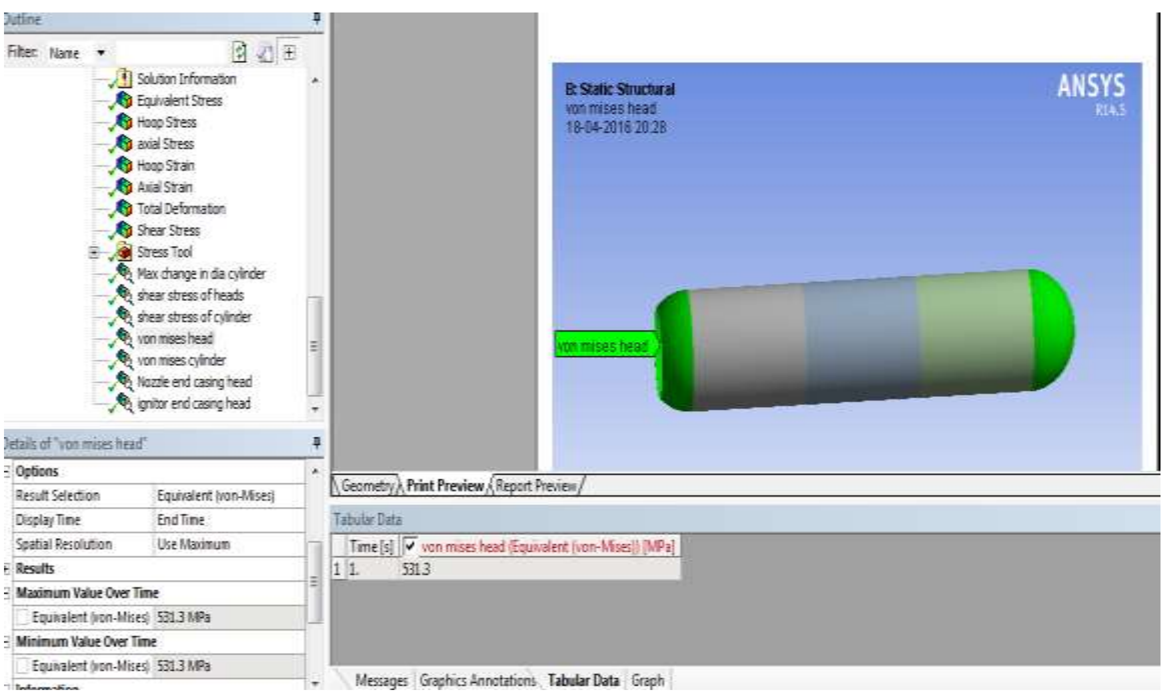

Figure 6: Overall Equivalent(Von Mises) Stress probe in head/Domes = 531.3 Mpa

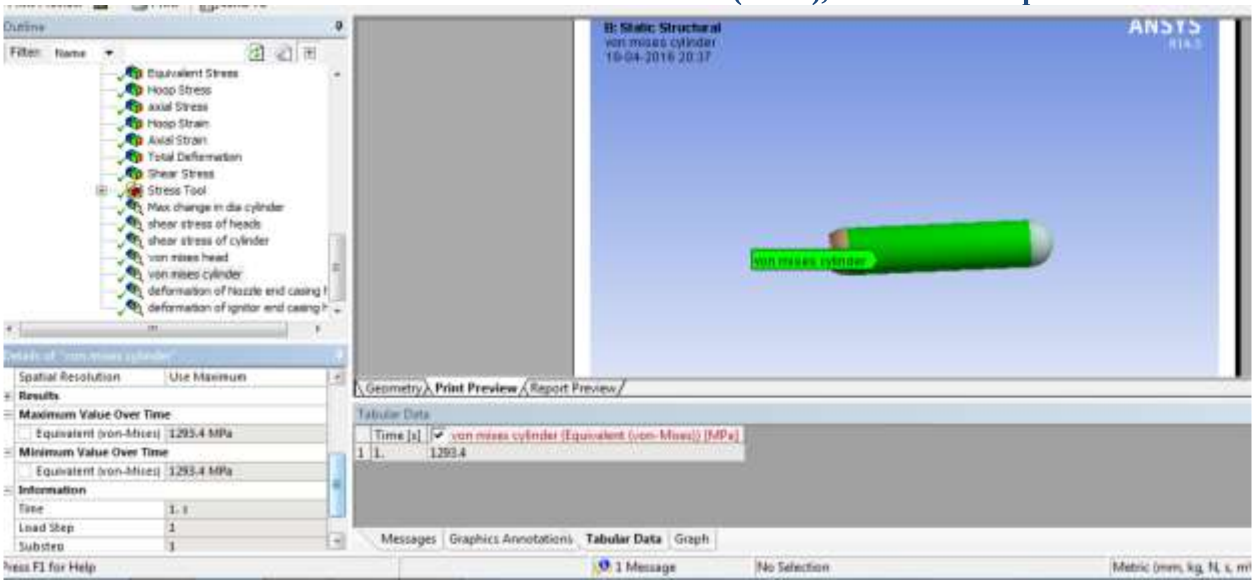

Figure 7: Overall Equivalent(Von Mises) Stress probe in cylinder = 1293.4 Mpa

1) Numerical Von Misses Stress for hemispherical Head/Domes

$$t = \frac{Pd \cdot Di}{(4\sigma all \cdot \eta) - 0.4Pi}$$

Upon calculation von mises stress in hemispherical Head/Domes = 515.15 Mpa

2) Numerical Von Misses Stress for cylindrical region

$$\sigma t = \frac{Pi(Do^2 + Di^2)}{(Do^2 - Di^2)}$$

Upon calculation von mises stress in cylindrical region = 2150.3 Mpa

Where,

P=Pi *1.05 = 6 * 1.05 = 6.3,

Do= Outer Dia = 800mm,

Di= Inner Dia = 797.66 mm,

ή= Weld Efficiency = 1

Table 2. Von mises stress comparision table.

TYPE OF STRESS	ANALYTICAL VALUE	NUMERICAL VALUE
Von Mises stress in cylindrical region	1293.4Mpa	2150.3Mpa
von mises stress in hemispherical head/Domes region	531.3Mpa	515.15Mpa

Hoop stress analytical and numerical analysis comparision

In mechanics, a stress distribution with rotational symmetry is considered as cylindrical stress.The cylindrical stress patterns include Hoop stress or circumferential stress a stress which is normal in the tangential direction.

This stress always acts in the X direction,but the forces acting in the Z direction can be avoided as their values will be negligibly small.Hence,Thin walled cylinders/Pressure vessels ususally wont have radial stresses as they are ignored.Hoop stress distribution by analytical evaluation is given in the below figure.

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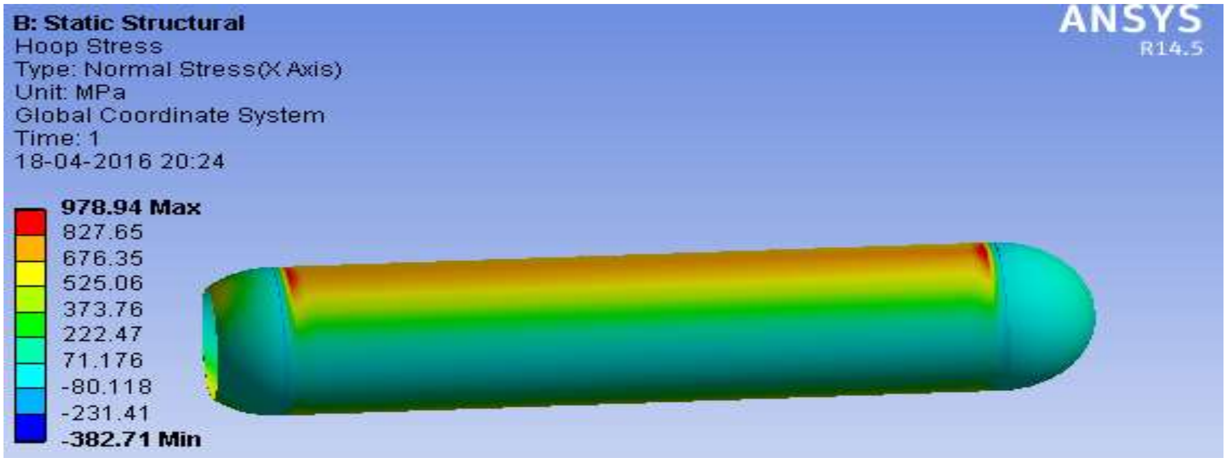

Figure 8: Max Hoop Stress= 978.94Mpa

The cylindrical stress patterns include Hoop stress or circumferential stress a stress which is normal in the tangential direction.For thin walled motor casing/pressure vessel the numerical formula for Hoop stress considered for numerical analysis and value obtained after substitution is

$$\frac{PD}{2t} = 983.63 \text{ Mpa}$$

Longitudinal/Axial stress analytical and numerical analysis comparision

Axial stresses are usually normal stresses which are usually parallel to the axis of cylindrical symmetry.In simple words these are the stresses which are usually acting along the Y direction of any model.Here axial stresses are also considered as longitudinal stress as the radial stresses are ignored as they have negligbly low values.Also these values are usually half of hoop stress based on the formula and the below figure shows the axial stress distribution in the motor casing having thin walls.

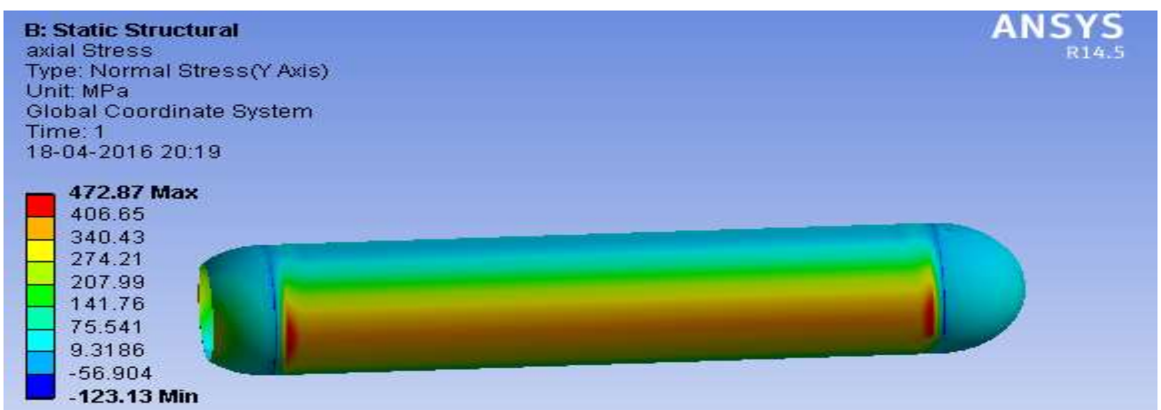

Figure 8: Max Hoop Stress= 472.87Mpa

For thin walled motor casing/pressure vessel the numerical formula for axial/Longitudinal stress considered for numerical analysis and value obtained after substitution is

$$\frac{PD}{4t} = 491.81 \text{ Mpa}$$

Table 3. Hoop and axial stress comparision table

TYPE OF STRESS	ANALYTICAL VALUE	NUMERICAL VALUE
Hoop Stress	978.94	983.63
Longitudinal/Axial Stress	472.87	491.81

Deformation of model using different materials by analytical and numerical analysis comparision

Analytical analysis of Maraging steel

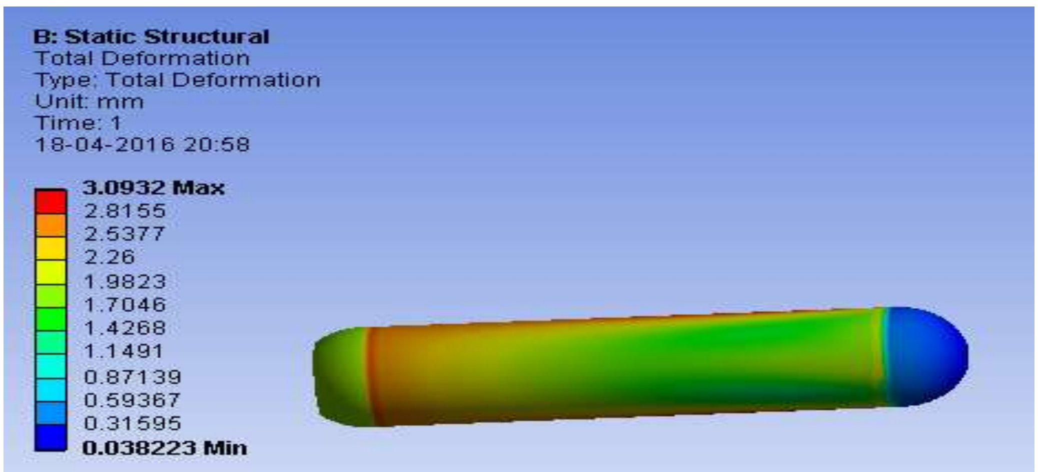

Figure 9:Maximum change in diameter/Deformation of cylinder=3.0932mm

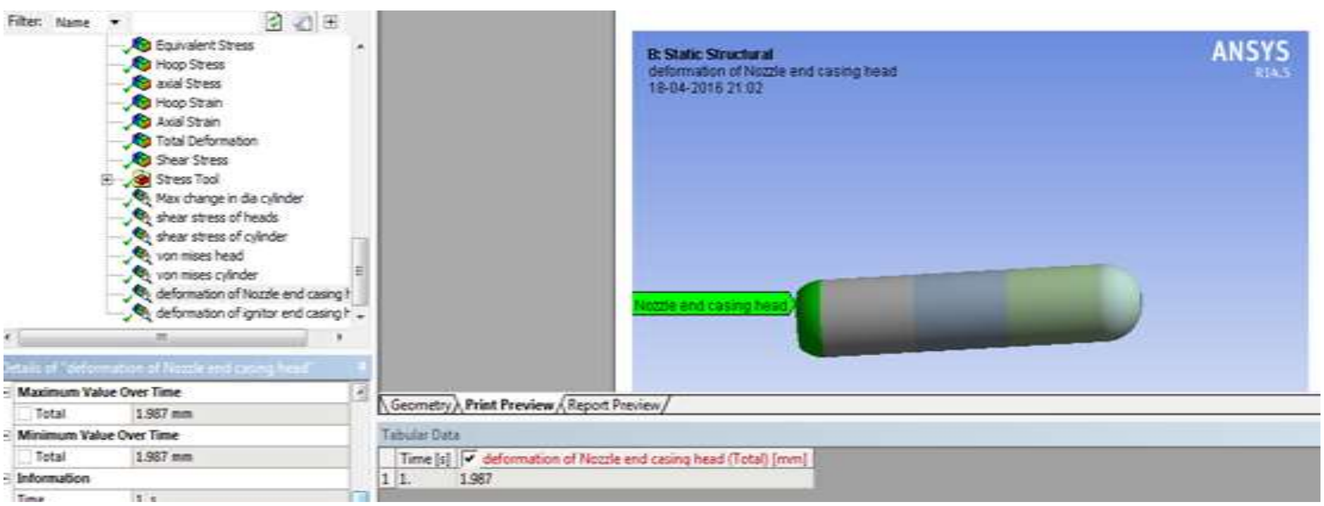

Figure 10:Change in length/Deformation of head/Dome at the nozzle end=1.967mm

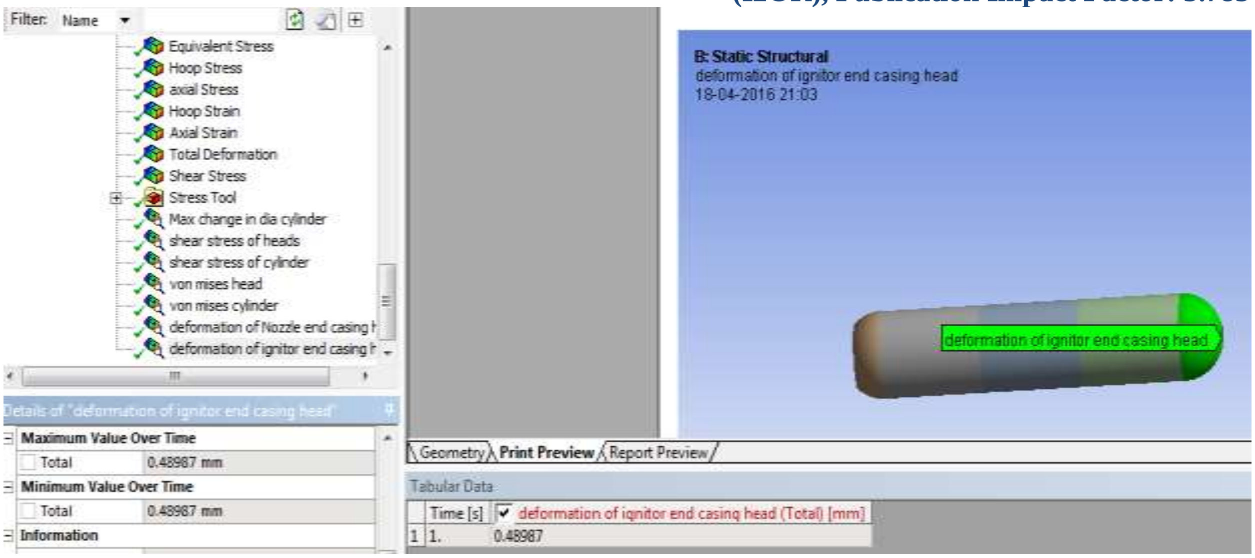

Figure 11:Change in length/Deformation of head/Dome at the ignitor end=0.489mm

Analytical analysis of D6AC Steel

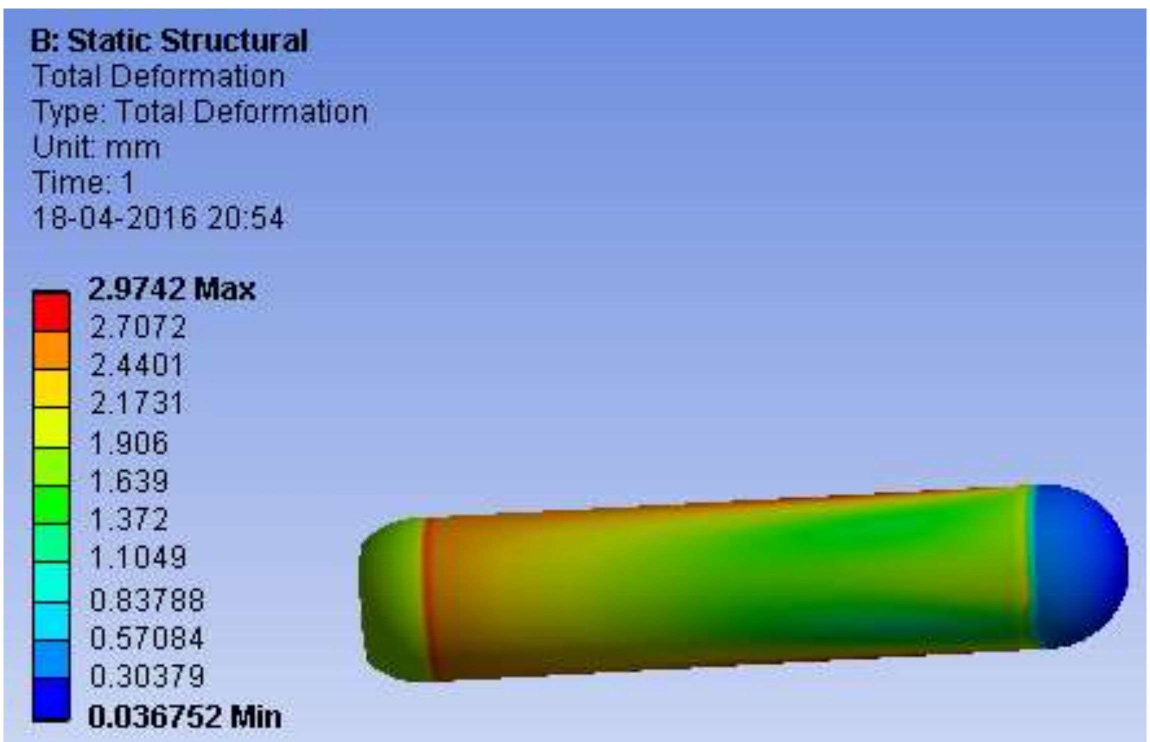

Figure 12:Maximum change in diameter/Deformation of cylinder=2.9742 mm

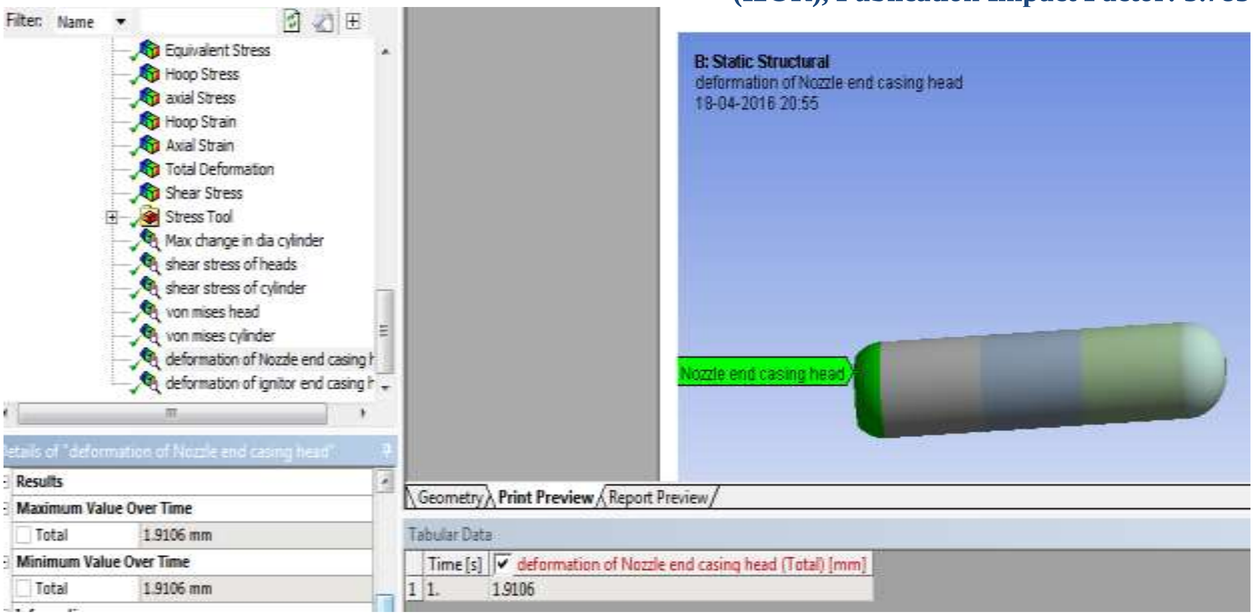

Figure 13:Change in length/Deformation of head/Dome at the nozzle end=1.9106mm

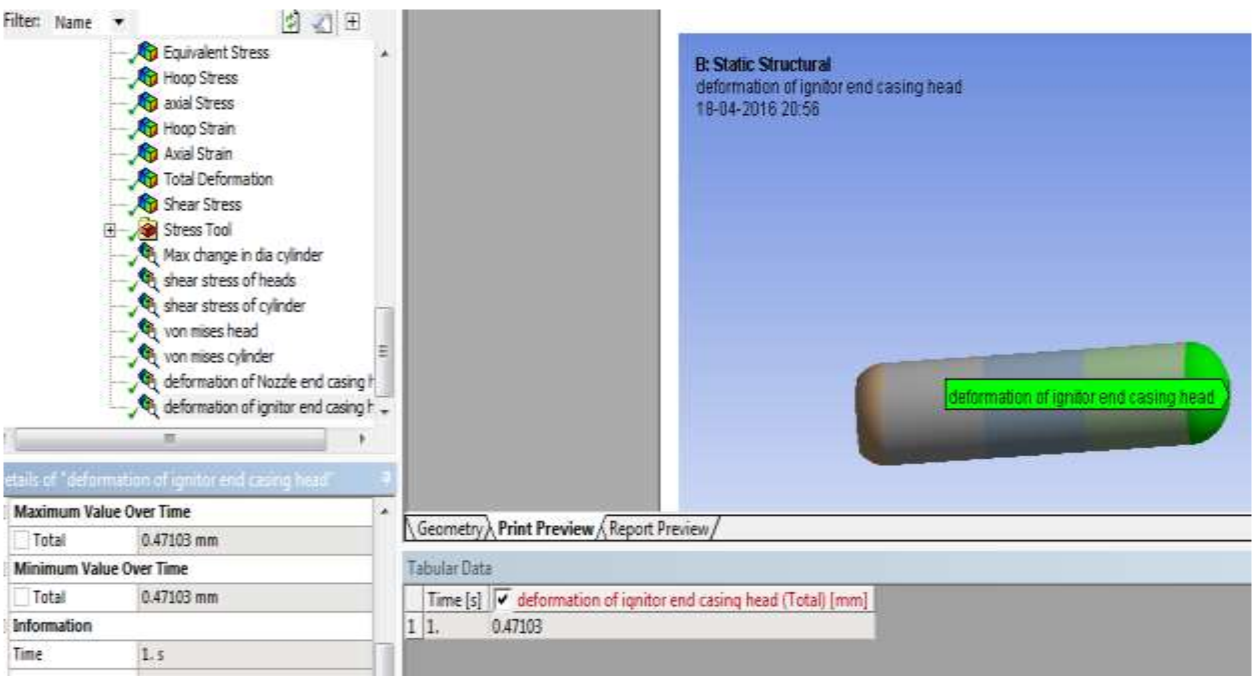

Figure 14:Change in length/Deformation of head/Dome at the ignitor end=0.47103mm

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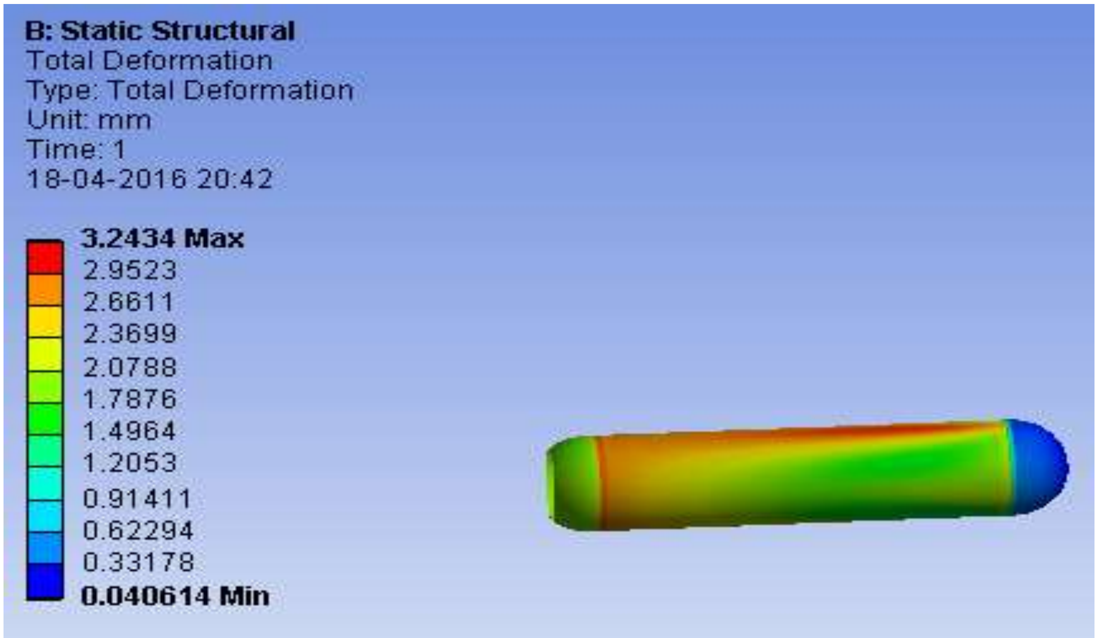


Figure 15:Maximum change in diameter/Deformation of cylinder=3.2434 mm

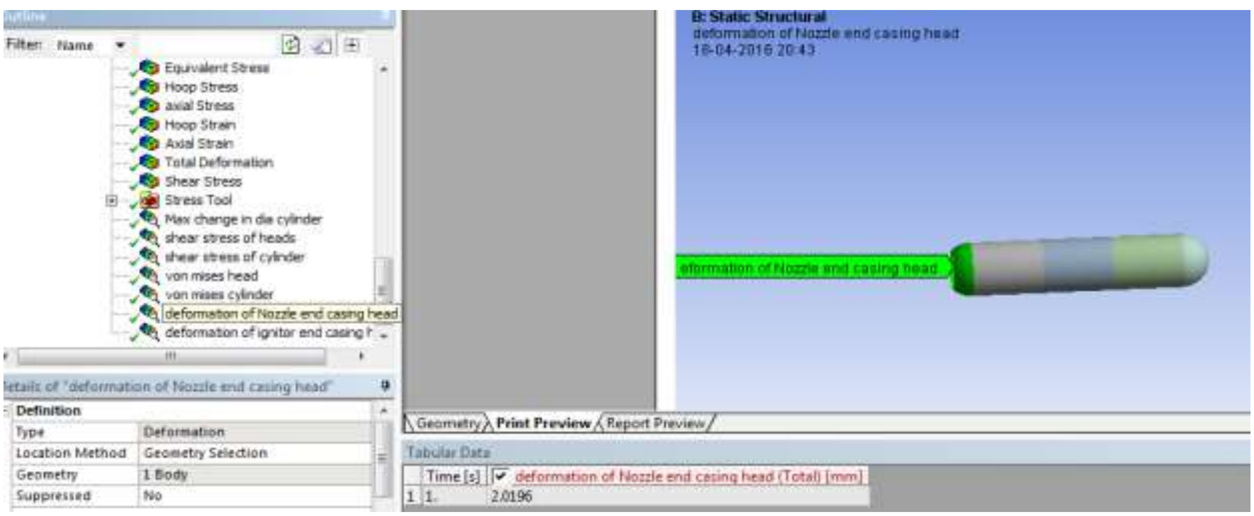


Figure 16:Change in length/Deformation of head/Dome at the nozzle end=2.0196mm

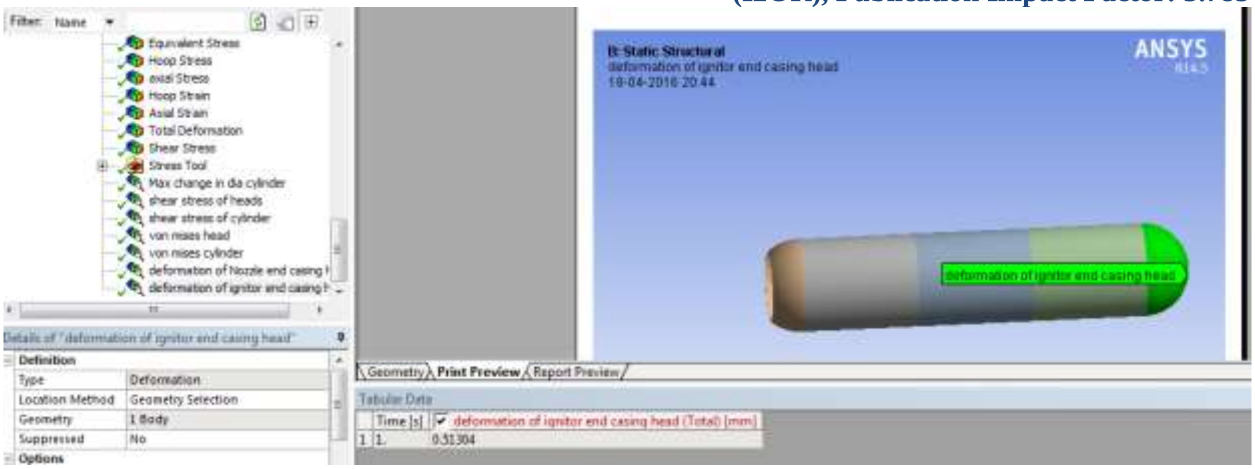


Figure 17:Change in length/Deformation of head/Dome at the ignitor end=0.5130mm

Analytical analysis of Martensitic steel

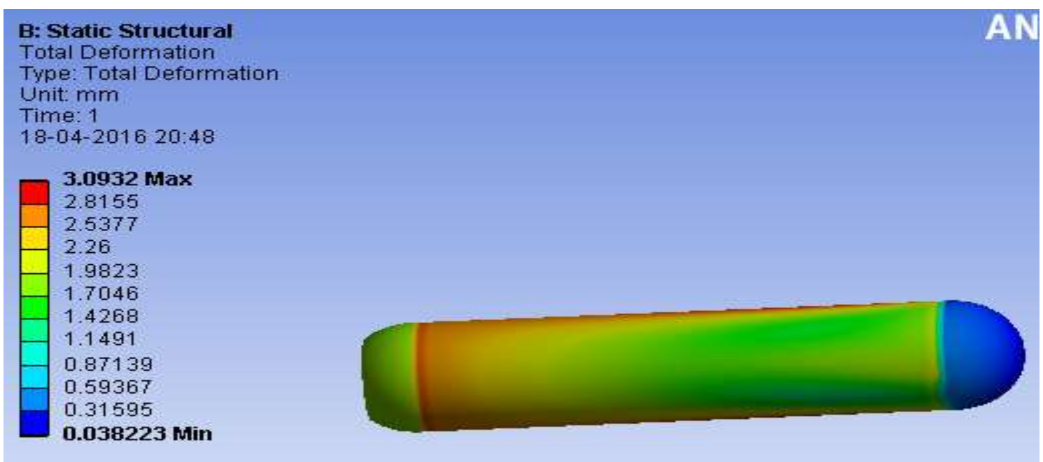


Figure 18:Maximum change in diameter/Deformation of cylinder=3.0932 mm

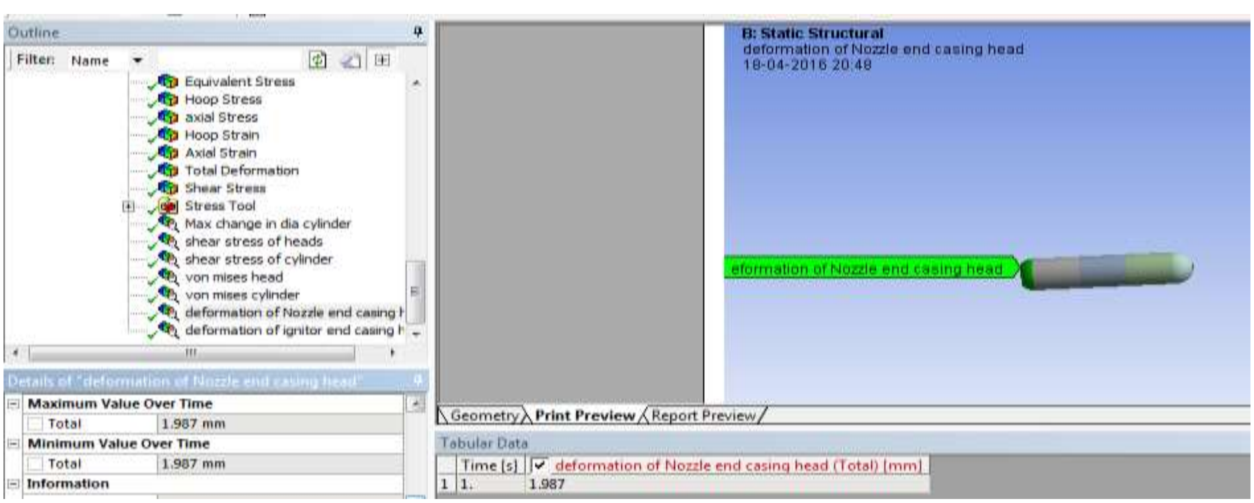


Figure 19:Change in length/Deformation of head/Dome at the nozzle end=1.987mm

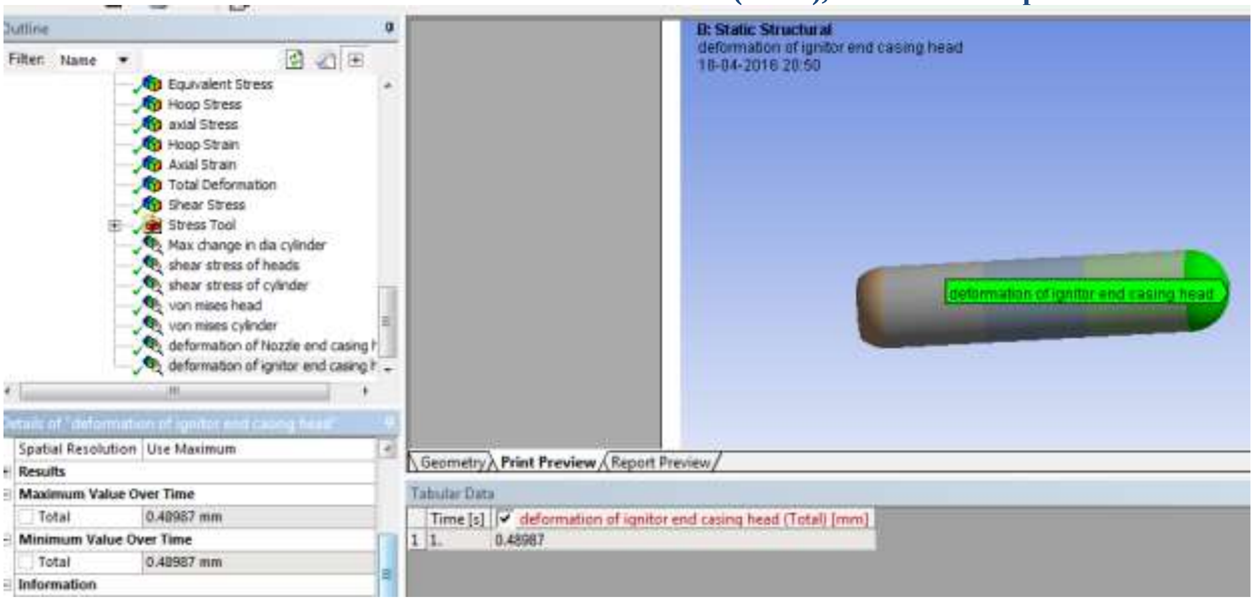


Figure 20:Change in length/Deformation of head/Dome at the ignitor end=0.4898mm

Numerical values and formulas

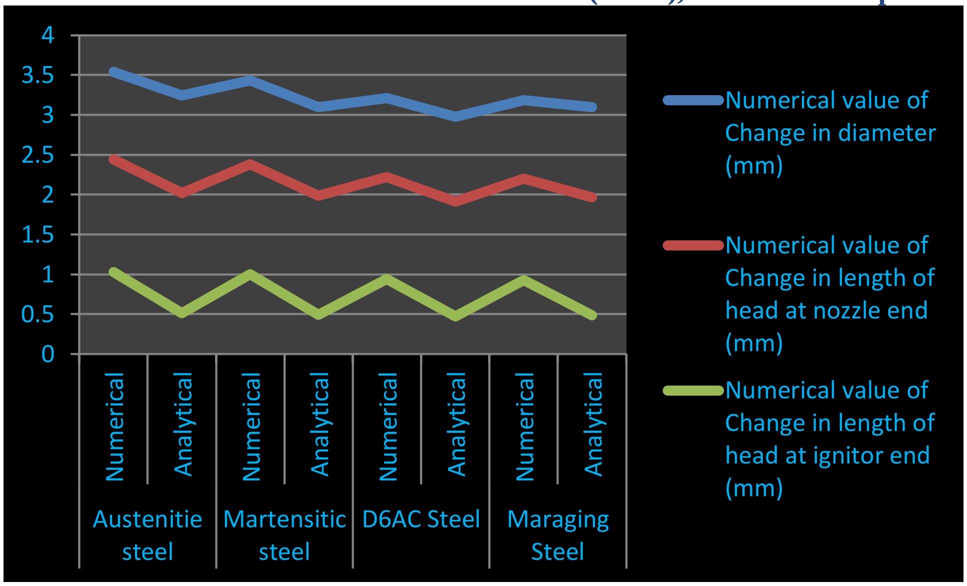
I) Formula for Numerical value of Change in diameter = Pd^2(2-´ϒ)/4tE
Where, ´ϒ = Poissons ratio

II) Formula for Numerical value of Change in length of head at nozzle end = dl = (Von mises stress of head/youngs modulus)*length of head at nozzle end

III) Formula for Numerical value of Change in length of head at ignitor end =dl= (Von mises stress of head/youngs modulus)*length of head at ignitor end

Table 4. Numerical analysis value table

	Austenitie steel	Martensitic steel	D6AC Steel	Maraging Steel
Numerical value of Change in diameter (mm)	3.54	3.43	3.21	3.185
Numerical value of Change in length of head at nozzle end (mm)	2.44	2.377	2.22	2.2
Numerical value of Change in length of head at ignitor end (mm)	1.03	1.003	0.941	0.93



Graph 1: Graph comparing the numerical and analytical values for different materials

CONCLUSION

1) Numerically calculated values using different formulae are very close to values obtained from analytical analysis of Von mises stress,Hoop and axial stresses from table 2 and 3 respectively.This shows that analysis is done with a valid model of motor casing.
2) It is concluded that smaller values of equivalent stresses are appearing in motor casing at hemispherical heads, and equivalent stress distribution is advantageous in case of head geometry and also for that of the cylindrical region comparative to that of numerical values.
3) Also, from graph 1 it can be clearly concluded that maraging steel is the best material to be considered for this sheet metal based rocket motor casing as it's overall deformation is the lowest among the four materials and also, D6AC steel can also be considered as an alternative since it has the least value for deformation of dome at the ignitor and nozzle end only in analytical analysis.

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