International Journal of
Emerging Research in Management &Technology
ISSN: 2278-9359 (Volume-4, Issue-12)

**Research Article** | **December 2015**

# Design and analysis of Hydraulic Actuator for Engine Gimbal Control of a Satellite Launch Vehicle (for 5000 psi Hydraulic System)

**Akhil V M**
Assistant Professor, Mechanical Engineering Department
Heera College of Engineering and Technology,
Kerala, India

***Abstract—***

***Hydraulic system design is a major area of interest for any aerospace actuation system primarily due to high power-to-weight ratio associated with it. Weight reduction is the one of the important considerations in any aerospace hydraulic system; redesigned components are replacing the existing. This article deals with the design of a higher pressure system with same power rating to replace an existing unit, thereby bringing about weight reduction of 35%. The detailed component design and its analysis, mentioning the equations for theoretical design consideration are given. The working pressure of existing hydraulic Engine Gimbal actuator is 207 bar (3000 psi), the new system is designed 345 bar (5000 psi). Design is based on theoretical aspects and verification is performed with Numerical solution. An experiment is conducted with the existing actuator system for validation of the Numerical solution.***

***Keywords— Hydraulic actuator, Engine Gimbal system, Aerospace hydraulic, hydraulic seals***

## I. INTRODUCTION

Thrust vectoring is a common name given to the technique used in achieving the directional control of 'flying objects' with all degrees of freedom by changing the angle of the thrust relative to the centre of gravity of the object. Thrust vectoring is usually the only method available for rockets and missiles which can work in conditions where the air density is so low that control surfaces like ailerons, elevators, flaps etc are rendered useless.

Thrust vectoring is achieved generally by using position controlled actuation systems with one end attached to the nozzle of engine generating the thrust and other end to the frame of the engine. It may be noted as far as the actuation systems are considered, they remain identical whether it issued for thrust vectoring or control surface deflection application. Actuators are key elements for any Aircraft and Space Applications [1].

A linear, hydraulic actuator is commonly used for changing the direction of thrust vector or control surface deflection where high force requirement exists. For use in rockets and missiles, the engine is given provision for being swivelled about a pivot point. In rocket stages using liquid engines, the engine may be mounted on a universal pivot joint – termed as Gimbal Joint and the propellants may be fed through flexible tubing. An actuator (position controlled, linear movement) may be employed with the stationary part fixed to the rocket stage and the moving part fixed to the rocket engine. Such an actuator can tilt the engine about the Gimbal joint. This actuator is termed as Engine Gimbal Actuator.

Universally, hydraulic system designers are aiming for higher pressures for making compact systems. Designing an updated component for retrofitting in an existing system places more constraints than a total system redesign. The various constraints have been considered and a hydraulic actuation system has been designed to operate on higher pressures with a weight reduction of 35% from existing system.

In this work, Modelling of the components was completed and analyses of the components were done using ANSYS software. Hydrostatic Proof Pressure test is conducted on existing actuator assembly with pressure of 1.5 times operating pressure. Results are captured with the help of strain gages and compared with numerical solution. Once the numerical solution is validated for the existing system, the same is applied for higher pressure rated system.

## II. HYDRAULIC ACTUATORS

The generation of higher pressure and distribution is beyond the scope of this article. This paper deals with the design of the hydraulic actuator part alone taking into consideration the mechanical interface, centre to centre distance, load capacity and power rating identical to the existing system, at the same time, reduction in weight of the actuation system. The basic hydraulic circuit has been adopted from the existing system.

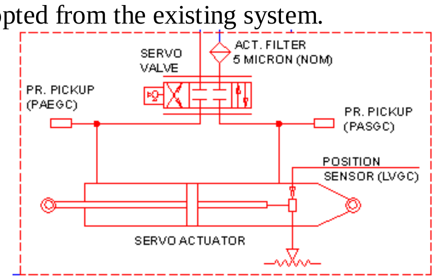



Fig1 .Circuit diagram of hydraulic actuator

**Pascal's Law**

Pressure applied to a confined fluid at any point is transmitted undiminished and equally throughout the fluid in all directions and acts upon every part of the confining vessel at right angles to its interior surfaces [2].

Since pressure P applied on an area A gives rise to a force F, given as, F = P×A --------(1)

## III. MATERIAL SELECTION

Material strength varies with composition, heat treatments, etc. Material selection also depends on factors other than high strength to weight ratio for eg.- environmental conditions, availability, service life, cost etc. Here 15-5 PH stainless steel (15%Cr 5%Ni Precipitation Hardening Stainless Steel) is selected for the design under the comparison with other four available materials.

Table I. Comparison of Materials

| Material | AISI 4340 | 18Ni Maraging | 13-8 PH Mo Stainless steel | 17-4PH Stainless steel | 15-5 PH Stainless steel |
|---|---|---|---|---|---|
| **Correction resistance** | Poor | Poor | Mild | Mild | Good resistance |
| **Stress corrosion cracking** | Medium | Medium | - | Susceptible | Good |
| **Toughness** | Good | Good | - | Medium | High |
| **Yield strength* (MPa)** | 880 | 1280 | 1200 | 1379 | 1385 |
| **Ultimate tensile strength* (MPa)** | 1000 | 1460 | 1310 | 1448 | 1460 |

*- Under special heat treatment condition

## IV. DESIGN OF ACTUATOR COMPONENTS

The design process also attempts to reduce the no. of components required for the assembly without sacrificing any of the functionalities of the existing design. Fig 2 shows the existing hydraulic actuator. Hydraulic fluid selection is also not discussed here as these are not absolutely required in the current structural design.

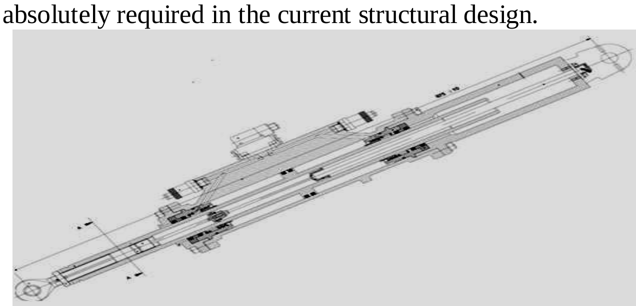


Fig 2. Existing 207 bar Hydraulic Actuator

### *A. Piston-rod*

Piston-rod is an important element in actuator design consideration because the load is transmitted fully by piston-rod. In addition to direct loads like tensile load& compressive load, the piston has to bear external pressure and bending loads. Design stress is selected and Optimization of thickness is done with thick cylinder and tensile load concept.

$\sigma_d$ = 210 N/$mm^2$

$\sigma_d = \frac{F}{Ar}$ (Tensile) ----------- (2)

$\sigma = \frac{2pD1^2}{D1^2 - D0^2}$ (Thick cylinder) ------------(3)

The optimized plot for thickness is given below.

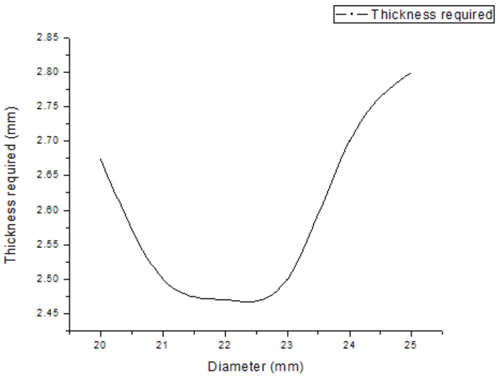



Fig 3. Optimized plot for thickness required for piston-rod

The minimum thickness based internal diameter $D_0$ is 22 mm. ie. $D_1$ =26.92 mm

Standardized diameter from seal data hand book [4]. $D_1$ =28.575 mm (1.125 inch)

Rod Seal, Wiper seal, Wear Ring were selected based on the outer diameter of piston rod and pressure condition.

Load capacity is checked with crippling equation

$$\frac{P_{cr}}{Ar} = \sigma_y - \left[\frac{\sigma_y}{2\pi} x \left(\frac{L}{K}\right)^2\right] x \frac{1}{c^2 x E} \quad \text{---------- (4)}$$

Factor of safety for crippling load is obtained as 2.92

Check for collapsing pressure

The collapsing pressure for steel tube [5] (Prof. A.P.Carman's formula)

$$P_{co} = 658.6 \frac{t1}{D_1} - 14.5 \quad \text{---------- (5)}$$

Factor of safety for collapsing pressure is obtained as 2.409.

### *B. Piston*

The pressure acting on piston area transforms into linear force. So piston area will depend upon how much force needs to be transmitted provided the pressure is fixed.

$$P = \frac{F}{A} \quad \text{---------- (6)}$$

Outer diameter of piston $D_2$ = 47.6619mm

Piston Seal is selected from the available standard diameter.

Standardized diameter $D_2$ = 49.149mm (1.935 inch)

### *C. Cylinder*

Cylinder inner diameter is selected based on outer diameter of piston and the clearance.

$D_2c$ = 50.8 mm

Based on pressure withstanding and load transmitting capability, outer diameter is selected.

Dilation

Change in outer diameter of cylinder due to internal pressure

$$\Delta d = \frac{pd2}{2tE}\left(1 - \frac{\mu}{2}\right) \quad \text{---------- (7)}$$

$\Delta d$ = 0.0341 mm

### *D. Cushioning*

To retard inertial loads and increase the fatigue life of actuators some form of internal cushioning is often used [7]. When the actuator outlet flow is directed through the restrictor, the pressure drop generated will create a back pressure on the actuator, thus causing it to be retarded. Fig 4 shows the cushioner in the actuator based on the radial clearance flow.

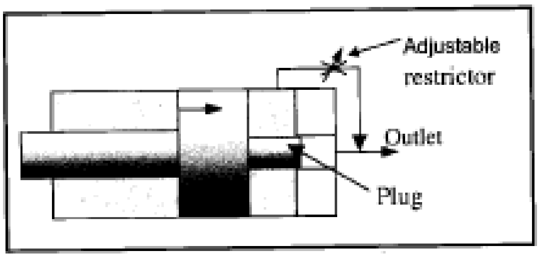



Fig 4. Cushioning In Actuator

Design is based on the inertial loads and pressure load coming into the system.

Equation for flow rate of fluid Q

$$Q = \frac{\pi\, ds\, h^3\, \Delta p}{12\, \mu d\, l} \quad \text{---------- (8)}$$

ds - Inner dia of cushioner

h – Radial clearance

$\Delta p$- Change in pressure

µd – Dynamic viscosity

l – Length

Total volume of fluid = Volume of cushioning

For 20mm length, radial clearance h = 2.7634 e -06 m

This value is considerably very small and variation of radial clearance and length of cushioner is plotted

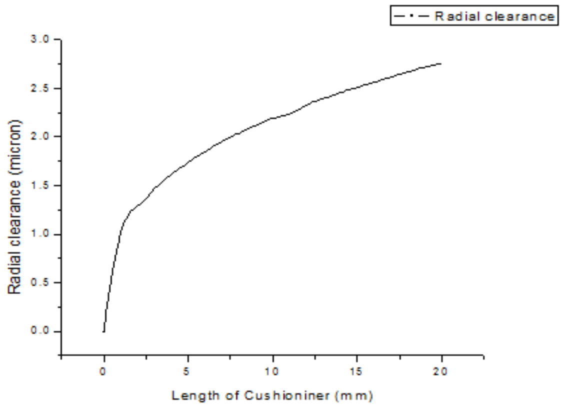
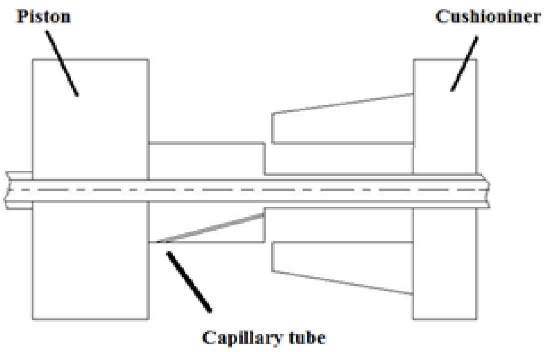



Fig 5. Variation of radial clearance with length of cushioner

From the manufacturing point of view, it is too difficult or impossible to make this one. This clearly shows that the radial clearance type cushioning system is complicated for small scale system. So we need an alternate design for cushioning.

Flow through the capillary tube $Q = \frac{\pi \ dc^4 \ \Delta p}{128 \ \mu \ l}$ --------- (10)



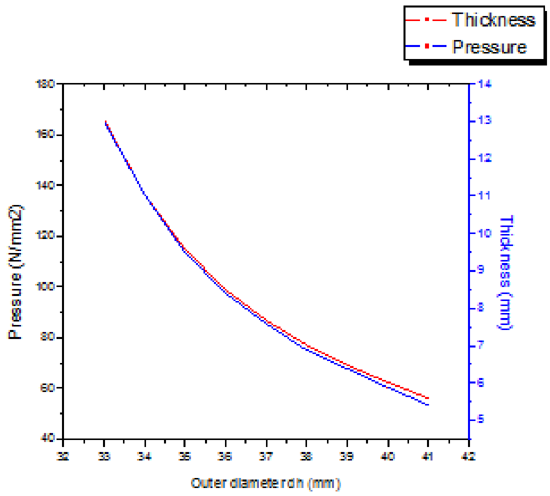




FIG 6.Cushioning Shaft and Hole Assembl Fig 7.Graph shows the variation of pressure and thickness with outer diameter.

Optimized thickness is got from the pressure and thickness consideration of the cushioner.The optimum diameter as 37mm selected from the above graph.
From the equation (10)
If the flow take place in 1/8 second diameter of the capillary tube dc= 0.302mm
Set the diameter of the capillary tube as 0.3mm.

### *E. Actuator cylinder flange*

Actuator cylinder flange need to transmitting forces without failure. Dimensional value for the actuator cylinder flange is taken from the classical equations.

$D_p$ = D2 + 2tc + 3 $d_B$ ----------- (11)

e = $\frac{D_p}{2} + (\frac{d_B}{2} - \frac{D3}{2})$ ---------- (12)

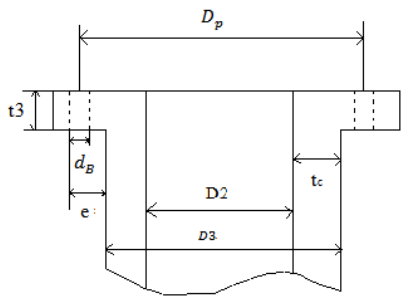



Fig 8. Actuator cylinder flange

From the above classical equation we get the flange thickness
t3 = 9.67 mm

## V. ANALYSIS OF HYDRAULIC ACTUATOR

Modeling of structural parts was done in PRO-E software. Finite element analysis for the parts is done using Ansys11 software.

Finite Element Details: Solid 95 finite element using Ansys was adopted for the analysis. The currently used element is a higher order version of the 3-D 8-node solid element ‘Solid 95’. It can tolerate irregular shapes without much loss of accuracy. Solid 95 elements have compatible displacement shapes and are well suited to curved boundaries of the model.

### *A. CYLINDER:*

For carrying out structural analysis, the bolt holes are selected and UX, UY degrees of the freedom are arrested. Uniform pressure of 350 bar (35 N/$mm^2$) is applied on the inner surface of the cylinder. In the analysis of cylinder, symmetric modeling is followed because one of the sections of the cylinder is symmetrical with respect to its YZ plan and the loading is symmetrical with respect to that plane.

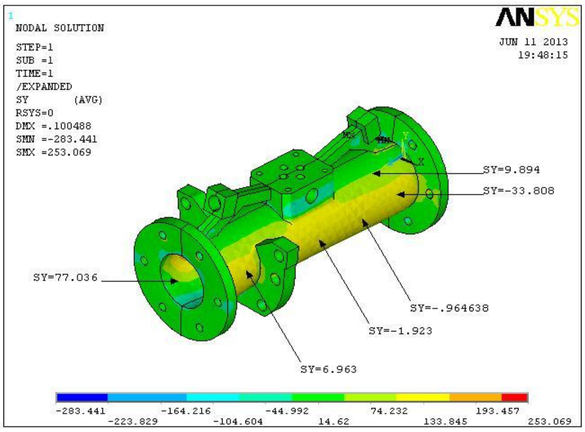



Fig 9 Stress distribution in cylinder

Deformation in y direction (radial) due to application of the pressure of 35 N/$mm^2$ on internal surface of cylinder get a max deformation is 87 micron at inner sharp edge. Inner Radial dilatation of cylinder is got within the expected limit. Stress value is also within the limit, max value is obtained as 283 N/$mm^2$.

### *B. PISTON-ROD:*

For carrying out structural analysis, one of the piston rod surfaces is selected and UX, UY degrees of the freedom are arrested. Uniform pressure of 350 bar (35 N/$mm^2$) is applied on the outer surface to inner of the piston rod.

The maximum stress value is obtained as 144 N/mm2 which is also within the acceptable limit of 210 N/$mm^2$ . The radial contraction is 7.1 micron.

The combined radial deformation (dilation for cylinder and contraction for piston) is 47 micron. This is well within the acceptable limit from the point of view of leak tightness of selected seals.

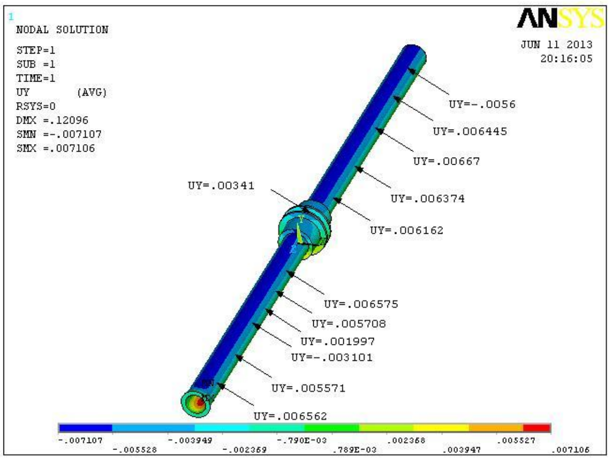



Fig10.Piston deformation in Y direction

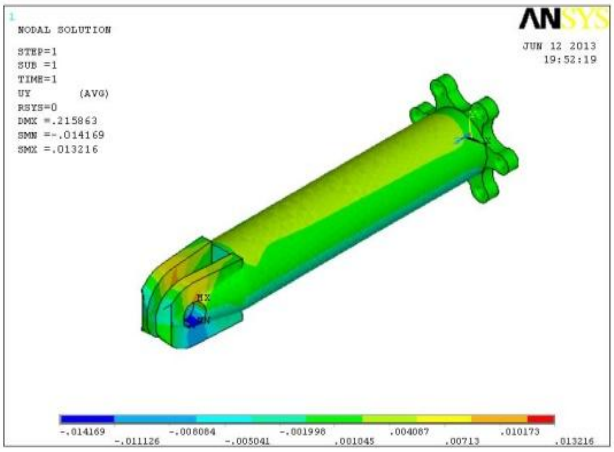



Fig 11. deformation in compression

### *C. ACTUATOR BODY:*

Bolt-hole area is selected and UX, UY degrees of the freedom are arrested. The load 40000 N is applied on the clamping end of body part.

The maximum stress value got as within the limit for tension and compression. The maximum stress values for tension is 70.368 N/$mm^2$ and for compression is 87.606 N/$mm^2$.

### *D. CUSHIONER*:

In this cushion part bolt-hole area is selected and UX, UY degrees of the freedom are arrested. Pressure 80 N/$mm^2$ is applied on the inner surface of the cushioning hole.

Cushion cone deforms up to 0.023mm, this is very small compared to dimension of cushioner. The maximum stress value is obtained at cone plate joining side as 166 N/$mm^2$ . This is within acceptable stress limit.

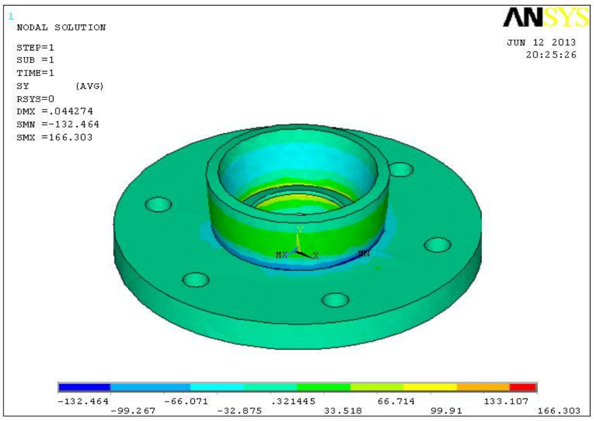



Fig 12 Stress distribution in cushioner

### *E. SEAL ANALYSIS*

Seals play a critical role in hydraulic and pneumatic power transmission elements. Failure of seals can cause the entire system to fail. Bi- directional seals are selected for this application.

Material property of seal is very important because it decides the capability of the seal to withstand tensile and compressive loads during pressurization and actuation in the system. An analysis of a bi-directional seal made of polyurethane material is carried out to find the effect of material deformation at varying pressure conditions.

*Seal dimension*

Standard available seals were selected. Profile structure considered is that of 'OK profile' type seals manufactured by Parker Hannifin Corporation. 'Ok profile' bi-directional piston seal are designed for heavy-duty hydraulic applications.

*Bounty condition*

Groove dimension and clearance between the piston and cylinder will act as the physical barriers. Pressure 35 N/$mm^2$ is acting on one side of seal during forward/ backward stroke. Other contact surfaces are treated as contact pairs. The effect of friction is neglected because in actual conditions; the seals will be immersed in oil.

*Material property*

Polyurethane is versatile material, which shows different behaviours with change in ratio of their constituent component [6]. Their behaviour can vary from an elastomeric to a rigid behaviour. They can be varied widely in composition and can be designed to possess either softer or stiffer mechanical characteristics according to the intended end-use.

The deformations at the top critical portion under the different pressure conditions are calculated. The deformation of critical nodes at varying pressure condition is plotted in the graph (fig13). Variation of pressure from zero to 35N/$mm^2$ is represented by UY.

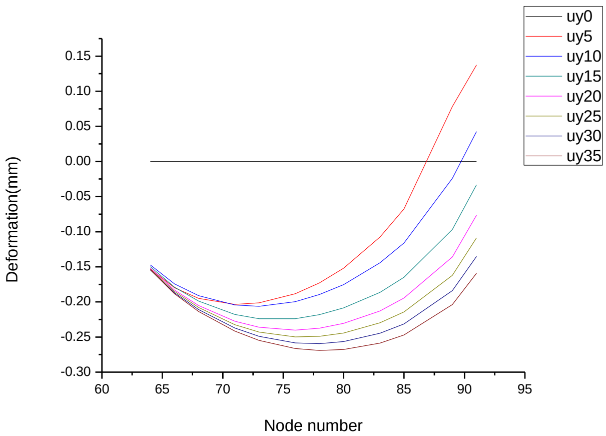



Fig 13.Nodal deformations at the top critical portion

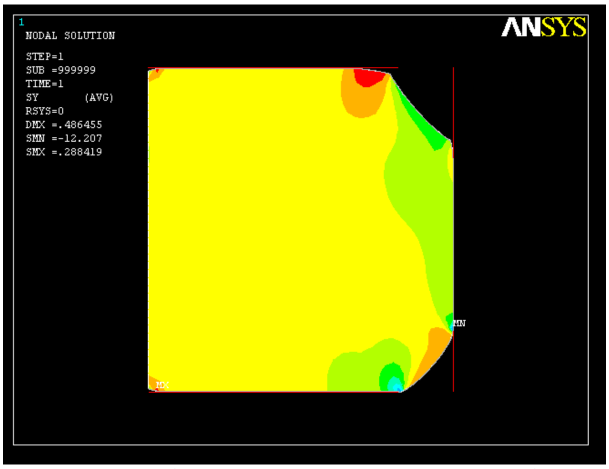



Fig 14.Stress distribution of piston seal while the application of pressure

Fig 13 above shows the deformation of the seal with varying pressure. Here we have considered the seal from no pressure condition to 35 N/$mm^2$ pressurized condition From above analysis we can clearly understand that due to the application of pressure the seal will get compressed and give a force to upward direction, this shows seal is act as leak proof.

## VI. VALIDATION OF ANALYTICAL RESULTS WITH EXPERIMENT

**Experimental setup**

The currently used engine Gimbal hydraulic actuator is 3000-psi (207bar) system The current system consists of cylinder, piston, body etc. The cylinder has to withstand the full operating pressure (207 bar) as well the proof pressure (315 bar). Strain gauges were bonded on identified locations where sudden change of geometry were noted. Fig 17 shows the position of the strain gage bonded on the 210 bar actuator. 8 strain gages were bonded in the different position of actuator cylinder part. 2 strain gages were in longitudinal direction and others were in radial direction. Strain gage number 6, 8 are mounded in longitudinal direction.

**Finite element analysis of existing actuator cylinder:**

The cylinder was modeled in the Pro-E software and transformed to Ansys software. Strain gage mounting points were captured with node number.

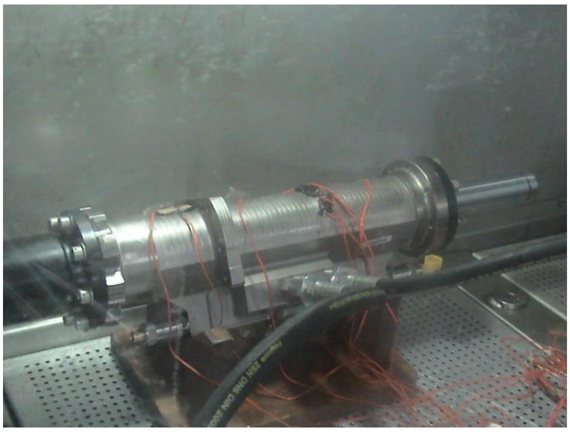


Fig 16.Actuator in pressurized condition in proof pressure test rig

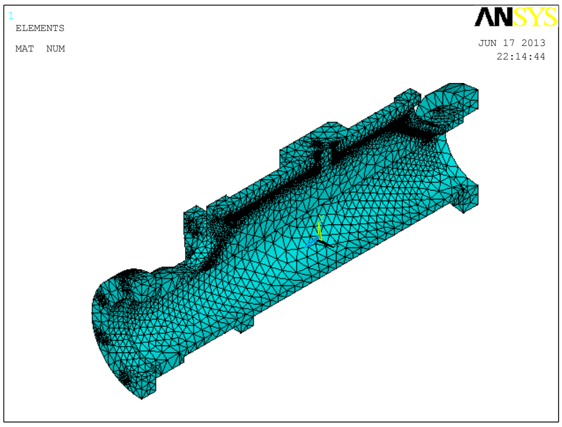



Fig17.Meshed view of half cylinder

In the half cylinder part, bolt-hole area was selected and UX, UY degrees of the freedom were arrested. Varying Pressure (80,160,220 and 320 bar) was applied on the inner surface of the cylinder part.

**Comparison of Experimental Vs Analytical Strains**

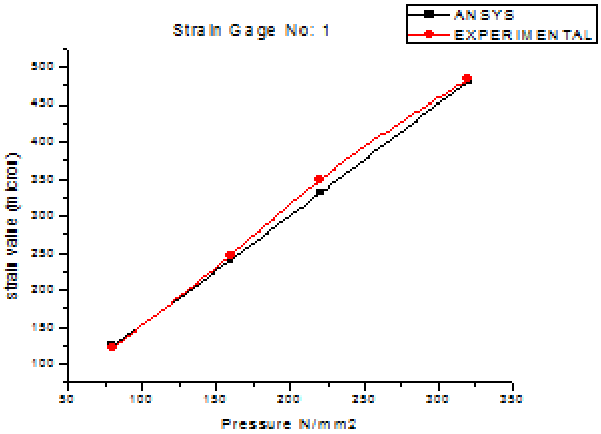



Fig 15. Strain gage no: 1

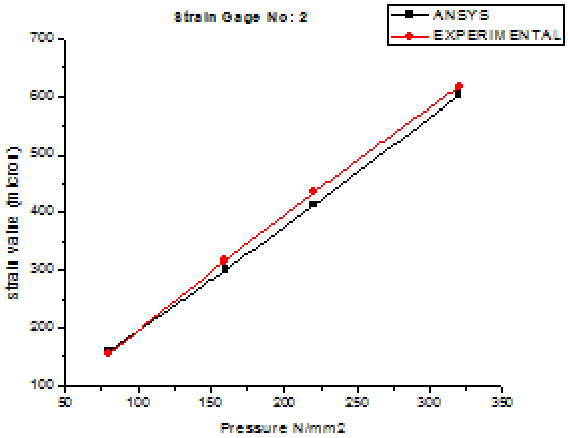



Fig 16. Strain gage no: 2

The result obtained from experiment and ANSYS values are considerably same. From this comparison we can understand that the relational error produced by the numerical software is very much less. So it is in the acceptable range to adapt with analysis.

The following graph shows the Comparison of Experimental Strains gage value and Analytical strain value in different nodal position with varying pressure condition.

## VII. WEIGHT COMPARISON

Table 2. Component name and its weight in kg

| Component | Current system Weight (kg) | Designed system Weight (kg) |
|---|---|---|
| Piston | 3.18 | 1.7 |
| Cylinder | 6.870 | 3.7 |
| Head end cover | 0.32 | 0.94 |
| Body | 1.92 | - |
| Rod guide | 0.52 | - |

| | | |
|---|---|---|
| Rod bearing | 0.16 | 0.16 |
| Cushion sleeve | 0.08 | 0.71 |
| End cover | | 0.41 |

Total weight of components including Mass fluid insider cylinder is 14.082 kg for current system and for designed system total weight of components including Mass fluid inside cylinder is 9.019 kg
Expected weight reduction 30% and we achieve percentage of reduction in weight is 35.9 %.

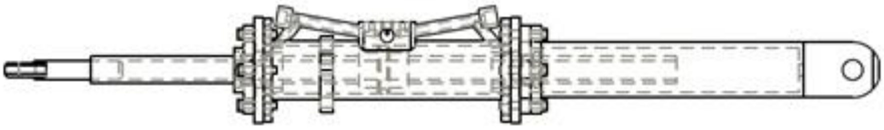


Fig 17 sketch of newly designed actuator

## VIII. CONCLUSION

New Design of the hydraulic actuator achieves weight reduction of 35%with current system. Analysis of the components shows they are capable of withstanding required load condition. In seal, it will become more leak proof due to the upward stress produced the seal. From the experiment it is clearly understand that the current system is safe in its proof pressure. Numerical results are closely related with experimental value in all strain gage locations.

**REFERENCES**

[1] P. Jänker, F. Claeyssen, "New Actuators for Aircraft and Space Applications" 11th International Conference on New Actuators, Bremen, Germany, 9 – 11 June 2008, page 346-354
[2] Peter. J. chapple, , "*Principles of hydraulic system design*" first edition 2003
[3] Yeaple "*Hydraulic and pneumatic power control",* 1966 McGrew-Hill
[4] Rotary Seal Design Guide, Parker Hannifin Corporation,2007
[5] K.Mahadevan,"Design data book for Mechanical Engineers",CBS publications2009.
[6] D.J. dos Santos, L.B. Tavares, G.F. Batalha, "Mechanical and physical properties investigation of polyurethane material obtained from renewable natural source", Journal of Achievements in Materials and Manufacturing Engineering, Volume 54 Issue 2 October 2012, page no:211-217