DESIGN AND ANALYSIS OF TELESCOPIC JACK

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Abstract - *The use of hydraulic jack in the industry is widespread as load lifting structures. Telescopic hydraulic jack is a special design of jack with a series of tubes of progressively smaller diameters nested within each other. They have long stroke from a compact initial package, which have attracted a lot of attention for their applications as load lifting structures.*

Design and analysis of telescopic hydraulic cylinder is very complex phenomenon that involves interactions between the two stages. In this project telescopic jack is designed for the industry "Hydrau Solutions" located in Belgaum. Principle parameters included the maximum lifting capacity of 7000 Kg and a stroke of 980 mm.

With CATIA, the parametric modeling of the two-stage hydraulic jack is carried out. They are subjected to large side forces. These forces, combined with the load being pushed, threaten to deform the telescopic assembly. Therefore the configuration design is analyzed, using finite element method, according to the different loading conditions and specification.

. As expected, results show that load carrying capability of the jack is significantly affected by different stress acting on it. Finite element results were in good agreement with the theoretical solution for the telescopic jack under different stress.

Key Words: *Telescopic cylinder, Hoop stress*

1. INTRODUCTION

A telescopic hydraulic jack uses a liquid to push against a piston. This is based on Pascal's Principle. The principle states that pressure in a closed container is the same at all points. If there are two cylinders connected, applying force to the smaller cylinder will result in the same amount of pressure in the larger cylinder. However, since the larger cylinder has more area, the resulting force will be greater. In other words, an increase in area leads to an increase in force. The greater the difference in size between the two cylinders, the greater the increase in the force will be. A hydraulic jack operates based on this two cylinder system.

1.1 Problem Statement

The purpose of this project is to study how different loads & stress act on a telescopic jack and design a 7000 kg telescopic jack. This project will focus on how the hoop, bending & pressure of fluid (combined loading) in the cylinder affect the design of the jack. Also in this project, the jack consists of a swivel plate at the end as shown in fig 1, which can tilt by 13°, so due to that side load would be imposed on the cylinder which has to be studied.

Bearing area is also to be checked as well as the jack has to be analyzed under minimum area loading conditions.

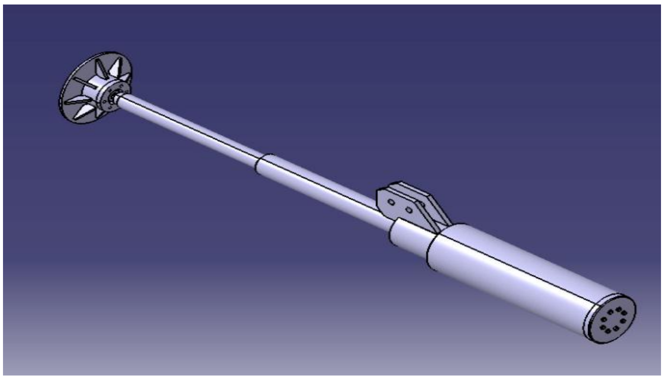

Fig -1: showing the swivel plate

1.2 Methodology

The finite element method is used to conduct the analysis for this project. The software used is ANSYS; ANSYS is a comprehensive general-purpose finite element computer program that contains over 100,000 lines of codes. ANSYS is capable of performing static, dynamic, heat transfer, fluid flow, and electromagnetism analyses. ANSYS has been leading FEA program for well over 20 years. A telescopic cylinder is modeled in CATIA. The element size used is determined by conducting a mesh density study. The largest element that produces accurate results is used to produce accurate results in a model that runs as quickly as possible. Once an element size is determined, static analysis is performed.

The resulting stress obtained from this analysis is validated by comparing it to a simple hand calculation using a simple equation derived using SOM.

2. LITERATURE SURVEY

Hydraulic jacks are typically used for shop work, rather than as an emergency jack to be carried with the vehicle. Use of jacks, not designed for a specific vehicle, requires more than the usual care in selecting ground conditions, the jacking point on the vehicle, and to ensure stability when the jack is extended. Hydraulic jacks are often used to lift elevators in low and medium rise buildings.
A hydraulic jack uses a fluid, which is incompressible, that is forced into a cylinder by a pump plunger. Oil is usually used for the liquid because it is self-lubricating and has stability compared with other liquids. The pressure of the liquid enables the device to lift heavy loads.

2.1 Telescopic hydraulic jack

The advantage of telescopic hydraulic is their ability to provide an exceptionally long stroke from a compact initial package as shown in fig 2. The collapsed length of typical telescopic cylinders varies 20% to 40% of their extended length. This finds application when mounting space is limited, and the application needs a long stroke.

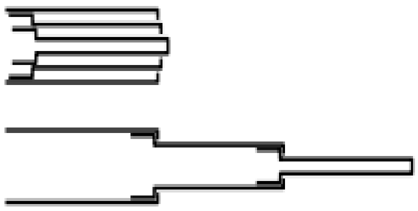

Fig -2 showing the telescopic principle, an object collapsed (top) and extended (bottom), providing more reach.

For example, assume a dump body needs to be tilted 60° in order to empty completely as shown in fig 3. If the body or trailer is fitted with a conventional rod-type cylinder with a one-piece barrel and stroke, long enough to attain that angle, the dump body could not return to a horizontal orientation for highway travel because of the cylinder's length, even when fully retracted. A telescopic cylinder easily solves this problem.

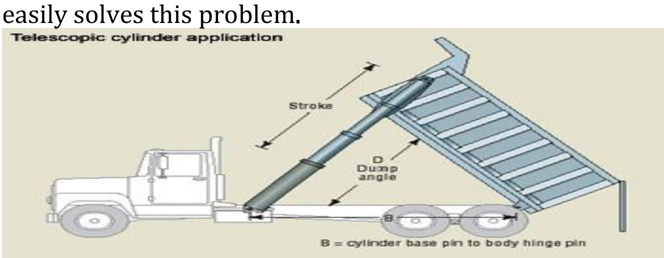

Fig -3 Vehicle with telescopic cylinder to lift the dump body

2.2 Single-Acting Telescopic Cylinders

High pressure oil from the pump is directed by the control valve through the port (A) to fill the cylinder. Any air in the system is trapped in the end of the cylinder (B) and may be bled off through the bleeder valve (C) as shown in fig 3.

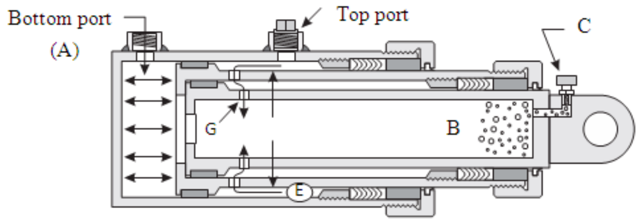

Fig-3 showing extension of single acting telescopic cylinder

Oil pushes on the bottom of the sleeve or plunger forcing it to move out. The outside diameter or sealing area of the moving sleeve or plunger (D) determines the effective area (fig. 2.13)

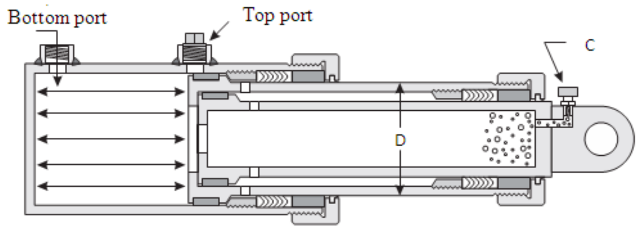

Fig-4 extension of 1st stage of single acting telescopic cylinder

As the sleeve or plunger moves out, the oil trapped between (E) the sleeve or plunger wall is released through transfer holes (G) which are drilled in the sleeve or plunger (fig 5). Under normal operating conditions the largest diameter moving sleeve extends first, then the next largest sleeve, etc.

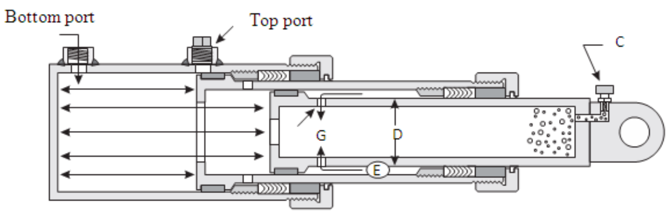

Fig-5 showing extension of single acting telescopic cylinder

So at a given PSI (pressure) and LPM (liters per min.) the cylinder will develop less force and increase in speed as it changes to the next moving stage.

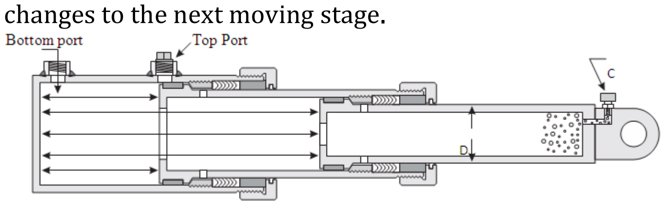

Fig -6 fully extended single acting telescopic cylinder

3. 3- D MODELING OF TELESCOPIC JACK

3.1 Solid Rod (2nd stage)

The solid rod is modeled to a diameter of 50 mm diameter and 734 mm length; it forms the 2nd stage of the jack (fig 7)

Fig -7 solid rod of 50 mm diameter

3.2 Hollow Piston Rod (1st stage)

The 1st stage consists of a hollow cylinder which acts as the piston for the extension of jack. It modeled with OD 72 mm & ID 55 mm. its total length is 674 mm (fig 8).

Fig -8 Hollow piston rod which forms the 1st stage

3.3 Main or the Barrel of Cylinder

The barrel is modeled with OD as 96 mm & ID as 80 mm. Its total length is 687 mm. Oil is fed first to the barrel from where it flow's to further stages. All stages are nested inside this barrel.

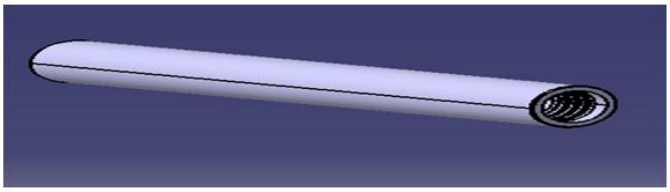

Fig -8 showing the isometric view of barrel

3.4 Swivel Plate

Swivel is of diameter 250 mm and thickness 8 mm. It consists of 8 ribs surrounding the plate as shown in fig 9. The thickness of the rib is 15 mm

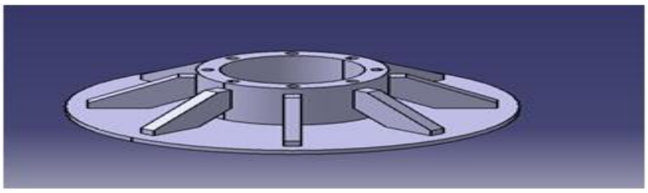

Fig -8 Swivel plate with ribs around it

4. FINITE ELEMENT ANALYSIS OF TELESCOPIC JACK AND ANSYS

4.1 Importing the Assembly Model

Since the telescopic hydraulic jack assembly model is modeled in CATIA, it is in 'inp' file format. It is imported to Ansys work bench.

4.2 Defining the material properties

Steel, specifically En-24, is used as the material in this project, since we wanted high yield strength so as to sustain high side loads. The material properties for EN-24 can be found in Fig 9. For Solid & Hollow piston rods En-24 material is used.

Typical chemical composition of En24

C	SI	MN	S	P	Cr	Mo	Ni
0.36/0.44	0.10/0.35	0.45/0.70	0.040 max	0.035 max	1.00/1.40	0.20/0.35	1.30/1.70

817M40T / EN24T Mechanical Properties

Size mm	Tensile Strength N/mm²	Yield Stress N/mm²	Elongation	Impact Izod J	Impact KCV J	Hardness HB
63 to 150	850-1000	680 Min	13%	54	50	248/302
150 to 250	850-1000	654 Min	13%	40	35	248/302

Fig -9 chemical composition and mechanical properties of En-24 material

4.2 Selecting the Suitable Element Type

10-Node Tetrahedral Structural Solid has a quadratic displacement behavior and is well suited to model irregular meshes (such as produced from various CAD/CAM systems). The element is defined by ten nodes having three degrees of freedom at each node: translations in the nodal x, y, and z directions. The element also has plasticity, creep, swelling, stress stiffening, large deflection, and large strain capabilities.

4.3 Meshing

The telescopic jack model is meshed as shown in fig 10 using the 10 nodded tetrahedral elements as this element is suitable for irregular meshes.

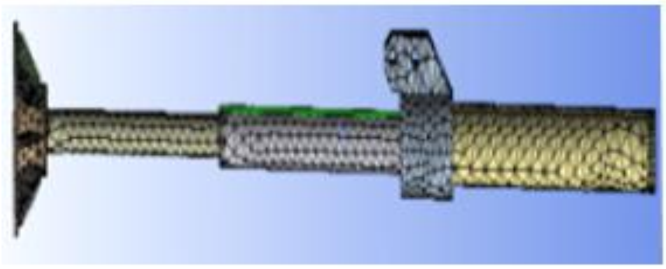

Fig -10 Meshed telescopic jack

3.4 Applying Boundary Conditions

The jack is fixed supported at the loading bracket and load of 70000 N is applied as shown in the fig 11, 12, 13.

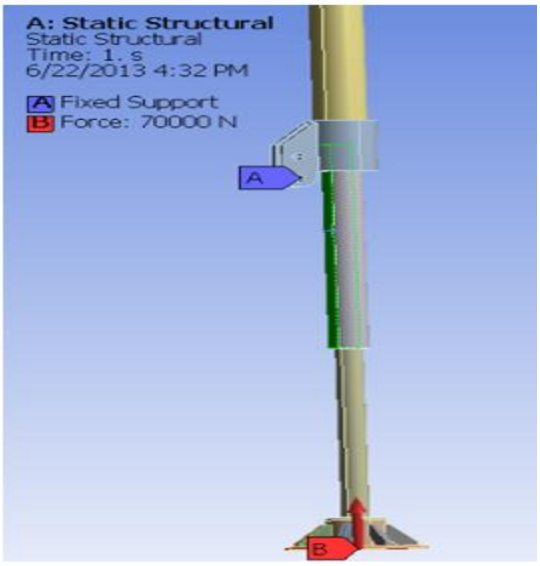

Fig -11 showing the boundary conditions

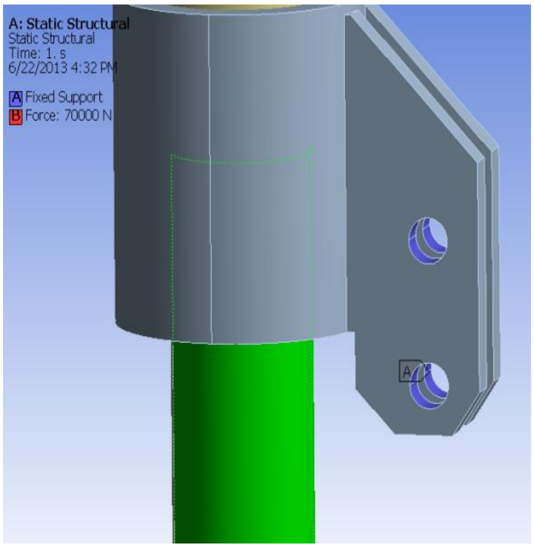

Fig -12 Zoomed image of fixed support

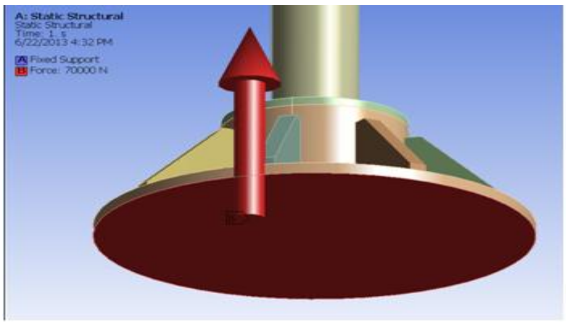

Fig -13 Zoomed image of load application point

In the case of minimum area loading conditions under the worst condition, the jack may be assumed to be resting on 3 points which forms equilateral triangle. Under this analysis two cases are considered, first when the 3 points are considered on the ribs as shown in the fig 4.8 and the second when they are in between the ribs as shown in fig 4.9.

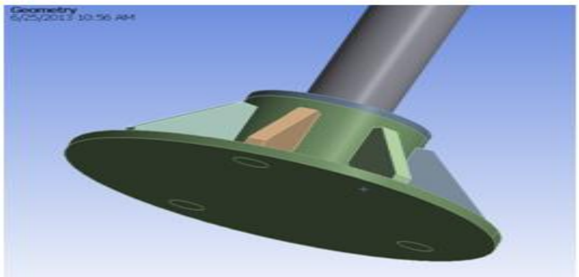

Fig -14 loading points on the ribs

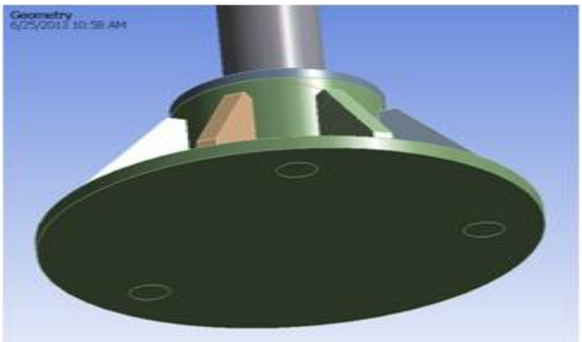

Fig -15 loading points in between the ribs

3.5 Theoretical Solution

By using value of D & Q, it is found out that the velocity with which the load is lifted is 3.77 mm/stroke of hand pump. Radial (P) and hoop (σc) stress in both the cylinder can be obtained easily by making use of lame's equation.
Substituting the boundary conditions (BC) of barrel, pressure at inner (Pi) fibre & outer fibre (Po) of cylinder as Pi = 300 at ri = 27.5 mm & Po = 0 at ro = 36 mm, we will obtain the values of a & b. Making use of a, b values the hoop stress at the inner fibre (σci) and hoop stress at outer fibre (σco) are obtained as 114.128 Mpa & 84.125 Mpa. Similarly for the 1st stage substituting the BC as Pi = 300 at ri = 40 mm & Po = 0 at ro = 48 mm the values of hoop stress is calculated as σci = 166.33 Mpa σco = 136.342 Mpa. At Solid piston rod, 3 points as A, B & C are considered on the rod at a distance of 532 mm from the plate and von misses stresses are found out on it. Bending moment (Mz) due to the side load & moment of inertia can be calculated using simple bending equation and moment of inertia equation. Results obtained are tabulated in the Table-1.

Table -1: showing the stresses at different points

S.no	points	Stress acting on the cylinder				Von misses stress
		Tensile stress due to bending	Compressive stress due to bending	Shear stress due to bending	Compressive stress due to load	
1	A	534.58	0	0	-35.65	498.9
2	B	0	0	554	-35.65	518.35
3	C	0	0	-534.58	-35.65	-570.2

On the 2nd stage, six points are considered, three on outer fibre (Ao1, Bo1, and Co1) of cylinder & three on inner fibre (Ai1, Bi1, and Ci1) as shown in fig.16

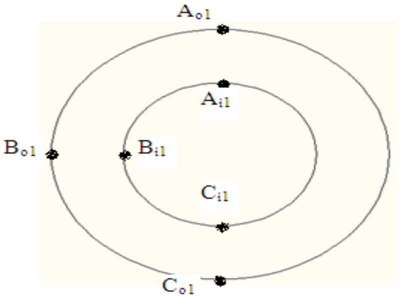

Fig -15 showing the points on outer and inner fibre

Table -2: showing the stresses at different points

S.no	points	Stress acting on the cylinder						Von misses stress
		Tensile stress due to bending	Compressive stress due to bending	Shear stress due to bending	Compressive stress due to load	Hoop stress	Radial pressure	
1	A_{o1}	529.4	0	0	0	84.12	0	492.7
2	B_{o1}	0	0	28.75	0	84.12	0	97.75
3	C_{o1}	0	-529.4	0	0	84.12	0	-570.2
4	A_{i1}	404.4	0	0	0	114.1	-30	576.0
5	B_{i1}	0	0	28.75	0	114.1	-30	140.8
6	C_{i1}	0	-404.4	0	0	114.1	-30	463.6

On the barrel also six points are considered 3 on the outer fibre of cylinder body (Ao2, Bo2, and Co2) and 3 on the inner fibre of the cylinder (Ai2, Bi2, and Ci2) as shown in the fig 16.

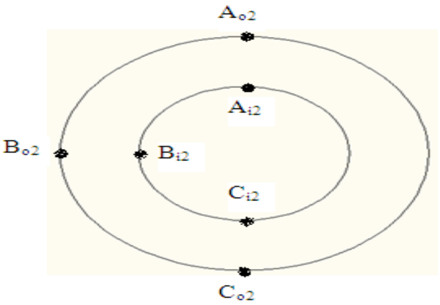

Fig -16 showing the points on outer and inner fibre

Table -3: showing the stresses at different points

S.no	points	Stress acting on the cylinder						Von misses stress
		Tensile stress due to bending	Compressive stress due to bending	Shear stress due to bending	Compressive stress due to load	Hoop stress	Radial pressure	
1	A_{o1}	529.4	0	0	0	84.12	0	492.7
2	B_{o1}	0	0	28.75	0	84.12	0	97.75
3	C_{o1}	0	-529.4	0	0	84.12	0	-570.2
4	A_{i1}	404.4	0	0	0	114.1	-30	576.0
5	B_{i1}	0	0	28.75	0	114.1	-30	140.8
6	C_{i1}	0	-404.4	0	0	114.1	-30	463.6

Up to Θ = 10^o the side load is within the limits i.e. Fs = 12.34 x 10^3 N. Beyond 10^o the side load exceed much and will result in failure of jack. By keeping the side load limited to 12.34 × 10^3 N the acceptable load at angle more than 10^o is calculated. By varying the angle of the plate (Θ) the Fc is calculated and all the result are tabulated in Table 4.

Table- 4: Angle of the plate and its corresponding load

S.no	Angle of the plate (Θ)	Acceptable load in Newton (N)
1	11°	63483.70
2	12°	58055.13
3	13°	53450.41
4	14°	49493.03

Table -5: showing the stresses at different points

S.no	points	Stress acting on the cylinder						Von misses stress
		Tensile stress due to bending	Compressive stress due to bending	Shear stress due to bending	Compressive stress due to load	Hoop stress	Radial pressure	
1	A_{o1}	529.4	0	0	0	84.12	0	492.7
2	B_{o1}	0	0	28.75	0	84.12	0	97.75
3	C_{o1}	0	-529.4	0	0	84.12	0	-570.2
4	A_{i1}	404.4	0	0	0	114.1	-30	576.0
5	B_{i1}	0	0	28.75	0	114.1	-30	140.8
6	C_{i1}	0	-404.4	0	0	114.1	-30	463.6

Due to the side load on the jack forces would act on the bearing rings as shown in the fig 17.

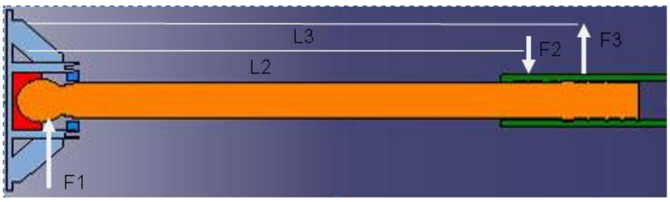

Fig -17 showing the loads acting on the bearing

These forces can be easily calculated using the moment equation. Substituting the value of L & D for both the stages bearing stress is obtained as 38.75 N/mm2 & 27.86 N/mm2 for 1st stage. The yield strength of bearing material is 340 N/mm2, so the design is safe. It can be seen that the bearing area is enough to sustain the side load. Forces would also be acting on the bearing rings located at the barrel and 2nd stage piston rod as shown in the fig 4.6. This load can be easily calculated using the moment equation. Substituting the value of L & D for both the stages bearing ring, stresses are obtained as 54.58 N/mm2 & 34.61 N/mm2. The yield strength of bearing material is 340 N/mm2, so the design is safe. It can be seen that the bearing area is enough to sustain the side load.

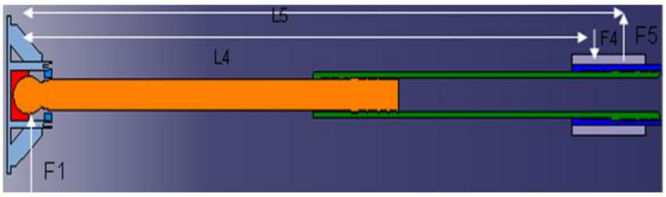

Fig -17 showing the forces acting on bearing in 2nd stage

5. RESULT AND DISCUSSION

The equivalent von misses stresses due to different loads are obtained by using ANSYS by considering few points on each stage.

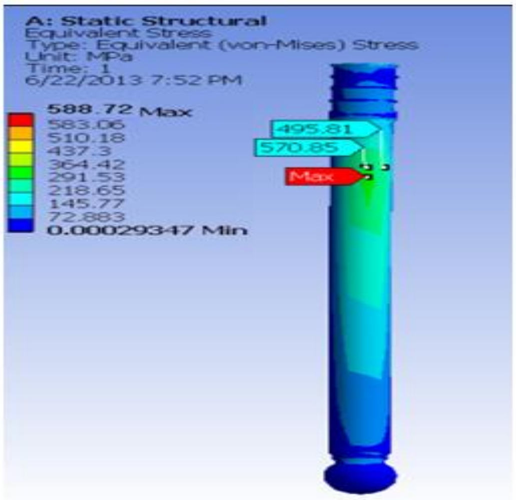

Fig -18 Equivalent von-mises stress induced in solid rod

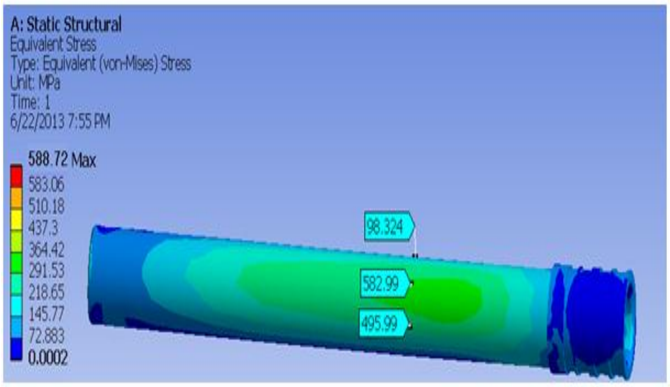

Fig -19 stresses induced on the outer fibre 2nd stage hollow piston rod

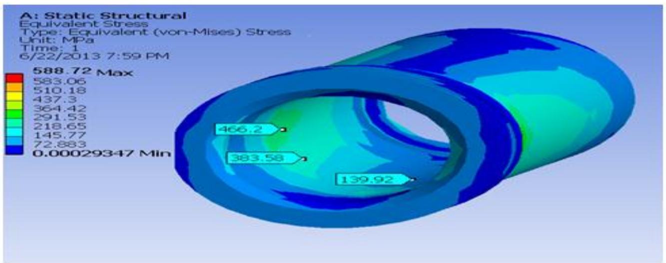

Fig -20 stresses induced on the inner fibre 2nd stage hollow piston rod

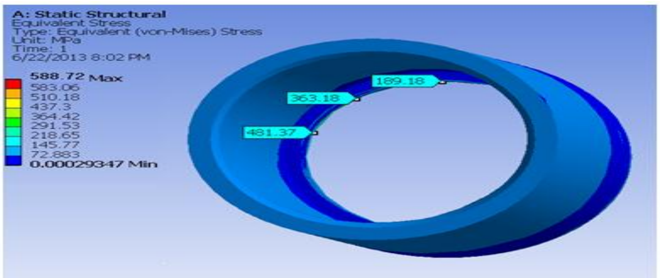

Fig -21 stresses induced on the outer fibre of barrel

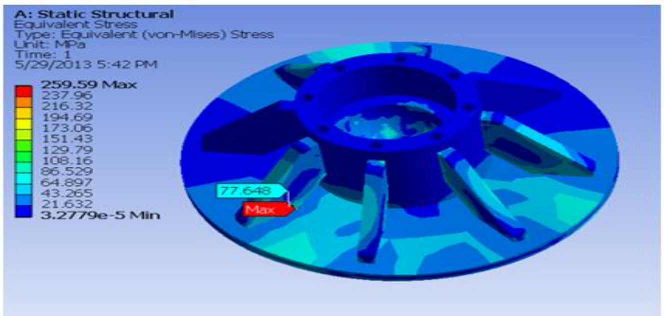

Fig -22 stresses induced on the outer fibre of barrel

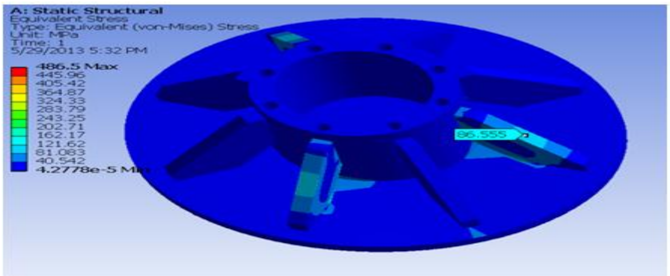

Fig -23 stress when the load is applied on the ribs

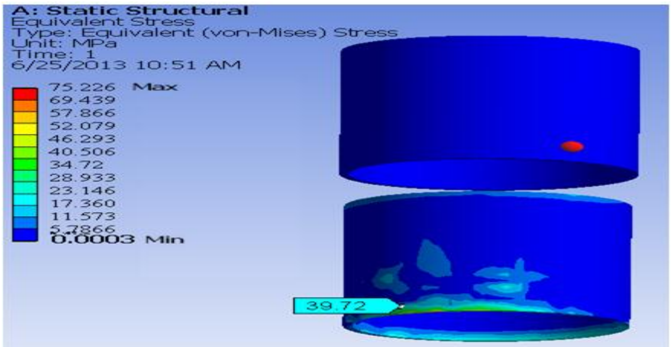

Fig -24 stress value on the bearing on 1st stage

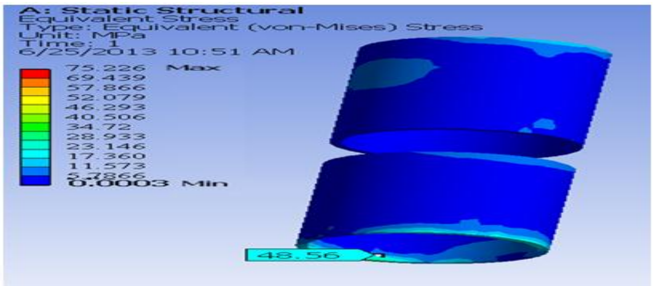

Fig -24 stress value on the bearing on 2nd stage

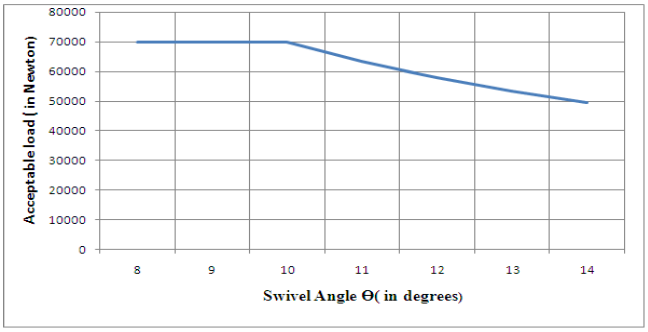

Fig -25 graph of acceptable load v/s swivel angle

If angle of the plate is kept at 10° and load of 70000 N is applied it is observed that, the stresses developed on the solid rod (1st stage) are within the yield strength of the material as seen from & fig 18. It can sustain the side load without any deformation. The stresses induced in the hollow piston rod (2nd stage) are below the yield strength of the material as seen fig 19, 20. The stresses induced on the barrel are also below the yield strength of the material as seen from fig 21.
Under the minimum area loading conditions it can be seen that the stresses developed are less than the yield strength of the plate. Also the stress induced in case of load acting on the ribs is higher than the case when it is acting in between the ribs. The bearing area selected is enough to sustain the side load. And the bearing material is able to withstand the stress acting on it. It can be seen from graph (fig 25) that as the angle of the plate is increased the load which the jack can sustain goes on reducing. It would work efficiently within 10°. All the parts in the jack give factor of safety (FOS) between 1.5 and 2.

6. CONCULSIONS

Telescopic hydraulic jack are a special design of jack which provide an exceptionally long output travel from a very compact retracted length. Static analysis of the telescopic hydraulic jack components, for the different loading conditions, can be performed by using the commercial finite element packages if the loads on the structure are properly defined. From the results presented in this study, we can summarize that the commercial finite element packages can be used to analyze the engineering structures that are subjected to combination of loads with the proper definition of the loading conditions. A telescopic jack subjected side load, hoop stress & radial pressure is analyzed with the finite element package ANSYS. Good agreement is found between the results obtained by ANSYS and Theoretical calculation. Hoop stress, pressure, compressive stress & side load have significant impact on the design of the hydraulic cylinder.

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