ANALYSIS AND OPTIMIZATION OF PINIONOF A JACK-UP RIG

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***Abstract:** -Jack-up platforms have been designed to operate at various locations with different sea-bed conditions and greater water depths. Due to good adaptability and working stability, jack-up platforms are widely used for offshore drilling. Jacking system is one which undertakes the weight of the platform and working load during the lifting, and its performance exerts a direct influence on the safety of the whole platform.Since rack and pinion of jacking system bears heavy loads. This paper attempts to improve the life of the pinion in jacking system by reducing the mean stresses. Since high stresses are occurring at the root of the pinion study of optimum fillet radii is beneficial for strengthening the root. A first initial crack appears at thegear tooth and it is affected, the most, by root stress concentration. Hence,this paper's research topic is focused on finding the optimal fillet toothroot radius to minimize the tooth root stress intensity.Therefore a practical model of the rack and pinion is modelled using CATIA V5 software and Analysis was done using ANSYS 16.0 for various fillet radii and validated theoretically.*

***Keywords:-** offshore drilling,jack-up rig, jacking system,rack and pinion, fillet radii, mean stress.*

1. INTRODUCTION

Jack-Up Units have been a part of the Offshore Oil Industry exploration fleet since the 1950's. They have been used for exploration drilling, tender assisted drilling, production, accommodation, and work/maintenance platforms. As with every innovative technology A Jack Up structure composed of a hull, legs and a lifting system that allows it to be towed to a site, lower its legs into the seabed and elevate its hull to provide a stable work deck capable of withstanding the environmental loads. A typical modern drilling Jack-Up is capable of working in harsh environments (Wave Heights up to 80ft, Wind Speeds in excess of 100 knots) in water depths up to 500 feet. Because Jack Ups are supported by the seabed, they are preloaded when they first arrive at a site to simulate the maximum expected leg loads and ensure that, after they are Jacked to full airgap and experience operating and environmental loads, the supporting soil will provide a reliable foundation. All Jack Ups have mechanisms for lifting and lowering the hull. The majority of Jack Ups in use today are equipped with a Rack and Pinion system for continuous jacking operations. One of the most common failures in jacking system is fatigue failure. The factor that determines fatigue failure is mean stress. As the mean stress decreases, the fatigue life increases. The defects on the contact surface will cause a decrease in the life of a material. As such, sharp corners which stresses concentrate on will probably be the first where cracks will occur and propagate. To increase the strength of the pinion of jacking system avoiding sharp corners is necessary. Therefore an increase of fillet radiican minimize the bending stresses causing at the root of the pinion. Bending stress at root of pinion is theoretically calculated for various fillet radii and model of rack and pinion is done in CATIA V5 software and is imported to ANSYS 17.0 and analysis was carried out

2. GEOMRTRIC PARAMETERS:

Primary parameters of pinion areNumber of teeth z = 7, Modulus m = 80 mm, pressure angle=20° Pitch p = 251.3 mm profile of pinion is involute and profile of rack is straight line. Material of pinion is SAE4340 whose Elasticity modulus e= 210 GPa, Poisson's ratio 0.3, Ultimate strength U_{ts}=745MPa, Yield strength Y_s= 470MPa and material of rack is ASTM A514 GrQ Elasticity modulus e= 200 GPa, Poisson's ratio 0.3, Ultimate strength U_{ts}=895MPa, Yield strength Y_s= 620MPa.

3. STRESS CALCULATIONS:

To calculate the maximum tooth stresses, to ensure that the teeth will not be damaged during the operation of the gear pair. However, the shape of a gear tooth makes it impossible to calculate the stresses exactly, using the theory of elasticity. Most of the approximate methods are based on beam theory (Lewis formula) for bending stress calculations and on Hertz theory for contact stress calculations. The stresses in a gear tooth depend primarily on the load and the tooth shape, but there are several other phenomena which must be taken into account. Stress which is often responsible for tooth damage is the tensile stress in the fillet, caused by a tooth load on the face of the tooth. If the tensile stress is too large, fatigue cracks will be formed in the fillet, and the tooth will eventually fracture. It is clear that both the contact stress and the fillet stress must always be calculated, and compared with values which the gear material can sustain without damage.

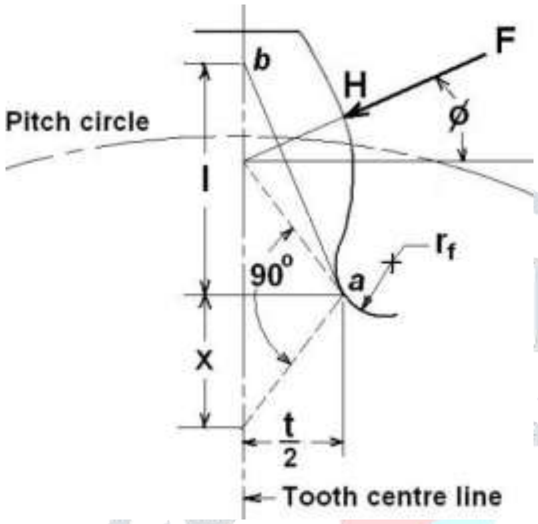


Fig.1 Bending Stress At Root

According to American Gear Manufacturing Association (AGMA) Bending stress at tooth root (σ) is

$$\sigma = \frac{F}{b\ m\ J} K_v K_o K_m$$

$J=\frac{Y}{K_f}$; $K_f = H + \{\frac{t}{r}\}^L + \{\frac{t}{l}\}^M$; where, $H = 0.340 - 0.4583662\,\phi$

$L = 0.316 - 0.4583662\phi$

$M = 0.290 + 0.4583662\phi$

Where, F is the nominal tangential force;b is face width; m is normal module**;** $K_v$is dynamic factor;K_o is overload factor; $K_m$is load distribution factor;J is Spur gear geometry factor; Y is modified Lewis factor; K_f=fatigue stress concentration factor,ϕ is pressure angle.

According to hertz theory, for contact between cylinder and plane, the contact stress can be calculated as

$$\sigma_H = 0.546\sqrt{\frac{\frac{F}{lR}}{\frac{1-v_1^2}{E_1}+\frac{1-v_2^2}{E_2}}}$$

where, l is length of contact line; R is radius of cylinder; v_1 v_2 are Poisson's ratio of cylinder and plane; E_1,E_2 are modulus of elasticity of cylinder and plane respectively

4. MODELLING:

Modeling and Assembly of rack and pinion of jack-up rig is done with the help of CATIA V5 software using the geometric parameters mentioned earlier was shown in fig2.

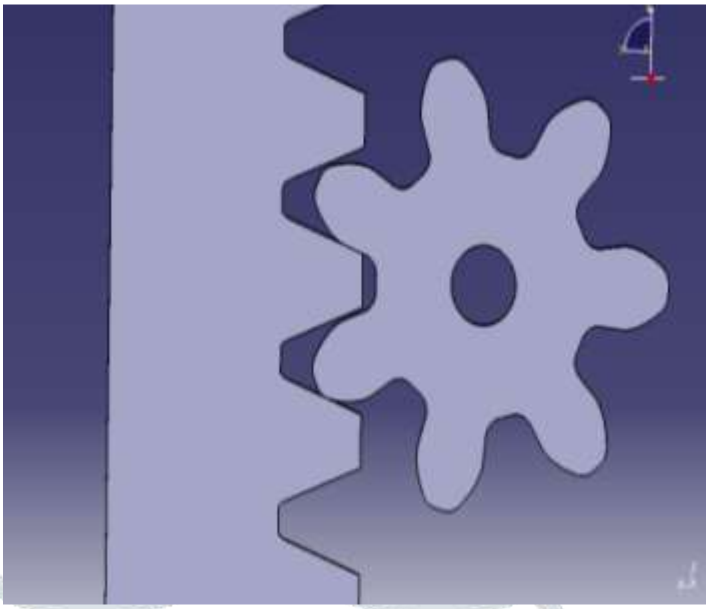


Fig2.CATIA Model of rack and pinion

5. ANALYSIS

In ANSYS workbench meshing is done using solid187element shown in fig3.Boundary conditions were given as per fig4.In this paper preload condition was chosen and load of 300T is acting on the pinion. Since maximum load occurs under preload condition. Different loads are applied under different working conditions such as jacking condition, storm survival, and preload condition and so on

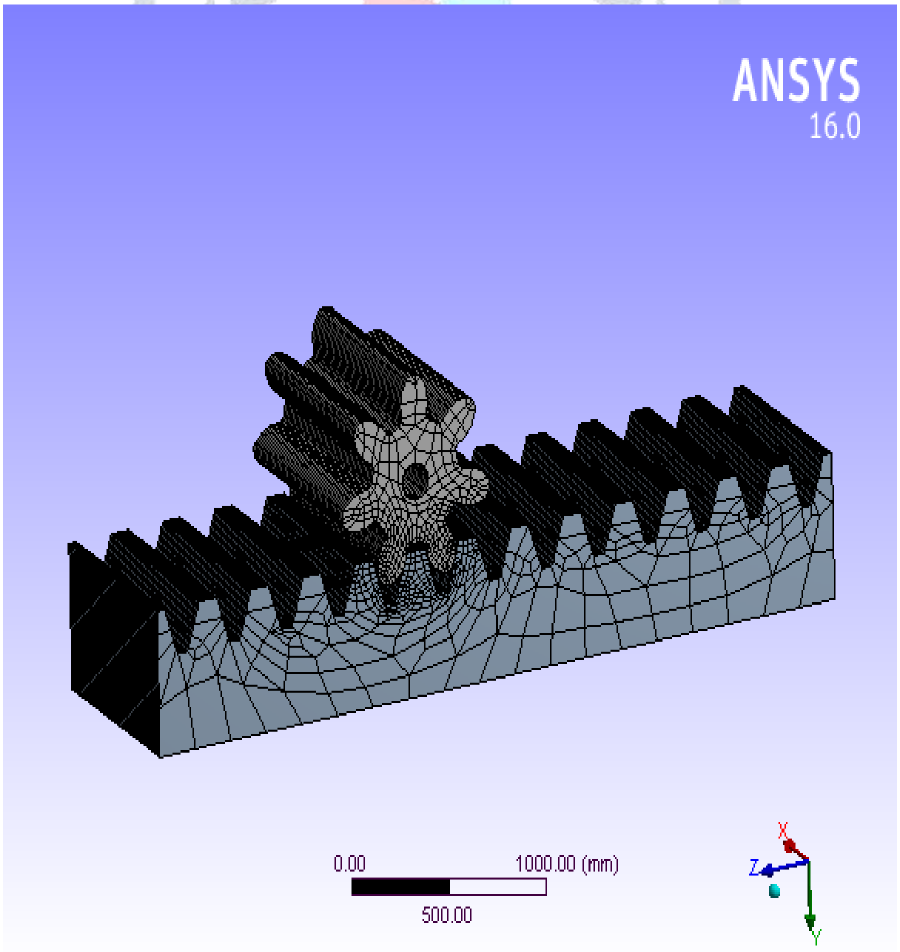


Fig3.mesh of ANSYS model

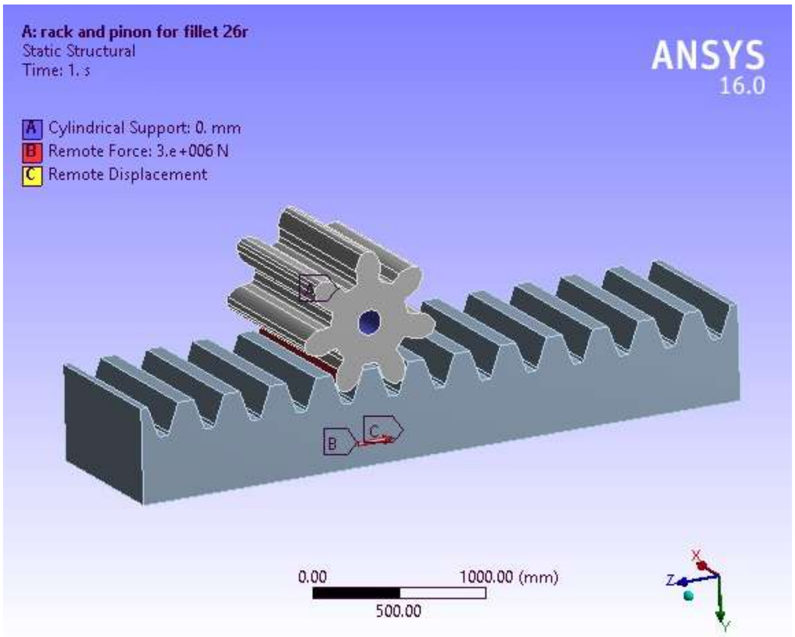


Fig4. Boundary conditions

6. ANSYS RESULTS:

The distribution of von-mises stress and bending stresses on the rack and pinion when they come into engagement for various fillet radii from 26mm to 42 mm with an interval of 2 mm each was shown in fig5. i.e, through parametric calculations in ANSYS,we can clearly obsereve that both von-mises stress and bending stress were minimum for fillet radius of 40mm further, increase in fillet radii causes increase of these stresses. Therefore 40mm radius fillet is considered to be optimum.FEM analysis was done for the following radii.

Name	P1 - FBlend1.FD1	P2 - Equivalent Stress Maximum	P3 - Normal Stress Maximum
Units	mm	MPa	MPa
DP 0 (Current)	26	616.63	523.74
DP 1	28	623.04	508.96
DP 2	30	641.92	498.94
DP 3	32	634.15	457.29
DP 4	34	613.27	427.9
DP 5	36	621.85	459.27
DP 6	38	619.32	453.94
DP 7	40	605.94	417.25
DP 8	42	614.68	431.45

Fig5. ANSYS paramentric results

For fillet radius of 26mm the von-mises stress obtained was 616.63MPa and it is maximum at the tooth root of the pinion and bending stress was 523.74Mpa

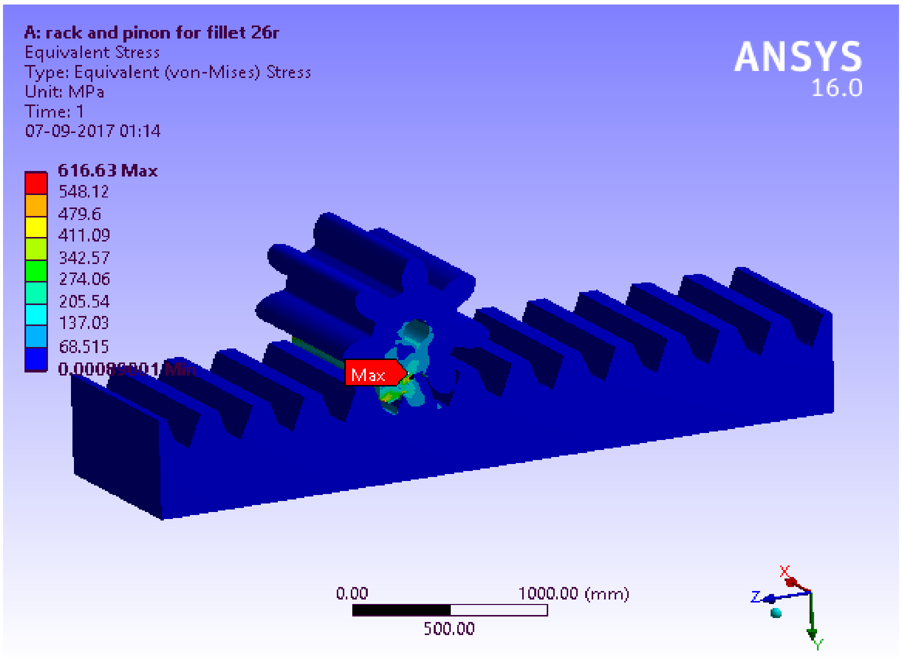


Fig 6. Von-Mises Stress

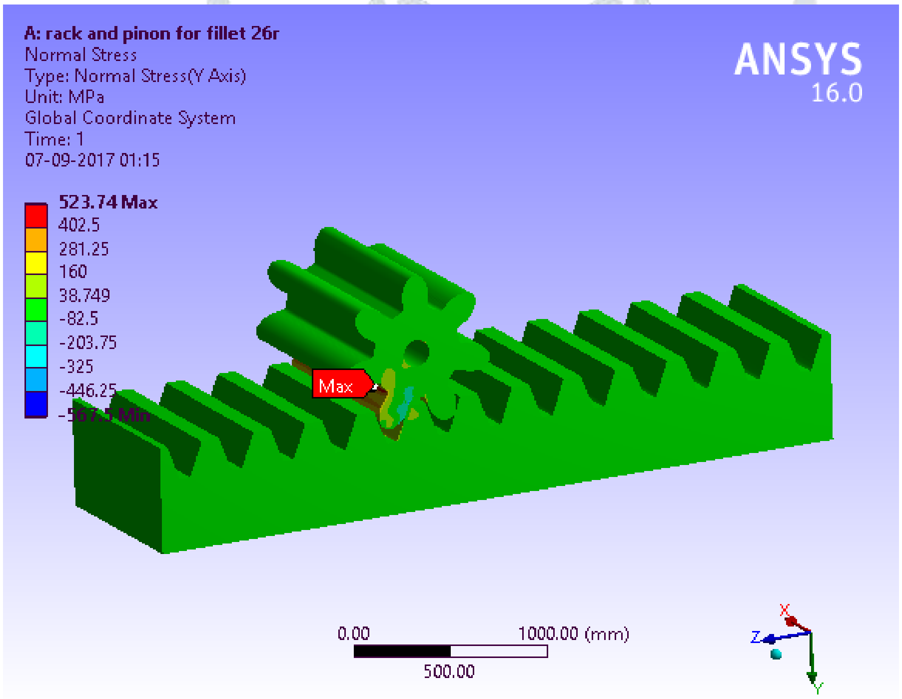


Fig7.Normal Stress

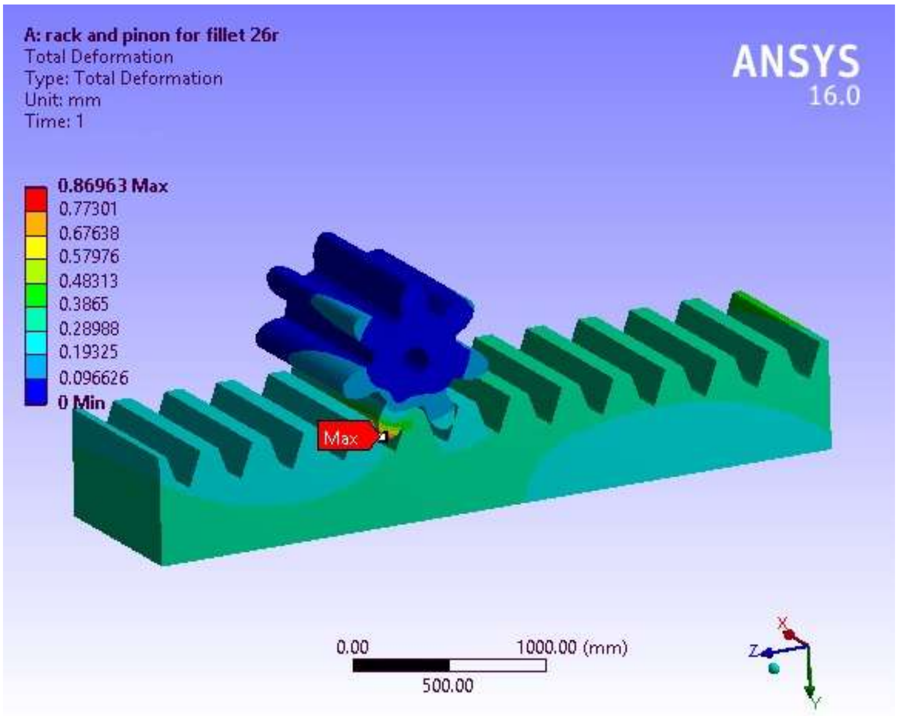


Fig8.Total Deformation

For Fillet radius of 40mm von-mises stress obtained was 605.93MPa and bending stress was 416.38MPa there is a considerable decrease in von-mises stress but bending stress at root has an appreciable amount stress reduction compared to bending stress at 26mm fillet radius therefore fillet radius of 40mm is considered to be optimum.The stresses for all fillet radii were tabuled in fig12.

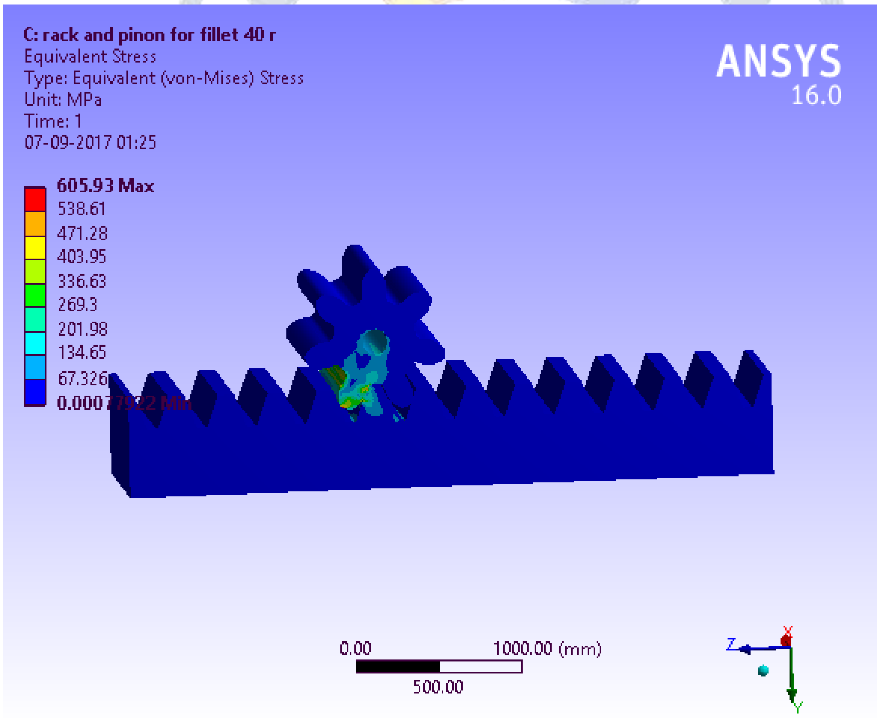


Fig9.Von-Mises Stress

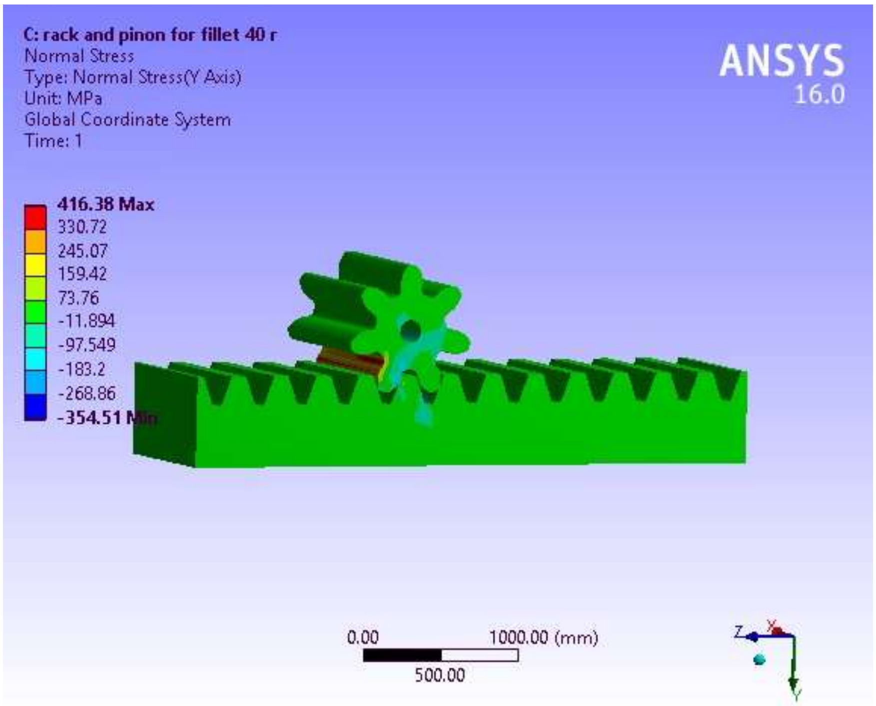


Fig10.Normal Stress

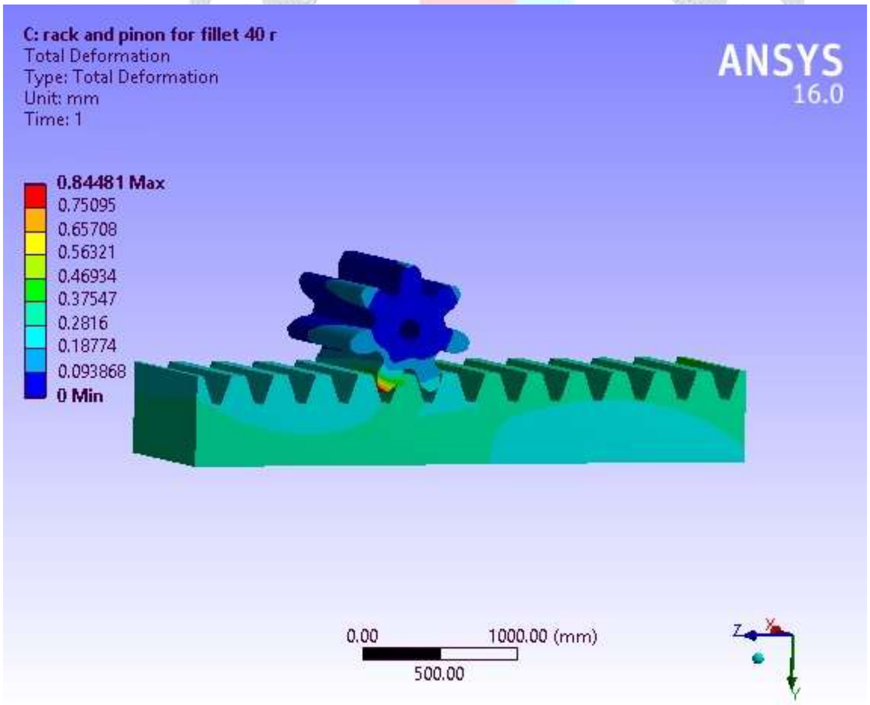
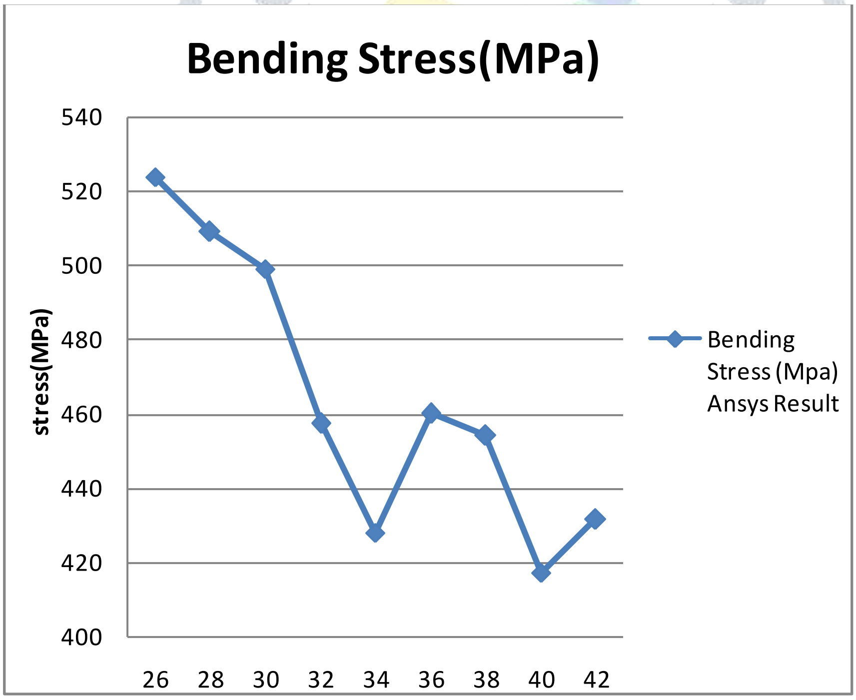


Fig11.Total Deformation

Fillet Radii(mm)	Maximum Equivalent Stress (MPa)	Bending Stress (Mpa)	
		Ansys Result	Theoritical
26	616.63	523.74	561.9512
28	623.04	508.96	557.561
30	641.92	498.94	555.80
32	634.15	457.29	552.73
34	613.27	427.9	557.4
36	621.85	459.94	548.7805
38	619.32	453.94	545.2683
40	605.93	416.38	542.1951
42	614.68	431.45	540

Fig12.Stress Variation for Increasing Fillet Radii



fillet radius (mm)

Fig13.Fillet Radius vs Bending Stress

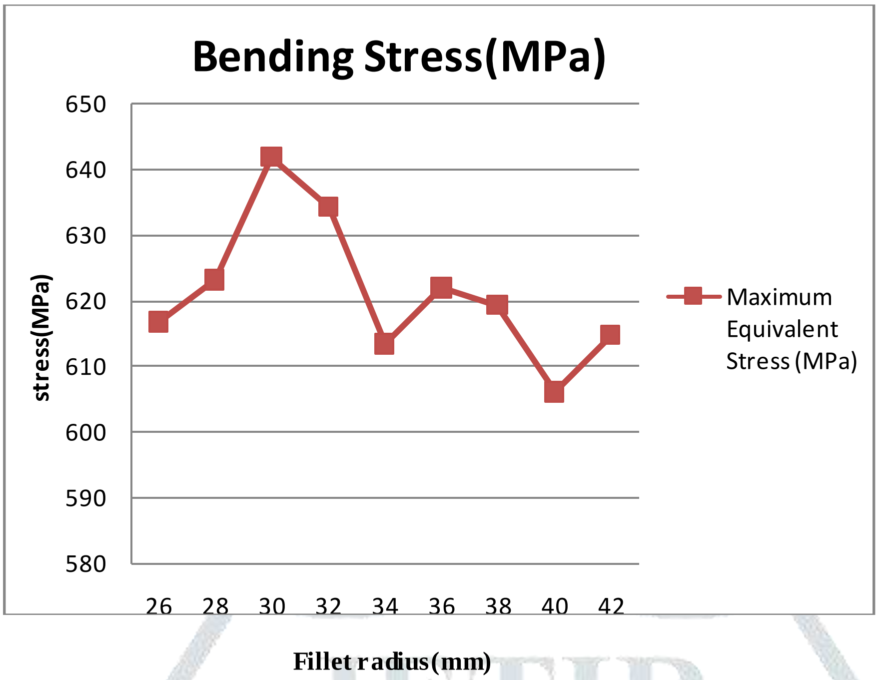


Fig14. Fillet Radius Vs Bending Stress

CONCLUSION:

- From this study, bending stresses at root can be reduced by increasing the radius of the fillet up to a considerable limit and we have found out an optimum fillet for the tooth root of pinion. It can be concluded from the studies that a fillet radius of 40mm is the optimum radius. It shows that with fillet radius of 40mm would yield the lowest bending stress of 416.38MPa.
- All of the results that are exhibited are static stresses produced from constant loads. In actual operation, the rack and pinion experiences dynamic loads variations. However the dynamic loads will be small as the rotational speed of the pinion will be slow when elevating the hull.
- This paper forms efficient and convenient numerical model for the determination of tooth root under static loads. Results in this research show good agreement with analytical results, so this methodology may apply in practice.

FUTURE SCOPE:

- By increasing the module and face width contact stresses can be decreased but that may depend on other factors
- In practical offshore industry, there is no record of filleting the edges of the rack. However from this study, it shows that by chamfering the edges of the rack, one is able to reduce the maximum contact stress, thus reducing mean stress. From there, the chances of fatigue failure can be reduced too.
- Further studies on gear coatings can help us to decrease the contact pressure.

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