



Structural Design Research Through Finite Element Method (FEM) for a Compressor Disc

Merin Christo J, Badarinath B G, Shivakumar S, Umesh V

Abstract: Compressor Discs are subjected to various loads like pressure load, tensile, compressive, buckling, temperature, gravity, etc. during its operation. There are load bearing members and non-load bearing members. The load bearing members are critical components of the disc. Due to these loading condition radial and hoop stresses are generated in the discs. When these stresses exceed yield strength of the material failure occurs. To determine the various stresses acting on a gas turbine compressor disc rotating with constant velocity, initially the Disc is modeled and analyzed using Commercial FEA codes focused on bolt holes and nominal stresses on the cracked disc (a corner crack at a hole and then a through-thickness crack between two holes) are determined. These Stresses can be accounted for the development of advanced crack growth model which can predict the propagation of crack from the critical location of the rotating disc.

Keywords: Compressor Disc, Load bearing members, Critical component, Failure, Stress, Deformation.

I. INTRODUCTION

Compressor disks are failure-critical components in aircraft engines that attract the most research attention in engine life management. Fatigue crack propagation is greatly relevant and need to be predicted from the critical location of the disc e.g., subjection of the bolt to a cyclic loading state. In this way for the prediction of the crack growth behavior, it is necessary to develop an advanced crack growth model that can actually take into account complex loading conditions and biaxial loading conditions. Rotating disks are often subject to biaxial or multi-axial loading conditions during the operation of the aircraft engine. The aim of this work is to examine the stresses acting in a tie bolt holes in a representative gas turbine compressor disc, with particular emphasis on the effect of biaxial stress inherent in a rotating disc. The nominal stresses on the disc (a corner at a hole, then a through-thickness & between two holes), are determined using three-dimensional finite element elastic studies. Therefore, Conventional Linear Elastic Fracture Mechanics

should be still valid for quantify the growth of short crack from the hole [1]. The objective of this work is to determine the various stresses acting on a gas turbine compressor disc rotating with constant velocity of 15000 RPM and 45000 RPM.

W.Z. Zhuang [1] studied the effect of bi-axial loading on the disc subjected to the cyclic rotational speed. He found the stress intensity factor ratio for both bi-axial and uniaxial loading condition. He studied the stress distribution around the bolt hole region for 8000 RPM and 37000 RPM. Hasan C et al. [2] studied the stress analysis on disc made of functionally graded materials (FGM). He studied the tangential and radial stress distribution and displacement of the disc by analytical and numerical methods. He found by reducing the tangential stresses the disc's rotational speed is increased. M. Kumar et al. [3] studied the fatigue crack growth in the nickel based super alloys at two temperature. He predicted crack growth in the aero-engine turbine disc. R.A Claudio et al. [4] studied the propagation of crack in a turbine disc without crack and with two types of crack. He found the critical component of the disc is the notch root and found high stresses developed in those location. Wieslaw Beres et al. [5] studied the residual life of a turbo prop engine by both experimental and numerical methods. Kumar R et al. [6] Performed the elasto plastic analysis on a turbine disc rotating with two different speeds to find the burst margins. He found that the burst margin decreases with increase in speed of rotation of disc and reduced the hoop stress. Sariti Lemi [7] done various design studies through finite element method for a gas turbine disc considering the structural and thermal effects made of Inconel 713LC Superalloy. Vyas, N.S et al [8] investigated the dynamic stress analysis of a turbine blade using life prediction algorithm and studied fatigue life prediction of blades. Horgon C.O et al. [9] investigated the stresses acting on a hollow cylinder with internal and external pressure made of functionally graded, isotropic, linearly elastic materials. Durodola, J. F.et al. [10] investigated the deformation and stress acting on a rotating disc made of functionally graded materials and compared them with theoretical results.

II. FEA ANALYSIS

To determine the state of stresses around the bolt hole region in the disc, a three-dimensional finite element analysis of the disc was carried out using ANSYS. The geometry of the disc is shown in Fig. 1 below;

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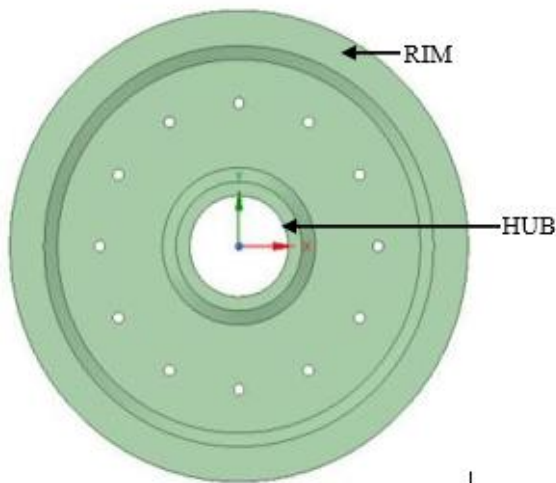


Fig. 1.Full disk model 3D

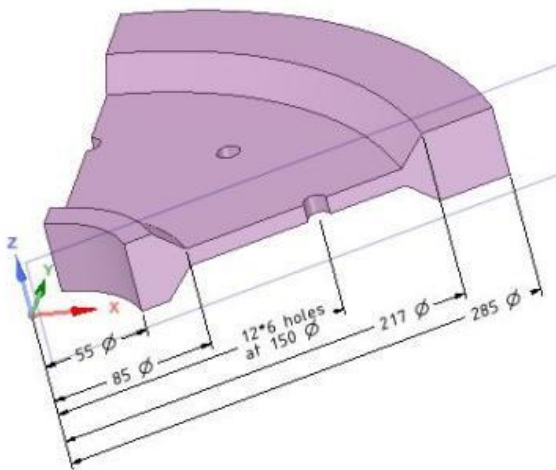


Fig. 2.Model Dimensions

The inner diameter of the disc is 55mm and the outer diameter of the disc is 255mm. The Pitch circle diameter is 150mm and the diameter of the bolt holes is 6mm. The thickness in the region of the bolt holes is 3mm, and the thickness at the hub and the rim is 15mm. The Dimensions are given in Fig. 2. Meshing is done in ANSYS. The finite element mesh of the model is shown in the Fig. 3. below;

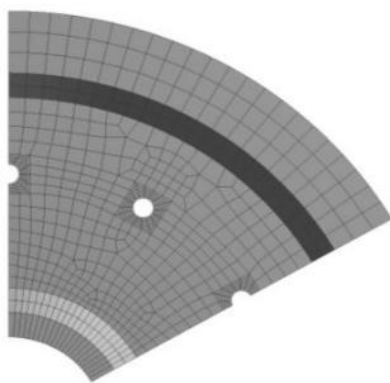


Fig. 3.Mesh model

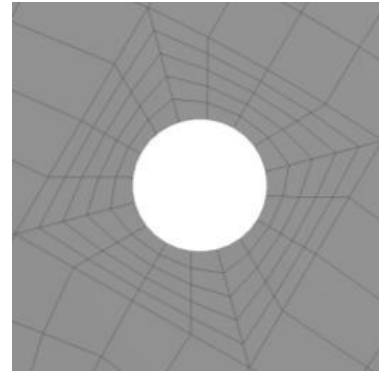


Fig. 4.Mesh around bolt hole region

The meshed model consists of 3225, 3D Brick elements. The mesh is densely captured in the bolt hole region as shown in Fig. 4. Five inflation layers are used around the bolt hole regions. The mesh is more concentrated in the bolt hole region where the stresses are to be studied.

The element quality checks were performed for the critical area in meshed model. The meshed model satisfies the acceptable parameters. The parameters and the actual values are given in Table- I. below;

Table- I: Quality check-critical areas

Parameters	Actual value	Acceptable value
Aspect Ratio	1.9	1 to 4
Jacobian	1.4	> 0.7

The element quality parameters are within the acceptable range in bolt hole region. The disc material used in the FEA analysis of a disc was titanium alloy. Its ultimate tensile stress σ_u 1070 MPa, and yield stress σ_y is 930 MPa. The elastic modulus E is 96 GPa. The Poisson's ratio is 0.36 and the density ρ is 4373 kg/m³.

The loads, Boundary conditions and the analysis cases is given below:

Case(i) 15,000 RPM (or) 1570.79 rad/s.

Case(ii) 45,000 RPM (or) 4712.38 rad/s.

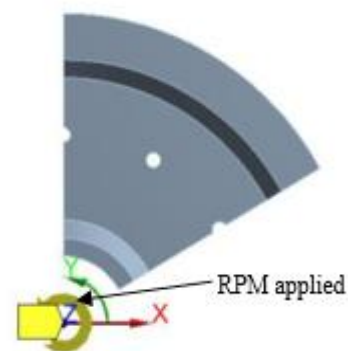


Fig. 5.Applying RPM

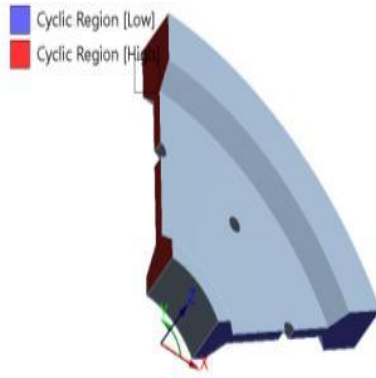


Fig. 6.Cyclic Symmetry Condition

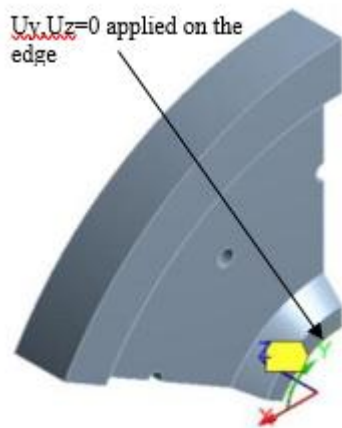


Fig. 7.In Cylindrical coordinate system

III. RESULTS

The following section consists of the detailed discussion of results for 15000 RPM and 45000 RPM. The stress and deformation plots are compared and analyzed for different rotational speed of 15000 RPM and 45000 RPM. The Stress variation across the surface of the disc between two bolt hole region and through the thickness of the disc in the bolt hole region is compared. Then the state of the stress in the bolt hole region is analyzed and discussed. Then the cyclic stress acting on the hole is discussed. Finally, the cyclic local stress strain response is discussed.

A. Results Discussions 15000 RPM

At 15000 RPM the following results were obtained. The Directional deformation and the various stress are obtained as shown in Fig. 8., Fig. 9., Fig. 10., Fig. 11., Fig. 12., Fig. 13. Below;

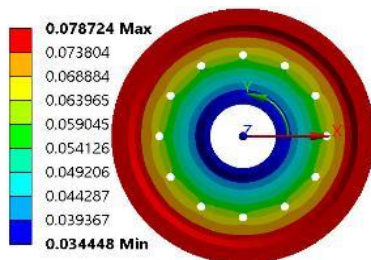


Fig. 8.Radial Deformation

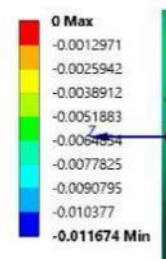


Fig. 9.Axial Deformation

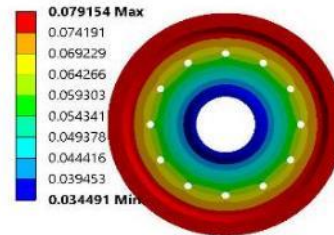


Fig. 10. Total Deformation

Maximum deformation at Rim in Radial direction is 0.078mm due to centrifugal force pulling the disc outward as shown in the Fig. 8. Maximum deformation in Axial direction is 0 mm as shown in Fig. 9. Maximum deformation of 0.079mm is observed on the rim of the disc as shown in Fig. 10.

Max stress of 198 MPa is observed

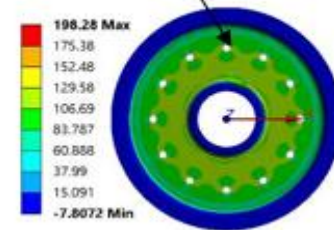


Fig. 11. Radial Stress

Max stress of 245.8 MPa is observed

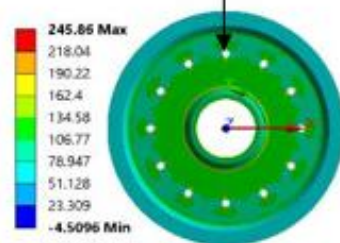


Fig. 12. Hoop Stress

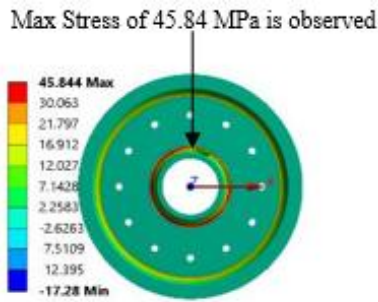


Fig. 13. Axial Stress

The radial stress is minimal in the hub and the rim. The maximum radial Stress is 198 MPa and it is observed in all bolt hole regions of the disc as shown in Fig. 11. The maximum hoop stress is 245.8 MPa and is observed in all bolt hole region as shown in Fig. 12. The maximum axial stress is 45.84 MPa and it is observed in the region as shown in Fig. 13.

1) Radial Stress Variation Along Hole Thickness & between two bolt holes

In the below section the variation of the radial stress through the thickness of the hole and between the bolt holes is discussed

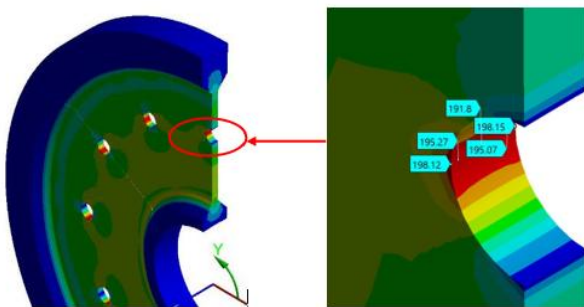


Fig. 14. Radial Stress region

Fig. 15. Radial Stress Variation through thickness

The maximum radial stress is found in the bolt hole region as shown in the Fig. 14. It is found that the stress 198 MPa is varying through-thickness of the hole and the minimum stress 191 MPa is found at the middle and is again gradually increasing towards the other edge of the bolt hole as shown in Fig. 15.



Fig. 16. Radial stress variation along the corner at 15000 RPM

The variation of stress between two bolt holes is seen in the Fig. 16. It is observed that the stress (198 MPa) is gradually decreasing from the corner of the bolt hole region and it is found minimal (103.68 MPa) at the middle and is gradually increasing again and found maximum (198 MPa) at

the edge of the other bolt hole region. The Average Remote stress is 104 MPa.

2) Hoop stress Variation through thickness of hole & along surface

A detailed study is done to understand the behavior of the hoop stress variation through the thickness of the bolt hole and between two bolt holes as shown in the Fig. 17 and Fig. 18.

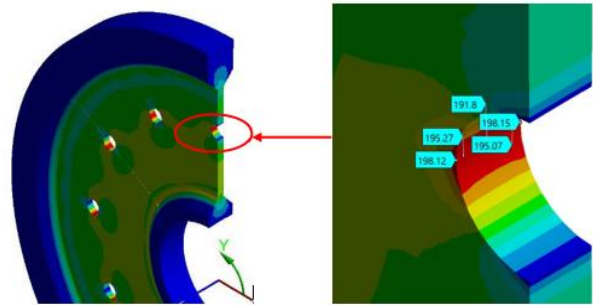


Fig. 17. Hoop Stress region

Fig. 18. Hoop Stress Variation through thickness

The maximum hoop stress is found in the bolt hole region as shown in the Fig. 17. It is found that the stress is varying through-thickness of the hole and the maximum stress (246 MPa) is found on the middle of the region as shown in Fig. 18.



Fig. 19. Hoop stress variation along surface at 15000 RPM

The hoop stress is gradually decreasing from the edge of the bolt hole towards the other side of the disc as shown in Fig. 19. The stress value at the edge of the hole is 234 MPa. The hoop stress is maximum when compared with the radial stress acting on the rotating disc. The Average Remote stress is 104 MPa.

B. Results Discussions 45000 RPM

At 45000 RPM the following results were obtained. The Directional deformation and the various stress are obtained as shown in Fig. 20., Fig. 21., Fig. 22., Fig. 23., Fig. 24., Fig. 25. below;

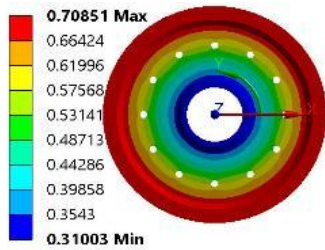


Fig. 20. Radial Deformation

Max stress of 412 MPa is observed

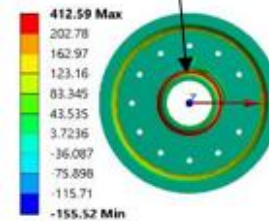


Fig. 25. Axial Stress

The Radial stress is minimal in the hub and the rim. The maximum radial Stress is 1784 MPa and it is observed in all bolt hole regions of the disc as shown in Fig. 23. The maximum hoop stress is 2212 MPa and is observed in all bolt hole region as shown in Fig. 24. The maximum axial stress is 412 MPa and it is observed in the region as shown in Fig. 25.

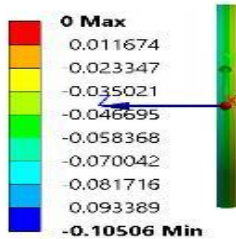


Fig. 21. Axial Deformation

1) Radial Stress Variation along the hole thickness & between bolt holes

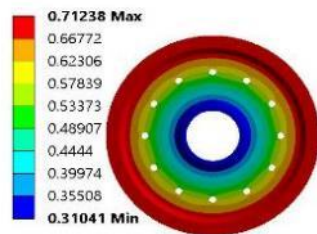


Fig. 22. Total Deformation

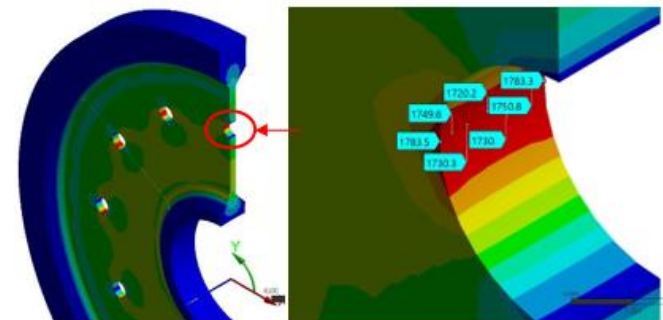


Fig. 26. Radial Stress region

Maximum deformation in Radial direction is 0.7 mm at Rim due to centrifugal force pulling disc outward as shown in Fig. 20. Maximum deformation in Axial direction is 0 mm as shown in Fig. 21. Maximum deformation of 0.7 mm is observed on the rim of the disc as shown in Fig. 22.

Fig. 27. Radial Stress Variation through thickness

The maximum radial stress is found in the bolt hole region as shown in the Fig. 26. It is found that the stress 1784 MPa is varying through-thickness of the hole and the minimum stress 1720 MPa is found at the middle and is again gradually increasing towards the other edge of the bolt hole as shown in Fig. 27.

Max stress of 1784 MPa is observed

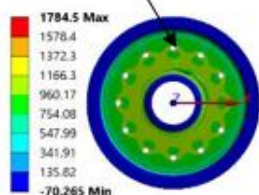


Fig. 23. Radial Stress



Fig. 28. Radial stress variation along the corner at 45000 RPM

Max stress of 2212 MPa is observed

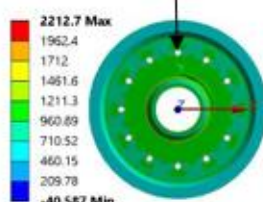


Fig. 24. Hoop Stress

The variation of stress between two bolt holes is seen in the Fig. 28. It is observed that the stress (1784 MPa) is gradually decreasing from the corner of the bolt hole region and it is found minimal (933 MPa) at the middle and is gradually increasing again and found maximum (1784 MPa) at the edge of the other bolt hole region. The Average Remote stress is 900 MPa.

2) Hoop stress Variation through thickness of hole & along surface

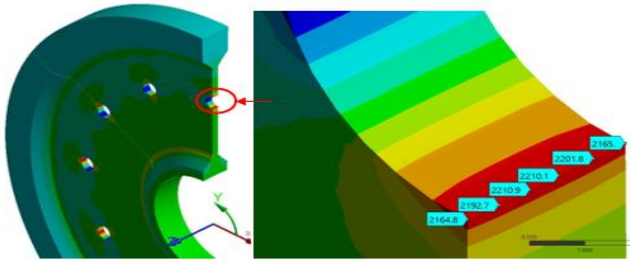


Fig. 29. Hoop Stress region

Fig. 30. Hoop Stress Variation through thickness
Fig. 31.

The maximum hoop stress is found in the bolt hole region as shown in the Fig. 29. It is found that the stress are varying through–thickness of the hole and the maximum stress (2212 MPa) is found in the middle region as shown in Fig. 30.



Fig. 32. Hoop stress variation along surface at 45000 RPM

The hoop stress is gradually decreasing from the edge of the bolt hole towards the other side of the disc as shown in Fig. 31. The stress value at the edge of the hole is 2132 MPa. The hoop stress is minimum when compared with the radial stress acting on the rotating disc. The Average remote stress is 900 MPa.

C. Results Detailed Discussions

In this section, the state of stress around the bolt hole region, the radial cyclic stress, the cyclic stress strain response of the disc is discussed.

The stress acting on the bolt hole region is equi–biaxial. At 45000 rpm, the radial/hoop remote stress is 900 MPa and $\lambda^*=1$. where λ is the ratio of stress along x-axis to that of Y-axis. At 15000 rpm, the radial/Hoop remote stress is 104 MPa, $\lambda^*=1$.

According to Theory of Elasticity,

- Hoop stress at the edge of hole (2212 MPa) is more than twice the Remote stress (900 MPa)
- Since the remote stress (900 MPa) is greater than half the yield strength (930 MPa) of the material the hole edge undergoes plastic yielding.

At the minimum rotation speed of 15000 RPM, the radial stress is equal to 104 MPa and at the maximum rotation speed of 45000 RPM, the radial stress is equal to 900 MPa.

$$\text{Stress Ratio} = \frac{\sigma_{min}}{\sigma_{max}} = \frac{104}{900} = 0.1$$

The cyclic stress amplitude is 796 MPa which is less than the material yield strength (930 MPa). Therefore, after plastic deformation, elastic shakedown occurs and there is no cyclic plasticity around the hole.

Further, the cyclic local stress strain response for elastic material and for the elastic plastic material is compared and analysed. The details are as follows,

During first half cycle, local plastic yielding would occur around the hole edge when the stress reaches 900MPa (Remote stress).

For elastic material the stress concentration factor is 2.3 and hence during cyclic loading the stress at the edge of the hole varies from 70 MPa to 1496 MPa during the rotation.

$$\text{Stress Concentration factor } K_t = \frac{\sigma_{max}}{\sigma_{min}} = \frac{2212}{900} = 2.4$$

Using Neuber's rule;

$$\text{Total strain} = \epsilon_0 = \frac{\sigma_{ys}}{E} = \frac{930 * 10^6}{96 * 10^9} = 0.0096$$

$$\text{Yield (Elastic) Strain} = \epsilon_{total} = \frac{(K_t * \sigma)^2}{(\sigma_{ys} * E)} = 0.0522$$

Thus, total strain (0.0522) is twice the yield(elastic) strain (0.0096) of the material.

IV. CONCLUSION

The compressor disc model with bolt holes is analyzed for Centrifugal force. When the Disc with a bolt hole Is made to rotate with constant velocity the hoop stress is higher than the radial stress. The maximum stress is found in the bolt hole region. The summary of the results obtained are tabulated in the Table-II given below,

Table- II: Results Summary

Rotational Velocity (RPM)	15000	45000
Max Radial Sress (Mpa) (in bolt hole)	198	1784
Max Hoop Stress (Mpa) (in bolt hole)	246	2212
Remote Stress (Mpa)	104	900

From the results summary it's clear that the stress acting on the bolt hole region is equi –biaxial. The hoop stress is more than the Radial stress for both the rotational velocity. According to Theory of Elasticity, Hoop stress at the edge of hole 2212 MPa) is more than twice the Remote stress (900Mpa). Since the remote stress(900MPa) is greater than half the yield strength (930MPa) of the material the hole edge undergoes plastic yielding. The cyclic stress amplitude is 796 MPa which is less than the material yield strength (930 MPa). Therefore, after plastic deformation, elastic shakedown occurs and there is no cyclic plasticity around the hole.

For elastic material the stress concentration factor is 2.4 and hence during cyclic loading the stress at the edge of the hole varies from 246 MPa to 2212 MPa during the rotation. Thus, total strain (0.0522) is twice the yield(elastic) strain (0.0096) of the material.

V. FUTURE WORK

Since the cyclic stress amplitude at the hole edge is equal to 796 MPa. Slightly above $2/3^{\text{rd}}$ material yield strength of 930 MPa, Conventional Linear Elastic Fracture Mechanics should be still applicable to quantifying the growth of short cracks emanating from the hole. However, elastic-plastic stress distribution along prospective crack path is required to determine max. stress intensity factors, while, the range of stress intensity factor can be determined using elastic solutions [1]. Regarding the validation of the results through experiments, some experimental tests are already conducted in literature. Because experimental validation of these type of disc is very expensive and hard to be conducted. Nevertheless, the Experimental results will be in correlation with the Finite element analysis. For a future studies one can perform Rig test for a given Disc model to compare the experimental results with the numerical results obtained from our studies.

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