Stress Analysis And Optimization of Rocket Gas Turbine Disk

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Abstract- *Turbopump is one of the main modules of liquid rocket engine and represents a large part of rocket engine overall cost. So improving the efficiency of turbopump and reducing its weight are effective ways to improve the performance of rocket engine. Turbopump includes group of blades and rotating disk. Rotating disks from the past are of great importance to designers. One of the best examples of such applications is gas turbine disks. Rotating disks undergoes mechanical load as well as thermal load. A disk experiences pressure internally because of being shrink-fitted onto its mounting shaft. Also the blade placed on its external boundary cause an outer load, which will increase the load on outer edge. Blades experiencing high temperature gases create a temperature field to vary on the disk. In present study, the rotating disk is optimized by numerical simulation method [ANSYS].*

Keywords- Turbopump, Liquid Rocket Engine, Rotating Disks, Numerical Simulation Method

I. INTRODUCTION

Rotating disks from the past are of great importance to designers. One of the best examples of such applications is gas turbine disks. Rotating discs undergoes mechanical load as well as thermal load. A disk experiences pressure internally because of being shrink-fitted onto its mounting shaft. Also the blade placed on its external boundary cause an outer load, which will increase the load on outer edge. Blades experiencing high temperature gases create a temperature field to vary on the disk. A finite element analysis should be carried out to investigate damage mechanisms of turbine disk along with consistent high levels of stress caused by enormous rotational speed. One more study utilized logical result to display intensity of circumferential stress element wrt radial stress element in rotating disk.

II. LITERATURE SURVEY

L.Witek[1] discussed about the failure analysis of the turbine disk of an aero engine, installed in a certain type of aircraft. A non-linear finite element method was utilized to determine the stress state of the disk/blade segment under operating conditions. A computation was also performed with excessive rotational speed. In this study, attention is devoted to analysis of the damage mechanisms of the turbine disc subjected to both operational and over speed conditions and also to indicate critical areas, from the point of view of the stress analysis. The additional goal of this analysis is to improve the safety and reliability of the aircraft and different planes, powered by the same type of engine.

H. Jaed, B. Fashi, J. Bidaadi[2] discussed about the minimum weight design of inhomogeneous rotating disks. Gas turbine disks have various applications in aerospace industry such as in turbojet engines. In general disks operate under intensive heat while experiencing extreme angular velocities. Decreasing the mass of disk in aeronautical applications will lead to improve such as reduced dead weight and costs. Enormous rotation speed causes the disk to experience huge centrifugal force and also the presence of extreme temperature decreases the strength of the disk material. Thus, in turn increases the disk deformations under the loads applied. As to attain a consistent disk analysis and close to the corresponding true stress-distribution, results should take into account changes in properties of material since the disk throughout experiences the temperature field. To succeed this aim, inhomogeneous disk model with varying thickness is to be taken into account. Utilizing the method of varying properties of material, stresses are realized for the disk subjected to rotation and steady temperature field. This is accomplished by modelling rotating disk as string of rings of diverse but continuous properties. The ideal disk profile is obtained by successively proportioning the thickness of every ring to accomplish stress needs. This procedure vice-versa a numerical programming method for improvising shows considered advantages. First thing, it is straight forward iterative integral in every design phase not needing numerical operations. In addition, due to easiness it increases the requirement of specific simplification is mutual in difficult mathematical methods. The solutions received, correlated to those published show superiority and agreement.

III. DESCRIPTION OF GAS TURBINE DISK

This section provides outlines of the disk geometry, material, and operating conditions.

A. Disk Geometry

Figure 1 shows turbine disk under study. This rotating disk has a varying thickness ranging from 0.008 m to 0.014 m. The outer diameter of the disk is 0.180 m. The complete measurements of the varying disk are illustrated in figure 2. In addition disk carries a hub where the shaft of the turbine is attached [3].

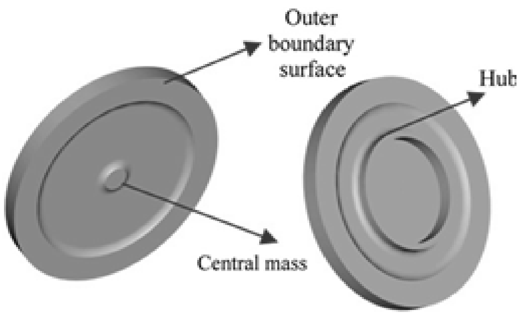

Fig. 1. Gas turbine disk

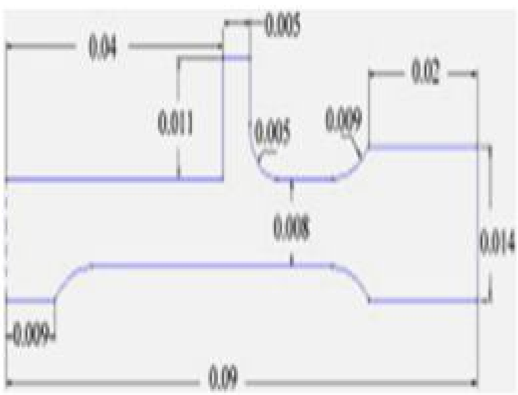

Fig. 2. Complete dimensions of disk cross section (unit: m)

B. Disk material

The material used for gas turbine disk is chrome-nickel steel alloy. The properties of material varied with temperatures ranging from O°c up to 8OO°c as shown in table I.

TABLE I
Engineering Constants

Data	Temperature/°C						
Data	0	100	200	300	400	600	800
K	23	23	23	23	25	25	27
***α*/10 −6**	9.23	9.35	9.55	9.68	9.78	9.88	10.0 5
Data	**Temperature/°C**						
	30	90	200	310	415	590	825
E	205	190	180	175	165	152	138
N	0.29 3	0.29 7	0.30 2	0.30 8	0.31 4	0.32 4	0.33 6

Where *E*: Young's modulus (GPa)
ν: Poisson's ratio
K: Thermal conductivity (W/ (m·°C))
ρ: Density=7835 kg/m^3
α: Coefficient of thermal expansion, α=10×10^{-6} °C
CP: Specific heat, *CP*=460 J/ (kg·°C)

C. Disk Operating Conditions

In operation turbine rotates at a speed of 38O rps and a power of 750 kW is generated. The blades present in the disk outer periphery are not shown in figure 6.1, but the disk experiencing thermal and mechanical effects due to the blades were considered. Due to the blades enormous rotational speed causes high centrifugal force, which is the prime reason for mechanical effects on the disk. The mechanical effect was simulated by pressure of 45 MPa uniformly distributed at the disk outer periphery. This simplification ignores stress concentration at the connection area between the disk and blades. Although it was appropriate in evaluating the strength ȯf the disc, a 3D elastic–plastic analysis should be employed for a fatigue calculation for the blade. However the thermal effect is resulted from hot blades attached to the disk was indicated at the disk outer periphery by enormous temperature of 600°c. Heats were transmitted from disk outer periphery to the disk center for successive 100 s because of thermal conduction indicating one life cycle operation of turbine. The shaft of the turbine was shrink-fitted into the hub of the disk for the transmission of torque. The interference amount between the shaft outer diameter and the hub inner diameter was 0.0001 m and extended 0.01 m along the hub length. Logically induced pressure contact of 26 MPa. Even though shaft not been shown to evaluate the stress levels rising from shrink-fitted load, value were simulated as a pressure load on the hub inner surface.

IV. FINITE ELEMENT MODELLING

A. Meshing

The disk was simulated by a three-dimensional (3D) section. The disk was modelled using the commercial multi-purpose FE software package ANSYS. The geometry of the 3D non-uniform disc was meshed with tetrahedron elements having three translational degrees of freedom (DOF) per node (UX, UY and UZ). The total number of elements used was 5934. The mesh and element shape are illustrated in figure 3.

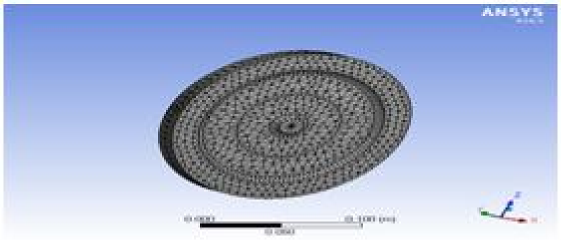

Fig. 3. Fine mesh of disk

B. Loads and boundary conditions

Four different types of loads were applied to the model: a rotational load, shrink-fitted load, blades outer boundary load, and thermal load. The rotational load was simulated by an inertial velocity in the z-direction of 380 rps. The shrink fit load caused by the shaft is represented by a pressure of 26 MPa, and it was applied to the inner surface of the hub. The load from the blades was represented by a uniformly distributed pressure of 45 MPa at the outer boundary surface of the disk. Finally, the thermal load was represented by a temperature of 600 ˚C at the outer boundary surface of the disk. For symmetrical boundary conditions for the displacement of the disk, all of the nodes at the inner vertical surface of the hub were fixed in the z-direction, which represented the actual fixation method for the disk to the turbine shaft.

1. Rotational Load:

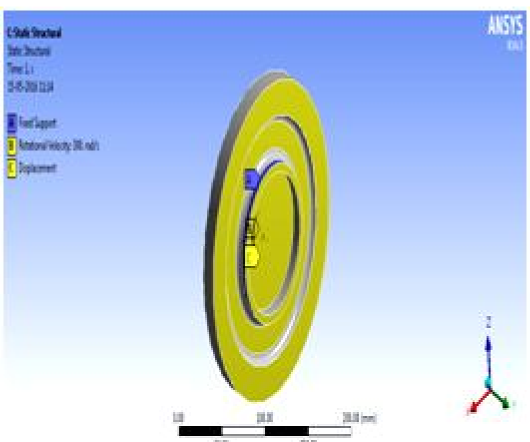

Fig. 4. Disk subjected to rotational load

The rotational load was simulated by an inertial velocity in the z-direction of 380 rps. Part A represents a fixed support, part B represents a rotational velocity of 380 rps and part C represents a displacement as shown in figure 4.

2. Shrink-fitted Load:

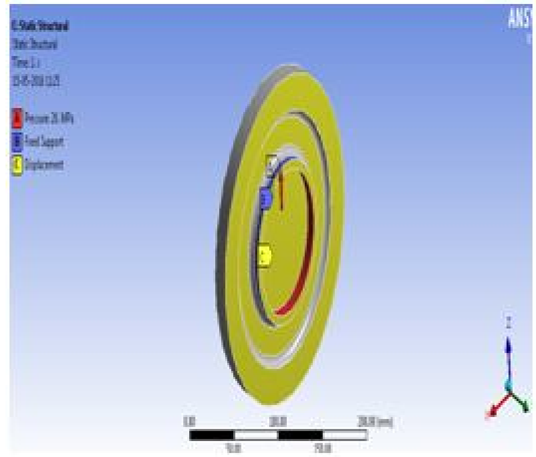

Fig. 5. Disk subjected to shrink-fitted load

The shrink-fitted load caused by the shaft is represented by a pressure of 26 MPa, and it was applied to the inner surface of the hub. Here part A represents pressure of 26 MPa, part B represents a fixed support and part C represents a displacement. As shown in figure 5.

3. Blade Load:

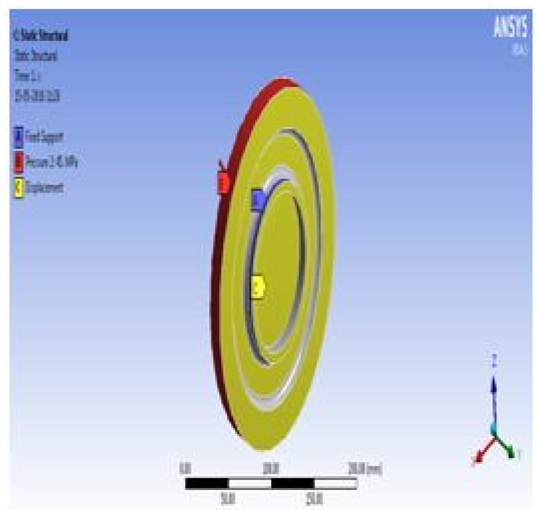

Fig. 6. Disk subjected to blade load

The load from the blades was represented by a uniformly distributed pressure of 45 MPa at the outer boundary surface of the disk. Here part A represents a fixed support, part B represents a pressure of 45 MPa and part C represents a displacement as shown in figure 6.

4. Thermal Load:

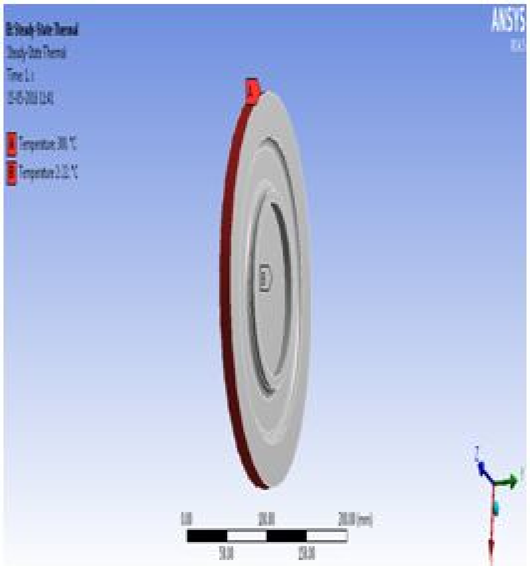

Fig. 7. Disk subjected to thermal load

Finally, the thermal load was represented by a temperature of 300 °C at the outer boundary surface of the disk. Here the part A represents the temperature at the outer periphery of the disk and the part B represents the temperature at the centre of the disk as shown in figure 7.

5. Combined Load:

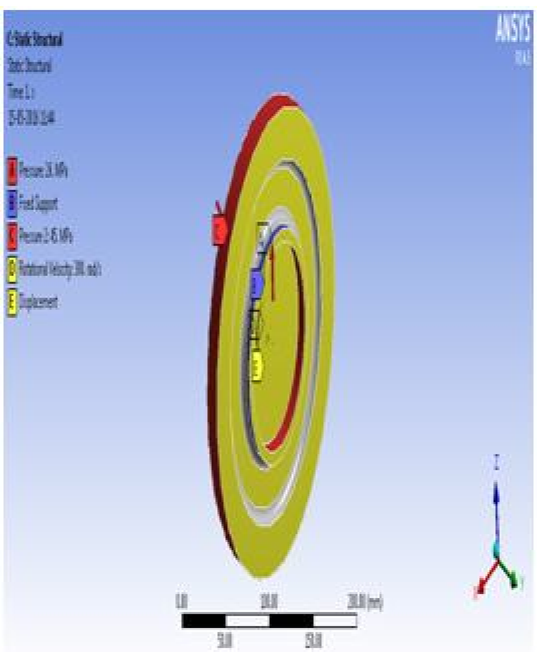

Fig. 8. Disk subjected to combined load

Here part A represents a pressure of 26 MPa that is shrink-fit load, part B represents a fixed support, part C represents a pressure of 45 MPa that is blade load, part D represents a rotational velocity of 380 rps and part E represents a displacement as shown in figure 8.

C. Goal driven optimization

Goal Driven Optimization is a set of constrained, multi-objective optimization techniques in which the “best" possible designs are obtained from a sample set given the goals you set for parameters. A Goal Driven Optimization study allows you to determine the effect on input parameters with certain objectives applied for the output parameters. To do this, you specify a series of design goals or objectives that will be used to generate an optimized design; you can define the optimization domain, specify values for input and response parameters, and weight goals in terms of their importance. Based on your specifications, Design Explorer then generates a set of sample designs from which you can select the most promising candidate designs. A response surface optimization system draws its information from its own response surface component, and so is dependent on the quality of the response surface. The available optimization methods (Screening, MOGA, NLPQL, and MISQP) utilize response surface evaluations, rather than real solves. Response surfaces are built from the design of experiments and quickly provide approximated values of output parameters throughout the design space [4].

D. Modal analysis

Any physical system can vibrate. The frequencies at which vibration naturally occurs, and the modal shapes which the vibrating system assumes are properties of the system, and can be determined analytically using Modal Analysis. Analysis of vibration modes is a critical component of a design, but is often overlooked. Structural elements such as chassis can be particularly prone to perceptible vibration or disturbing sensitive equipment. Inherent vibration modes in structural components or mechanical support systems can shorten equipment life, and cause premature or completely unanticipated failure, often resulting in hazardous situations. Detailed fatigue analysis is often required to assess the potential for failure or damage resulting from the rapid stress cycles of vibration. Detailed modal analysis determines the fundamental vibration mode shapes and corresponding frequencies. This can be relatively simple for basic components of a simple system, and extremely complicated when qualifying a complex mechanical device or a complicated structure exposed to periodic loading. These systems require accurate determination of natural frequencies and mode shapes using techniques such as Finite Element Analysis.

V. RESULTS AND DISCUSSIONS

The FE package was used to perform linear static analyses to solve five different load cases. These load cases were the rotational load (load case 1), shrink-fitted load (load case 2), blades outer boundary load (load case 3), temperature gradient as a thermal load (load case 4), and combination of all the loads (load case 5). The results in terms of the Von Mises stress distributions are shown below.

A. Rotational Load

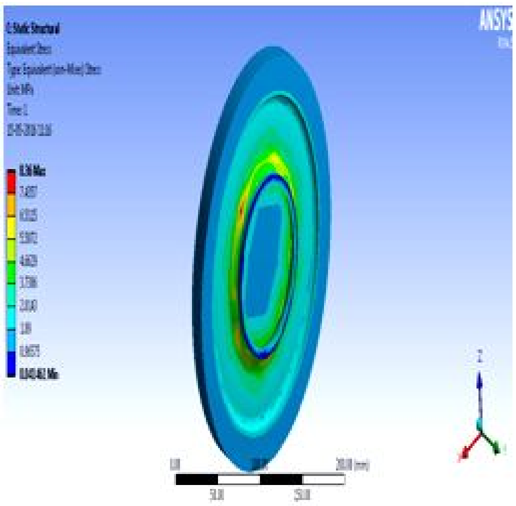

Fig. 9. Equivalent stress of disk subjected to rotational load

For the case of rotational load, an equivalent stress has a maximum of 8.36 MPa and a minimum of 0.041 MPa was recorded. The maximum equivalent stress occurs at the centre of the disk as shown in figure 9.

B. Shrink-fitted Load

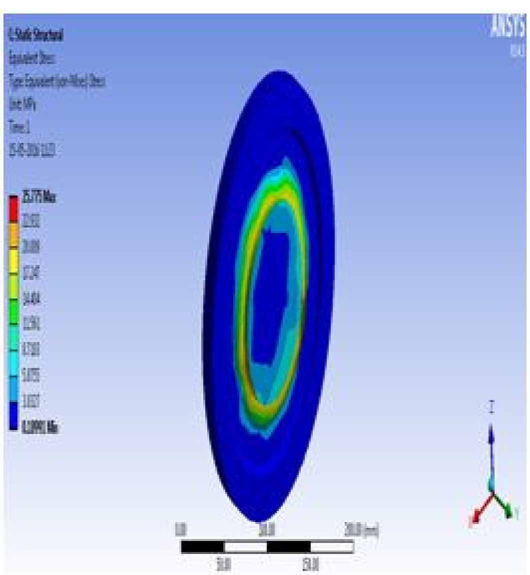

Fig. 10. Equivalent stress of disk subjected to shrink-fitted load

For the case of shrink-fitted load, an equivalent stress has a maximum of 25.775 MPa and a minimum of 0.18991 MPa was recorded. The maximum equivalent stress occurs at the outer radius of the hub as shown in figure 10.

C. Blade Load

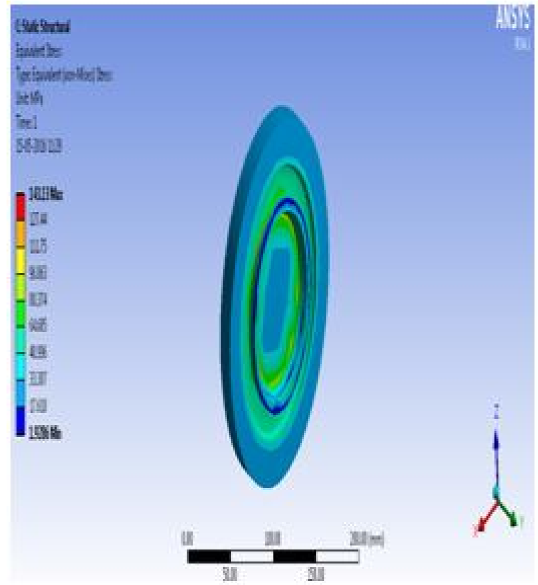

Fig. 11. Equivalent stress of disk subjected to blade load

For the case of blade load, an equivalent stress has maximum of 143.13 MPa and a minimum of 1.9286MPa was recorded. The maximum equivalent stress occurs at the outer radius of the hub as shown in figure 11.

D. Thermal Load

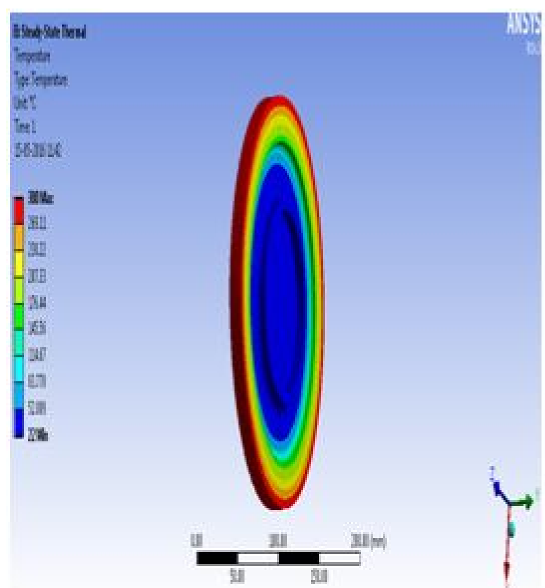

Fig. 12. Temperature distribution of disk subjected to thermal load

For the case of thermal load, maximum temperature of 300˚c is found at the outer periphery of the disk and minimum temperature of 22˚c is found at the centre of the disk as shown in figure 12.

E. Combined Load

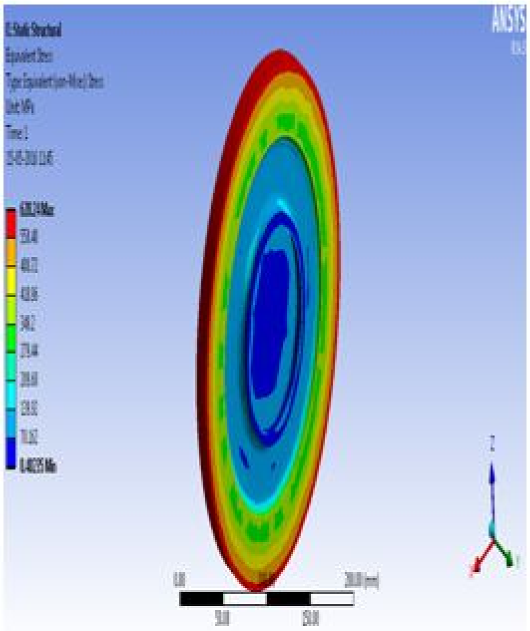

Fig. 13. Equivalent stress of disk subjected to combined load

For the case of combined load, an equivalent stress has a maximum of 628.24 MPa and a minimum of 0.4023 MPa was recorded. The maximum equivalent stress occurs at the outer periphery of the disk as shown in figure 13.

F. Optimization of the disk

1. Design of Experiments (DOE):

TABLE II
DOE possible design points

Name	Web Thickness	Fillet Radius	Solid Mass (kg)	Equivalent Stress Maximum (Pa)
1	5	4.75	2.4569	6.278ė+08
2	3	4.75	2.3179	6.2518ė+08
3	7	4.75	2.6003	6.3143ė+08
4	5	4	2.4522	6.2834ė+08
5	5	5.5	2.4625	6.2766ė+08
6	3	4	2.3133	6.2436ė+08
7	7	4	2.5954	6.3027ė+08
8	3	5.5	2.3233	6.2541ė+08
9	7	5.5	2.606	6.3101ė+08

In the DOE workspace, we have to set the bounds for input parameters i.e., web thickness and fillet radius respectively. We have set the lower and upper bounds for web thickness as 3 mm and 7 mm and for fillet radius as 4 mm and 5.5 mm respectively. The table of design points show a total of 9 automatic design points. Here solid mass has a minimum value of 2.3133 kg and a maximum value of 2.606 kg. Similarly fillet radius has a minimum value of 6.2498E+08 Pa and a maximum value of 6.3092E+08 Pa. The optimized value of the disk weight is 2.3133 kg.

TABLE III
Comparison of before and after optimization with respect to equivalent stress and weight of the disk

Object name	Equivalent stress in Mpa		Weight of the disk in kg
	Minimum	Maximum	
Before optimization	0.40235	628.24	2.5
After optimization	1.3717	624.36	2.3133

2. Equivalent stress before optimization of the disk:

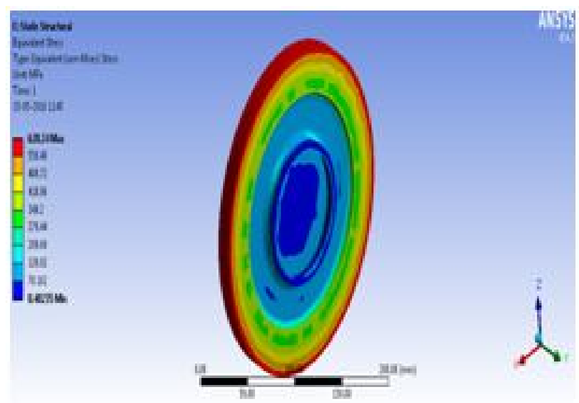

Fig. 14. Equivalent stress of the disk before optimization

Before optimization, an equivalent stress has a maximum of 628.24 MPa and a minimum of 0.4023 MPa was recorded. The maximum equivalent stress occurs at the outer periphery of the disk as shown in figure 14.

3. Equivalent stress after optimization of the disk:

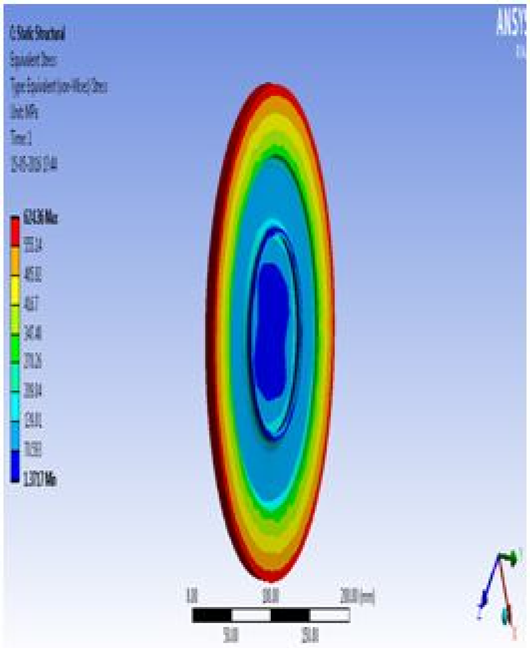

Fig. 15. Equivalent stress of the disk after optimization

After optimization, an equivalent stress has a maximum of 624.36 MPa and a minimum of 1.3717 MPa was recorded. The maximum equivalent stress occurs at the outer periphery of the disk as shown in figure 15.

4. Modal Analysis:

The modal analysis is carried out on disk to determine the natural frequency. The induced natural frequency is shown in TABLE IV.

TABLE IV
Modal analysis frequency list

Mode No	Natural Frequency
1	2159.2
2	2164.7
3	2349.7
4	2893.7
5	2903.7
6	5213.1

The first natural frequency is observed at 2159.2Hz and mode shape is shown in Fig. 16 to Fig. 21.

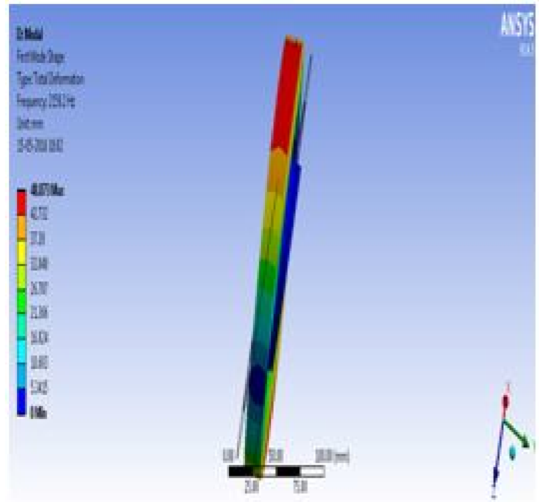

Fig. 16. Mode 1 at 2159.2Hz

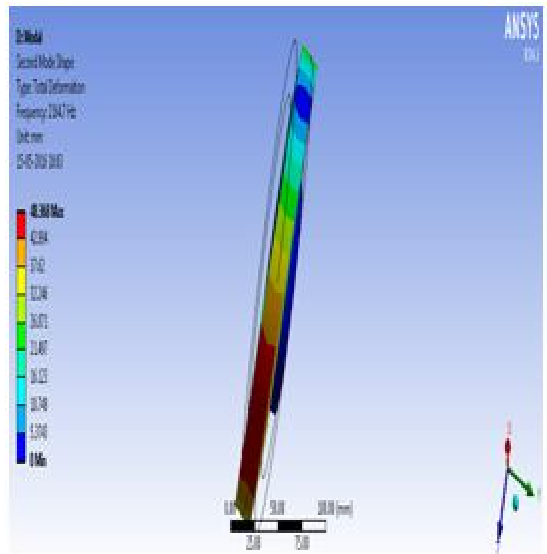

Fig. 17. Mode 2 at 2164.7Hz

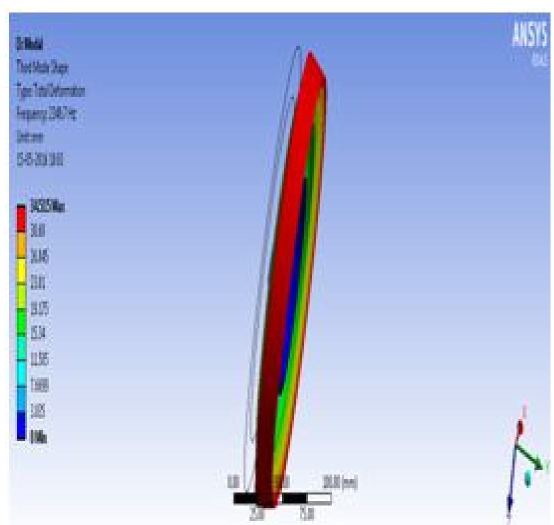

Fig. 18. Mode 3 at 2349.7Hz

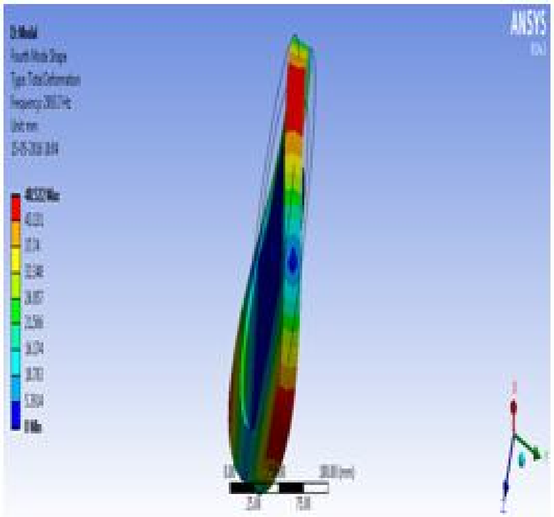

Fig. 19. Mode 4 at 2893.7Hz

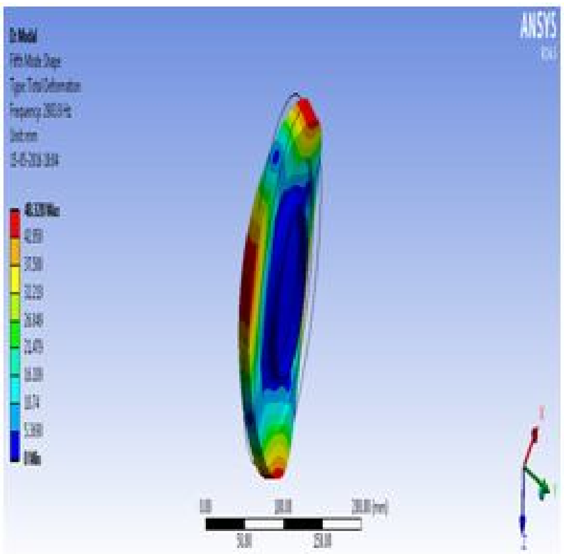

Fig. 20. Mode 5 at 2903.7Hz

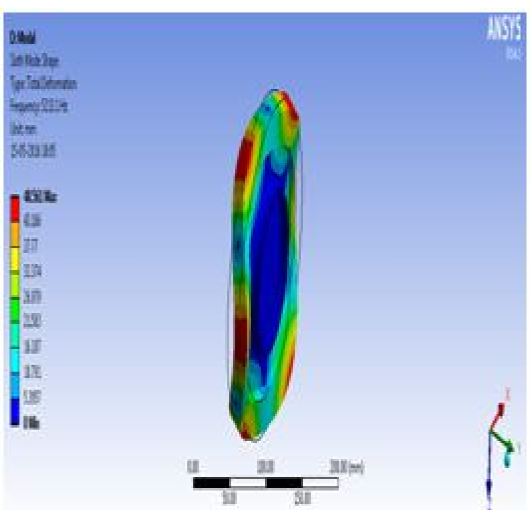

Fig. 21. Mode 5 at 5213.1Hz

VI. CONCLUSION

It is concluded from the above analysis, the disk is optimized to minimize its mass without sacrificing its strength against all loads such as rotational, shrink fit, blade load, thermal load and combination of all loads. Goal driven optimization was carried to determine the best candidate points which give minimum mass of the disk. Also for the disk with a minimum mass after optimization, static structural analysis considering all the loads are done again to verify its strength. Then the modal analysis was carried out to determine the first six natural frequencies and its corresponding mode shapes which helps to come to a conclusion that the disk is not experiencing a resonance phenomenon during its regular operations.

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