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Weight Optimisation and Burst Margin of Aero Engine Compressor Disc

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Abstract: Rotating disks are heavy, highly stressed components used in gas turbine mainly due to over speed and operating temperatures. In this work compressor disc are concentrated. The objective of this work was to develop finite elemental prediction to find burst margin and mass reduction of compressor disc. Ansys APDL V17 is used for analysis. The FE method for mechanical analysis is used for the determination of burst margin and allowable mean hoop stress for high speed aero engine compressor disc for the speed of 14000 RPM and 15000 RPM at temperature of 400 C. The result shows that the magnitude of hoop stress is maximum. The burst margin for the compressor discs are predicted and obtained for 14000rpm and 15000rpm as 12% to 21% of ultimate tensile strength. Mass of the compressor disc has been reduced from 13kg to 11kg, by comparing all the cases.

Key Words: Hoop stress, Radial stress, Average weighted mean hoop stress, burst margin.

I. INTRODUCTION

A gas turbine engine is a type of IC engine. Gas turbine engine produces a great thrust and causes the aircraft to propel forward at high speeds. Gas turbine engines work on the principle of Brayton cycle. The engine transforms chemicalsenergy of fuel into mechanical energy with the help of hot gases coming out of the combustion chamber to power the turbine, compressor and propeller, which in turn propels the aircraft. The engine sucks air through the diffuser which is in the front of the engine. The sucked air is used as the working fluid. The air is then sent to the compressor. The compressor compresses the air coming through diffuser according to the requirement. The compressor is constructed with by attaching a disc to shaft. The blades will be attached to disc. The blades spin at high speeds and compress the air due to arrangement of rotor blades and stator vanes. The high pressure air is then sent to combustion chamber. Along with the compressed air fuel

will be sprayed in to the combustion chamber with the help of fuel injectors. With the help of spark plugs the air fuel mixture will be ignited. Due to the combustion of the air fuel mixture the temperature increases in the combustion chamber. Along with the temperature entropy also increases. After combustion of air fuel mixture the hot gases come out of combustion chamber and impinge on turbine. Due to the impingement of hot gases on turbine, turbine rotates. Turbine also consists of disc and blades. The compressor is coupled with turbine using a shaft. The hot gases produced in the combustion chamber will hit on the turbine which is coupled with the compressor. Due to the coupling turbine rotates the compressor. After the hot gases pass through turbine, the hot gases escape the engine through exhaust nozzle.Compression of inlet air is achieved in a centrifugal flow engine by accelerating air outward perpendicular to the longitudinal axis of the machine. The axial-flow engine compresses air by a series of rotating and stationary aerofoils moving the air parallel to the longitudinal axis. The centrifugal-axial flow design uses both kinds of compressors to achieve the desired compression. The path the air takes through the engine and how power is produced determines the type of engine. There are four types of aircraft turbine engine- Turbojet, Turboprop, Turbofan, and Turbo shaft.

II. LITERATURE SURVEY

Maruthi B H et al have concentrated on burst margin and over speed limit on gas turbine rotating disc. The present paper was concentrated on optimisation of the model of disc for burst margin over speed, along with that disc speed correction was also done and it has been verified with the help of FE model. For different thermal load conditions and different speeds radial stress, burst margin and hoop stress were calculated using both mathematical model and FE method. Using ANSYS 12.0 linear analysis tool studies are executed based on non-linear problem. Under real operating conditions disc or blade segment were analysed for stress state using a non-linear FEM. The paper demonstrates that the altered mathematical model and FE model permits automated analysis that can be used for large scale simulation.M. Venkatarama Reddy et al have worked on over speed and burst limit. The aim of this work is to develop FE prediction to find over-speed and burst-margin limits. For the determination of over speed burst margin limit for high speed aircraft engine turbine disc for 10,000 RPM – 22,000 RPM, FE method for mechanical analysis and the thermal analysis is used for variable temperatures. The result demonstrates that the amount of the tangential stress constituents is more than that of the radial stress constituents for all the discs under different temperature dissemination. The tangential stress constituents are more at inner surface and reduce near outer surface.

In the present work the application of process modelling is discussed to support the fabrication of critical rotating components of gas turbine engine. Main aim of this work was to build FE models of the phenomena occurring during disc failure due to over speed in thermal environment. In the present analysis evaluation of over-speed and burst margins for a gas turbine disc is subjected to combination of thermal, blade and centrifugal loads. It is observed that not only centrifugal load plays a major role but also thermal and blade load dominating on stresses and burst margin of the disc. At the end of the work, the maximum speed limits gas turbines for aero-engine applications are determined. For three high-speed gas turbine disc for the speeds 18,500 rpm, 19,000 rpm and more than 22,000 rpm for thermal+ blade+ centrifugal load, thermal + centrifugal load and centrifugal load respectively for their maximum mechanical powers. However the radial deformation of the disc at burst speed limits tends to be exponential indicating the possibility of burst speed. The study gives an insight understanding the component stresses coming on the rotating disc which are very essential in determining the structural integrity of a gas turbine disc.

K.Kumar et al, has focused on reduction of weight and published a paper named “Computational Technique for Weight Reduction of Aero- Engine Rotor”. This paper however gives an overview of the critical rotating structures of compressor spool which were optimized using a general purpose linear optimization program and also highlights additional unique features, specific to the requirements of aero-engines, which were incorporated into the program. The components, which were analysed and optimized and discussed in this paper with reference to enhancement of the versatility of optimization program, are High Pressure Compressor Rotor (HPC) It has been possible to measure the versatility and use existing linear software quite successfully not only in the area of weight reduction but also in designing feasible components and structures, starting from designs that initially violated critical design contraits.

The optimization that was carried out converged in 30 design cycles. The final weight was reduced by 10 per cent. Simultaneously and most importantly, all the constraints were satisfied and a feasible design obtained. Based on highest local plastic strain, at the appropriate temperature, indicates a lower range of burst speed. General purpose linear optimization software can be used to handle optimization problem with multiple design constraints effectively reducing the experimental cost.

III. PROBLEM STATEMENT

Due to high centrifugal forces, RPM and temperature working conditions the aero engine compressor disc are experiencing high stresses such as hoop stress and radial stress as well as vibrations. Due to the high stresses and vibrations in aero engine compressor disc there may be chance of failure or change in the contour of gas turbine compressor disc. To prevent the failure or variation in the shape of the aero engine compressor disc we should know the amount of stresses acting on the gas turbine compressor disc. High speed rotating machinery stocks huge amount of kinetic energy. When one of these rotating components burst, energy will be released. This energy may be equal to the explosion of some bombs. Not only major components even small rotors spinning at high speed can lead to terrible damage during the burst. The problem with the burst of a rotating disk is that the pieces in which it breaks have huge inertia: they can destroy the engine and, eventually, pass through the nacelle. In the present work aero engine compressor disc has been analysed for over speed burst margin.

IV. PROJECT OBJECTIVE AND METHODOLOGY

A. Objective

1) To understand the loads acting on the aero engine compressor disc and to obtain the geometric details from the literature survey.
2) To model the geometry of aero engine compressor disc using the data obtained from literature review in Ansys.
3) To modify the compressor disc model for burst margin of over speed and verify the same with FE model.
4) The main aim is to reduce the final weight of the compressor disc.
5) Burst margin, radial stress and hoop stress were done at dissimilar speeds using finite element model.
6) To predict the best design for the aero engine compressor disc.

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B. Methodology

1) The geometry was created in Ansys APDL.
2) The geometry was meshed.
3) The boundary conditions were applied to the aero engine compressor disc which was obtained from literature review.
4) Design of the aero engine compressor disc was optimised and analysed for 16 iterations for 14000 and 15000 RPM.
5) Investigations are performed on linear analysis tool Ansys APDL.
6) The results obtained were validated.

V. GEOMETRY DETAILS

The cross-section geometry of the aero engine compressor disc was obtained from a reference. The geometry was created in Ansys. The use of 2D axisymmetric solid elements to design a cross-section of the disc utterly simulates the real time conduct of the 3D axisymmetric construction of the disc. The loads which are acting on the disc are also axisymmetric. Only a two-dimensional section of the aero engine compressor disc is taken for the analysis of over speed burst margin. The two-dimensional cross-section of the aero engine compressor disc taken for the analysis is shown in the figure 1. The details of the fixed dimensions of 2D compressor disc model are tabulated in table 1. The remaining parameters which are varied for 16 iterations are tabulated in the results section. Figure 2 shows the meshed model of compressor disc.

Parameter	Value	Unit
Bore outer radius	150	mm
Rim outer radius	285	mm
Rim width	20	mm
Rim height	10	mm
Rim angle	10^0	deg

Table 1: Fixed dimensions of 2D compressor disc model

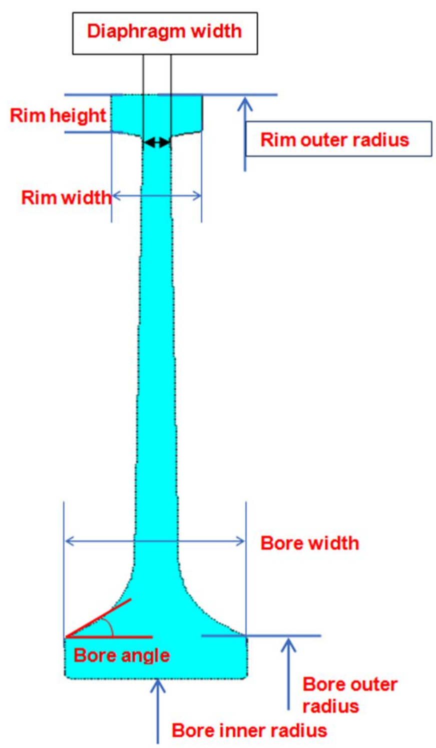

Figure 1: 2D cross-section of the aero engine compressor disc

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Material Property	Units	Ti 6Al 4V
Density(ρ)	Kg/m^3	4420
Young's Modulus(E)	GPa	120
Poisson's Ratio(μ)	-	0.34
Tensile Yield Strength	MPa	880
Ultimate tensile strength	MPa	1000
Thermal Conductivity(k)	ω/m-K	7.3

Table 2: Materials properties

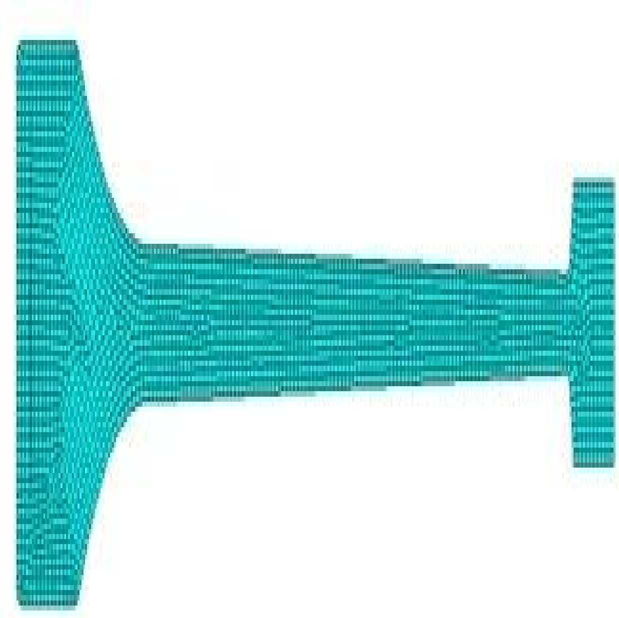

Figure 2: Meshed model of aero engine compressor disc

VI. BOUNDARY CONDITIONS

For simulating the real time working condition of rotation of the aero engine compressor disc with blade, the disc is given the rotational velocity of 14000 and 15000 RPM. The rotational velocity in terms of angular velocity applied to the disc. In an axisymmetric model there is no need of constraining rigid body motion in the x and z. To prevent rigid body motion of the disc one node must be constrained in the y. Centrifugal forces are created due to the rotation of the disc and blades. Due to the centrifugal forces of blade slots and blades, radial pressure load acts on the rim of the compressor disc and are calculated to be 323.6KN for rotational velocity of 14000 RPM and 371.59 KN for rotational velocity of 15,000 RPM.

VII. RESULTS AND DISCUSSIONS

This section deals with the results of over speed burst margin of gas turbine engine compressor disc. The results of burst margin under the effect of temperature, RPM and forces for Titanium Ti 6Al 4V alloy for the aero engine compressor disc are shown below for 14000 and 15000 RPM.

A. For 14000 RPM

Parameter	Case 1	Case 7	Case 9	Case 16
Bore inner radius	130	130	135	135
Bore width	40	45	40	45
Bore angle	35^0	45^0	35^0	45^0
Diaphragm width	7	7	7	8
No. of elements	2429	2566	2174	2429
Mass of disc	11.9235	12.8298	11.1367	12.4125

Table 3: Geometry parameters of 14000 RPM

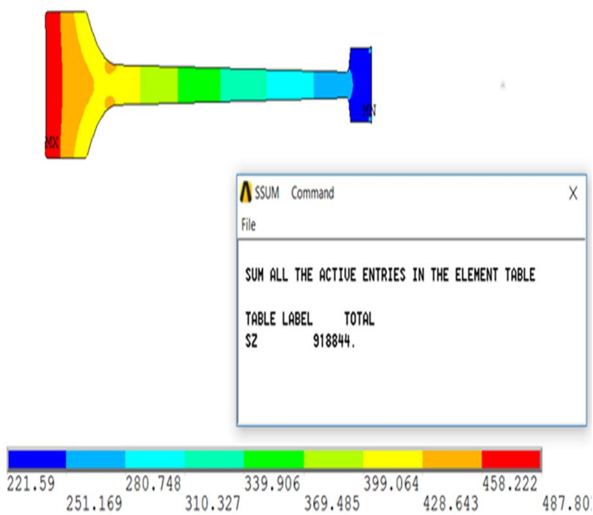

Figure 3: Average weighted mean hoop stress of case 1 for 14000 RPM

Figure 4: Average weighted mean hoop stress of case 7 for 14000 RPM

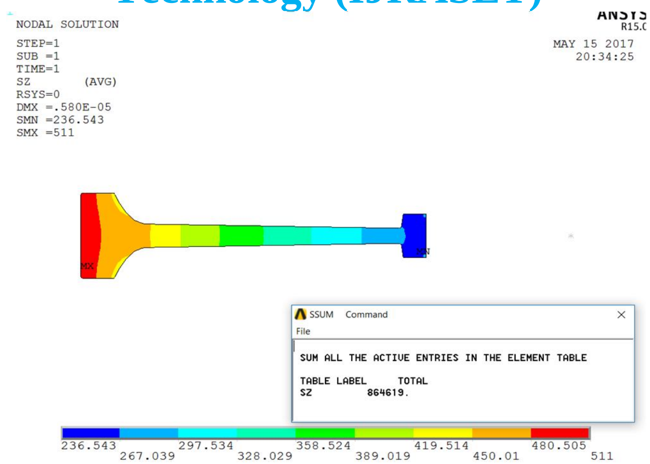

Figure 5: Average weighted mean hoop stress of case 9 for 14000 RPM

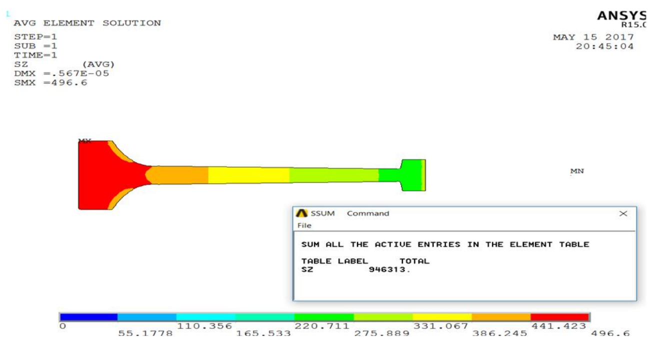

Figure 6: Average weighte mean hoop stress of case 16 for 14000 RPM

Case no.	No. of elements	AWMHS (MPa)	mass of the disc (Kg)	% of allowable stress
1	2429	376.281	11.9235	16.4304
2	2468	380.6	12.2657	15.7088
3	2404	375.07	12.0641	16.81083
4	2502	379.88	12.7814	15.9319
5	2496	370.44	13.3127	20.9372
6	2597	374.01	13.0322	19.7282
7	2566	369.21	12.8298	21.34015
8	2632	369.92	13.2369	21.1072
9	2174	395.89	11.1367	12.5939
10	2260	400.13	11.855	11.9636
11	2256	395.17	11.3833	12.7980
12	2332	396.96	12.1033	12.9345
13	2179	391.01	1.5403	14.5662
14	2367	394.84	12.2078	13.4636
15	2353	386.59	11.8549	14.2556
16	2429	389.58	12.4125	14.9956

Table 4: Results of 14000 RPM

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B. For 15000 RPM

Parameter	Case 1	Case 7	Case 11	Case 15
Bore inner radius	130	130	135	135
Bore width	40	45	40	45
Bore angle	35^0	45^0	45^0	45^0
Diaphragm width	7	7	7	7
No. of elements	2429	2566	2256	2353
Mass of disc	11.9235	12.8298	11.3833	11.8549

Table 5: Geometry parameters for 15000 RPM

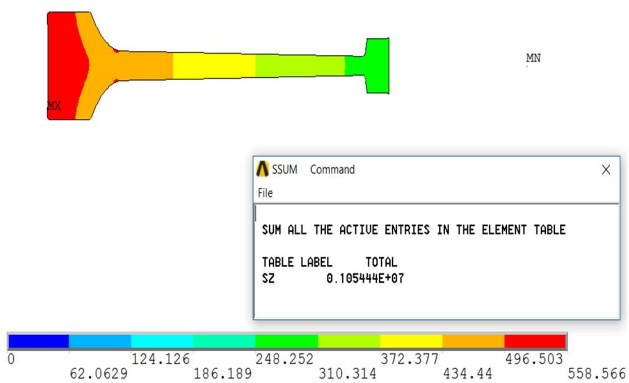

Figure 7: Average weighted mean hoop stress of case 1 for 15000 RPM

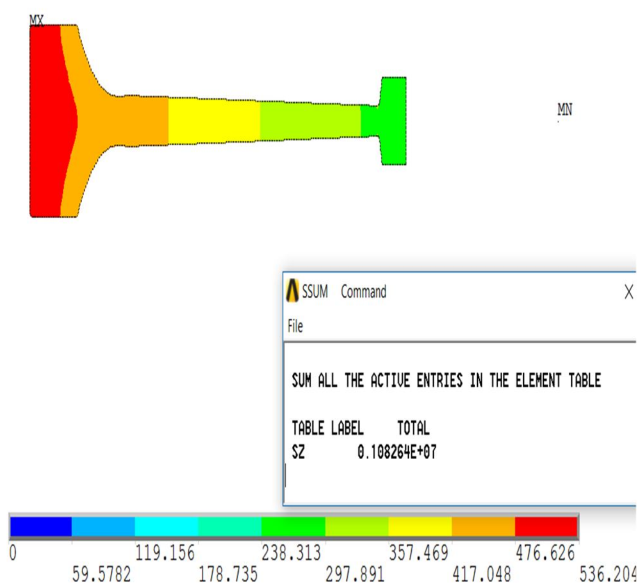

Figure 8: Average weighted mean hoop stress of case 7 for 15000 RPM

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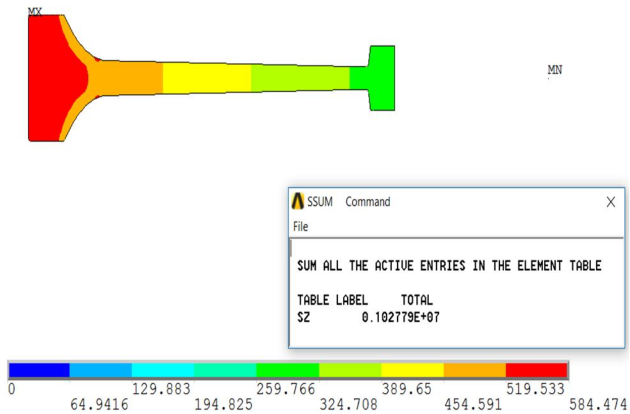

Figure 9: Average weighted mean hoop stress of case 11 for 15000 RPM

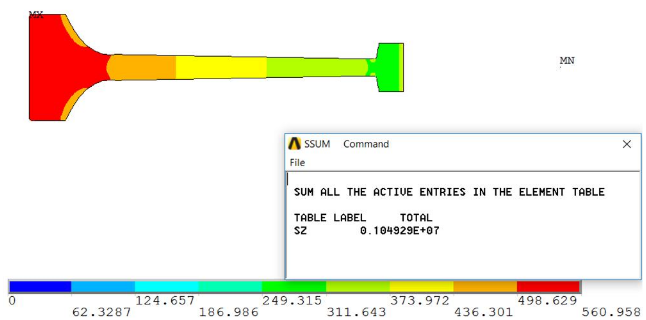

Figure 10: Average weighted mean hoop stress of case 15 for 15000 RPM

Case no.	No. of elements	AWMHS (MPa)	mass of the disc (Kg)	% of allowable stress
1	2429	434.105	11.9235	3.200839
2	2468	436.776	12.2657	2.569738
3	2404	432.728	12.0641	3.529238
4	2502	434.955	12.7814	2.7629
5	2496	424.116	13.3127	4.383001
6	2597	429.222	13.0322	4.374892
7	2566	421.917	12.8298	6.182022
8	2632	424.528	13.2369	4.528964
9	2174	456.604	11.1367	-1.88435
10	2260	461.154	11.855	-2.85241
11	2256	454.782	11.3833	-1.7074
12	2332	454.244	12.1033	-1.59123
13	2179	448.749	12.5403	-0.16691
14	2367	453.117	12.2078	-1.12929
15	2353	444.937	11.8549	0.462621
16	2429	445.093	12.4125	0.202866

Table 5: Results of 15000 RPM

VIII. CONCLUSION AND FUTURE SCOPE

A. Conclusion

1) By observing the burst margin and mass reduction iterations for 14000 RPM case 7 is best for burst margin and case 9 is best for weight reductin.
2) For 15000 RPM case 7 is best for burst margin and case 15 is best for weight reductin.
3) The mass of the compressor disc is reduced.
4) The burst margin is increased with decrease of average mean hoop stress.

B. Future Scope

1) To get best and more approximate results, number of iterations can be performed.
2) Thermal analysis and dynamic analysis on the compressor disc can also be done.
3) Burst speed of a compressor disc can be calculated.

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