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Stress and Vibration Analysis of Turbine Rotor

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Abstract— This Abstract presents details of design and analysis of a Turbine Rotor carried out as part of establishment of Wide Ranges of Applications. The rig is to evaluate the performance of Rotor with a maximum power rating of 4 kW with a speed up to 50,000 RPM. The scope of work includes the design and analysis of Rotor; A Rotor in the form of blisk has been designed for a design speed of 30,000-50000 rpm. Initially a semi-3D model using axi-symmetric element with blade load simulated at the disc periphery is carried out for disc optimization. Then the turbine is exactly simulated and analyzed using cyclic symmetric sector model for the actual stresses induced.

Keywords—Rotor & Disc;Stresses;Modal;; optimization; Speed; Design Margin; Generator Test Rig;Turbine

I. Introduction OF TURBINE Rotor

The turbines Rotor that are designed need to be validated for their performance. Several computation codes are available to predict performance of the turbines. But results from computational tools are satisfactory considering limitations in simulating exact working environment. So, an experimental testing is always mandatory to validate the performance of the turbine as a certification process. The results obtained from the experiments can be used to benchmark computational results. Alternator Turbine Test Rig to study the performance of gas turbine stages up to 4 kW with a speed range of 30,000 to 50,000 rpm to support Medium , small gas turbine developmental activities, presents work done by past researchers on structural analysis of turbine discs and blades, bearing arrangement and various aspects of rotor dynamics. The structural analysis of Blisk is presented from Parallel Disc to Optimized Disc. And also predict the critical speeds of the rotor-bearing system.

II. Description of Turbine Rotor Setup

For the Development of Rotating machinery can find range of applications from Aerospace to Power generation for Major Industries, hence the development of gas turbines needs a test facility where the performance of the turbine and Rotor can be evaluated. To develop a Generator Test Rig to meet the testing requirement of turbine stages up to 4 kW-10 kW with a speed range of 30000 to 100,000 rpm. Several computational tools have been deployed for design and develop Air turbines or Rotor is available worldwide, but very limited experimental data is available in open literature. The experimental data generated by Air turbine manufacturers is kept confidential. The test rig that would be developed can be used to generate experimental data and validate the computed results from various commercial quotes. In this Project actives attempted to modify an existing air turbine to improve for a Power Generation application is presented in this project. Alternator (Power Generator) Consists of basically 2 part Stator and rotor.

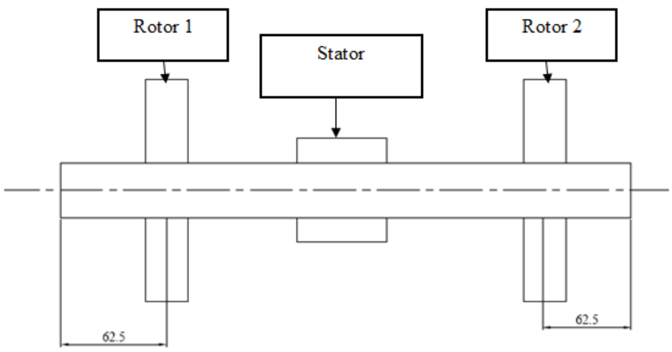

Fig 2-1: Schematic of Rotor and Stator Setup

In high Speed Alternator (Power Generator), that Carrier the Stator (Permanent Magnet) Concentrically Position inside stator. The stator is essentially made out of soft iron plates or laminates slots. Coil of wire is placed inside the slot of stator core. When the rotor is rotate at high speed, the Coils Subjected to Varying magnetic field. The coil or electric conductor placed in a Varying magnetic field produces the electric current. The rotor Core with embedded magnet is mounted at the middle of rotor shaft. Two air turbines is attached at either side of the rotor core. Compressed air Nozzle is directed at the Blades of the air turbine. Air at the nozzle gains velocity by loosing pressure. The high velocity of air hits the blade and the turbine thus rotates. The energy is further converts into electrical energy at the stator. The present work, the development of the Turbine Rotor for High speed alternator (Power Generator) is Operating at very high speed results in smaller size that is an essential requirement in aerospace application. High speed drive is an essential part of Alternator development program.

III. Flow Chart for the Structural Design and Analysis of Rotor

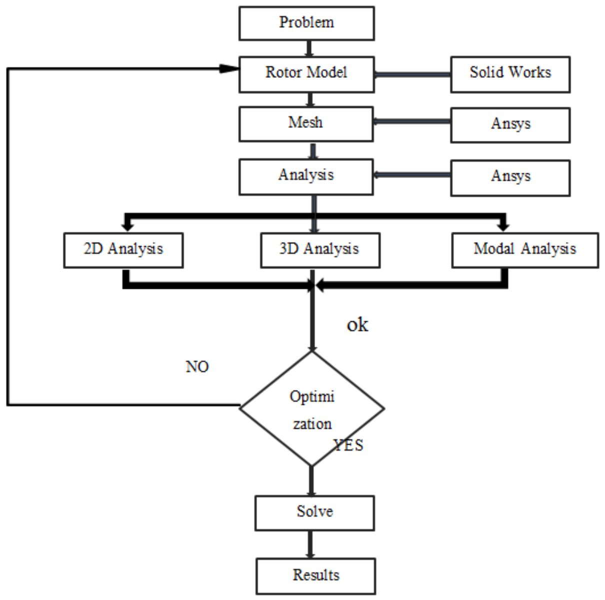

Fig 3-1 FLOW CHART FOR THE STRUCTURAL DESIGN AND ANALYSIS OF ROTOR

IV. MATERIALS USED

The decision of material assumes an essential part in the outline of Rotor. Jürgen Broede compared two discs made of Titanium alloy and Ni alloy with similar dimensions [21]. Firstly, the thermal stresses are compared. For a given temperature profile the thermal stresses in the Nickel disc are 2.7 times higher compared to the Titanium disc due to the differences in the coefficient of thermal expansion and the Young's modulus. Further, the weight of the Nickel disc is compared with Titanium for various design criteria and the results observed are as follows:

- 1.9 times higher for the same geometrical dimensions,
- 1.4 times higher for the same burst margin,
- 1.6 times higher for the same low-cycle fatigue life at the bore and
- 3.4 times higher for the same hoop stress at the bore.

Material	σ, Tensile Strength, MPa	ρ, Density, kg/m³	Specific Strength MPa-m³/kg(σ/ ρ)
Al-Alloy	510	2800	0.1821
Ti-Alloy	**890**	**4600**	**0.1935**
Inconel 718	1100	8190	0.1343
Rene 41	793	8240	0.0962
HS Steel	1667	8000	0.2084
Maraging Steel	2450	8100	0.3025

Table-4-1

V. METHODOLOGY

In view of the Methodology, The disc analysis is carried out using a 2D Model Using Plane Axi-symmetric Element to evaluate the Rotor Model geometry, loads such has Inertia Load Due to Density of a material with respect to Speed of the Rotor and boundary conditions are symmetric about the axis of the turbine. The axi-symmetric model needs much lesser computational effort compared to a 3D model.

Axi-symmetric element is chosen to mesh the model and the details of the element are given below in Table 4-3.

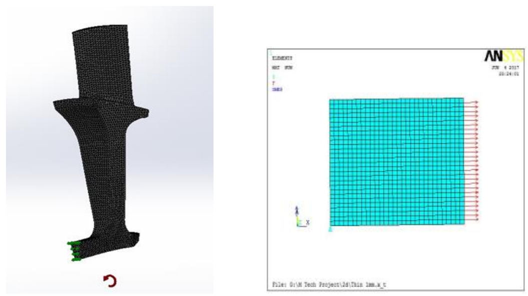

Fig 5-1 THE AXI-SYMMETRIC 3D MODEL AND PLANE 2D NEEDS MUCH LESSER COMPUTATIONAL EFFORT COMPARED TO A 3D MODEL

VI. RESULTS AND DISCUSSIONS

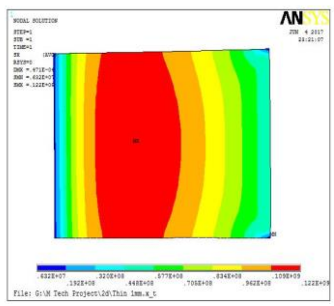

FIG. 6-1A. RADIAL STRESS ALONG DISC RADIUS, RECTANGULAR CROSS-SECTION

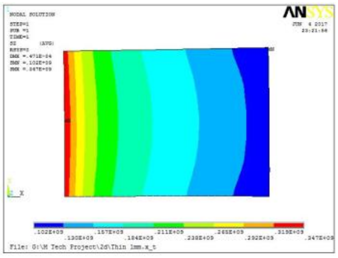

FIG. 6-2 B. HOOP STRESS ALONG DISC RADIUS, RECTANGULAR CROSS-SECTION

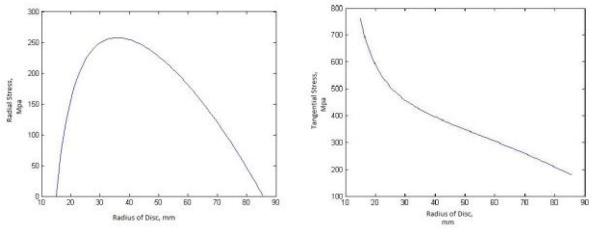

FIG. 6-3A, B. THEORITICAL HOOP STRESS ALONG DISC RADIUS AND RADIAL STRESS PLOTS

To assess the variety of spiral anxiety and digressive worry along the sweep of the circle are plotted in Fig. 6-1a and Fig. 6-2B likewise Standard Theoretical Equations Stresses are Obtained to Compare push plot acquired from Ansys is appeared in Fig. 6-1a which is like the chart appeared in Fig. 6.3a,b. The most extreme spiral worry of 122.0 MPa is seen at around 33% the circle range though the outspread anxiety is observed to be a base at the drag and edge of the plate. The outcomes from Ansys for distracting stress are appeared in Fig. 6.2b which is likewise like the chart appeared in Fig. 6.3a,b.

VII. RESULTS AND DISCUSSIONS(Optimized)

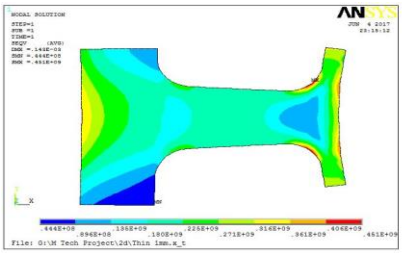

FIG. 7-1. VON-MISES STRESS ALONG DISC RADIUS, OPTIMIZED CROSS-SECTION

From The Previous Iteration Disc are made as Rectangular to assess Theoretical Evaluations, in this Iterations Disk made decreased to oblige the turbine cutting edge. The cross-area of the circle is made decreased along the range from the center point to the edge. This brought about a uniform anxiety design around 374 MPa over the web area and greatest worry of 451.0 MPa at the plate bore. Rotor is assessed to meet the Required Design Margin.

VIII. DYNAMIC ANALYSIS OF ROTOR SYSTEM

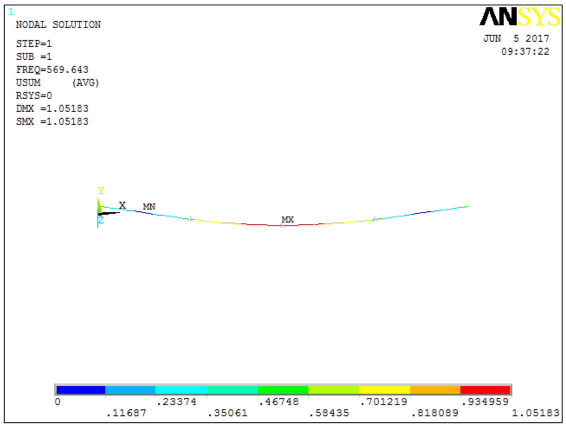

Fig. 8.1. First Bending Mode Shape of Lateral Vibration of Rig Rotor Bearing System

Roto-dynamics plays Major role in Designing and Evaluating Rotor Components which should be balanced and The natural frequencies of rotor-bearing system must be kept away from the operating frequencies to avoid resonance or critical speed problems.

***** INDEX OF DATA SETS ON RESULTS FILE *****

SET	TIME/FREQ	LOAD STEP	SUBSTEP	CUMULATIVE
1	569.64	1	1	1
2	569.64	1	2	2
3	1673.8	1	3	3
4	1673.8	1	4	4
5	2969.0	1	5	5
6	3190.9	1	6	6
7	3190.9	1	7	7
8	8044.7	1	8	8
9	12699.	1	9	9

Table 8.1. Critical Speeds values:

From the Table found that the rotor components which are running at high Speed, The first critical frequency from Ansys result was found to be 569.64 Hz (34,176 RPM). The Ansys prediction of critical speed ((34,176 RPM) is reasonably away from the operating speed of 50,000 RPM with a margin of over 35%. Hence the chosen configuration is considered to be safe.

IX. CONCLUSIONS

A Turbine Rotor Test apparatus to assess the execution of a turbine with a most extreme power rating of 4 KW-10KW @ 30000 to 100,000 RPM has been composed.

The blisk has been intended to withstand spiral and distracting worries at an outline speed of 30000 to 100,000 RPM. The circle has been improved from a rectangular area that brought about huge weight sparing of 50 %.

High anxiety focus was seen at edge root and the worry at sharp edge root was observed to be around 430 Mpa at a plan speed of 30000 to 100,000, which is Lower than the material yield worry of 860 MPa. In this way, the sharp edge root connection has been altered without influencing the streamlined execution of the cutting edge to diminish the anxiety fixation levels at the edge root.

A shade turbine Rotor is arranged to outline Comfortable get together and dismantling of turbine Rotor without a need to dismantle the direction. The rotor framework is upheld by two rakish contact metal balls with a heading range of 187.5 mm. The basic speed of the rotor framework is anticipated utilizing Rayleigh-Ritz and Ansys. The outcomes from Rayleigh-Ritz and Ansys were observed to be in sensibly agreeable assention. Anticipated Number of first basic speed (34,176 RPM) from Ansys was observed to be altogether far from the operational speed of 50,000 RPM

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