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Coupled Field Analysis of Turbine Rotor Blade

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ABSTRACT:- *Rotor blade is one of the important components in a gas turbine. Its function is translating the thermal energy of gas into the mechanical energy. The working environment of blade is very terrible. It has to suffer from centrifugal force and thermal stress due to temperature variation. According to statistics, the proportion of failures of rotor component in total failures of gas turbine arrives to 80%. The main objective of this paper is to suggest a suitable material for the manufacturing of rotor blades, which would practically reduce the cost of production and more durable than the already existing materials. There are many materials currently being used to manufacture gas turbines. In this project, two materials viz., Nimonic Alloy 80A and Udimet 500 (U500) were selected. A rotor blade was modelled accordingly and by applying the properties of both the above mentioned materials, static thermal and structural analysis was carried out. The results thus obtained were verified with original results.*

Keywords:- *Gas turbine rotor blade, Modeling, Meshing, Thermal Analysis.*

I. INTRODUCTION

A gas turbine is a machine delivering mechanical power or thrust. It does this using a gaseous working fluid. The mechanical power generated can be used by, for example, an industrial device. The outgoing gaseous fluid can be used to generate thrust. In the gas turbine, there is a continuous flow of the working fluid. This working fluid is initially compressed in the compressor. It is then heated in the combustion chamber. Finally, it goes through the turbine. The turbine converts the energy of the gas into mechanical work. Part of this work is used to drive the compressor. The remaining part is known as the net -work of the gas turbine.

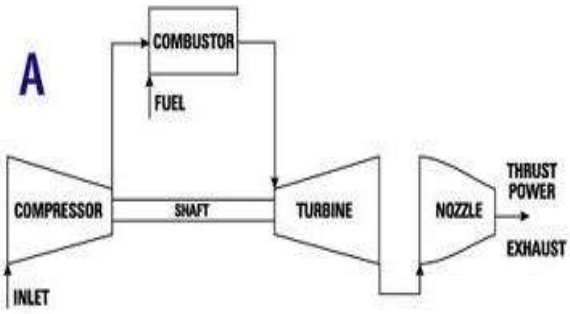

Fig.1. Schematic for an Aircraft Engine.

There are two important types of gas turbines. They are industrial gas turbines and jet engine gas turbines. Both types of gas turbines have a short but interesting history. Industrial gas turbines were developed rather slowly. This was because, to use a gas turbine, a high initial compression is necessary. This rather troubled early engineers. Due to this, the first working gas turbine was only made in 1905 by the Frenchman Rateau. The first gas turbine for power generation became operational in 1939 in Switzerland. It was developed by the company Brown Boveri. Back then, gas turbines had a rather low thermal efficiency. But they were still useful. This was because they could start up rather quickly. They were therefore used to provide power at peak loads in the electricity network. In the 1980's, natural gas made its breakthrough as fuel. Since then, gas turbines have increased in popularity. The first time a gas turbine was considered as a jet engine, was in 1929 by the Englishman Frank Whittle. However, he had trouble finding funds. The first actual jet aircraft was built by the

German Von Ohain in 1939. After World War 2, the gas turbine developed rapidly. New high-temperature materials, new cooling techniques and research in aerodynamics strongly improved the efficiency of the jet engine. It therefore soon became the primary choice for many applications. Currently, there are several companies producing gas turbines. The biggest producer of both industrial gas turbines and jet engines is General Electric (GE) from the USA. Rolls Royce and Pratt &Whitney are also important manufacturers of jet engines.

Jiang-jiang Zhu, et al. [1], research of life prediction and damage control for improving the reliability and machinability of a gas turbine. The common modes of failure in gas turbines are creep and fatigue. S. Suresh, et al. [2], the first stage rotor blade off the gas turbine has been analyzed using ANSYS 9.0 for the mechanical and radial elongations resulting from the tangential, axial and centrifugal forces. The gas forces namely tangential, axial were determined by constructing velocity triangles at inlet and exist of rotor blades. Manohar K, et al. [3], made an attempt to understand the design criteria used for the design of gas turbine disc running at maximum speed of 12000 R.P.M and operating at a temperature of 1300deg centigrade. Ujjawal A. Jaiswal [4], presented the design of Axial flow compressor for a given mass flow rate and required pressure ratio by using mean line method. The parameters include thermodynamic properties of the working fluid, number of rotor and stator blades , tip and hub diameters, stage efficiency, blade dimensions (chord, length and space) for both rotor and stator, flow and blade angles (blade twist), Mach number . Josine George, et al. [5], described about gas turbine blades will subject to high tangential, axial and centrifugal forces during their working conditions. While withstanding these forces gas turbine blades may subjected to elongation. Summarizes the design, analysis and modification of the cooling passage in the gas turbine blade design.

V.Raga Deepu, et al. [6], presented that the first stage rotor blade of the gas turbine is created in CATIA V5 R17 software. The material of the blade is NI-CR alloys. This model has been analyzed using ANSYS11.0 for the couple field (static and thermal) stresses. P.V.Krishnakanth, et al. [7], specified that to analyse the complex turbine blade geometries, how the program will make effective use of the ANSYS pre-processor and applying boundary conditions to examine steady state thermal & structural performance of the blade for Haste alloy x, N 155 & Inconel 625 materials. Candelario Bolaina, et al [8], investigated about the thermo-mechanical stresses in gas turbine blades. The study reveals that in the central region of blade transversal cross-section stresses were increased by blocking channels. R.D.V.Prasad, et al. [9], presented that the temperature has a significant effect on the von Mises stress in the turbine blade. Maximum elongation and temperatures were observed at the blade tip section and minimum elongation and temperature variations at the root of the blade.

II. MODELING

For designing the rotor blade, first of all, the profile of the rotor blade has to be designed. The dimensions of the blade are 90 mm in height and 40 - 50 mm in width. This should be followed by the root of the blade. The profile of the blade is as shown in the fig. 2.1. The root was designed for the profile. The type of root is fir root shown in fig. 2.2. The rotor blade after assembly is shown in fig. 2.3.

Fig 2.1. Top View for Profile of gas turbine rotor blade with cooling pipes

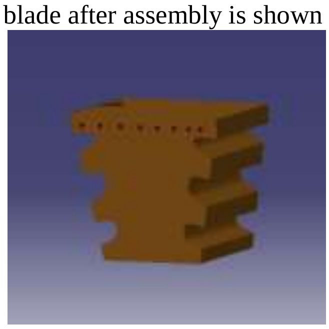

Fig.2.2. Fir root of rotor blade

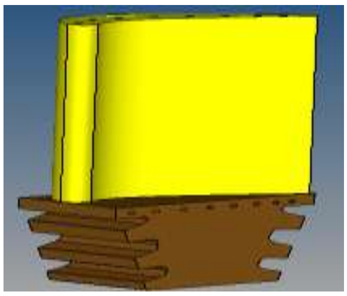

Fig.2.3. Assembled view of the rotor blade

The rotor blade was modified by creating 1.6 mm diameter cooling pipes to supply air there by reducing the temperatures as shown in fig. 2.4. The rotor blade, after designing was imported to Hypermesh by converting the file into .stp format. Then, it is meshed as shown in fig. 2.5 and fig. 2.6.

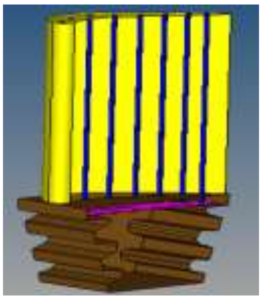
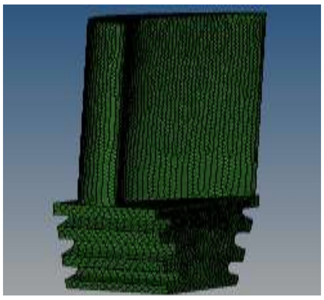
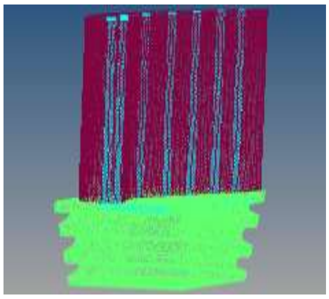

Fig. 2.4. Rotor blade Fig.2.5. Meshing of Rotor Blade Fig.2.6. Meshed structure for with 1.6mm pipes the cooling pipes

The type of meshing used was hexa- mesh. Element type used was **solid 90**. This has been used for thermal analysis.

Number of elements = 133568
Number of nodes = 202968.

III. ANALYSIS AND RESULT

3.1. **Thermal Analysis**

The rotor blade file is imported from Hypermesh to Ansys in .igs format. The following conditions are taken during this process.

Table. 3.1. Heat Transfer coefficient and temperature in different parts of the rotor blade.

Name of the part	Heat Transfer Coefficient(H)	Temperature (T)
Root	$500\ e^{-6}$ W/mm^{20}c	750^{0}C
Profile	$500\ e^{-6}$ W/mm^{20}c	1000^{0}C
Internal pipe	$500\ e^{-6}$ W/mm^{20}c	350^{0}C

Table. 3.2. Material properties of Nimonic alloy 80A and U500.

Material properties	U500	Nimonic Alloy 80A
Young's modulus	$2.19e^{5}$	$4\ e^{5}$
Poisson's ratio	0.29	0.3
Density	$7850e^{-6}$ kg/m^{3}	$8165e^{-5}$ kg/m^{3}
Specific Heat	0.14 J/g-^{0}C	0.11 J/g-^{0}C
α	$1.75e^{-5}/^{0}$C	$12.7e^{-5}/^{0}$C
k	0.0162W/m^{2}K	11.2

3.1. 1. For U500

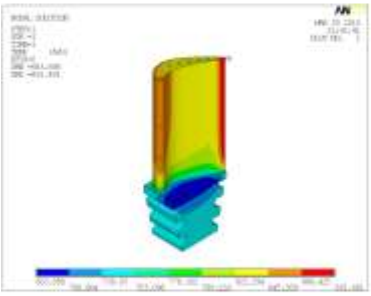
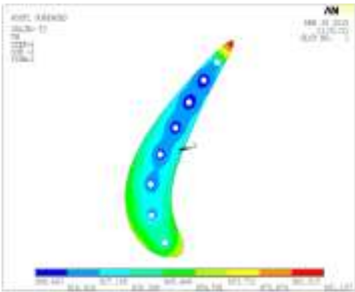
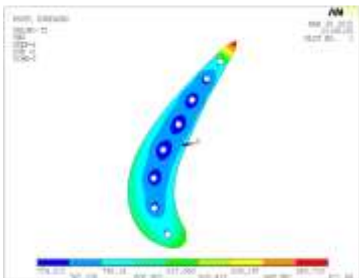

Fig. 3.1.1.1. Thermal Analysis results of U500 Fig. 3.1.1.2. Thermal Stress distribution at the center of the blade Fig.3.1.1.3. Thermal distribution at the top

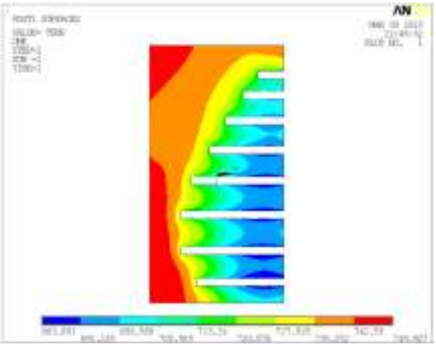

Fig.3.1.1.4. Thermal Stress distribution at the bottom of the blade near the root.

3.1.2. For Nimonic Alloy 80A

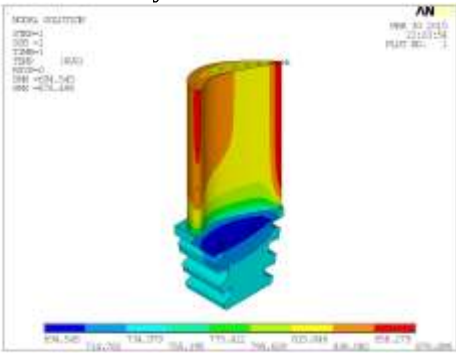

Fig. 3.1.2.1. Thermal stress distribution in the blade.

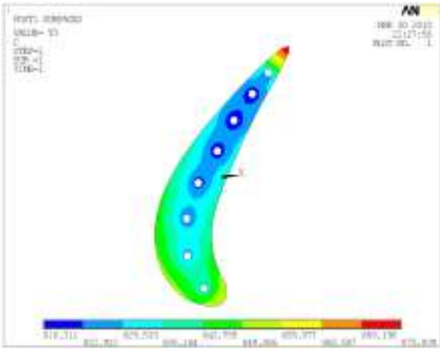

Fig. 3.1.2.2. Thermal distribution at the top of blade.

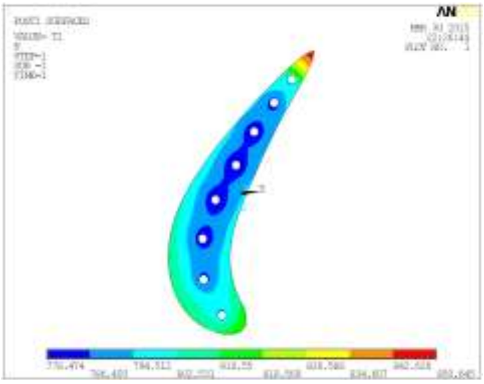

Fig. 3.1.2.3. Thermal distribution at the middle of blade.

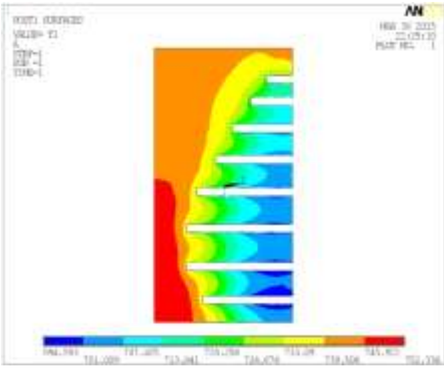

Fig. 3.1.2.4. Thermal distribution at the bottom of blade

Udimet 500 or U500 can resist a temperature upto 1360 degrees centigrade. The temperature we obtained after the analysis is 891.5 degrees centigrade. Similarly Nimonic 80A can resist upto 1170 degrees centigrade. From the result, we have 876.5. Hence the results are justified.

3.2. **Strutural Analysis**

Neuber's Rule

One of the objectives of durability analysis is to predict the magnitudes of the local cyclic stresses and strains experienced at the hot-spot of many components subjected to fatigue loading. Most estimates of component stress however, have been calculated elastically, by means of either traditional manual calculations or elastic FEA methods. It is then necessary to translate the elastic calculated stress at the critical locations into estimates of elastic-plastic stress and strain behaviour. Of the several methods of accomplishing this translation, the one most popularly adopted by most software methods is the Neuber [1-3] plasticity correction.

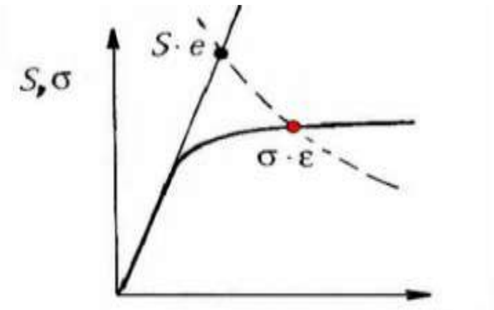

Fig. 3.2.1. The application of Neuber's rule.

As depicted in Fig.3.2.1, the Neuber correction can be set into three steps:

1. Using elastic calculation methods compute the stress and strain at the fatigue hot-spot.
2. Compute the energy or product of elastic stress multiplied by elastic strain.
3. Using the stabilized cyclic stress-strain curve of the material at the hot-spot find the cyclic stress and cyclic strain that give the same energy product as in Step 2. One can find examples of the stabilized cyclic stress-strain curve in fatigue databases.

After completing the thermal analysis, the file is changed from thermal to structural and load conditions are applied. The results are as follows:

3.2.1. For U500

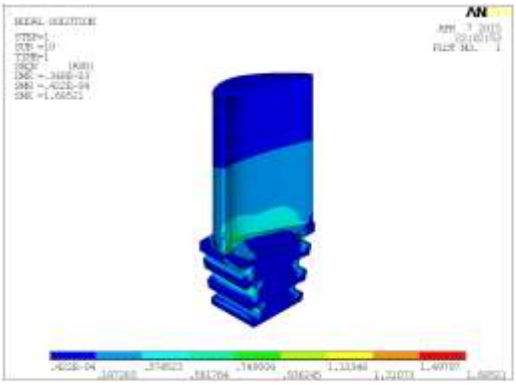

Fig. 3.2.1.1. Stress analysis of rotor blade for U500

3.2.2. For Nimonic Alloy 80A

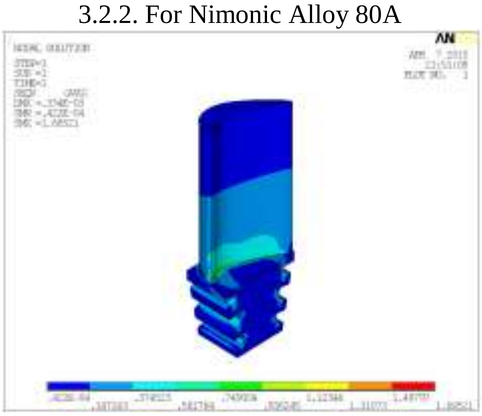

Fig. 3.2.2.1. Structural loads applied on Nimonic 80A.

3.3. Both Thermal and Structural Analysis

After the application of both the structural and thermal loads, the results are as follows. As we have two materials, we will deal with each of the material in detail there by understanding the essential concepts of Neuberation.

3.3.1. For U500

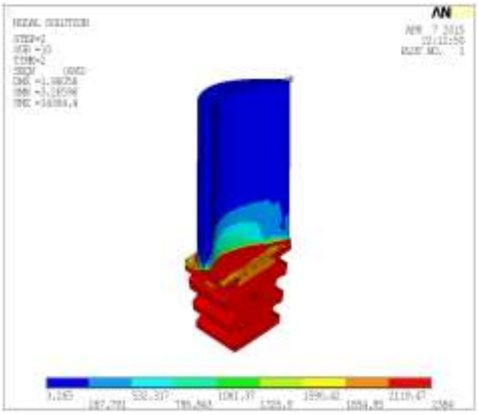

Fig. 3.3.1.1. The distribution of both thermal and structural

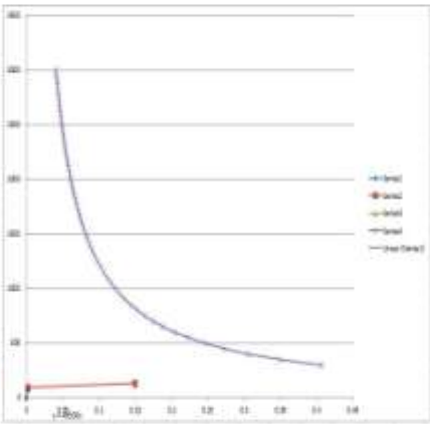

Fig. 3.3.1.2. The Neuber's curve for U500 stresses.

The minimum stress generated in this process is 1.67MPa and maximum stress is 1937MPa. Acoording to the properties of Nimonic 80A, the ultimate tensile strength of this material is 1250MPa. So at this maximum stress, the design will ultimately fail. The above curve as shown in fig. 3.3.1.2, is plotted in the following manner. The initial point in the graph is considerd as (0,0). The first point is (0.02, 780). The second point of the line is (0.3, 1250). With the help of these points, the lines have been drawn and the line quation of the first line is generated. Here 0.02 is the proof stress. 780 is the yield strength. 0.3 is the percentage of elongation. 1250 is the ultimate tensile strength. To plot the curve, the following actions are performed.

: Y = 39000x – e-13 (1)

If we subsitute the value of y as 1937 and find out 'x', we can have that value as 0.0496. We can see that once the graph was plotted, the curve cuts the line at some point. That point after taking coordinates is around 950 to 970 Mpa which is well less than the 1250 Mpa. Hence the design is safe.

3.2.2. For Nimonic Alloy 80A

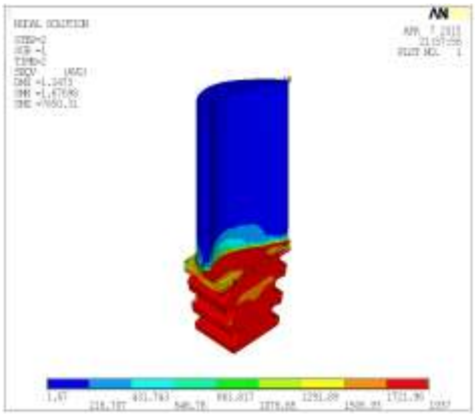

Fig. 3.2.2.1. Shows the application of both thermal and structural loads on Nimonic 80A.

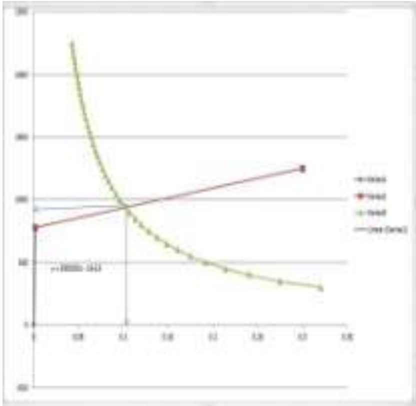

Fig. 3.2.2.2. The neuber's curve for Nimonic 80A.

After the application of both thermal and structural stresses (coupled filed) the maximum load on the design is 2384 Mpa, which is considerably very large for the material. Hence Neuber's curve to it can b applied. Here, though the Neuber's curve is plotted, it is found out that the curve does not intersect the two lines and hence, we can assume that the product fails in this case. Hence, U500 fails under the given conditions.

V. CONCLUSION

Rotor blade is an important part in a gas turbine, whether we are using it for aerospace or power generation purpose. In this, a rotor blade has been designed which is specifically used in the engines of airplanes and aircrafts. Two super alloys, namely Nimonic 80A and U500 have been considered and their properties have been individually applied on the design. The entire idea of the project is to suggest a suitable alloy to replace the conventional Titanium alloy as this would be very much cost effective and more or less displays the same amount of material properties. From the beginning, we have been focussed on the Nimonic 80A which we believed a better alloy than U500 and the same has been proved.

ACKNOWLEDGEMENTS

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