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# NON-LINEAR CONTACT STRESS ANALYSIS OF L P COMPRESSOR BLADE-DISC ASSEMBLY FOR GAS AND CENTRIFUGAL LOAD

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## ABSTRACT

*Modern military aero engines set very high technology standards for the component design, as they operate in an increasingly hostile thermo-mechanical environment. With rapid advancement in materials, manufacturing processes and analytical tools, there is an increase in demands on the aero engine performance and thrust-to-weight ratio. The rotary parts like blades, disks, shafts of compressor or turbine have taken into very important consideration in gas turbine engines. As they are subjected to very high centrifugal loads and gas loads.*

*In the present work, finite element analysis of typical conventional compressors bladed disc assembly has been carried out to study the stress levels at critical locations in discussing standard commercial finite element software (ANSYS 10.0). Conventional bladed disc's assembly are already operating at high stress level, because of high speed and thermal gradients across the disc bore and rim, weight optimization of disc is possible only by resorting to other techniques. A significant part of the disk weight is at blade attachment region, because the additional disk material required withstanding the centrifugal pull of the blade, an attempt has been made to find the contact stress for the weight optimized compressor bladed disc assembly.*

*Determination of yield strength and effective comparison with the input yield strength has been done. This observation clearly shows that compressor blade and disk are in safe position from the crack initiation.*

*The peak values of compressive stresses also induced near the region of contact stresses are also well within the design limit1. The peak value of von-misses stresses induced near the region of contact stresses are also well within the design limit.*

**Keywords:** *Aero Engine, Finite Element Analysis, Gas Turbine Engines , Thermo-Mechanical Environment, Stress Analysis.*

## 1. INTRODUCTION

The stress analysis of the dovetail or the derivative fir-tree root type of turbine blade fixing has received the attention of several experimental investigators employing the techniques of photo elasticity since the introduction of this method as a general stress analysis tool. The emphasis of these investigations however has been on the stress values on the unloaded boundaries of the component, particular attention having been paid to the fillet radius at the junction of the dovetail and the radial shank of the blade. Consideration of the failures occurring in practice however suggests that the fillet region is not always the site for fatigue cracks and that the region below and adjacent to the contacting flanks of the joint is just as likely to provide a site for crack initiation.

A requirement to obtain internal stress distributions leads the experimental stress analysis to the photo elastic method and the theoretical stress analysis now a days to the finite element or boundary element methods of numerical analysis. It was decided that it would be interesting to compare the photo elastic results for this problem with those obtained from using a commercially available finite element package.

## 2. MATERIALS

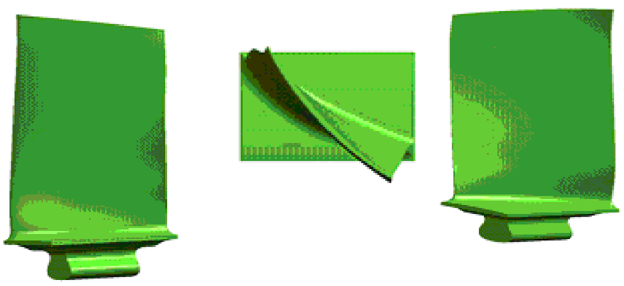
### 2.1 BLADE MATERIAL

The materials that have found to be suitable for use in blades are steels, titanium alloys and nickel-based alloys. All the three types of alloys, which are mainly used, have varying proportion of chromium and aluminum to improve the strength and corrosion at high temperature. Titanium alloys are preferred to steel because of its lower density (nearly 50%). Titanium has superior oxidation resistance.

### 2.1 DISC MATERIAL

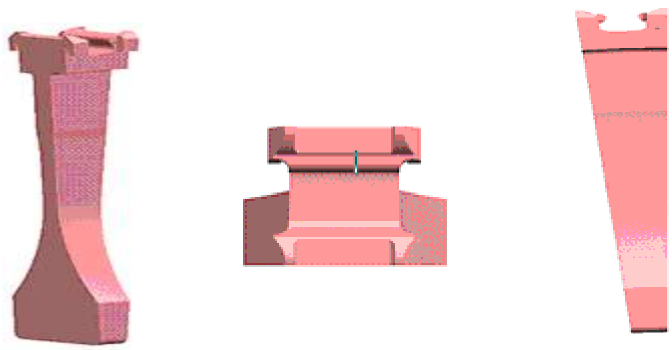
Disc carries a series of blades, so great care must be taken in design and material selection to avoid the catastrophic failure, based on yield and ultimate strength at approaching temperatures, resistance to creep relaxation and good fracture toughness at ambient temperature conditions as well as operating temperatures. For blisk, Titanium based alloy (Ti-64) is used.

## 3. Methodology: Geometric modelling of Compressor Rotor Blade and Disc



**(a) Front view (b) Top view (c) Rearview**

**Fig. 3.1 Aero engine compressor rotor blade**



**(a) Front view (b) Top view (c) Rear view**

**Fig. 3.3 Aero engine compressor rotor Disc**

## 3.4 Compressor blade disc assembly

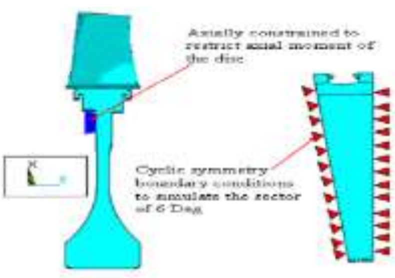
**Fig.3.4 Aero engine compressor rotor Assembly**

### 3.5 MATERIAL PROPERTIES

**Table1. Material property.**

| Material Properties | Ti-64Al-4V | |
|---|---|---|
| Physical Properties | Value | Units |
| Density | 4.43E+03 | Kg/m³ |
| Mechanical Properties | | |
| Modulus of Elasticity | 1.14E+05 | Mpa |
| Compressive Yield Strength | 970 | Mpa |
| Poisson's Ratio | .342 | |
| Fatigue Strength | 240 | Mpa |

## 4. Boundary conditions



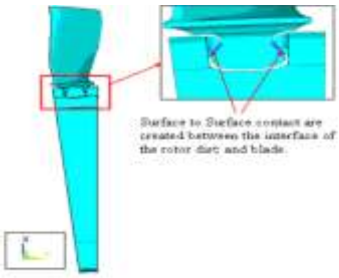




**Fig.4.1 Boundary condition for the analysis of the Disc Fig.4.2 Boundary condition for the Contact analysis**

## 5. Results and Analysis

Displacement distribution results in the ROTOR BLADE

Deflections are in mm

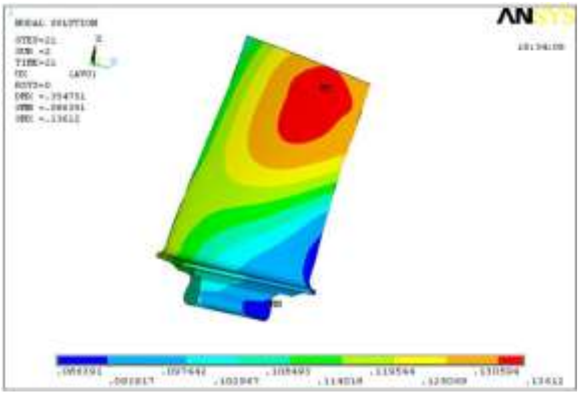


**Fig 5.1 Radial Deflection along X-direction of a Rotor Blade**

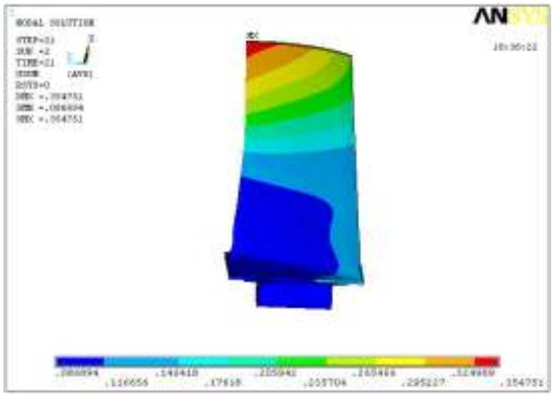


**Fig 5.2 Total Deflection of Rotor Blade**

Fig5.1 shows that radial deflection along X-direction which signification is that the radial clearance between compressor blade and casing (0.134mm) compared to the existing standard value of clearance (0.5mm) by analyzing the deformation distribution in the compressor blade and disk assembly. Whenever deflection starts radial clearance between compressor blade and casing reduce.

The main observation is that analyzing deflection from Table 5.1 and Fig 5.3, The Optimum value of Radial and Total deflection is well below the standard value, so whole compressor is safe from the radial deflection in compressor blade. If exceed the standard value, again repeat the same analysis by having different loads. The following Table 5.1 and Fig 5.3 shows the variation of Radial as well as Total Deflection for various Centrifugal Loads.

**Table.5.1 Deflections in Blade due to varying Speeds**

| Speed (rpm) | Radial Deflection (mm) | Total Deflection (mm) |
|---|---|---|
| 7500 | 0.08446 | 0.1321 |
| 8500 | 0.09471 | 0.1514 |
| 9500 | 0.11543 | 0.2645 |
| 10500 | 0.13612 | 0.3547 |
| 11500 | 0.15514 | 0.4615 |

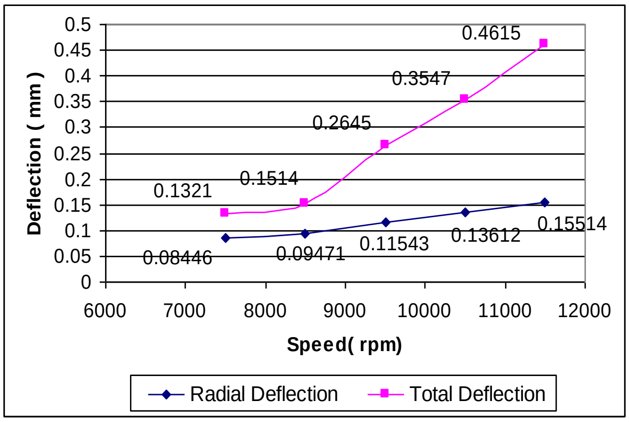



**Fig.5.3 Variation of Radial and Total Deflection of Blade due to Varying Speeds**

### 5.2 Typical stress distribution results in the ROTOR BLADE

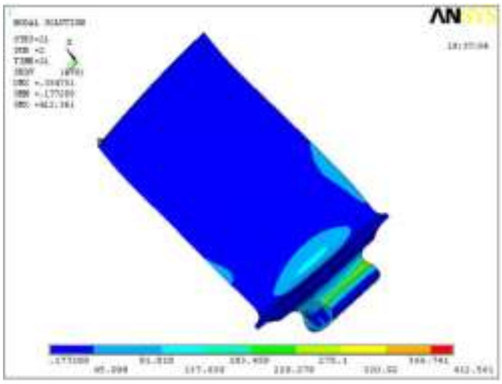


Fig 5.4 Equivalent stress distribution in rotor blade

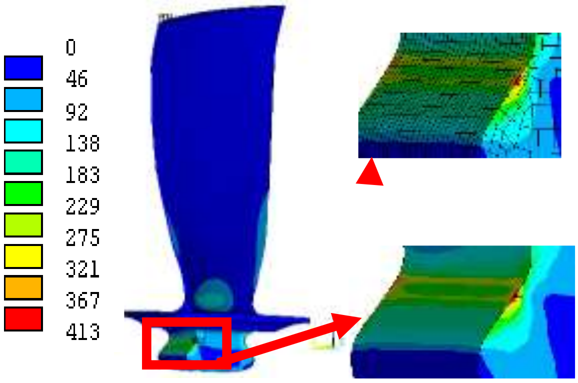



Fig 5.5 Equivalent stress distribution in rotor blade with mesh

### 1st Principle Stress

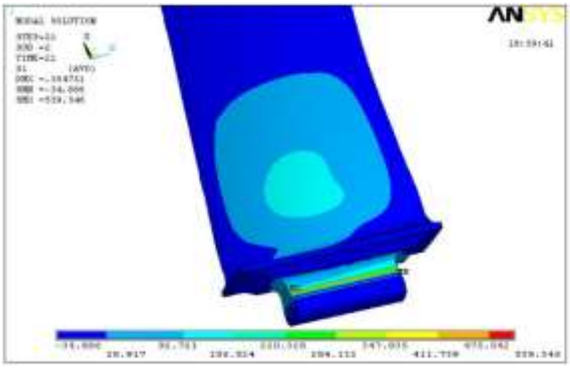


Fig 5.6 First Principle stress of a Rotor Blade along x-direction

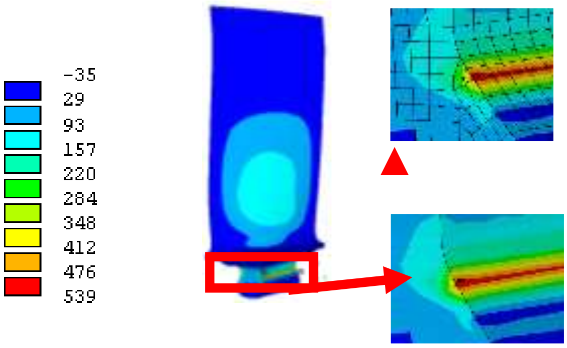



Fig5.7 First principle stress of a Rotor Blade X-direction with mesh

From the analysis of the compressor blade maximum stresses are induced due to existence of contact between compressor blade roots and dovetail groove of the disc in a running speed of 10500rpm. The main observations from the analysis are Equivalent stresses and 1st Principle Stresses are effectively compared with the standard material yield strength (970mpa, material property). The yield strength means stress at which the plastic deformation takes place after the elastic limit. If it exceeds the standard value then there may be chances of fracture initiation at the contact position. But observation clearly shows that the value is within the design limit so, compressor blade and disc assembly are in safe position from the crack initiation (breaking point) at 10500rpm. But when the speed increased to11500rpm we can notice the reduce in the stress by fig.6.8 which indicates the initiation of crack at the contact region

**Table.5.2 Stresses in Blade due to varying Speeds**

| Speed (rpm) | 1st Principle Stress (Mpa) | Equivalent Stress(Mpa) |
|---|---|---|
| 7500 | 399.66 | 311.112 |
| 8500 | 422.45 | 356.66 |
| 9500 | 498.59 | 391.22 |
| 10500 | 539.346 | 412.561 |
| 11500 | 510.79 | 398.62 |

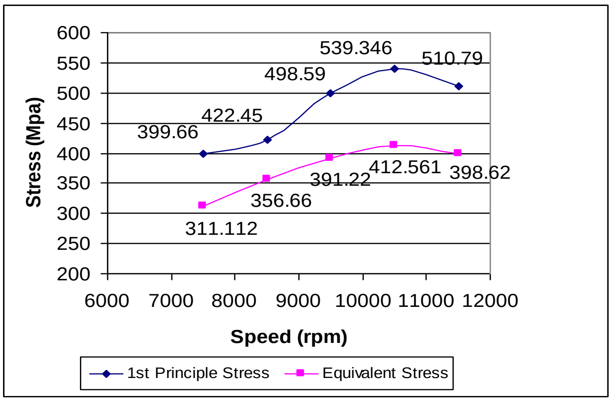



**Fig.5.8 Variation of Stresses in Blade due to varying speeds**

## 5.3 Typical deformation distribution in compressor disc

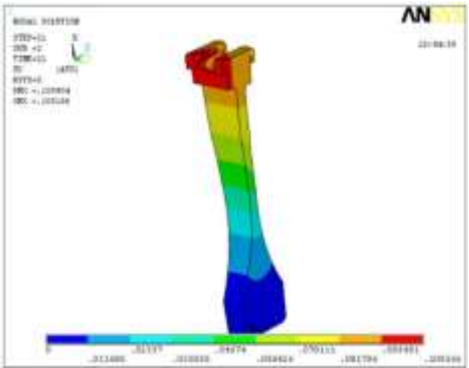


**Fig.5.9 Deformation distribution along UX**

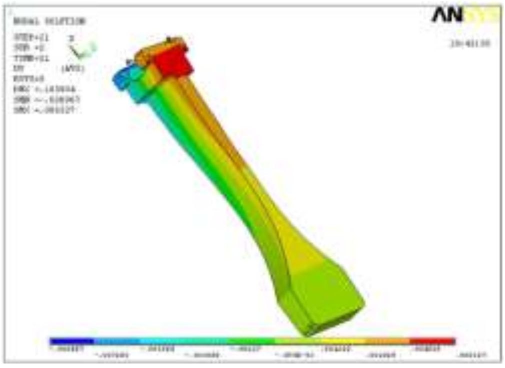


**Fig.5.10 Deformation distribution along UY**

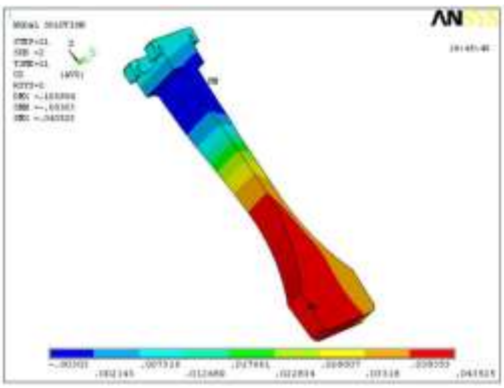


**Fig.5.11 Deformation distribution along UZ**

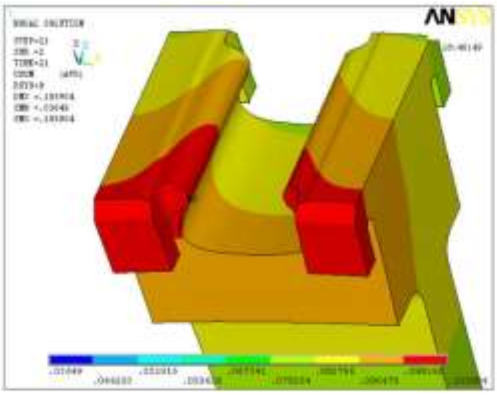


**Fig.5.12 Deformation distribution along USUM**

**Table.5.3 Deflections in Disc due to varying Speeds**

| Speed (rpm) | Radial Deflection (mm) | Total Deflection (mm) |
|---|---|---|
| 7500 | 0.10322 | 0.10354 |
| 8500 | 0.10378 | 0.10397 |
| 9500 | 0.10412 | 0.10441 |
| 10500 | 0.10516 | 0.105904 |
| 11500 | 0.10756 | 0.10798 |

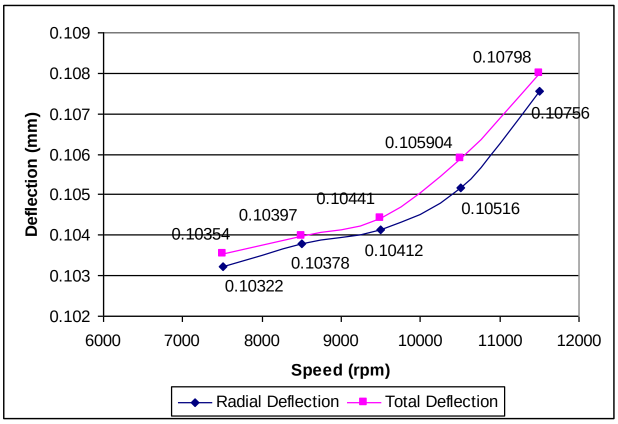


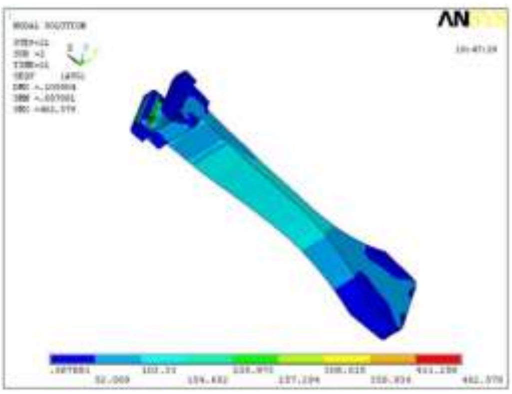

**Fig.5.13 variation of Radial and Total Deflection of Disc due to Varying Speeds**

Fig5.9 shows that radial deflection along X-direction which signification is that the radial clearance between compressor Blade-Disc assembly and casing is (0.134 mm + 0.105166 mm =0.239165 mm) compared to the existing standard value of clearance (0.5mm)

By analyzing the deformation distribution in the compressor blade and disk assembly. Here radial clearance is that “clearance between compressor blade and casing of the compressor. Whenever deflection starts radial clearance between compressor blade and casing reduce.

The main observation is that analyzing deflection from Table 5.3 and Fig 5.13, The Optimum value of Radial and Total deflection is well below the standard value, so whole compressor is safe from the radial deflection in compressor Disc. If exceed the standard value, again repeat the same analysis by having different loads. The following Table 5.1 and Fig 5.13 shows the variation of Radial as well as Total Deflection for various Centrifugal Loads.

## 5.4 Typical Stress distribution results in the ROTOR DISC



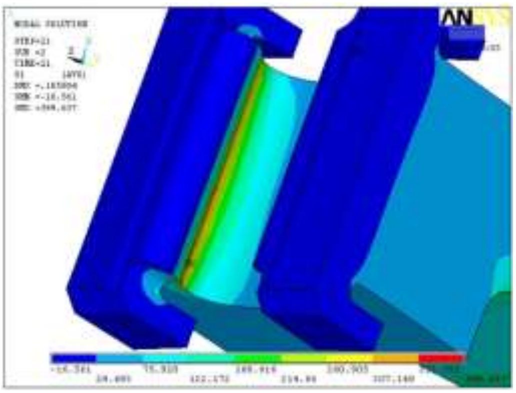


**Fig.5.14 Equivalent stress distribution along xyz direction Fig.5.15 First principal stress along x- direction**

**Table.5.4 Stresses in Disc due to varying Speeds**

| Speed (rpm) | 1st Principal Stress (Mpa) | Equivalent Stress (Mpa) |
|---|---|---|
| 7500 | 301.61 | 356.58 |
| 8500 | 348.54 | 391.61 |
| 9500 | 380.15 | 421.61 |
| 10500 | 399.639 | 462.579 |
| 11500 | 380.98 | 440.56 |

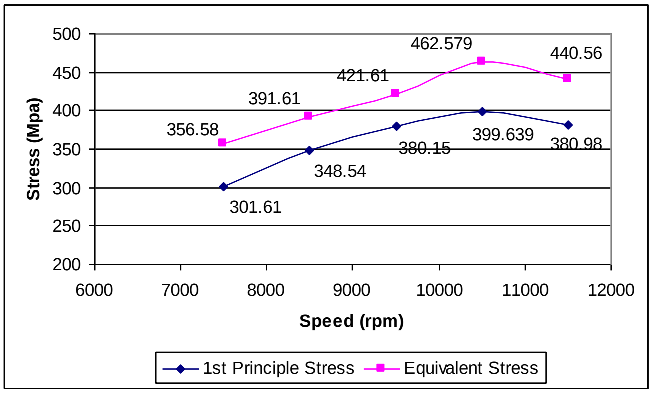



**Fig.5.16 Variation of Stresses in Disc due to varying speeds**

From the above analysis of the Compressor Disc maximum stresses are induced due to existence of contact between compressor blade roots and dovetail groove of the disc in a running speed of 10500rpm. The main observations from the analysis are Equivalent stresses and 1st Principle Stresses are effectively compared with the standard material yield strength (970mpa, material property). At the maximum running speed of 10500rpm the stress values are within the standard limit, which indicates the disc is free from crack. But when the speed is increased to 11500rpm we can observe the decrease in the stress values as shown in fig.5.16. Which indicates the initiation of crack at the contact region of Blade and Disc assembly, which in turn leads to the failure if the material.

## 5.5 Contact Pressure Distribution in compressor blade and disc

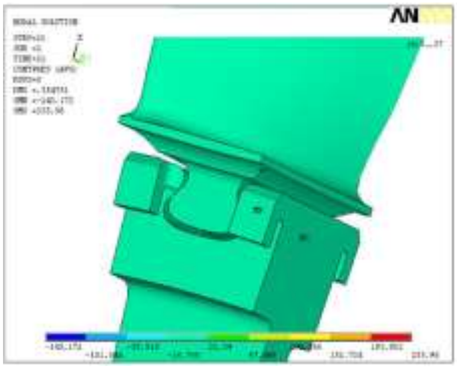


**Fig.5.17 Contact Pressure Distribution of Assembly**

Contact pressure distribution in the compressor blade disc assembly formed by the pressure load applied on the blades (80N). This pressure load distributes (0.0445N/mm$^2$) uniformly at constant rate. The above figure shows that pressure distribution of overall assembly.

**Table.5.5 Contact Pressure distribution due to varying Speeds**

| Speed(rpm) | Contact Pressure(Mpa) |
|---|---|
| 7500 | 198.61 |
| 8500 | 209.51 |
| 9500 | 221.13 |
| 10500 | 235.98 |
| 11500 | 220.65 |

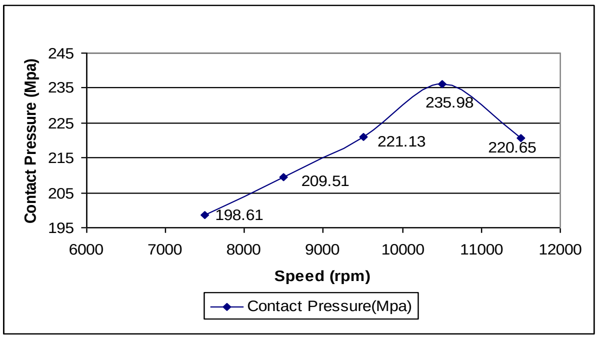



**Fig.5.18 Variation of Contact Pressure due to varying speeds**

## 6. CONCLUSIONS

In the present work an attempt was made to carry out contact stress analysis of compressor blade - disc assembly for gas loads and centrifugal loads by using commercial Finite Element package ANSYS.

The three types of elements SOLID-45, CONTA-173 and TARGET-170 were used for the analysis. The contact stress of bladed disc was done for static analysis.

The following Conclusion is obtained from the analysis:

- The radial clearance between compressor blade and casing (0.134mm) compared to the existing standard value of clearance (0.5mm) by analyzing the deformation distribution in the compressor blade and disc assembly.
- Determination of yield strength (413 Mpa) and effectively compared with the input yield strength (970mpa, material property).This observation clearly shows that compressor blade and disc are in safe position from the crack initiation (breaking point) at 10500rpm.
- The peak values of compressive stresses induced in the contact region are well within the design limit.
- The peak value of von-misses stresses induced near the region of contact stresses are also well within the design limit.
- The result and summary figure shows the variation of different stresses and deformation with maximum and minimum values.

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