Analysis and Improvement of Static behavior of an Impeller Blade for a Compressor of Aero Engine

PRAVEEN KUMAR K [1], KURUBARA RAGHAVENDRA[2], KHALANDAR AZAD[3]

[1,2,3]Lecturer in Mechanical Engineering

[1] Government Polytechnic, Karatagi, Koppal, Karnataka, India

[2] Government Polytechnic, Kushtagi, Koppal, Karnataka, India

[3] Government Polytechnic, Bellary, Karnataka, India

Abstract: Mixed flow compressor module is employed in machines such as gas turbine engines, steam turbines marine applications and power generation purpose. Mixed flow compressor impeller is cad modeled and analyzed for thermo-mechanical stress for improvement in the structural ability for load carrying capacity.

In this work mixed flow compressor impeller is cad designed based on the centrifugal compressor design methodology for given operating conditions. The geometric modeling of the impeller was done using the modeling tool catia and it was imported to Hyper mesh for meshing then exported to ANSYS through input file in 'in' file format for the subsequent analysis. The impeller was analyzed for different loads such as inertial and thermal loads and cyclic symmetric boundary conditions. The impeller shaft analyzed for natural frequency and mode shapes to avoid resonance conditions. Thermal barrier coating materials are used to reduce the heat transfer to the system and its structural ability is estimated. A layer of composite thermal barrier coating is created using hyper mesh and the effect of this on stress behavior is analyzed. The results shows reduction of deflection and stress with low thermal expansion coefficient material. Even composite material has lesser thermal conductivity along with density compared to the parent titanium material. This character of composites can be effectively utilized to increase the load carrying capacity of the impeller. The impellers can operate still at higher frequencies with this composite cladding. Modal analysis is carried out to find the resonant frequencies. The results show the first natural frequency of 1332 Hz is much higher compared to the operational frequency of 783Hz. So resonance will not takes place in the problem.

Index Terms – Impeller, Blade, Model Analysis, FEM.

I. INTRODUCTION

Overall size of an aircraft engine should be as small as possible in order to reduce drag and to obtain optimum thrust per unit mass flow rate through the engine. Therefore, much care is taken to optimize frontal area and axial length of each engine component including compressors.

Axial compressors are mainly used in aircrafts as they have higher flow capacity per unit frontal area, high efficiency and ease of staging. But they are expensive, heavy and have small pressure ratio per stage. The radial discharge centrifugal compressors (henceforth, radial compressors) have small axial lengths ad can provide comparatively higher pressure ratios per stage. Both they have relatively low efficiency and larger frontal area as compared to axial compressors.

At low mass flow rates, the exit blade height required for axial compressor becomes small and the efficiency benefit of the axial compressors reduces rates due to prominent detrimental effect of tip clearance. Efficiency of axial compressors is comparable to that of radial compressors at low mass flow rates. In such cases, radial compressor prove to be more beneficial due to their capability of providing high pressure ratio per stage. However, the benefit is accompanied by larger frontal area of engines. Compressors for conventional short-range missiles with small turbojet or turbofan engines and those for helicopters require moderate pressure ratios and mass flow rates small overall size is a critical requirement. The problem of larger frontal area associated with radial compressors can be solved if the compressors is used with split diffuser system. But such arrangement gives still lower efficiencies due to sudden turning of flow in 90^0 band. It is estimated that mixed flow centrifugal compressors (henceforth, mixed flow compressors) based on radial compressors design methods can meet this pressure ratio and mass flow rate requirements with higher efficiencies and lower frontal areas.

A Centrifugal compressor is an arrangement consisting of blades arranged, such that flow enters and leaves radially. Centrifugal compressors are mainly used in aircrafts, refrigerators marine applications and others. They can be used to produce higher-pressure ratios. The centrifugal compressors consist of impellers with main and splitter blades, which help to raise the pressure ratio. Due to the high –pressure ratios developed a high thrust is produced which causes an axial and radial thrusts. To balance this thrust load a balancing device called balancing disc must be used, so that this disc may take some part of the load coming on to the bearing. In the absence of the balancing disc one should go for a bearing that takes the whole thrust coming on to the bearing. Compressor shaft also designed.

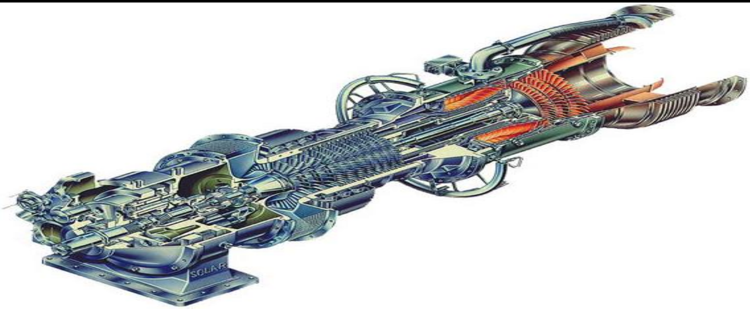


Fig 1 : Sectional View of the Gas Turbine Engine

As the bearings are supplied with lubricating oil, there is chance of oil leakage from the ends of bearing; seals are used with very close tolerances mounted in the stationary member. Mechanical seals are being used on both the ends of bearings to avoid the leakage of lubricating oil to the other ends. The other purpose of using the mechanical seals is to avoid the mixing of air with lubricating oil. Labyrinth seals are being used to avoid the leakage of air re-entering the impeller and are also used to obstruct the leakage of lubricating oil past the mechanical seal.

Mixed flow compressors

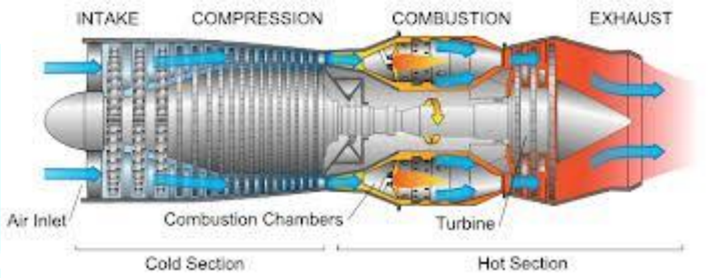


Fig.2: Terminology of Gas Turbine sections

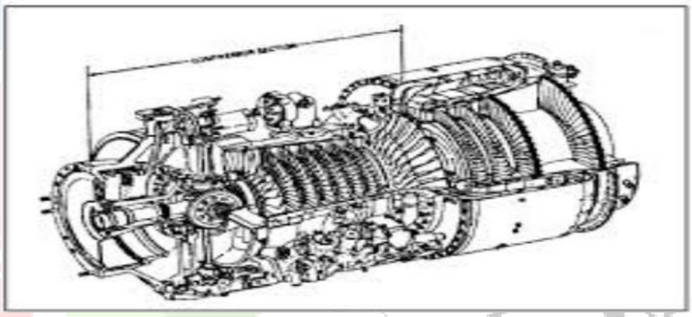


Fig 3: Mixed Flow Compressor

A **mixed flow compressor** combines the axial and radial components to produce a diagonal flow unit. The exit mean radius is greater than at the inlet, but the flow tends to exit in an axial rather than radial direction. This eliminates the need for a relatively large diameter exit diffuser associated with centrifugal compressors, The impeller can be machined from solid using NC machines, in much the same way as that of a centrifugal compressor.

Diagonal or mixed-flow compressor is effectively a cross between a centrifugal and axial-flow compressor. The American term diagonal-flow is very apt, because these compressors combine both axial and radial velocity components. The prime advantage is the relatively small diameter across the exit diffuser, compared with that of the equivalent centrifugal compressor Axial compressor an approximate comparison between radial and mixed flow compressors with same mean impeller exit diameters regarding frontal areas is shown in figure.

Action at critical speed: Extremely few shafts are in perfect balance. Owing to slight machining inaccuracies, the center of mass and the geometric center of rotation do not coincide, although the difference may amount to only a few ten-thousandths of an inch. Rotation produces a centrifugal force of the mass center, which is balanced by the springing action of the shaft. The axis of rotation is changed at the critical speed from the geometric center to the mass center.

Lord Rayleigh found that a body vibrating at its natural frequency does so with simple harmonic motion and that all points on the body come to rest at the same time and attain their maximum velocity simultaneously, even though they are completely out of phase. A vibrating body contains energy i.e., kinetic energy and potential energy. Since total energy in the system remains constant the maximum potential and kinetic energies may be equated to determine the frequency. If a shaft has a number of weights W_1, W_2, etc., placed along it and the corresponding maximum deflections of the weights are y_1, y_2, etc.,

Maximum potential energy = $\frac{1}{2}W_1y_1 + \frac{1}{2}W_2y_2$ +etc.

Maximum kinetic energy $= \frac{1}{2}\frac{W_1}{g}\omega^2 y_1^2 + \frac{1}{2}\frac{W_2}{g}\omega^2 y_2^2 + etc.$

Equating

$$\omega = \sqrt{\frac{\frac{1}{2}W_1y_1 + \frac{1}{2}W_2y_2 + etc}{\frac{1}{2}\frac{W_1}{g}\omega^2 y_1^2 + \frac{1}{2}\frac{W_1}{g}\omega^2 y_2^2 + etc}} = \sqrt{g}\sqrt{\frac{\sum Wy}{\sum Wy^2}}\ rad/\sec ond$$

or

$$F = \frac{60\omega}{2\pi} = \frac{60\sqrt{g}}{2\pi}\sqrt{\frac{\sum Wy}{\sum Wy^2}} = 187.5\sqrt{\frac{\sum Wy}{\sum Wy^2}} Cycles\ per\ \sec ond$$

Lord Rayliegh developed an important principle, which is of great use in the vibration work. This principle states that in a vibrating system the displacements of the system are such that, when the maximum potential and kinetic energies are equated to each other, the lowest frequency is obtained.

If the displacements are taken any other way, the frequency will be higher, i.e., the system deflects naturally so that frequency is a minimum. This principle is applicable to any of the possible modes of vibrations.

The critical speed is approximately inversely proportional to the square root of the static deflection. For beams or shafts, the static deflection is proportional to $\frac{WL^3}{EI}$ so the critical speed is proportional to $\frac{1}{\sqrt{W}}, \frac{1}{L^{\frac{3}{2}}}, \sqrt{E}, \sqrt{I}$ or D^2 These rations are quite useful in estimating the changes, which may be placed either above or below the operating speed. If the unit is to operate at high speeds, which do not vary widely the critical speed, may be below the operating speed and the shaft is said to be flexible. If the operating speed is low or must vary through wide ranges the critical speed is placed above it and the shaft is called rigid or stiff.

II. Literature on Suspension:

A.k.Singh and Narayana Sarma presented the dynamic analysis of a coupled rotor-bearing foundation system. A 200 MW turbo generator rotor on its bearings and foundation of a thermal power plant is considered. The model is constructed by sub structuring the rotor and foundation separately. The dynamic stiffness of the foundation is obtained by frequency response analysis with harmonic excitation applied at bearing locations. The critical speeds of rotor alone on its bearings and with the foundation are compared.

D.L. Rhode and **M.J.Murry** made a parametric study of geometrical effects on labyrinth seal leakages. Further an enhanced basic understanding of the effect of sharp corners on cavity friction factor was obtained from contours of intense turbulence generation.

W.W.Gardner presented a test data on the effect of circumferential pivot position on bearing temperature for two tilting pad thrust-bearing designs. Pivots at 50,60,70 and 80 percent of the are length were used. The results indicate that the pad temperatures decrease significantly as the pivot is advanced from the central (50 percent) position to at least the 70 percent position, and then ten to level, with some indications that a minimum is reached at about 75 percent.

C.Bouchoule, M.Fillon, D.Nicolas in their work a test machine, experimental results and comparison between theoretical results and experimental data are presented.

The test bearing are located in two speed increasing and reduction gearboxes. The shaft is driven by a 1 MW motor. The bearing diameter and the bearing length are equal to 160 mm. The rotational speed varies from 2700 r.p.m to 11,800 r.p.m. applied load is up to 88,000N. Temperature in the bearing, power losses and oil flow are measured. The influence of the bearing design and the pivot position on the pad is analyzed.

John. M. vanes predicted that first two or three critical speeds for industrial machinery can be reliably predicted with errors no larger than seven percent provided that care is taken to optimize the computer program used and to ensure that accurate values are used for the mass- elastic model, bearing properties and foundation impedance. Reliable predictions of critical speeds also depend on the proper choice of the bearing and foundation parameters.

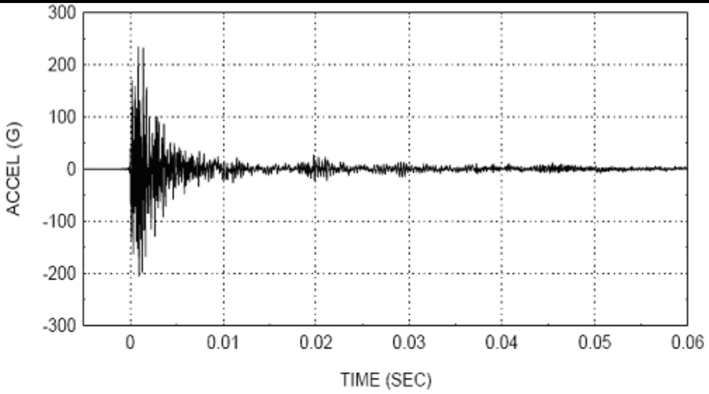


Figure 1.

Fig: 4 Typical Space Vehicle Pyro-technique shock

III. Mathematics of Rotating Disks of Uniform Thickness

Consider a thin rotating circular disk. Assume that over thickness, the radial and circumferential stress does not very and the stress in axial direction σ_z is zero. The stress produced in a disk rotating at high speed due to centrifugal forces of the rotating disk. The body force is the centrifugal forces which is $F_r = \rho\omega^2 r$ where ρ is the mass density of the disk material and ω is angular velocity of the rotating disk. Then the stress distribution in the disk symmetrical with respect to the axis of rotation i.e. z axis is the axis of rotation.

The equilibrium equation is

$\partial\sigma_r/\partial r + (\sigma_r - \sigma_\theta)/r + \rho\omega^2 r = 0$

Since r is the independent variable, the above equation can be written as,

$d(r\sigma_r)/dr - \sigma_\theta + \rho\omega^2 r = 0$

The strain components are,

$\varepsilon_r = \partial u_r/\partial r$

and

$\varepsilon_{\theta} = u/r$

From Hook's law, if $\sigma_z = 0$

$\varepsilon_r = 1/E(\sigma_r - \nu\sigma_\theta)$

$\varepsilon_\theta = 1/E(\sigma_\theta - \nu\sigma_r)$

From equation 2.3a and 2.3b

$\varepsilon_r = d(r\varepsilon_\theta)/dr$

Using the Hook's law,

(Substituting the equations 2.4a and 2.4b to equations2.5)

$1/E(\sigma_r - \nu\sigma_\theta) = 1/E\ \{d(r\sigma_\theta - \nu r\sigma_r)/dr\}$

Let $r\sigma_r = y$

Then from equation 2.2

$\sigma_\theta = dy/dr + \rho\omega^2 r$

Substituting these in equations 2.6

$r^2(d^2y/dr^2) + r(dy/dr) - y + (3+\gamma)\rho\omega^2 r$

The solution of the above differential equation is

$y = C_1 r + C_2/r - \{(3+\gamma)/5\}\rho\omega^2 r$

Where C_1 and C_2 are the constant of integration to be determined from the boundary condition, then from equation

$\sigma_r = C_1 + C_2/r^2 - \{(3+\gamma)/5\}\ \rho\omega^2 r^2$

From equation 2.5

$\sigma_\theta = C_1 + C_2/r^2 - \{(1+3\gamma)/5\}\rho\omega^2 r$

Annular Circular Disk

- If there are no external forces applied at the inner radius ‘a’ and outer radius ‘b’ which are the boundaries,
- Then
- $(\sigma_r)_{r=a}= (\sigma_r)_{r=b}=0$
- From which we find
- $(\sigma_r)_{r=a}= C_1 + C_2/r^2 - \{(3+\gamma)/5\}\rho\omega^2r^2=0$
- $(\sigma_r)_{r=b}= C_1 + C_2/r^2 - \{(3+\gamma)/5\}\rho\omega^2r^2=0$
- Solving equations 2.13 and 2.14 for C_1 and C_2
- $C_1= \{(3+\gamma)/5\}\rho\omega^2(b^2+a^2)$
- $C_2=-\{(3+\gamma)/5\}\rho\omega^2b^2a^2$
- Substituting these into equations 2.11 and 2.12
- $\sigma_r=\{(3+\gamma)/5\}\rho\omega^2\{b^2+a^2-(a^2b^2/r^2)-r^2\}$
- $\sigma_\theta= [(3+\gamma)/5\}\rho\omega^2\{b^2+a^2+(a^2b^2/r^2)-\{(1+3\gamma /(3+ \gamma)\}r^2]$
- The radial stress σ_r reaches to maximum at $r = \sqrt{ab}$
- Then $(\sigma_r)_{max}= \{(3+ \gamma)/8\}\rho\omega^2(b-a)^2$
- The circumferential stress σ_θ reaches to maximum at the inner boundary when r=a
- $(\sigma_\theta)_{max}= \{(3+ \gamma)/4\}\rho\omega^2\{b^2+(1+\gamma)/(3+ \gamma)a^2\}$
- The displacement in the disk is calculated

IV PROBLEM DEFINITION

Coupled field analysis to find the temperature effect on stress generation and resulting stability of the impeller is the main objective of the problem.

The objectives include

- Modelling of the impeller (Cyclic segment)
- Meshing and analysis
- Checking for the dynamic stability
- Finding the effect of composite material on structural performance

Requirement of the Problem: Since impellers rotates at very high rpm along with thermal loads, it is always desirable to have better thermal stability. Even though temperature values are small, the resultant stresses are more. So coupled analysis (Load transfer of one type of load to other type of load) is very essential for better structural stability.

Methodology:

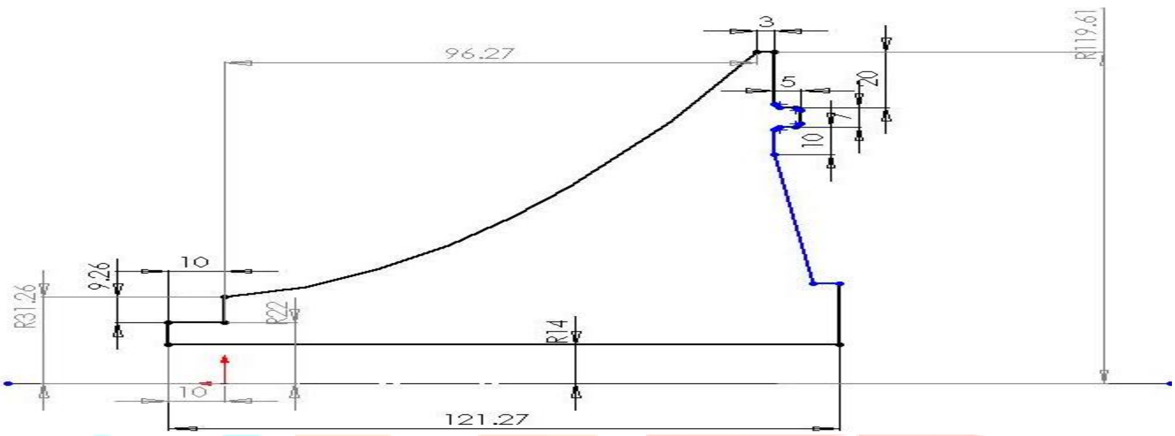
- Literature on Impellers and corresponding problems.
- Three dimensional modelling
- Meshing with Hypermesh for good quality elements to obtain better results
- Analysis for static inertia and temperature loads
- Improvement analysis by providing the thermal resistant materials

Material:

Table3.1: Material Properties

Material	Titanium	Kevlar UD	Std CF Fabric	HMCF Fabric
Young’s Modulus(GPa)	110	75	70	85
Poison’s ratio	0.35	0.34	0.1	0.1
Thermal Expansion Coefficient	5e-6	4	2.1	1.1
Density	4500	1400	1600	1600

Geometrical Modeling:



Blade profile:

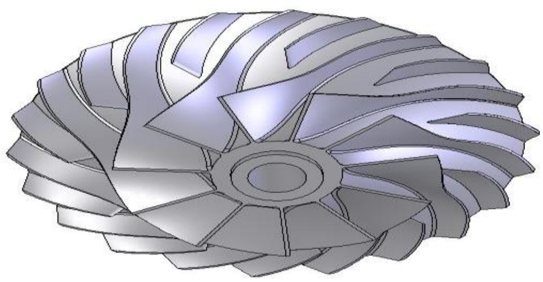


Figure 5: Mixed flow impeller

NO.OF BLADES: 11Main and 11 Splitter 32.73^0 Sector

Since the component is cyclically symmetrical it is modeled as a 32.73^0 sector. Cyclic symmetry is specialized condition where features that are repeated about an axis can be modeled by a single instance (sector) of the feature. More precisely cyclic symmetry can be defined as follows;

Many structural components possess geometric characteristics, which are repeated about an axis of symmetry for example disks, gears, and impellers. These structures can be defined in terms of a primary segment, which is repeated at equally spaced intervals about the symmetry axis. This technique is called cyclic symmetry. The main advantage of cyclic symmetry boundary conditions is large savings in CPU time and computer resources. The 32.73^0sector of the component is as shown below;

V.RESULTS FOR ANALYSIS

Cyclic symmetry analysis is carried out to find the temperature and stress generation in the titanium impeller. Generally, a problem with some kind of symmetry, if it is used properly, it will reduce lot of execution time and memory requirements. Also it allows the user to use better mesh with low size for the element. One can satisfy the requirements for better quality mesh (Aspect ratio, warpage, skew angle and Jacobian) which is the base for good convergence for finite element results.

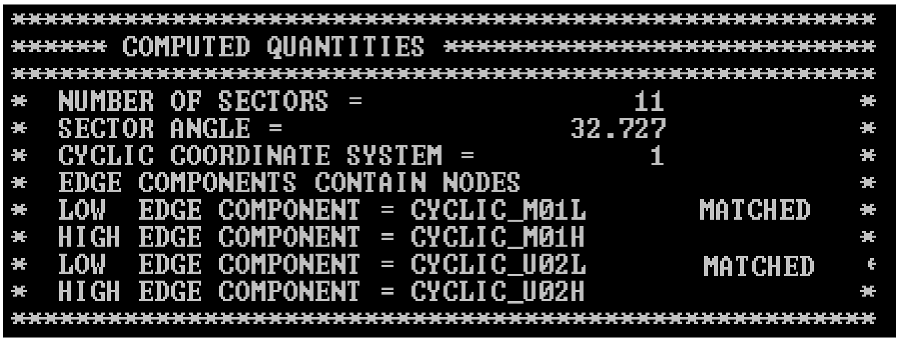


Fig 6: Solution Printout for Cyclic Segments

If geometry is symmetrical, then only ansys gives a printout option as shown above. Almost 11 segments and required to form the full impeller. Each segment angle is 32.727 degrees. A variable temperature across the blade sections are applied along with a pressure load. The results are obtained through ansys software.

Analysis Results:

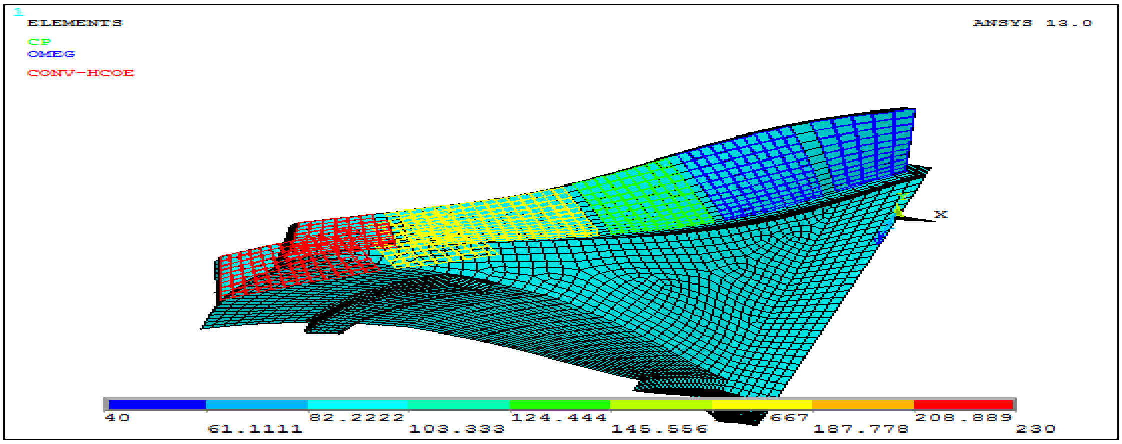


Fig7: Temperature Boundary Conditions

The applied boundary condition on the impeller blade. The temperature is varying from 40 degrees to 230 degrees. The inlet temperature is less and the outlet temperature is 230 degrees. The temperature load is applied on both main and splinter blade.

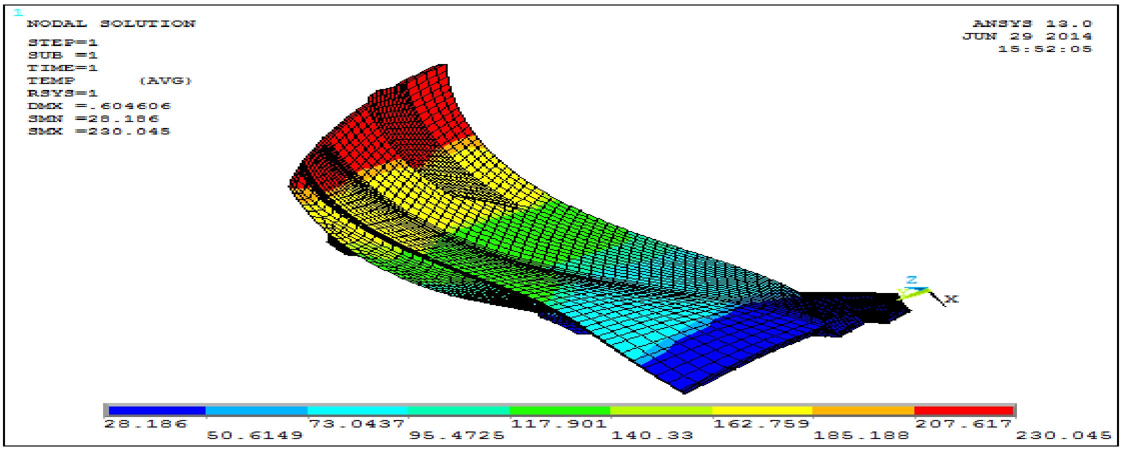


Fig 8: temperature plot

The figure 8 shows solution obtained after doing thermal analysis. The results shows variation of temperature along the blade path. Minimum temperatures at the inlet region and higher temperatures at the outlet.

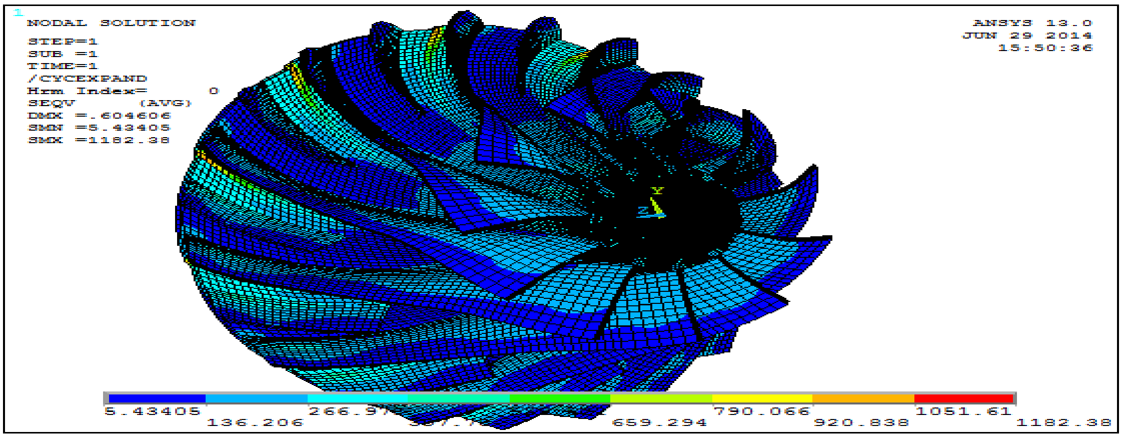


Fig9: Vonmises Stress Plot

The figure9 shows vonmises stress in the impeller section. The stress value is increasing to 1182.38 Mpa. The stresses are maximum at the splinter blade. The status bar at the bottom shows variation of stress in the impeller blade.

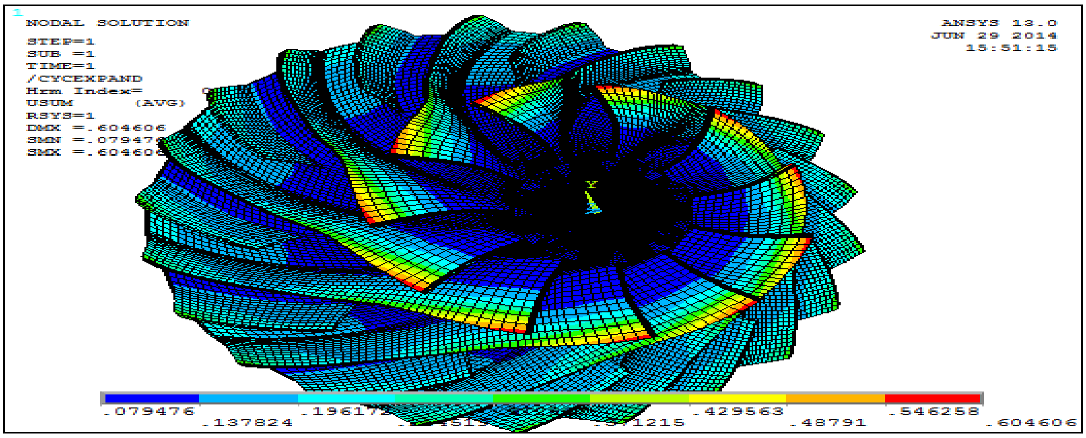


Fig 10: Overall displacement plot

The figure 10 shows overall displacement plot in the problem. Maximum displacement is around 0.6mm at the inlet main blades. This can be mainly attributed to the height of the blade. It is almost having cantilever type arrangement, by which the deflection is maximum at the main blade.

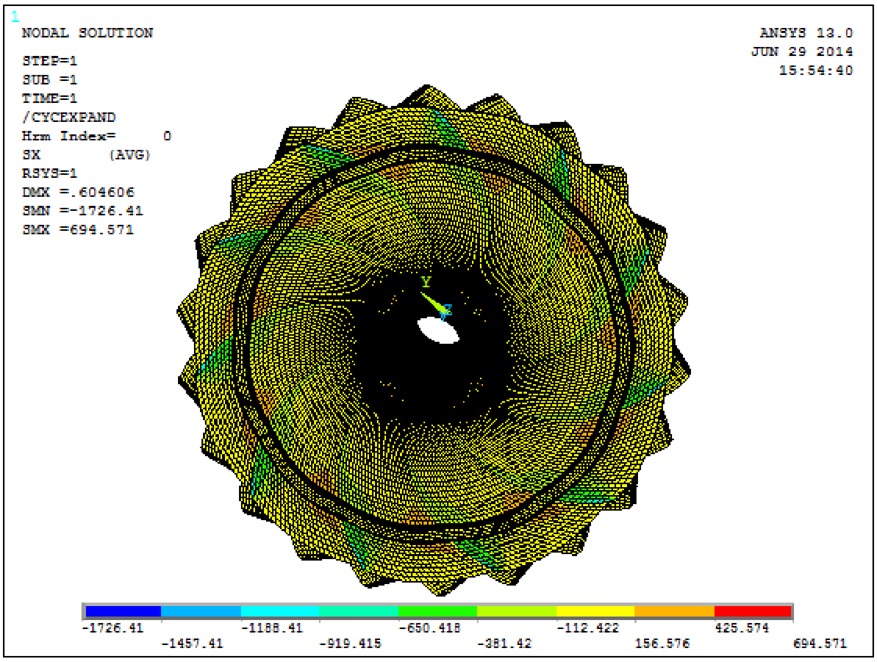


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Fig 11: Radial Stress Plot

The figure 12 shows radial stress in the impeller section. Maximum stress is around 694.571Mpa.

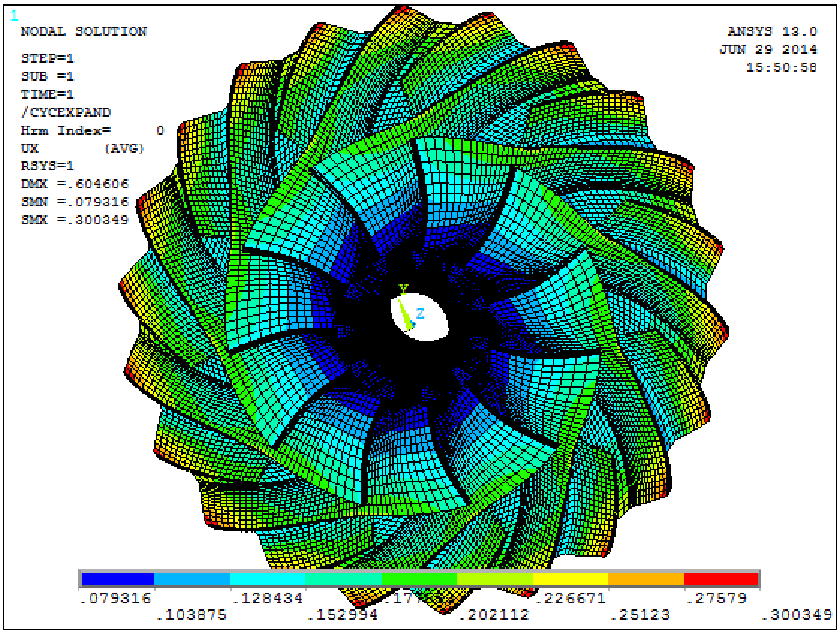

Fig 12: Radial Displacement plot

The figure 12 shows radial displacement in the impeller. Maximum displacement is around 0.3mm. Maximum radial displacement is observed at the main blade outlet section.

5.3 Result summary

Table 2 : Result Summary (Comparison for Different Thermal barrier Coatings)

Description	Radial Displacement(mm)	Resultant Displacement(mm)	Vonmises Stress (MPa)
Titanium	0.3	0.6	1182
Kevlar Ud	0.189	0.318	931
Standard CF Fabric	0.109	0.244	777.824
HMCF Fabric	0.099	0.233	695.24

The table 2 shows results variation with the use of coatings on the impeller. The stress values are reducing along with displacements. This can be mainly attributed to the reduced density along with thermal expansion coefficient. Even thermal conductivity value of the above composite materials are less by which transfer of heat will be less to the core impeller. So improvement is possible with thermal barrier coatings on titanium impellers.

VI Conclusion

Mixed flow impeller blade has been analyzed for structural integrity under thermo-mechanical loads. Coupled field analysis has been carried out to find the effect of thermal effects on the structural stability. The overall summary of the problem is as follows.

- The geometry of the impeller is built for one segment due to cyclic symmetry advantage
- Initially using sketcher, the two dimensional profile is built and later extruded using part modeller of catia. The main blade and splinter blade geometry are further extruded on the existing three dimensional geometries.
- The geometry is imported to hyper mesh for brick meshing. The geometry is split to ease brick meshing. The elements are maintained same on both the sides of the cyclic symmetry which is the primary requirement for cyclic symmetry analysis. Aspect ratio and Jacobin is maintained on the mesh for better convergence.
- The imported mesh in 'inp' format is applied with varying thermal and pressure loads which are represented on the figures. An intertia load of 4467 rad/sec is applied along with the thermal loads. The results for radial displacement,

resultant displacement and vonmises are captured. Here radial displacement is required to know whether the impeller radial growth with touch the outer casing and vonmises stress is for structural integrity.

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