

NON-LINEAR TRANSIENT AND STRUCTURAL ANALYSIS OF ROTOR ASSEMBLY OF A MICRO GAS TURBINE

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Abstract: Miniaturization of gas turbines is not straightforward and is not simply the scaling down of large gas turbines. The small dimensions have implications on the efficiency of the components due to the increased viscous friction. Also limits in fabrication accuracy cause additional flow losses. For a 120sec typical accel-decel mission, non-linear transient thermal & structural analyses were performed. It is important to note that this mission resembles true test conditions of gas turbine engines. An axisymmetric model was considered for analysis and mass approximations were carried out using real constants. Heat transfer coefficients are calculated using appropriate correlations and transient thermal scenario of system was captured. Structural analysis was performed at several time points in the mission cycle with all thermal & structural loads. Stress & temperature history of component was examined & fatigue life calculation was performed for all critical locations of the component. It is observed that component can run for a minimum 100000 cycles. Also, clearance calculation was performed for rotor & stator parts in system and observed that current method of assembly was safe. Regarding bearing design, at higher shaft speeds (up to 100000rpm); stainless steel ball bearings cannot withstand the stresses. So, ceramic ball bearings are used as the best feasible solution for the given temperature & pressures.

Keywords: Micro Gas Turbine, Autodesk Inventor Professional, Ansys V13, FE Analysis, Fatigue Life.

1. INTRODUCTION

Micro turbines are becoming widespread for distributed power and co-generation (Combined heat and power) applications. They are one of the most promising technologies for powering hybrid electric vehicles. They range from hand held units producing less than a kilowatt, to commercial sized systems that produce tens or hundreds of kilowatts. Electronic power switching technology eliminates the need for the generator to be synchronized with the power grid. This allows the generator to be integrated with the turbine shaft, and to double as the starter motor. They accept most commercial fuels, such as gasoline, natural gas, propane, diesel, and kerosene as well as renewable fuels such as E85, biodiesel and biogas. The machines generally rotate over 50,000 rpm. The bearing selection (oil or air) is dependent on usage. A single shaft micro turbine with high rotating speeds of 90,000 to 120,000 revolutions per minute is the more common design, as it is simpler and less expensive to build.

1.1 Overall aim and Specific Objectives:

The overall aim of the present work is to carry out Transient Thermal and Structural analysis (Centrifugal Stress analysis, fatigue life of shaft and also to justify the clearance between rotating components of rotor assembly of a micro gas turbine. The problem is intendable by continuum thermal – structural analysis method. Even though the assembly is available for testing, experimental investigation will be expensive and time consuming.

In the present paper, ANSYS software is employed.

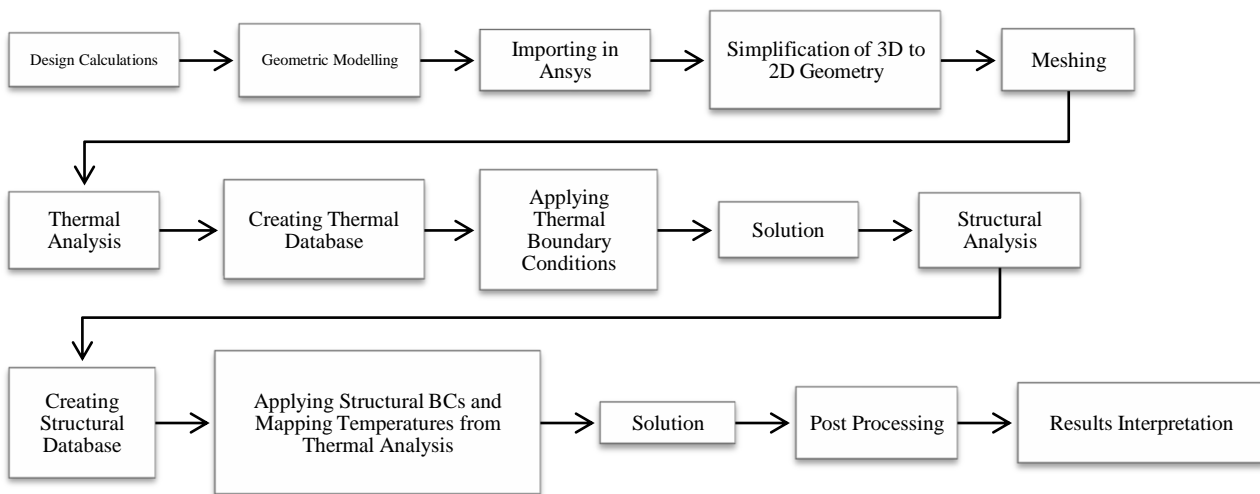


Figure.1.1 Flow chart showing procedure of analysis.

2. Literature Review

J.C Bruno and A Coronas [1] summarizes that the use of micro gas turbine (MGT) for onsite small scale energy production offers a great opportunity for primary energy saving and reduction of pollutant and greenhouse gas emissions. These benefits mainly come from the possibility to locally recover waste heat from exhaust gas especially for air conditioning thermally activated technologies and also for their capability to use low heating value gases such digester gas.

P.A. Pilavachi [2] stated that micro-turbines offer a number of potential advantages compared to other technologies for small-scale power generation, particularly for distributed power generation, there is an uncertainty about their market potential but they could be used for power generation in the industrial, commercial and residential sectors.

Nageshwara Rao Muktinutalapati[3] suggests ceramic material for micro gas turbine components in his paper Materials for Gas Turbines – An Overview: he says that Increase in turbine inlet temperatures, beyond what is possible with super alloys, can be conceived if ceramic materials can be used in place of super alloys in gas turbine engine. Turbines would then operate at higher temperatures, yielding higher power with smaller engine sizes.

R.A. Van den Braembussche[4] stated that Good micro gas turbines are not miniaturized copies of large ones. The very high rotational speed that is needed to obtain the enthalpy and pressure changes prescribed by the gas turbine cycle, is the major mechanical problem.

Mehdi Tofighi Naeem,[5] mentioned in his journal Failure Analysis of Gas Turbine Blades, that blade failures can be caused by a number of mechanisms under the turbine operating conditions of high rotational speed at elevated temperatures.

Mohammed Javed Hyder [6] mentioned in his paper Stress Distribution of the Gas Turbine Blade that, Many kinds of stresses do come into play when it comes to turbo machinery especially turbine where the temperature changes have also become vital. The important stresses in the designing of gas turbine include Thermal stresses, Centrifugal stresses and Creep phenomena.

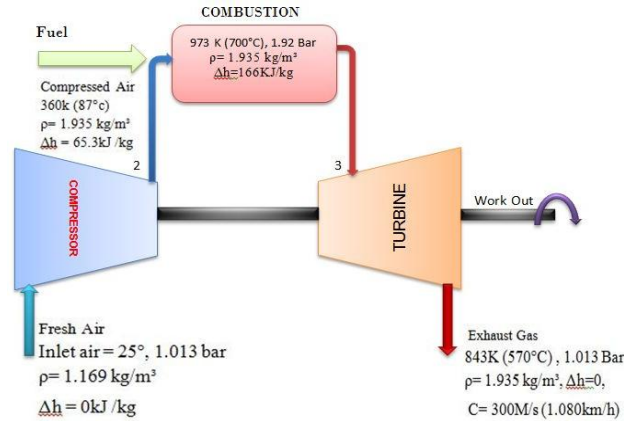
R Christmann [7] conclude that frequent causes of failure on the compressor wheel are fatigue fractures, whereby a distinction can be made between Low-Cycle Fatigue (LCF) und High-Cycle Fatigue (HCF) as regards the tolerable number of cycles and the damage effects occurring. The LCF range is characterized by the dominant occurrence of plastic strain.

Siddarth.A, S.R.Shankapal and S.Ramamurthy[8] analyzed Radial turbine rotor of a micro gas turbine rotating at 1,00,000 rpm. They says there are many rotating components in a micro gas turbine. Among them the radial turbine rotor is a critical component since it is subjected to high centrifugal loads and temperatures. Radial turbine rotor is analyzed for different loads such as inertial and thermal loads and cyclic symmetry boundary conditions. The analysis indicated that the centrifugal stresses developed in the rotor were exceeding the yield limit.

J Peirs, T Waumans, P Vleugels [9] concluded that miniaturization of gas turbines is not straightforward and is not simply the scaling down of large gas turbines. The small dimensions have implications on the efficiency of the components due to the increased viscous friction. Also limits in fabrication accuracy cause additional flow losses. K. Shoghi , M. Becker , T. Dicker , D. Metz , C. Oberste-Brandenburg [10] presents the theory and the method of determining the

required torque and angle values applied to the nut on the rotor part of the turbocharger to provide sufficient clamping force during the operation.

3. Calculations



3.1 Detailed Values of the process;

3.1.1 Inlet to the Compressor

Inlet air = 25°
 Pressure = 1.013 bar
 Density = 1.169 kg/m³
 Enthalpy = $\Delta h = 0$

3.1.2 Outlet of the compressor

Temperature = 360K = 87°C
 Pressure = 2Bar
 Density = 1.935 kg/m³
 Enthalpy = 65.3KJ/kg

3.1.3 Combustion Chamber

Temperature = 973k = 700°C
 Pressure = 1.92bar
 Density = 0.69 kg/m³
 Enthalpy = 166KJ/kg

3.1.3 Exhaust Gases

Temperature = 843K = 570°C
 Pressure = 1.013bar
 Density = 0.42Kg/m³
 Enthalpy = 166KJ/kg
 Speed = 300M/s (1.080 km/h)

3.2 Engine Thrust.

$$F = C_m \times \dot{m}$$

$$F = 218 \frac{\text{m}}{\text{s}} \times 0.18 \frac{\text{kg}}{\text{s}}$$

$$F = 39 \text{ N}$$

3.3 Centrifugal load on rotor wheel.

$$F = \frac{m \times U_m^2}{r_m}$$

F = Centrifugal force in newton
 Um = Average peripheral speed in m/s
 Rm = dm / 2, average radius in m
 m = blade mass in kg
 F = 1780 N

3.4 Critical rotational speed.

$$w = \sqrt{\frac{3 \times E \times I}{m \times (I + C) \times l^2}}$$

l= Bearing Spacing
 d= Material Thickness
 ρ= Material Density, 7800 kg/m³
 Wheel = 21597 c/s

$$\frac{1}{w^2} = \frac{1}{W_{compressor}^2} + \frac{1}{W_{Turbine}^2} + \frac{1}{W_{Shaft}^2}$$

$$\frac{1}{w^2} = \frac{1}{38046^2} + \frac{1}{33760^2} + \frac{1}{21597^2}$$

W=16406.679 cycles/sec
 Critical Speed = 156683.78Rpm
 Designed Rpm = 1, 00,000

4. Modeling

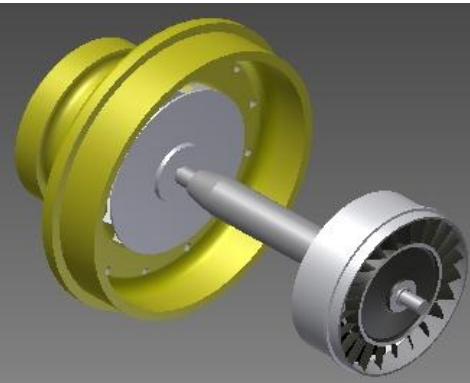
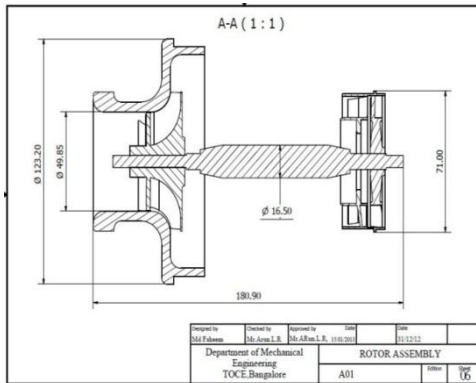


Figure.4.1 2D representation of Rotor Assembly Figure. 4.2, 3D model of Rotor Assembly.

4.1 Materials selected for rotor assembly components

Sl No	Component Name	Material
1	Compressor	A1 - 6061
2	Compressor Casing	EN-8
3	Shaft	En-24
4	Nozzle Guide Vane	Inconel-718
5	Turbine	Inconel-718

Table 4.1 Materials selected for Rotor Assembly Components

5. Analysis

5.1 Procedural Steps for Thermal Analysis.

1. Geometry cleanup
2. Data cleanup of boundary conditions i.e., scaling of flow vs. time data for one blade.
3. The geometry is divided into thermal zones.
4. The parasolid geometry is imported to ANSYS and the geometry is meshed in the valid meshing criterion. Surf152

elements have created for all thermal zones.

5. The heat transfer coefficients are calculated for all thermal zones transiently using appropriate correlations.
6. Input files are created with time vs HTC & time vs temperature tables for transient analysis in ANSYS.
7. A BC file is created for solution run. This contains all commands to apply HTCs & bulk temperatures and solution logic to run non-linear transient simulation. Writing of this command snippet will reduce human effort for other iterations.
8. Performing of non-linear transient thermal analysis.
9. Post processing of these results.

5.2 Geometric Details

- An axisymmetric model is considered for the analysis.
- All non-axi features(like blades of compressor and turbine) are simulated based on real constants using “plane with thickness” option.
- Calculation of thickness is given below in table 5.1.

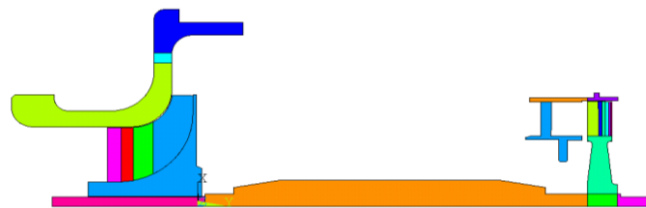


Figure 5.1 Axisymmetric model considered for the analysis.

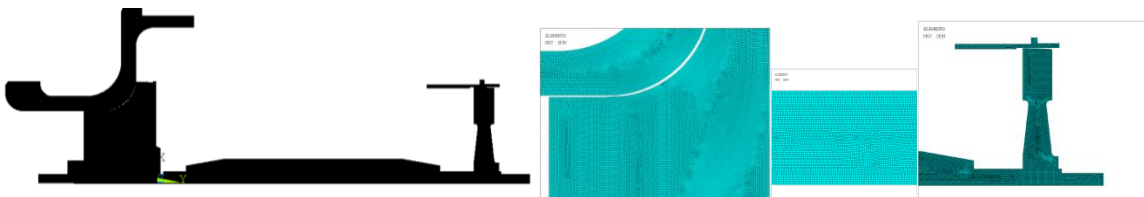
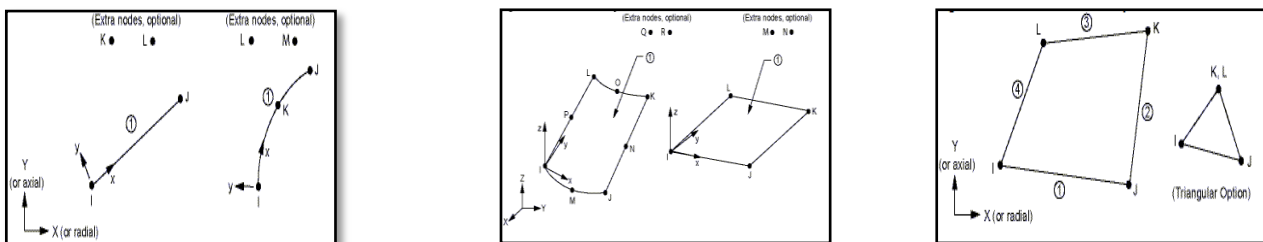


Figure 5.2 Finite Element Model which is Finely Meshed in ANSYS

5.3 Elements used for analysis.



5.3.1 Plane55 Element 5.3.2 SURF151 Element: 5.3.3 SURF152 Element:

PLANE55 can be used as a plane element or as an axisymmetric ring element with a 2-D thermal conduction capability SURF151 may be used for various load and surface effect applications. It may be overlaid onto a face of any 2-D thermal solid element (like plane55).

SURF152 will be used for various load and surface effect applications like convection & radiation loads.

5.4 Engine parameters:

- Out of many engine parameters, temperature, pressure, mass flow rate at the outlet of each component & shaft speed (N1) are considered as primary variables.
- Below are the corresponding variables with respect to time.

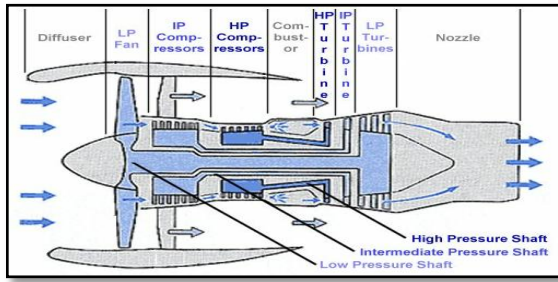


Figure. 5.4 Schematic of Micro gas Turbine

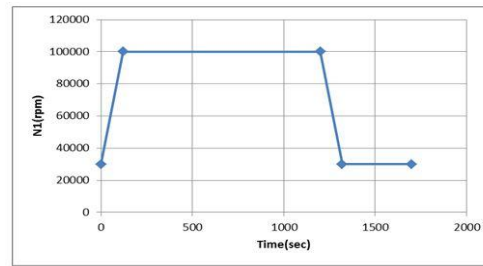


Figure. 6.5 Accel – Deccel Mission (N₁ Vs Time)

Where, N₁= Speed of shaft

- An accel-decel mission runs mainly based on acceleration & deceleration of high-shaft speed.
- Accel time is the time taken to ramp up the rpm of high shaft from idle to T/O. Similarly, decel time can be defined.

Here, ramp up/down time=120sec.

In other words, this is the time taken for (N₁)_{idle} to (N₁)_{T/O}.

Both accel & decel ramp time can be same or different. By observing GE and Pratt & Whitney’s traditional GT engine mission cycles, as an initial estimate, it is decided to consider a 120sec of ramp up & ramp down timing. So, based on this, speed of the shaft is going to increase & decrease based on accel & decel. Refer Figure 6.5 Air temperatures & mass flow rate into the system will also change in line with shaft speed. As per the design calculations, Max temperature of air at T/O is 88degC. Refer Figure 6.6 for temperature variation over the mission cycle. Similarly, refer Figure 6.7 for flow rate variation across the mission cycle.

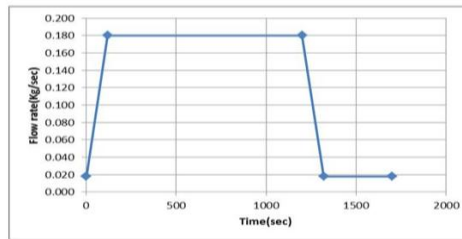
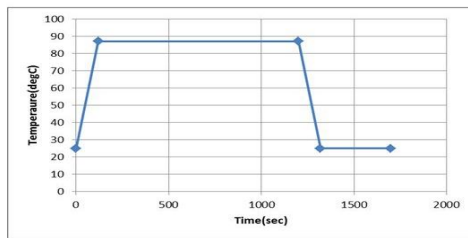


Figure. 6.6 Accel – Deccel Mission (AirTemp_{Compressor} Vs Time) Fig. 6.7 Accel – Deccel Mission Air Flow Rate_{Compressor} Vs Time

5.4 As blade’s thickness varies along its length, entire blade is divided into three parts.

Thickness Calculation for Part 1	<ul style="list-style-type: none"> • No of blades = 16 • Average Thickness of Blade Section = 0.85 mm • Thickness of Each Element = $16 \times 0.85 \times 10^{-3} = 0.013$
Thickness Calculation for Part 2	<ul style="list-style-type: none"> • No of blades = 16 • Average Thickness of Blade Section = 0.934 mm • Thickness of Each Element = $16 \times 0.934 \times 10^{-3} = 0.014$
Thickness Calculation for Part 3	<ul style="list-style-type: none"> • No of blades = 16 • Average Thickness of Blade Section = 1.024 mm • Thickness of Each Element = $16 \times 0.001 = 0.016$

Table. 5.1 Thickness Calculation

5.6 Calculating Real Constants for Compressor Blades

- 8big blades & 8small blades are observed.

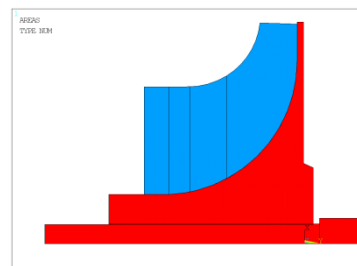
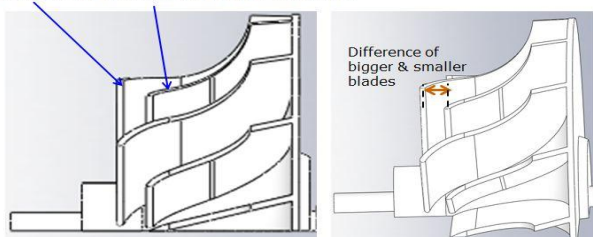


Figure.6.8 An equivalent area of blades is taken in axisymmetric plane.

Figure.6.9 Element shape in Ansys.

While creating the geometry of blades in 2D axisymmetric plane, it is systematically divided into multiple parts (areas) for increasing the accuracy. Using multiple areas, variable thickness can be input to system which can capture the actual variable thickness blades.

5.8 Thickness calculation for small blade of compressor.

- No of blades=8
- Average thickness of blades section=1mm
- Thickness of each element=8*0.001=0.008
- Good discipline is followed in creation of real constants & its values.
- A macro is written to create real set name equal to its element number.
- Thickness is given at 1st location of real set.
- No of blades & each blade thickness are provided at 9th& 10th location of real set for future reference.

5.9 Calculating Real Constants for Compressor Blades:

As blade’s thickness varies along its length, entire blade is divided into three parts. Below table gives the corresponding thickness calculation. After assigning all required real constants for compressor blades, effect of these real constants can be visualized by using element shape options (/eshape). Figure 6.10 for the comparison of element shape on and off. By this, it can be understood that varying thickness blades were captured in a very accurate manner. Similar methodology is used for modelling turbine blades also.

5.10 Element Shape in ANSYS after Assigning Real Constants for turbine.

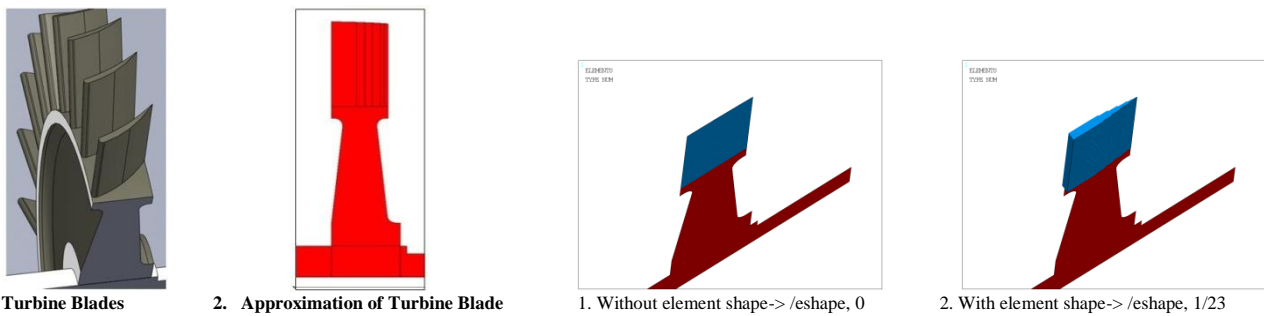


Figure. 6.10 Blade’s element shape with on & off

In case of turbine blades, to capture its thickness effectively, turbine blade is divided into 6 areas. So, six thicknesses are calculated & fed in the form of real constants. This calculational procedure is similar to of compressor blades. Figure 6.10 shows the blade’s element shape with on & off. It can observe that bladed thickness is captured in a right manner.

- Element’s shape can be visualized based on its thickness.
- Variable thickness in compressor blades can be visualized.
- A total of 23 turbine blades were observed. Figure-6.9.1 shows the variable blade thickness in solid works.
- To consider the variable thickness, in ANSYS, total blade area is modeled as 5 areas with variable blade thickness.
- Figure 6.10.1 & 6.10.2 shows the turbine blade geometry with “element shape” off & on.
- Same macro mentioned above is used to assign real constants.

5.12 Definition of Thermal Zones in Compressor

Zone	Correlation	Description
1	Flat Plate	OD of the shaft
2	Rdisk	LE of compressor
3	Flat Plate	Compressor slot
4	Flat Plate	Compressor slot
5	Rdisk	Compressor slot
6	Flat Plate	OD of compressor
7	General	Ambient

8	Flat Plate	ID of casing
9	Flat Plate	ID of casing
10	Rdisk	Cuter ID of casing
11	Flat Plate	ID of casing
103	Flat Plate	Zone for compressor blade
203	Flat Plate	Zone for compressor blade
104	Flat Plate	Zone for compressor blade
105	Flat Plate	Zone for compressor blade

Table 5.2 Thermal Zones based on Geometry

Given geometry is divided into several thermal zones based on geometry, flow variation:

5.13 Visualization of Convection on Zone 3, 4, 5 & Zone 103, 104, 105, 106:

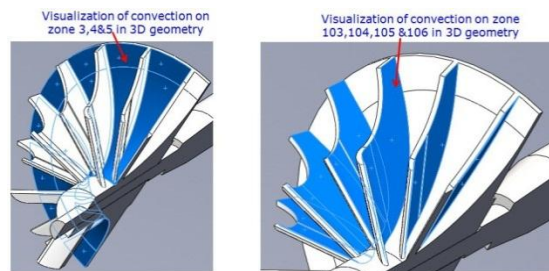


Figure.6.12 Visualization of Convection zone in Compressor

- Always HTC of surf152 zones (z103-106) will be applied with scaled HTC of surf151 zones (zone3, 4&5).
- This scale factor is based on no of blades.

6. RESULTS AND DISCUSSIONS

6.1 Thermal Analysis Results:

By running a non-linear transient thermal analysis for the given mission cycle, thermal scenario of components at different time points can be obtained. Below are the temperature contours at several time points. It is clearly visible that during ramp up (Accel), heating phenomenon of compressor & turbine can be observed. Effect of increased air temperature in T/O will last for another 350sec, which backs up the fact that metal will take sufficient time to heat up. Similarly, in decel, lag b/w air & metal temperature can be observed. This detailed analysis predicts temperature scenario which can be used for structural analysis & in further fatigue calculation.

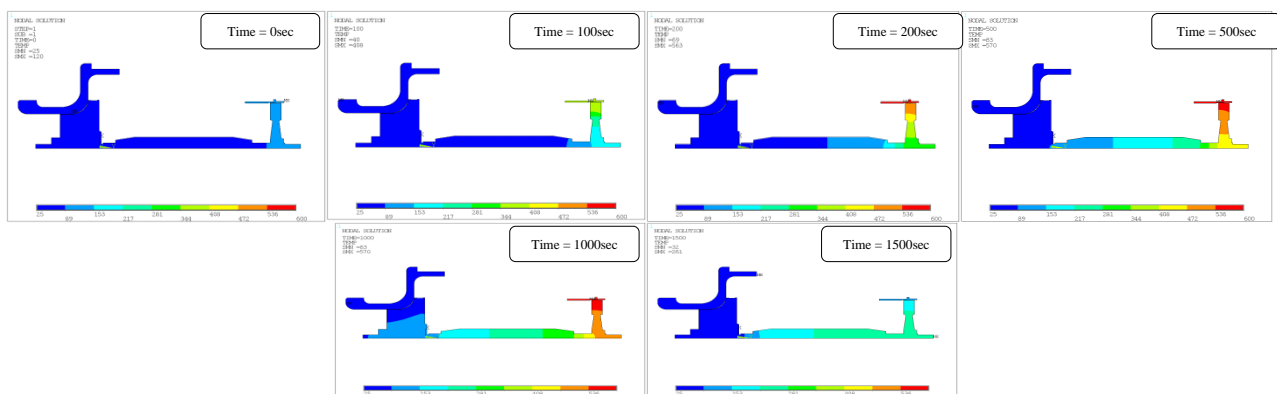


Figure.6.13 Variation of Temp along Time sec.

6.2 Structural Analysis.

6.2.1 Modelling of Structural Database.

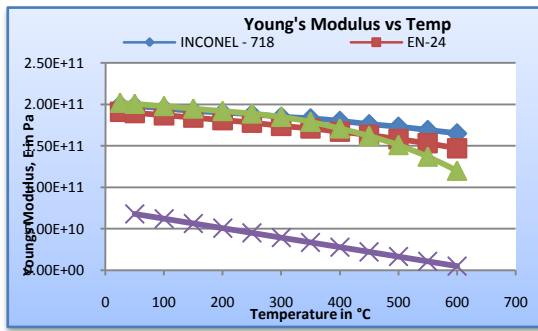


Figure.6.14 Plot Showing Young's Modulus vs. TEMP

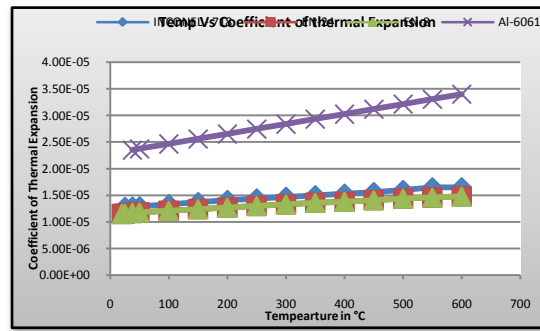


Figure.6.14 Plot Showing Temp vs. Coefficient of Thermal Expansion

6.2.2 Creation of Structural data base.

For generating structural database, current thermal db. can be used. Using the option "ETCHG" shown in Figure. 6.15, a thermal database can be converted to structural database. This option will convert all thermal elements to corresponding structural elements. It is important and mandatory to modify the key options for all the elements based on their behavior (axisymmetric/non-axisymmetric).

Current thermal model will be converted to equivalent structural model & can be used for structural analysis

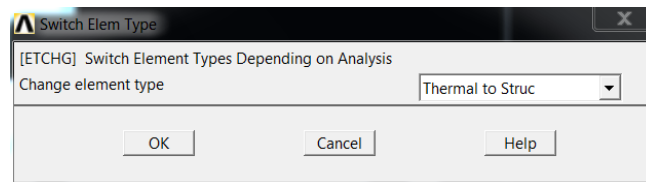


Figure 6.15 Element type switching from thermal to structural

6.2.3 Structural mass approximation

Based on the axisymmetric & non-axisymmetric features, structural mass will be approximated. As thicknesses & real constants were calculated & assigned in thermal analysis, this step is no longer required. All features are approximated in a right manner.

6.2.4 Structural constraints

To avoid rigid body motion in the component, model has been constrained. Axial direction is constrained for a single node in shaft, compressor & turbine casings.

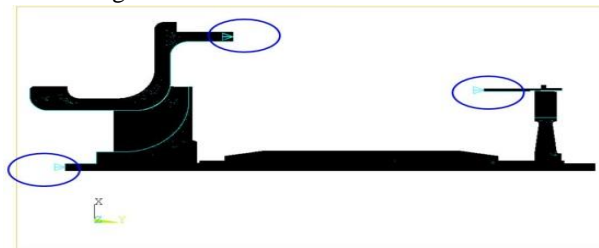


Figure.6.16 Structural Constraints in Axial Direction

6.2.5 Preparation of Load Case files for structural run

To run structural analysis at multiple time points, a macro is written to generate "*.s" files. All structural loads (pressure, angular velocity for rotating loads) were applied using this macro. Also, all thermal loads (from thermal analysis) were applied using "ldread" command. These thermal loads will be applied as body forces in the model.

6.2.6 Von-Mises stress at different time points:

Structural simulation is performed at 55 different time points. These time points cover all the important time points in entire Accel & decel mission. Von-mises stress for all time points is checked and time points of max von-mises stress for different components will be considered. Figure.6.17 gives the von-mises contours at mission pts.

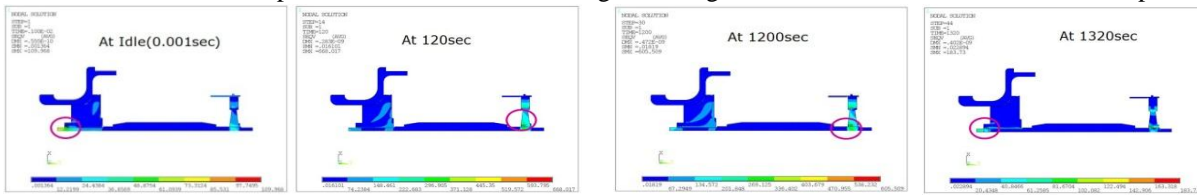


Figure. 6.17 Von Mises Stress at different time points.

6.3 Factors Influencing Life Calculation.

6.3.1 Effect of stress concentration in life calculation:

- As stress is taken from FE, it is with Kt effect captured. So un-notched curve (kt=1), & life can be evaluated directly from FE stress (conservative approach).

6.3.2 Effect of Temperature:-

- As temperature increases, allowable stress will come down. It needs to be accounted using certain de-rate factor when we are using fatigue curve at room temperature. Empirical relationship for De-rate factor can be arrived at from fatigue tests.

6.3.3 Effect of stress ratio:-

- Effect of stress ratio is captured in the equivalent stress equation.

6.4 Life Calculation at Max Stress Location of Turbine:

To calculate the fatigue life for max stress location of turbine, it is required to identify the max stress in the component. For turbine, max stress of 582Mpa is observed at 169sec. This stress is primarily because of higher temperature and rotational speeds. Figure 6.17 give detailed von-mises contour plots at 169sec. Figure 6.18 gives time vs. temperature and time vs. stress for the max stress location. It can be observed that temperature at 169sec is not the max temperature of system. The max stress is primarily because of thermal gradient across the component and differential thermal expansion in the system. It is important to note that after this, stress is in decreasing trend till the end of decel. To calculate fatigue life, it is required to estimate stress range rather than max stress of ANSYS. Even though, there is a residual stress observed at 0sec, always it is customary to consider min stress in the component is 0MPa. So, max stress from ANSYS can be regarded as stress range. Fatigue curves from FAA were used to estimate the fatigue life. It is observed that these curves are in FPS units. So, stress in MPA will be converted to ksi and life will be estimated. Below is the calculation for estimating life of the turbine.

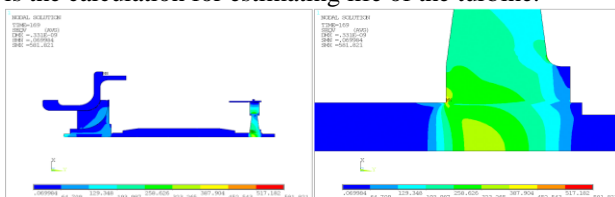


Figure.6.18 Variation of stress at different time.

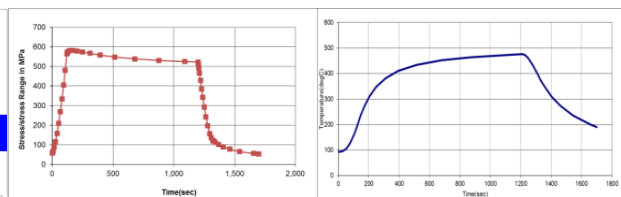


Figure6.19 Time vs. Max Von-mises Stress at turbine:

Equivalent Stress Equation

$$\log N_f = 24.49 - 9.62 \log (S_{eq}) \quad (6.1)$$

$$S_{eq} = S_{max} (1 - R)^{0.42}$$

$$\text{Std. Error of Estimate, } \log (\text{Life}) = 22.71 \left(\frac{1}{S_{eq}} \right)$$

$$\text{Standard Deviation, } \log (\text{Life}) = 0.985$$

$R^2 = 90\%$ Where, $N_f \rightarrow$ No of life cycles & $S_{max} =$ Max stress range from ANSYS & $S_{eq} =$ Equivalent alternating stress As $R=0$, $S_{eq} = S_{max} = 581.8 \text{MPa}$ (at 169sec) Temperature at 169sec = 269degC Assuming thermal knockdown of 0.8, S_{eq} after thermal knockdown = $582/0.8 = 727 \text{MPa}$ (105ksi) Based on the above formula, $N_f = 10^{(24.49 - 9.62 * \log (Seq))} = 106452 \text{cycles}$

In decel, similar phenomenon will occur in reverse manner. In overall, the trend of curve given in Figure 6.23 is sensible. Similar explanation applies to turbine as well. Refer Figure.6.24 for the clearance of turbine. A Macro is written to calculate the clearance in turbine and compressor. Below are the clearance curves for both compressor and turbines.

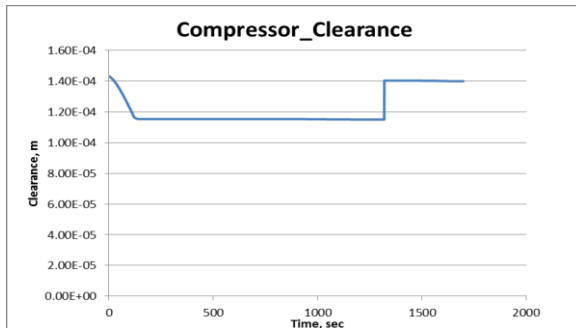


Figure. 6.23 Compressor Clearance with respect to time.

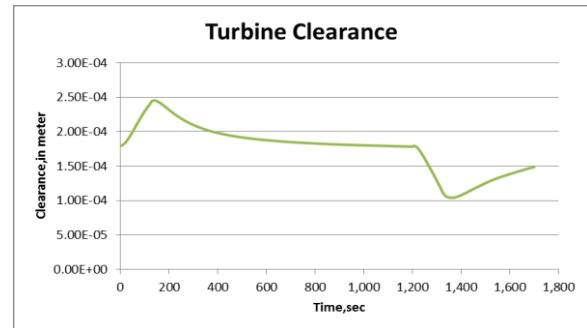


Figure. 6.24 Turbine Clearance with respect to time

CONCLUSION

- Mass approximations were carried out using real constants.
- Heat transfer coefficients are calculated using appropriate correlations.
- Non-linear transient thermal analysis is carried out for the 120sec Accel-decel mission.
- Structural analysis is carried out for 55 transient time points.
- Fatigue life calculation is performed at several locations in the assembly.
- Critical locations like fillets in turbine & turbine shaft are come under the “infinite life” category because stress at this location is less than 60ksi.
- All clearances are within the tolerance & any rotor will not touch corresponding stator component.
- Maximum estimated fatigue life at the turbine is 1, 06,234 cycles.
- Maximum Von mises Stress at 1200 sec is 320Mpa in compressor.
- Maximum Vonmises stress at the turbine part is 581Mpa at 169 sec.
- With respect to figure 6.21 fatigue life at 3 locations has been calculated, Life at shaft ID will be 1e8 cycles, Life at fillet 1 will be 1.02e8 cycles Life at fillet2 will be 3.90e7 cycles

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