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CFD, Structural and Vibration Analysis of a Steam Turbine Rotor Blade

Ashwani Thakur
ME (Mechanical Engineering)
PEC University of technology
Chandigarh
thakur.ashwani317@gmail.com

Arun K Lall
Professor
PEC University of technology
Chandigarh
lallkarun@gmail.com

Sandeep Sharma
Technical Head
Aerosphere
Chandigarh
sandeeps.aero@gmail.com

ABSTRACT
In steam turbines, the thermal energy of steam is converted into mechanical energy by impinging high velocity steam onto rotor blades and thus changing its momentum. In present work a steam turbine rotor blade is used that has been in operation for many years. The performance of the rotor blade is analyzed to know whether it meets the design requirements or not. Analysis is done using the methods of computational fluid dynamics and structural analysis, because of availability of computer powers and efficient numerical algorithms and to avoid costly experimental setups. In present work, the effects of thermal loads, centrifugal and steam impact loads on the blade are analyzed in terms of stresses and deformations produced. The temperature distribution for calculating the thermal loads is obtained by doing conjugate heat transfer analysis of the blade using CFD in ANSYS CFX 15.0 software. The structural loads i.e. centrifugal and steam impact loads are calculated analytically and applied on the model in static structural module of ANSYS 15.0 software and stresses and deformations are noted. Also to check the blade failure due to excessive resonance response the modal analysis of the blades is done to get different mode shapes and corresponding deformations at various operational speeds. Campbell diagram is also made at various excitation frequencies to calculate the critical speeds. The results showed satisfactory results except at few root edges which has excessive values of stress and need to be redesigned. The vibration results are also satisfactory as there is no critical speed at synchronous excitation speeds however critical speeds exists for higher level of excitation frequencies.

Keywords
Steam turbine rotor blades, CFD, ANSYS-CFX, Conjugate heat transfer, Centrifugal force, Axial force and Tangential forces, Resonance, Campbell diagram, Critical speed, ANSYS 15.0.

1. INTRODUCTION
A steam turbine is one of the most versatile and oldest technologies used in power generation because of its higher efficiencies and lower costs. The capacity of the steam turbine can range from as small as 0.75 KW to as large as 1500000 KW power. The steam turbine works by impinging the pressurized steam on the rotor blades that further rotates the shaft. Thus the rotor blades are the heart of the steam turbine and its shape or profile greatly affects the output, efficiency and usability of the steam turbine. They are the most critical component of the steam turbine in which the failure occurs frequently.

The performance of the steam turbine over a period of time depends on the parameters like blade geometry, blade material, operating conditions and surface finishing of the blade, etc. Blade design is a multi-disciplinary task. It involves the thermodynamic, aerodynamic, mechanical and material science disciplines. The major cause of break down in turbo machine is the failure of rotor blade. The failure of the rotor blade may lead to catastrophic consequences both physically and economically. During the service life, the rotor blades undergo many degradations and deviation from its designed profile.

The major challenge in designing rotor blades is its adverse operating environment that includes high steam temperature and high rotational velocities of the steam turbines that produces thermal and structural stresses which further leads to creep and fatigue phenomenon and finally failure of the blades. To withstand these conditions, steam turbine blades are made from super alloys now days for service at high temperatures. Super alloys consist of nickel-base super alloys, cobalt-base super alloys and iron base super alloys. The steam turbine blades are principally made of Nickel-base superalloys. The excellent thermal stability, tensile and fatigue strengths, resistance to creep and hot corrosion, and micro structural stability possessed by Nickel-base superalloys render the material an optimum choice for application in turbine blades.

Turbine blade failure also occurs due to excessive resonant response. Resonance occurs when the excitation frequency matches the natural frequency of the blade and the amplitude of vibration becomes very high. The most common excitation source is the non-uniform flow field generated by inlet distortion, wakes and/or pressure disturbances from adjacent blade rows. The natural frequencies and deformations at various modes shapes can be calculated at various rotational speeds of the blade. Campbell diagram is drawn to get the values of critical speeds at various excitation frequencies.

2. MATERIAL PROPERTIES AND MODELING
The steam turbine rotor blade belongs to an impulse reaction turbine. The rotor blade has been in service for many years.

2.1 Material Properties
The rotor blade is made of a nickel super alloy Inconel 718. The material properties of INCONEL 718 are given in Table 1.

Table 1: INCONEL 718 properties

Density (Kg/m^3)	8220.00
Melting point (K)	1609.15
Modulus of elasticity (KN/mm^2)	204.90
Modulus of rigidity (KN/mm^2)	77.20
Yield strength (N/mm^2)	1172.00
Tensile strength (N/mm^2)	1407.00
Coefficient of thermal expansion(/°C)	$12.8 * 10^{-6}$
Poisson's ratio	0.30
Thermal conductivity (W/m.K)	25.00
Specific heat (J/Kg.K)	586.20

The properties of steam are given in Table 2.

Table 2: Steam properties

Specific heat capacity (J/Kg.K)	2180.980
Reference temperature (K)	698.150
Reference pressure (bar)	15.000
Dynamic viscosity (Pa.s)	$25.0746 * 10^{-6}$
Thermal conductivity (W/m.K)	0.058

2.2 Modeling

The modeling of the rotor blade is done by reverse engineering using a white light scanner and Geomagic design X software. The solid model is shown in Figure 1.

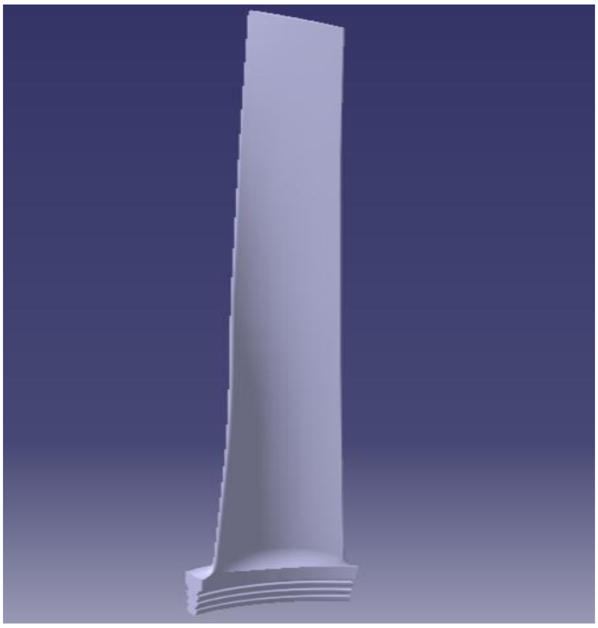

Figure 1: Solid model of rotor blade

3. FINITE ELEMENT ANALYSIS

3.1 Computational Fluid Dynamics

A conjugate heat transfer analysis is performed in ANSYS CFX software to get the temperature distribution by solving heat convection equations in fluid domain (Navier stroke solver) and heat conduction equation in solid domain (Finite element solver).
A fluid domain is defined around the solid blade to define the flow of steam. The Fluid domain consists of shroud, hub, inlet, outlet and periodic boundaries to define the operating conditions accurately. The solid domain consists of blade model.

3.1.1 Modeling

The rotor blade solid model is imported in CATIA V5 R 19 software and the fluid domain is also modeled in same software. The solid and fluid domains are shown in Figure 2.

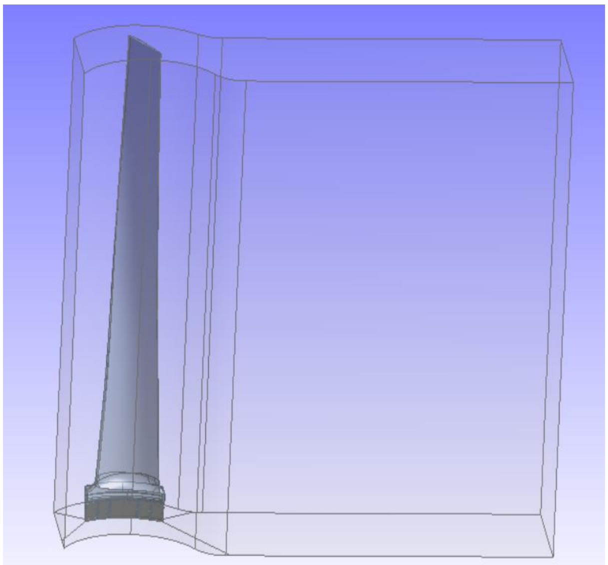

Figure 2: Solid and fluid domain

3.1.2 Meshing

Meshing of both solid and fluid domain is done in ANSYS 15.0 CFX module only. An element size of 8mm is used for blade and 12 mm for the fluid domain. The meshed solid and fluid domains are shown in Figure 3.

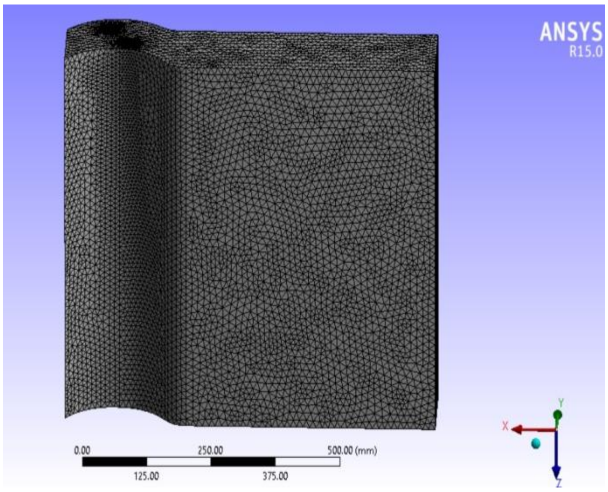

Figure 3: Meshed solid and fluid domains

3.1.3 Operating conditions:

Solid domain:
Material: Inconel 718
Domain: Stationary
Heat transfer: Thermal energy
Initial temperature: 298.15 K

Fluid domain:
Material: Dry steam
Domain: Stationary
Heat transfer: Total energy
Turbulence model: K-Epsilon

Inlet:
Total pressure: 17.64 bar
Total Inlet temperature: 698.15 K
Flow direction: Normal to inlet boundary

Outlet:
Static pressure: 14.52 bar
Hub, shroud and periodic walls: Smooth adiabatic walls with no slip condition
Heat transfer between solid and fluid domain: Conservative interface flux method.
Max. no of iterations: 100
Residual RMS value: 0.00055

Solver controls:
Advection scheme: High resolution
Min. no of iterations: 1
Max. no of iterations: 100
Residual RMS value: 0.00055
Time scale control: Automatic
The solution is performed in parallel run mode and standard run priority.
The solid and fluid domains with boundary conditions are shown in Figure 4.

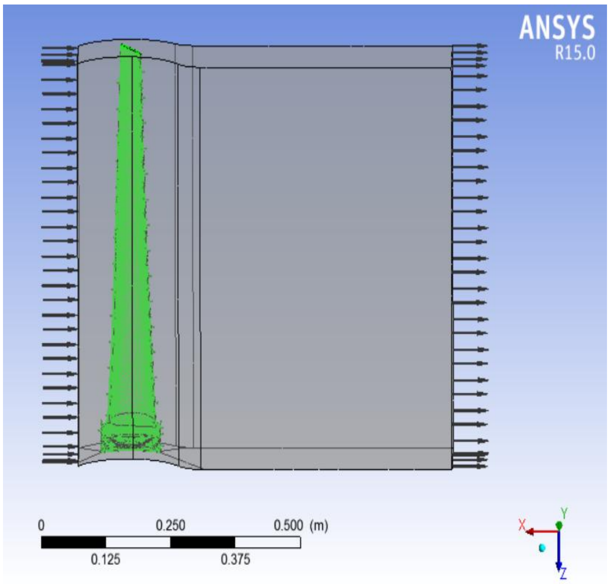

Figure 4: Solid and Fluid domains with boundary conditions

3.1.4 Results and discussion

In CFD post, the temperature distribution of the rotor blade is obtained as shown in Figure 5.

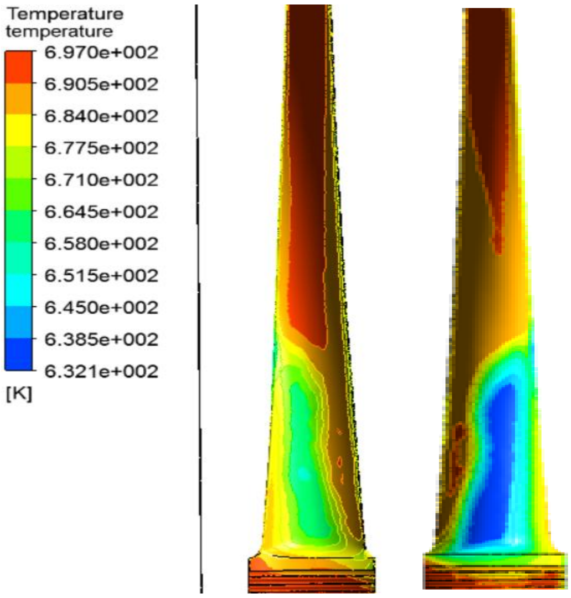

Figure 5: Temperature distribution along the rotor blade length

The temperature of the blade ranges from maximum of 697 K to a minimum of 632 K.
Since the blade is curved long its length, so the steam strikes different sections of the blade at different time, so there is different temperature values along the blade periphery. The maximum temperature occurs at the root and at the blade tip. The temperature distribution is required to calculate the thermal stress on the blade.

3.2 Structural Analysis

The structural analysis of the blade is done in ANSYS static structural module. The geometry and solution are imported from ANSYS CFX module. The fluid model is suppressed and solid blade model is meshed again.

3.2.1 Meshing

The meshing of blade is done in ANSYS 15.0. The meshed blade is shown in Figure 6.
Number of nodes: 60583
Number of elements: 34480

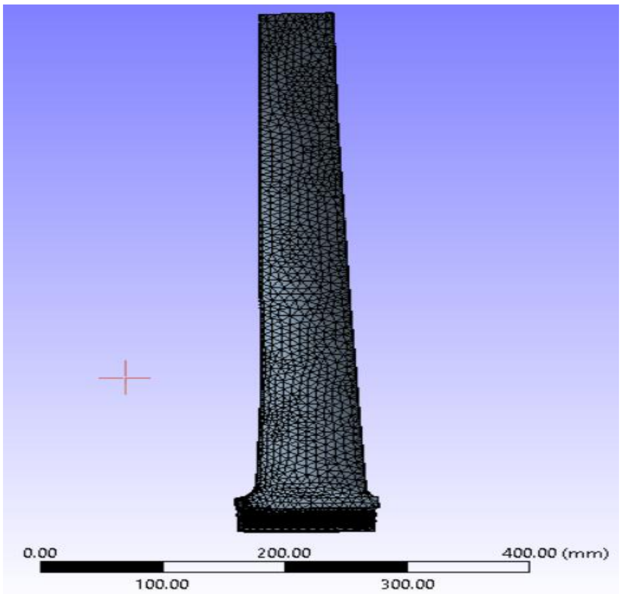

Figure 6: Blade meshing in ANSYS

3.2.2 Boundary conditions and loads

Boundary conditions:
Root portion of the blade is fixed
Blade RPM: 3000

Loads:
Thermal load: Imported from ANSYS CFX
Centrifugal load: 205726 N
Axial load: 54.90 N
Tangential load: 282.11 N

The centrifugal, axial and tangential loads are calculated from operating conditions, blade dimensions and inlet and outlet velocity triangles. The rotor blade specifications and operating conditions are given in Table 3.

Analysis settings:
No. of steps: 10
Step end time: 1 second
Time stepping: Automatic
Solver: Program controlled

Table 3: Blade specifications and operating conditions

Mean radius of blade centroid (mm)	700.00
Mean diameter of blade centroid (mm)	1400.00
RPM	3000.00
Blade mass (Kg)	2.98
Mean speed (m / s)	219.94
Inlet velocity (m / s)	480.00
Steam entrance angle (in degree)	23.85
Blade outlet angle (in degree)	20.40
Steam mass flow rate (Kg/s)	0.58
Friction coefficient	0.90

3.2.3 Results and discussions

Applying above given thermal, centrifugal, axial and tangential loads, the stress and deformation values are determined for the rotor blade in the solution section of static structural module. The stress values are given in Figure 7 and total deformation values in Figure 8.
The stress varies from a maximum value of 6652.2 MPa to a minimum value of 0.18508 MPa. Overall the stresses in the blade are below the yield point except at few portions of the root edges of the blade. Thus if this blade is to be reused the root section has to be redesigned or the blade has to be replaced.
The total deformation under this loading varies from maximum value of 22.205 mm to minimum value of 0 mm. The deformation results are as expected, maximum at the tip and zero at the root portion where the blade is fixed.

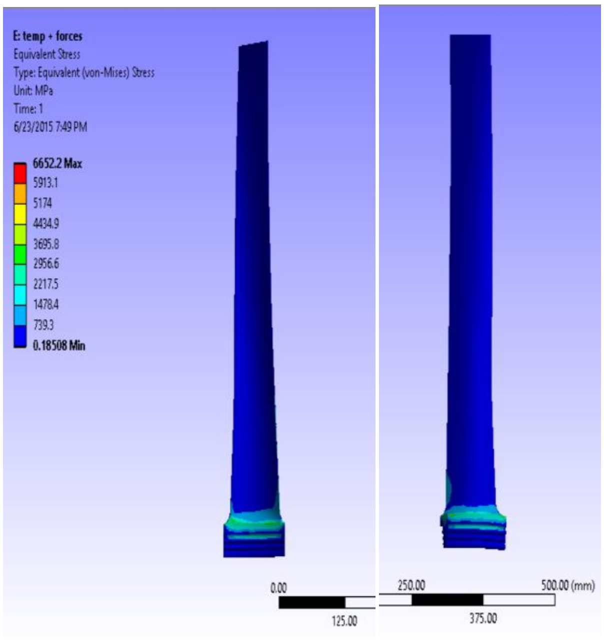

Figure 7: Von-Mises Stress distribution along the blade

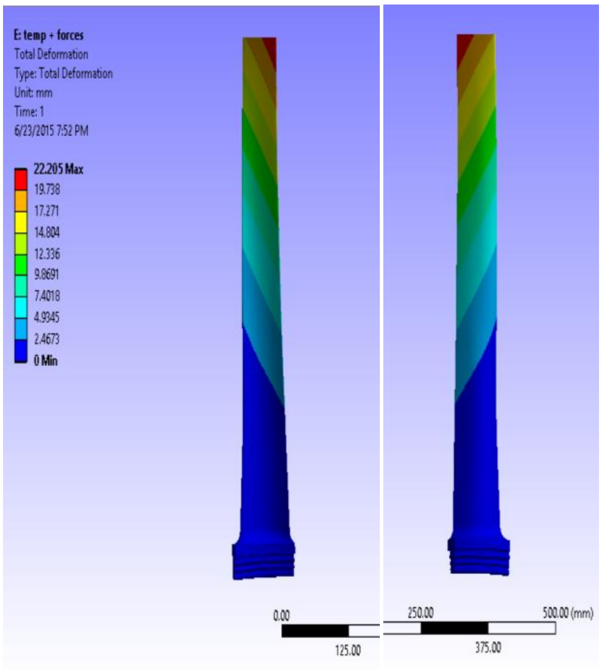

Figure 8: Deformation distribution along the blade

3.3 Vibration Analysis

A pre-stressed or forced vibration analysis of the rotor blade is performed in the modal analysis module of ANSYS 15.0. For pre-stressing the solution results of previous static structural analysis are imported into the setup of modal analysis.
The thermal, axial and tangential load remains the same but the centrifugal force changes for different rotational speeds of the turbine. The rotational speeds of the turbine used were 0, 10, 20, 30, 40 and 50 Hertz. Corresponding to each value of rotational speed or centrifugal force and other forces, the mode shapes, natural frequencies and deformations are calculated. The values of Centrifugal, axial and tangential forces at different values of operational speeds are given in Table 4.

Table 4: Centrifugal, axial and tangential force at different operational speeds

Operational Speed (Hz)	Centrifugal force (N)	Axial force (N)	Tangential force (N)
0	0.00	54.90	282.11
10	8229.04	54.90	282.11
20	32916.16	54.90	282.11
30	74061.37	54.90	282.11
40	131664.70	54.90	282.11
50	205726.00	54.90	282.11

The natural frequencies at different rotational speeds are given in Table 5 and the total deformations in Table 6.

Table 5: Natural frequencies at different rotational speeds

NATURAL FREQUENCY (Hz)						
	OPERATIONAL SPEED (Hz)					
	0	10	20	30	40	50
MODE 1	52.18	55.14	63.06	74.13	86.93	100.65
MODE 2	123.55	125.84	132.37	142.25	154.39	167.86
MODE 3	286.00	288.18	294.54	304.47	316.81	329.47
MODE 4	331.50	332.22	334.41	338.30	344.51	354.54
MODE 5	437.27	439.97	448.00	461.07	478.73	500.42
MODE 6	680.93	683.19	689.88	700.66	715.08	732.64

Table 6: Deformations at different rotational speeds

DEFORMATION (mm)						
	ROTATIONAL SPEED (Hz)					
	0	10	20	30	40	50
MODE 1	61.34	60.90	59.68	58.01	56.20	54.46
MODE 2	67.40	67.54	67.78	67.92	67.76	67.24
MODE 3	74.81	75.57	77.93	82.20	88.47	94.15
MODE 4	105.46	105.70	106.42	107.51	107.90	103.60
MODE 5	68.24	68.50	69.24	70.37	71.75	73.29
MODE 6	85.50	85.65	86.06	86.71	87.55	88.54

The natural frequencies and total deformations at different modes at operational speed of 50 HZ are shown in Figure 9.
Finally Campbell diagram is drawn to identify the critical speeds of the rotor blade at which resonance can occur. Critical speed occurs when the excitation frequency coincides with the natural frequency of the blade.
The Campbell diagram at various excitation frequencies are plotted as shown in Figure 10.
The results obtained from the Campbell diagram shows the value of critical speed at various excitation frequencies as shown in Table 7.

Table 7: Campbell diagram results

EXCITATION	CRITICAL SPEED (Hz)
1 X	No
2 X	50.0
3 X	22.0
4 X	14.5 and 37.0
5 X	11.5 and 27.5

4. CONCLUSIONS

In the present paper the heat transfer and structural strength of the rotor blade is studied using conjugate heat transfer analysis and finite element analysis applying centrifugal and impact loads along with the vibration analysis of the blade. The main conclusions obtained are as below:

1. Conjugate heat transfer analysis is carried out to simulate the heat transfer between the rotor blade and steam to obtain the temperature distribution of the rotor blade making use of CATIA V5 R19 for solid modeling and ANSYS CFX 15.0 for analysis.
2. The overall stress obtained along the blade is well below the yield strength of the blade except at few root edges, where it is very high. This is due to high value of thermal stress. Thus thermal stress along with stress due to centrifugal force and steam impact force gives more valuable information about the blade failure.
3. Based on the stress results, the rotor blade shows failure at root portion and thus needs to be re-designed at root portion to meet the design requirements.
4. The results of vibration analysis and Campbell diagram are satisfactory, as there is no critical speed at synchronous excitation speeds, however critical speeds exist for higher order excitations.

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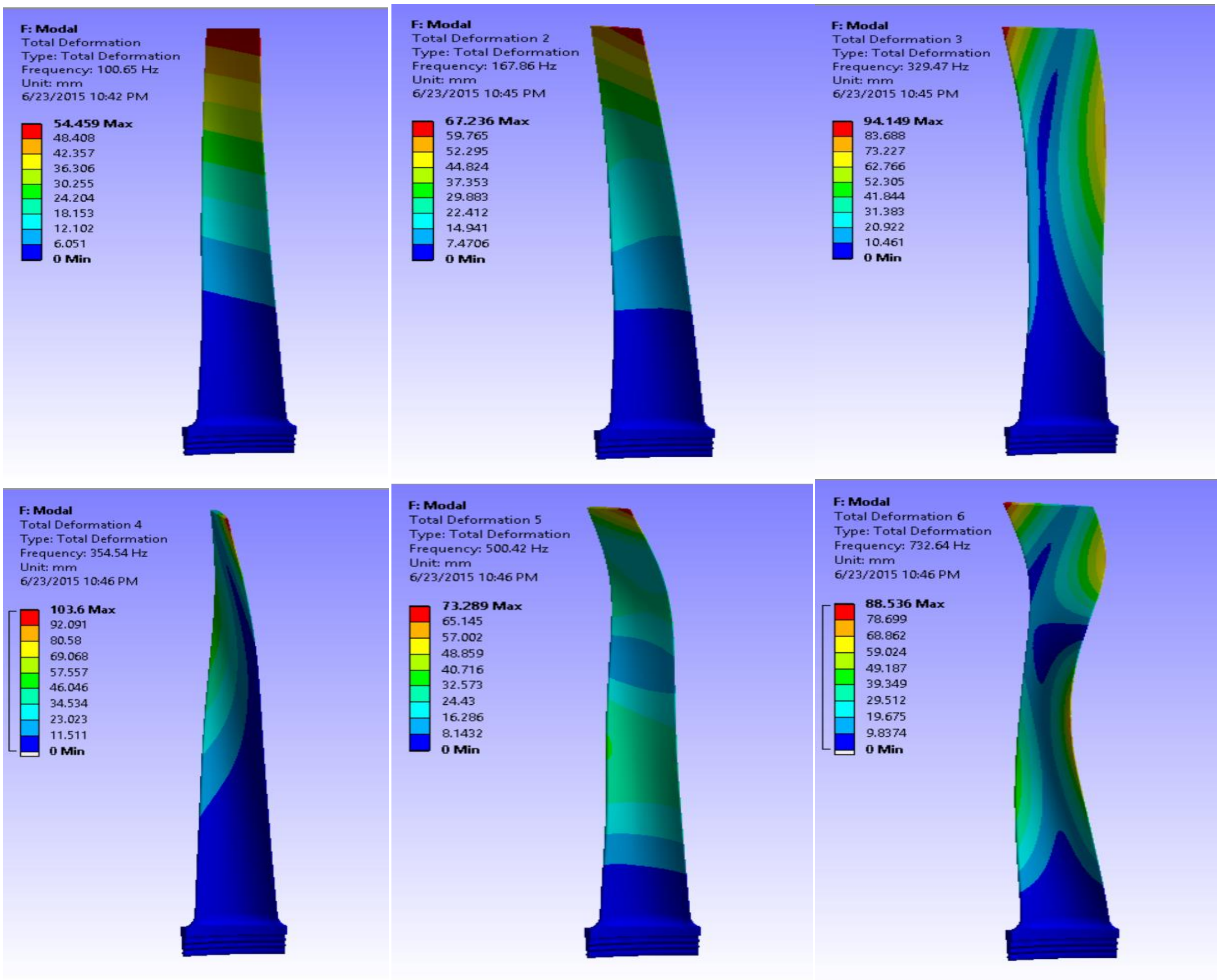

Figure 9: Natural frequencies, Deformations at various mode shapes at operational speed of 50 Hz

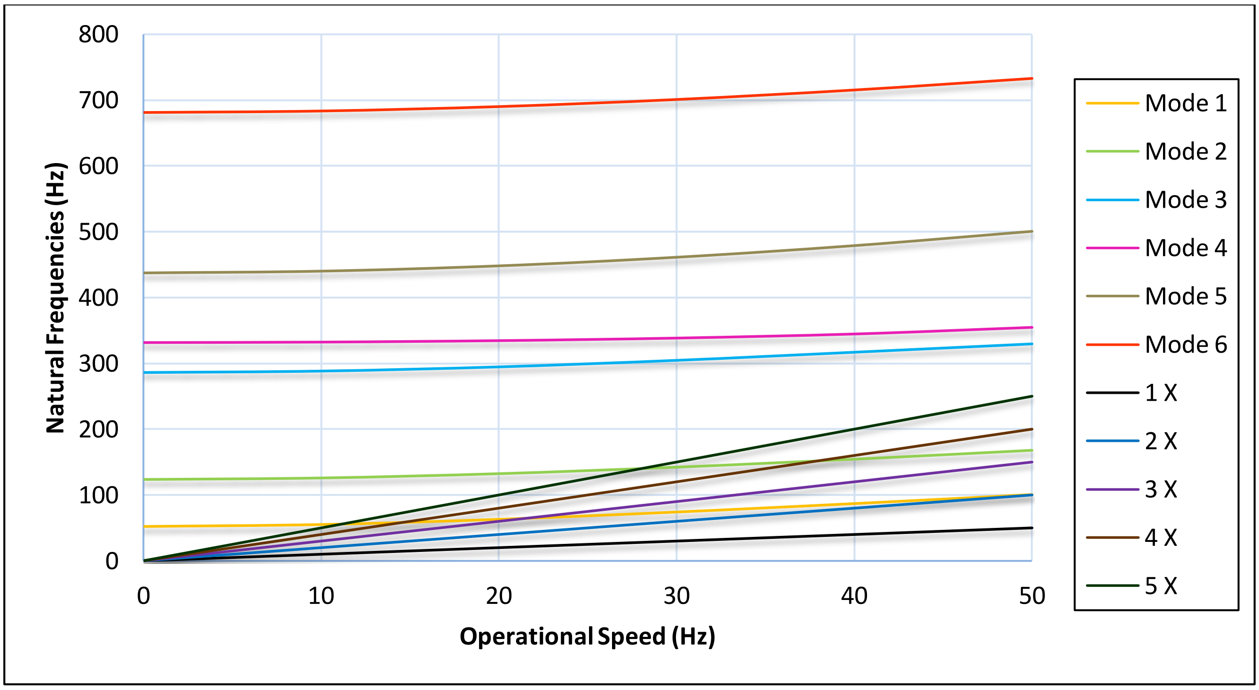

Figure 10: Campbell diagram

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