# EVALUATION OF THE TURBINE BLADED DISC DESIGN FOR RESONANT VIBRATION EXCITATION

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**Abstract -** *The project work is to study the vibration behavior of a bladed disc system, understand independent vibration of the blades, also to understand the criticality in assessing the structural integrity of a bladed disc system and further make an attempt to arrive at the integrated bladed rotor system which in recent years is the state of art in science and technology. The present work comprises of two parts mainly to establish a customized methodology or step by step procedure followed in the assessment of vibration characteristic of a turbine rotor system and extend the same procedure of assessment to the proposed integrated bladed rotor system to clearly distinguish the behavior of blades independently, in combination with bladed disc and as an integral. The study involves blending of mathematical models in combination with displacement modes to finite element analysis following employing the procedure established by Murari P Singh through Campbell and Singh's Advanced Frequency Evaluation diagram in design of typical turbine bladed disc system. Focusing mainly on the failures pertaining to vibrations adopting commercial finite element tool analysis ANSYS.*

***Key Words***: structural integrity, bladed disc system, integrated bladed rotor system, Campbell diagram, F E Analysis

## 1. INTRODUCTION

For the conversion of energy of flowing fluid into mechanical energy, a rotating turbine is used. Turbine blades failure during operation results in safety risk, non operational revenue loss and repair cost. Thus reliability of turbine blades is a major factor for the successful operation of the turbo machineries. From the point view of possibility of a resonant vibration excitation, the suitability of the bladed disk design can be evaluated on the basis of several approaches. Knowledge of natural vibration characteristics such as its natural frequency and mode shapes is must for applying these approaches. By using the SAFE diagram for the evaluation of the suitability of the bladed disk design most of the information can be found but the possibility of exciting the bladed disk resonant vibration can also be evaluated from the CAMPBELL diagram. Another criterion for the evaluation of design suitability of the bladed disk is

on the basis of critical speed, which is also one of the causes for the breakdown of relatively flexible disk. It's been found that fatigue is the predominant cause for the failure of turbine blades by carrying out metallurgical experiments; fatigue failure is caused by fluctuating forces in combination with steady forces.

Various techniques have been developed at the design stage to prevent excitation of bladed disc modes. Campbell diagram is the most commonly used analysis tool introduced in 1924 by Wilfred Campbell to examine the blade failures due to axial vibrations. Since then it has been found useful in different fields such as rotor dynamics and acoustics.

Interference diagram also called the SAFE diagram i.e. Singh's Advanced frequency evaluation diagram, was introduced by Murari Singh in 1984 and it was developed to examine spatial agreement between mode shapes of bladed disc and excitation patterns. Phase cancellation between excitation patterns and mode shape will prevent excitation of many bladed disc modes even when the excitation frequency is near a natural frequency.

### 1.1 BLADE EXCITATION FORCES
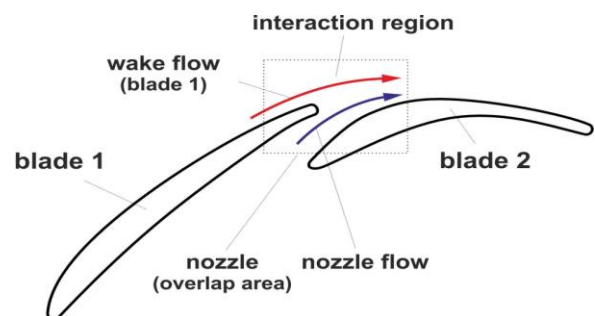


For any discussion about the natural frequency calculation, it is important to know about the excitation forces in the steam turbine. Two major source of excitation which could cause blade vibration leading to failure and both are associated with the flow of steam. Estimation of magnitude of these forces is difficult but the frequencies can be found out precisely.

A row of stationary blades between row of moving rotor blades, attached to the turbine outer casing, these stationary blades are known as nozzles, stator blades or diaphragm blades. The flow of steam causes pressure vibrations called nozzle wakes behind each stationary blade. When the rotor blades passes through the nozzle wakes they experiences series of impulses. The product of the number of stationary blades and rotational speed is the frequency of these impulses. This is called "Nozzle Passing Frequency". Excitation from the nozzle wakes at the blade passing frequency is the most concern mechanism where the blades with high natural frequency. The nozzle passing frequency can be calculated using the following formula –

$$\text{Nozzle Passing Frequency (NPF)} = \text{No. of nozzles } (N_Z) \times \frac{N \text{ in rpm}}{60}$$

Where, N = 100% operating speed of the turbine
Similarly, the Blade Passing Frequency is given by the formula –

$$\text{BPF} = \text{No. of Blades } (N_B) \times \frac{N \text{ in rpm}}{60}$$

Where, N = 100% operating speed of the turbine
Similarly, Running Speed Harmonics can also be calculated using the following formula –

$$\text{RSH} = \frac{N}{60}$$

Where, N = 100% operating speed of the turbine
The value obtained is taken as 'x' and similarly 2x, 3x, 4x....$N_B$x.

Any of the mode shapes with the natural frequency ranging near to any of these sources of excitation will cause resonance. Evaluation of separation is must in this range else over time 'T' the possibility of catastrophe is expected.

## 1.2 MATERIAL PROPERTIES

The material used for turbine blade is 12% chromium steel

| Quality | value | Unit |
|---|---|---|
| Young's modulus | 200000 | Mpa |
| Tensile strength | 650-880 | Mpa |
| elongation | 8-25 | % |
| fatigue | 275 | Mpa |
| Impact strength | 90-95 | J/cm |
| Yield strength | 350-550 | Mpa |

## 1.3 OBJECTIVES

The objectives are to identify the turbine blade failures due to vibrational issues, to understand the vibration behavior of a bladed disc system of a turbine to demark cause and effect, to establish preliminary design criteria for rotor blade design to address vibrational issues, Propose a customized methodology and extend the similar procedure to integrated bladed disc rotor to check the performance compared to conventional bladed disc.

## 2. METHODOLOGY

1. Preliminary design consideration
2. Cyclic symmetric analysis
3. Mesh convergences
4. Displacement Preliminary model formulation
5. Pre-stress modal analysis for
    (a) Independent free standing blade
    (b) Bladed disc assembly
    (c) Integral bladed rotor
6. Stress stiffening
7. Spin softening
8. Campbell diagram
9. Role of friction in bladed disc analysis

## 2.1 PRELIMINARY AND DETAIL DESIGN CONSIDERATIONS

A small exercise was carried out in the present work to understand the sensitivity of geometrical parameters with respect to free standing blade analysis. Assuming the blade to be a cantilever structure, pre-stress modal analysis is carried out to arrive at the fundamental modes such as bending mode, twisting mode, flip mode edge bending mode, etc. Referring to the notable literatures and practical best practices followed during the blade design by revered authors following observations are made.

1. Based on the source of excitation either from running speed harmonics or from stator wanes, the mass flow rate flowing through the aerofoil blade, single cantilever blade analysis is made to understand the modes and corresponding frequencies.
2. Running speed harmonics to be calculated and its associated excitations noted as 1x, 2x, 3x, 4x.....
3. Assuming 100% fixity at the root the free standing analysis provide the first cut results to the designer to life the blade based on its frequencies and number of cycles estimated for the given loading conditions.
4. The best practice followed by most of the designers for rotor blade design is the first fundamental bending mode frequency should be greater than 4x harmonics.
5. The operating range to be considered between 85 to 120% of 100% speed of the rotor for vibration assessment.
6. ± 5% variation to be considered at 100% speed and we should have free flow margin to the left and right. This range of frequency should not have resonance during operating condition specifically on 1rst mode.
7. If at all there is resonance during this range of frequencies for 2nd mode to 6th mode vibratory stress assessment would be recommended to life the blade for high cycle fatigue, in the frequency

range of resonance assuming 0.2% damping due to friction in the bladed disc assembly for the mode which is undergoing at the resonance conditions.

8. The first six modes should have the margin of minimum 10% frequency range from the 1[rst] nozzle passing frequency.

9. If the nozzle passing frequency falls between the 5th or 6th mode frequency then separation margin expected from each independent mode should be 10%.

10. The 2nd nozzle passing frequency will normally be far away from first 6 fundamental modes however a sufficient margin of 20% is expected to avoid resonance.

11. Each mode at operating range will be considered along with running speed harmonics as exciter box and a margin of 10% range frequency is expected to the left and the right margin of the operating speed to assure mechanical integrity. Similarly 5% frequency margin is expected at the top and bottom portion of the exciter box.

## 2.2 CYCLIC SYMMETRY

Turbine blade disks, impellers, fans, pulleys, and flywheels are the machine elements which have cyclic symmetric properties. In these machine elements the geometry and in some cases the boundary conditions also repeats themselves in a cyclic manner. Modeling of any one sector is sufficient is this class of problems, it is not just efficient also effective to consider only one sector for analysis. The displacement constraints applied for one sector model will be identical to those applicable for 360º model. The displacement fields are with respect to the axis of cyclic symmetry in a cylindrical co-ordinate system.

### 2.2.1 Cyclic Symmetry analysis for Integrally Bladed Disc (Blisk)

Blisk or bladed disk or integrally bladed rotors [IBR] is a component used in turbo machinery which consists of both the blade and disk as a single component instead of the usual assembly of the blade and rotor separately. They are usually manufactured by integrally casting, machined from a single solid piece of metal or by welding individual blades to a rotor disk.
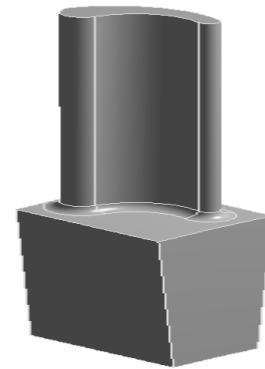


Fig: Blisk model used for the simulation and the Finite Element model

Some of the specifications of the blade and rotor assembly are tabulated in the table below. These are some of the important parameters that determine the geometry of the blade. This again varies depending on what type of blade is being used. For example, the blade length is different for different stages in an aero engine. Compressor blades are usually very short whereas the large stage blades are longer.

| Parameter | |
|---|---|
| Blade Length | 90 mm |
| Number of Blades | 60 |
| Blade Platform Radius (R2) | 225 mm |
| Rotor and blade platform thickness | 30 mm |

In the case of Blisk, there are no contacts at the blade and root region. Hence there is only one condition where analysis is performed for. The natural frequency of the blade is obtained by performing a modal analysis on the body. This gives us the natural frequencies of the structure. The first 6 natural frequencies are shown in the table below

| Mode | Frequency (Hz) |
|---|---|
| 1 | 765.47 |
| 2 | 2092.5 |
| 3 | 3712.5 |
| 4 | 3750.5 |
| 5 | 4452.6 |
| 6 | 9319.9 |

Fig: Natural frequencies of the blisk

Using these natural frequencies and the calculated Running Speed Harmonics, a Campbell diagram is plotted as shown in the figure below. The points at which the Running Speed Harmonic, the mode lines and the vertical speeds lines are intersecting, resonance takes place. These points of resonance are shown inside the red box called exciter box.

The exciter box is called so because; it is where the maximum of the blade excitations take place.

Campbell diagram is a graph which represents the response spectrum of a system as a function of its rotation regime. The graph is plotted by running the geometry through modal analysis in ANSYS workbench under the specified boundary conditions and plotting the obtained natural frequencies with respect to their speeds. The X-axis represents the speed in rpm and the Y-axis the frequency in Hz. The natural frequencies for different speeds are collected and the graph is plotted for the individual modes of each speed. This shows the variation in the frequency for the mode for different speeds. Then by plotting the Running Speed Harmonics, the speeds for which resonance is occurring can be observed. The Campbell diagram for the blisk is shown below
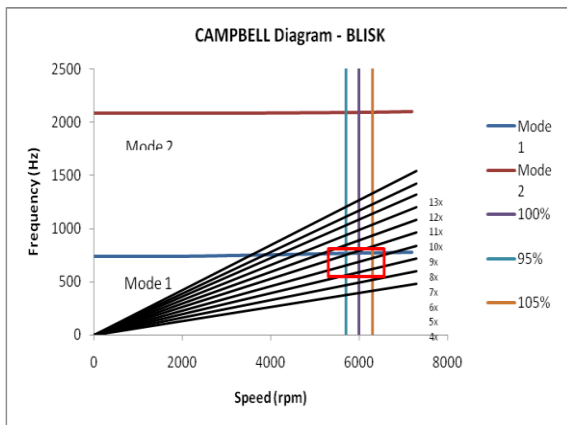


**Chart -1**: Campbell diagram for blisk at 100% speed (6000 rpm)

The exciter box for Mode 1 is shown below. The exciter box is represented in the Campbell by the red rectangle
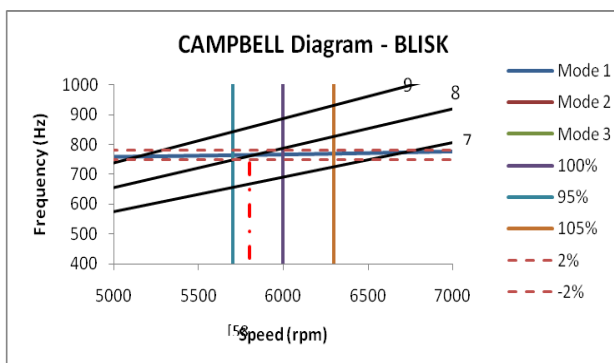


**Chart -2**: Exciter box for the Campbell diagram for the blisk

The point where the Running Speed Harmonics, the Mode lines and the vertical speed lines intersect is called the resonance point. As there is a ±2% above the Mode 1

frequency line, there might be multiple resonance points. The red line shown in the above figure is the critical speed.

From the exciter box, it is observed that for the 6050 rpm there is resonance present in Mode 1. Convert the speed in rpm into number of cycles per second or Hz and similarly to other chronologies. They are represented in the table below –

| Chronology | No. of Cycles {Hz} |
|---|---|
| 1 Second | 96.66 |
| 1 Minute | 5800 |
| 1 Hour | 34800 |
| 1 Day | 8.3e6 |
| 1 Week | 5.8e7 |
| 1 Month | 2.3e8 |
| 1 Year | 2.8e9 |

Table: Table showing the blade cycles in different chronologies for the blisk

## 2.2.2 Cyclic Symmetric Analysis of blade and disk assembly with bonded contact
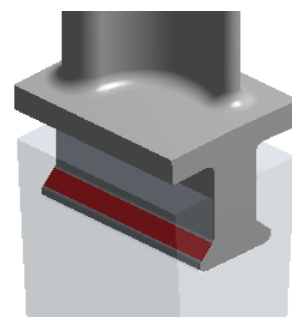


Fig: Contact region

A blade and disk assembly is designed as a sector and cyclic symmetry is applied to it as the stresses and its effects are repetitive across all the blades and the rotor. The contacts are assumed to have 100% fixity condition that is with bonded contacts.

A pre-stressed modal analysis is performed for 100% speed that is for 6000 rpm with all the necessary boundary conditions and the natural frequencies are obtained. A Campbell Diagram is drawn for this speed to evaluate the different frequencies for which resonance is occurring at this speed.

The Campbell diagram for 6000 rpm or 100% speed is shown below-
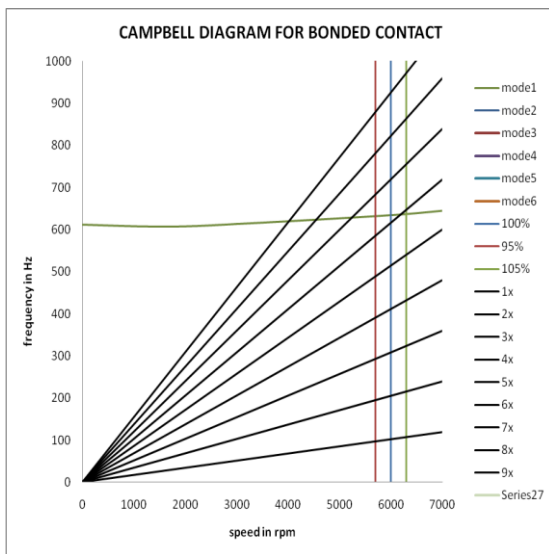
**Chart -3**: Campbell Diagram for 100% speed blade disk assemble with bonded contacts

The above Campbell diagram shows the regions or speeds for which resonance could take place. It can be observed in the Campbell Diagram that only Mode 1 and Mode 2 are in focus. It can also be observed that for Mode 1 there is interference for the 6th Running Speed Harmonic or 6X. Hence, an Exciter Box is drawn for that mode and the corresponding Running Speed Harmonic. The position for which the exciter box is drawn is represented in the red rectangle in the Campbell diagram. The exciter box for the Campbell diagram for Mode 1 is shown below –
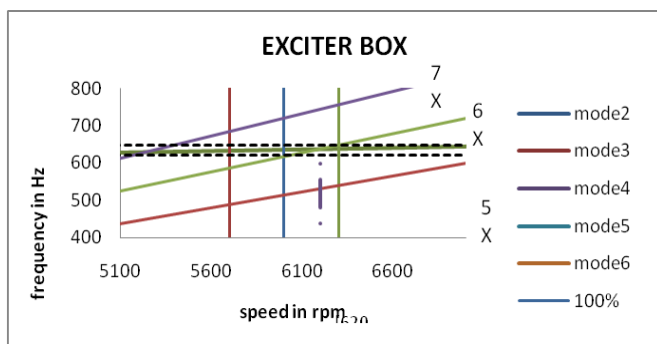


**Chart -4** Exciter box for Mode 1 of Campbell Diagram for Bonded contacts

The point(s) where the Mode lines, the Running Speed Harmonics and the vertical speed lines intersect are called Resonance Points. From the exciter box plotted, it can be observed that resonance is taking place at a 6200 rpm. The dotted lines represent the 2% allowance which is given to the Mode frequency in order to accumulate any minute errors or variation in the frequencies of the different blades. Therefore, at 6200 rpm resonance will occur at 6200 rpm. Now, to obtain the number of cycles per second we need to

convert rpm to rps which is given by dividing the speed in rpm by 60.

Therefore for one second, the blade will undergo 103 cycles or 103 Hz. Similarly, taking this forward we can calculate the number of cycles the blade will undergo in one day, a week or even a year. The values have been calculated and tabulated below

| Chronology | No. of cycles (Hz) |
|---|---|
| 1 second | 103 |
| 1 Minute | 6,200 |
| 1 Hour | 37,200 |
| 1 Day | 89,28,000 |
| 1 Week | 6.24e7 |
| 2 Weeks | 1.24e8 |
| 1 Month | 2.67e8 |
| 1 Year | 3.2e9 |

Table: Table showing the blade cycles in different chronologies

## 2.2.3 Blade and disk with frictional contact

In a blade and rotor assembly, friction acts as a damping agent. Hence, in the presence of friction the frequencies tend to dampen a little. This is because the vibrational energy is converted into heat at the contact regions. This results in the decrease in the vibrational energy which would otherwise result in vibration of the blade or rotor which would probably lead to catastrophe. Even in real life conditions, no metallic surface would have zero friction co-efficient; hence performing an analysis with friction co-efficient we would obtain results close to real life values.
Friction Co-efficient 0.1

The natural frequency of the blade and rotor assembly with 0.1 friction contact is

| Condition | Frequencies for 6000 rpm | | | | | |
|---|---|---|---|---|---|---|
| 0.1 | 465.54 | 1039.6 | 2642.2 | 2921.2 | 3329.2 | 4302.5 |
| Bonded | 634.38 | 1424.6 | 3444.8 | 3758.4 | 7195.5 | 8586.2 |
| % Variation | 26.61 | 27.02 | 23.29 | 22.5 | 53.73 | 49.89 |

It can be observed that there is a decrease in the frequencies from that of bonded contacts because of the friction introduces between the blade and the rotor. This is due to the energy is gone into converting the vibrational energy into heat, which is dissipated. The Campbell diagram plotted for the bladed disk with 0.1 friction co-efficient is given below –
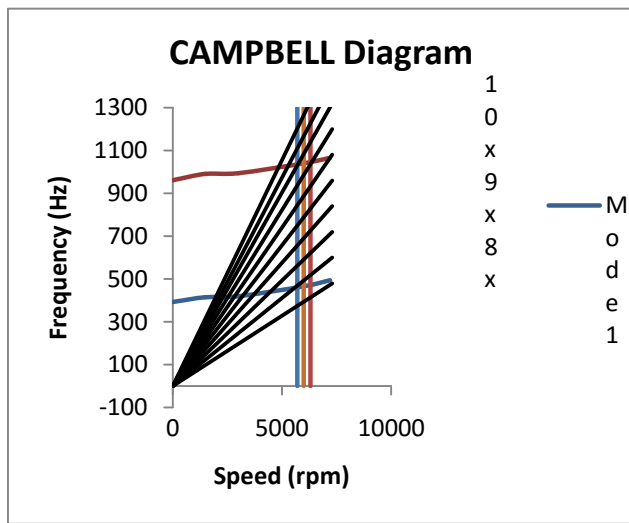
**Chart -5** Campbell Diagram showing Mode 1 and Mode 2 of blade and disk with 0.1 friction co-efficient

From the Campbell diagram, it is observed that for the 5770 rpm there is resonance present in Mode 1, Converting the speed in rpm to number of cycles per second or Hz and similarly to other chronologies. They are represented in the table below

| Chronology | No. of Cycles {Hz} |
|---|---|
| 1 Second | 96 |
| 1 Minute | 5770 |
| 1 Hour | 346200 |
| 1 Day | 8.3e6 |
| 1 Week | 5.8e7 |
| 1 Month | 2.3e8 |
| 1 Year | 2.8e9 |

Table: Table showing the blade cycles in different chronologies for 0.1 friction co-efficient

## 2.2.4 Blade Life Estimation using Goodman Diagram

Goodman diagram is a graph obtained by plotting mean stress and alternating stress which shows the number cycles at which the material fails.

The Goodman Diagram is obtained by plotting the alternating stress along the Y-axis and the mean stress along the X-axis. The Alternating stress is obtained using the Stress vs. Number of cycles [S-N] Curve of the material. The curve is the material data and is obtained by experimental data results. A sample S-N curve for steel is shown in the figure below-
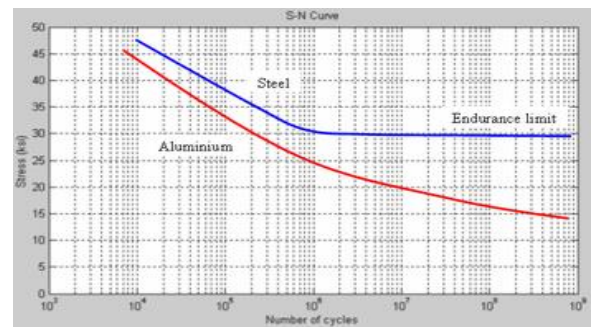


**Chart -6** S-N curve for steel and Aluminum

Now, infinite life is represented by 1e7 cycles. Hence, using the S-N graph for the material, the alternating stress of the material corresponding to 1e7 cycles is taken. The mean stress for the material in this case the yield strength of Chrome Steel is plotted along the X-axis and 80% of the yield stress [550 MPa] is considered as alternating stresses i.e. 440 Mpa thus leaving 20% as a safety factor. The alternating stress or the endurance stress is taken from the S-N curve for 1e7 cycles. This is then plotted to obtain the Goodman Line in the graph.

Now, the mean stress is obtained by performing a static structural analysis on the model under cyclic symmetry loading and the necessary boundary conditions along with a rotational velocity of 100% of the rotating speed i.e. 6000 rpm. The stress levels are observed at the base of the blade at which a section plane is taken. The average stress resulting is then taken as the mean stress. In this case it is around 93.857 MPa.
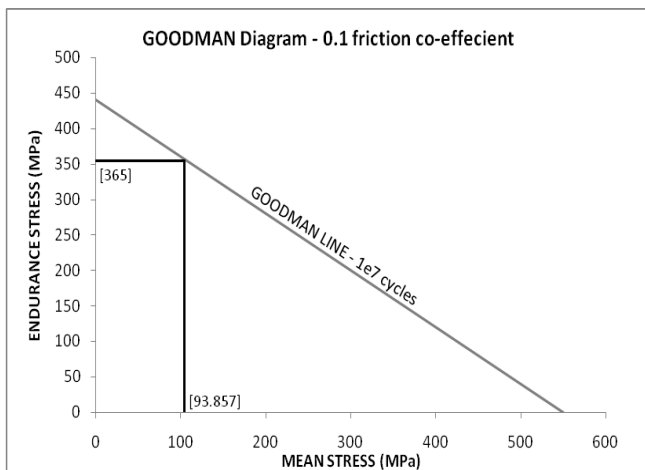
Plotting this on the Goodman Diagram, the obtained alternating stress is found using the graph.
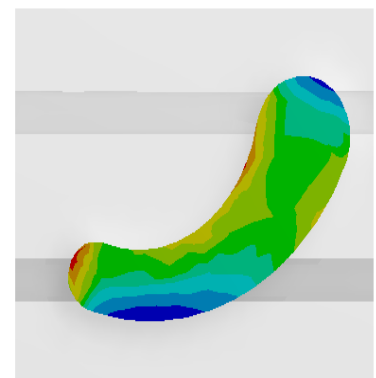


Fig: Stresses obtained at blade base for 0.1 friction co-efficient

**Chart -7** Goodman diagram for blade and disk assembly with 0.1 friction co-efficient

The 1e7 cycles is compared with the table of the number of cycles in various chronologies. By comparing it is determined that under resonance the life of the blade is around 28.9 Hours.

## 3. Comparison of Results

In this chapter, the results of all the analysis performed between the blisk, blade and rotor assembly with bonded contact and with varying friction co-efficient.
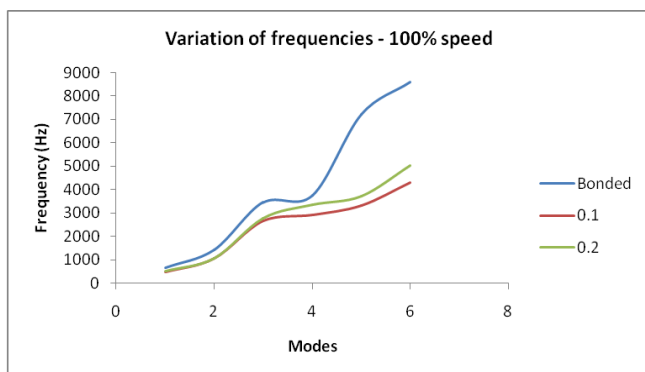


Fig: Variation of frequencies for different contacts

The above figure shows the variation of the frequencies for the various contact conditions. It is observed that the frequency drops as the stiffness in the structure is decreased. The frequencies for the bonded structure are found to have the highest while the ones with frictional contacts have lower frequencies. It is also observed that as the friction co-efficient increases, the frequencies decrease.
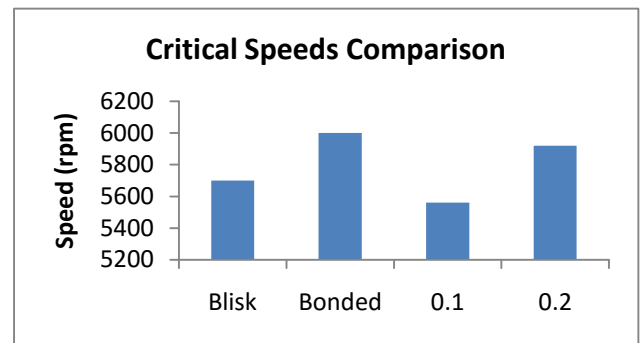


**Chart -8** Comparison of the critical speeds for different contact conditions

The above graph clearly represents the variation in the critical speeds for the various contact conditions to which analysis was performed for. It can be observed that for bonded condition, the critical speed is found to be the highest**.**

## 4. CONCLUSIONS

The study involved strength evaluation w.r.t vibrations in rotating machinery, the causes and effects of vibrations in rotor blades for two conditions 1. Bladed disc assembly 2. Blisk. Following observations are made as concluding remarks

1. The physics behind the disc mode vibrations and its participation in bladed disk assembly is understood for both the cases.
2. The source of excitations in turbo machines and its evaluation through Campbell diagram
3. Customized methodology to evaluate the separation margins in blisk and bladed disc, considering nozzle passing frequency  running speed harmonics and through stimulus approach HCF life estimation is successfully established.
4. Role of friction in modal analysis and its effects on frequency evaluation is also carried out

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